DOI: 10.7251/JEPM2102042P
UDC: 004.42MATLAB:661.321.31

Journal of
Engineering & Processing Management

SCIENTIFIC PAPER

Numerical simulation of soda ash drying process in pneumatic drying system with industrial scale

Edisa Paprаćanin[1] | Nisad Avdić[2] | Abdel Đozić[3] | Ervin Karić[1]

[1]Department of Chemical Engineering, Faculty of Technology, University of Tuzla, Urfeta Vejzagića 8, 75000 Tuzla, Bosnia and Herzegovina

[2]Sisecam soda Lukavac, Prva Ulica br.1, 75300 Lukavac, Bosnia and Herzegovina

[3]Department of Environmental Engineering, Faculty of Technology, University of Tuzla, UrfetaVejzagića 8, 75000 Tuzla, Bosnia and Herzegovina

Correspondence
Edisa Paprаćanin, Department of Chemical Engineering, Faculty of Technology, University of Tuzla, Urfeta Vejzagića 8, 75000 Tuzla, Bosnia and Herzegovina
Email: edisa.papracanin@untz.ba

Abstract

In this paper a mathematical model for the soda ash drying process in a pneumatic dryer was presented. The model presents a macroscopic aspect of the drying process, for a two-phase, gas-solid system. The model is based on mass and heat transfer between the gas phase and the particle, movement of air and particles through the system, and geometric characteristics of the drying system (fan, air heater, pneumatic dryer, and cyclone). The effects of the process parameters, such as airflow, inlet air temperature, and relative humidity, temperature at the inlet of the dryer, etc., have been studied by solving the model. Also, the model was tested for different values of the capacity of wet soda and different values of the operating parameters of the heating medium. The model was implemented in MATLAB and solved with a nonlinear equations solver. Data obtained by the model were compared with industrial pneumatic dryer data for drying wet soda ash particles with good agreement.

Keywords: industrial scale, mathematical model, pneumatic dryer, soda ash, numerical simulation

1. INTODUCTION

The drying process is an energy-intensive process that is of great importance for both the chemical and food industries. In the chemical industry, it is most often used for the production of detergents and glass, in the food industry, sodium bicarbonate is used as an additive marked E-500, while in the pharmaceutical industry it is used for cosmetic purposes. The selection of an adequate drying system is a complex process and requires complete knowledge of the characteristics of the starting material as well as the expected characteristics of the final powder product (Arsenijević 2006). Sodium bicarbonate or baking soda is produced by the Solvay process (E. Solvay). The Solvay process is a well-known and efficient process for the production of sodium bicarbonate from sodium chloride (NaCl) (Luna & Martínez 1994). In the production of soda ash, the drying process is one of the basic processes, and the quality of the final product as well as energy efficiency of the process both depend on the drying process. Due to the characteristics of wet soda ash for drying, pneumatic dryers are most often used. Pneumatic dryers are characterized by simultaneous momentum, heat and mass transfer processes between the dispersed material and the drying agent. The large surface area for heat and mass transfer results in higher drying rate and higher drying capacity (El-Behery, El-Askary, Hamed, & Ibrahim 2012). During pneumatic transport, the pressure difference along the pipe is created to move the bulk material from a higher-pressure region to a lower-pressure region by using a vacuum inducer, blower or compressor. The average particle size of the drying material can be 0.05 – 2.00 mm. The circulation speed of the heated agent of drying (air or gas) in the dryer is 10 – 30 m·s^{-1} (Rajan, Srivastava, Pitchumani, & Surendiran 2010). The drying time of this dryer type is very short (0.5–10 s at the drying section) with the application of appropriate temperatures, which can vary from 100°C to 650°C.The system for the drying process usually consists of the following process units: fan, heat exchanger, dryer, cyclone and scrubber. (Josimovic, Prvulovic, Tolmac, & Mihajlovic 2020) approved in the conventional heat exchanger, the area of heat transfer is fixed but in the pneumatic heat exchanger the heat transfer area depends upon the concentration of particles. The most important aspect of drying technologyis is mathematical modeling of the drying processes and the equipment (Górnicki & Kaleta 2011). Full scale experimentation for different products and systems configurations is sometimes costly and even not possible. Mathematical modeling of drying process highly depends on the relations that describe mass phenomena, air properties, energy consumption, pressure drop, etc. Literature review offers research dealing with the kinetics of the process of narrowing, or the phenomena of mass and heat transfer in the material together with geometry of process units for thedrying system. Patro, Patro, and Murugan (2014) numerically investigated the influence of particle on gas-solid heat transfer and pressure drop. Heat transfer and fluid dynamic study in a vertical pneumatic bed was studied by (Narimatsu, Ferreira, & Freire 2007). Neugebauer et al. (2017) proposed a dynamic two-zone model relating to the formation of granulation and fluidized bed drying zones, with internal product classification. The model is used to study the influence of different process parameters on the dynamics and stability of the process. Literature review provides data on research dealing with mathematical modeling of mass and heat exchange in a pneumatic dryer, determination of drying speed, pressure drop, etc., (Chapuis, Precoppe, Méot, Sriroth, & Tran 2016; Korn 2001; Li & Mason 2000). Satpati, Koley, and Datta (2019) investigated economic aspects of process control of pneumatic conveying dryer for drying of food grains. Jamaleddine and Ray (2011) simulated the drying of sludge in a pneumatic-cyclone dryer. The results showed that the combined pneumatic-cyclone drying mechanisms were successful, but quantitative predictions must be improved and validated using experimental data. Experimental research and verification of the proposed models for pneumatic conveying dryer is very limited. In order to verify the capacity of a drying process of soda ash in a pneumatic drying system, it is necessary to carry out the project and exploitation calculations, which refers to the verification of existing operating parameters and capacity increase in industrial scale. The main goal of this paper is to develop a simple mathematical model, to obtain data of the soda ash drying process in a pneumatic dryer. Also, the goal is to evaluate the model data with the data obtained from an industrial plant.

2. MATERIAL AND METHODS

2.1. PROCESS PARAMETERS AND INDUSTRIAL DRYING SYSTEM

In this study, data from the industrial plant for the drying of sodium bicarbonate (Sisecam factory of Soda Lukavac, BiH) was used as input for the mathematical model. Since 2014, the production of baking soda in this factory has amounted to 500,000 tons per year. The drying process of wet soda ash is performed in a pneumatic dryer with accompanying process units and together forms a drying system. The drying system (in this factory) consists of two lines, each of which consists of a fan, air heater, pneumatic dryer, cyclone and scrubber. Pneumatic dryers are with a diameter at the inlet DN500, and DN1200 at the outlet, with total length of 12m. Cyclone diameter is 1800 mm, with tangential inlet connection dimensions 700x400 mm and drain connection 650 mm, and a total height of 7720 mm. Drying is performed with hot air (160°C) which is heated with superheated steam with a pressure of 12 bar and a temperature of up to 270°C. Heating is performed in a heat exchanger with ribbed tubes with the geometric characteristics given in Table 1.

Parameters were measured at the inlet of the air heater: air temperature, relative humidity, airflow, pressure, and the temperature of the steam. After the heater, the air temperature at the inlet and the outlet of the dryer was measured. Also, the moisture content and the temperature of the soda ash at inlet and outlet of the dryer were measured.

2.2. THE DEVELOPMENT OF THE MATHEMATICAL MODEL

The mathematical model was developed in relation to the pneumatic drying system of Sisecam Soda Lukavac factory. Figure 1 shows the process scheme of the drying system. This model is presented as deterministic and consists of several submodels as described below. The advantage of this approach to mathematical modeling of the drying process is the fact that it is primarily a simple description of the transfer phenomenon based on the heat and mass conservation law. Given this fact, it is not difficult to conclude that this approach makes it possible to describe in detail the drying process, where this model can be solved as a design and as exploitation, depending on needs. The model shows the relations of mass and heat balance for each process unit, as well as the relations for calculating the geometric characteristics of process units. Also, relations for calculating the thermo-physical properties of air and soda ash are built into the model. The model is developed by the experience of the authors in combination with theoretical equations.

Table 1. The geometry of the heat exchanger for air heating (data from industrial scale "Sisecam.factory of Soda Lukavac").

	Unit	Value
Height of heater	mm	1430
Width of heater	mm	1245
Length of ribbed pipe section	mm	1400
The outer diameter of the steel pipe	mm	21.3
Wall thickness of the steel pipe	mm	2.3
The outer diameter of the heaters rib	mm	49
The thickness of the heater ribs	mm	0.6
The distance between the ribs	mm	2.5
Number of ribs per 1 m of pipe length	-	400
Number of ribed tubes in a row	-	24
Coefficient of the number of rows of pipe eq. (6)	-	0.98

2.2.1. *A mathematical model of heat exchangers*

The mathematical model of the air heater is related to the mass and heat balance and geometric characteristics of the heater. As mentioned earlier, for air heating, which is inserted by a fan, a ribbed tube heat exchanger was used.

The heat balance of the heat exchanger related to the moist air can be the simplest to introduce with a heat-balance equation (stationary condition):

$$\dot{m}_a \cdot \hat{h}_{ina} + Q_h = \dot{m}_a \cdot \hat{h}_{outa} \tag{1}$$

where $\dot{m}_a$- mass flow of air(kg/s), $\hat{h}_{ina}, \hat{h}_{outa}$- specific enthalpies of moist air, at the inlet and outlet of the heat exchanger depending on the temperature and moisture content (kJ/kg) (Ražnjević 1989), Q_h- quantity of heat that is necessary to heat the moist air (kJ/s).

The amount of heat supplied by the heating fluid can be calculated from the following relation:

$$Q_t = \dot{m}_p \cdot \widehat{H_w} \tag{2}$$

Wherein:$\dot{m}_p$- mass flow rate of steam (kg/s), $\widehat{H_w}$- the specific heat of steam as a function of temperature and pressure (kJ/kg) (Hopmans 2006).

The amount of heat for a steady state with the overall heat transfer coefficient for heat transfer is calculated by the expression:

$$Q_t = K \cdot A \cdot t_m \tag{3}$$

Wherein: K – overall heat transfer coefficient for ribbed tube, W/m^2·K,A – surface for heat exchange, m^2,t_m- mean temperature difference in the heat exchanger.

The overall heat transfer coefficient in the finned tube exchanger can be calculated by the relation:

$$K = \frac{1}{\frac{1}{\alpha_1} \cdot \frac{\phi \cdot d_v}{d_u} + \frac{\phi \cdot d_v}{2 \cdot \lambda} \cdot \ln \frac{d_v}{d_u} + \frac{1}{\alpha_2}} \tag{4}$$

Where: $_1$-convective heat transfer coefficient for steam side of tube wall (W/m^2°C), dv- outer diameter of steel pipe (m), d$_u$-inner diameter of steel pipe (m), ϕ- coefficient of the fin tube, - coefficient of thermal conductivity of the material (W/mK), $_2$ - convective heat transfer coefficient for air side of tube wall (W/m^2·°C).

Mean logarithmic temperature difference in the heater:

$$t_{lm} = \frac{\left(t_1^{"} - t_2^{"}\right) - \left(t_1^{'} - t_2^{'}\right)}{\ln \frac{t_1^{"} - t_2^{"}}{t_1^{'} - t_2^{'}}} \tag{5}$$

Where t1"- steam temperature at the inlet to the heater (°C), t1'-steam temperature at the outlet of the heater (°C), t2"- air temperature at the heater outlet (°C), t2'- air temperature at the inlet to the heater (°C).

Mean temperature difference in the heat exchanger:

$$t_m = \varepsilon_t \cdot t_{lm} \tag{6}$$

Where $_t$ - temperature correction coefficient, which is obtained based on the coefficient of the number of rows $\varepsilon$and temperature difference.

Relations for calculating the geometric characteristics of heat exchangers with ribbed tubes and the relations for the convective heat transfer coefficientare as follows:

Number of ribs per 1 meter of pipe length:

$$n_r = \frac{1000}{x} \tag{7}$$

Where x is distance between ribs (mm).

Coefficient of reduction of the light surface of the heater section:

$$\chi = 1 - \frac{1}{s_1}\left(d_o + 2 \cdot \frac{h \cdot y}{x}\right) \tag{8}$$

Where: s$_1$ - the distance between the tubes (width) of the heater (m), d$_0$ - inner diameter of the tube (m), h- rib height (m), y- the thickness of the ribs heater (m).

Air velocity in the intercostal space:

$$w_2 = \frac{w_g}{\chi} \tag{9}$$

Wherein: w_g -air velocity in cross section in front of the heater section (m/s).

Heater tube finning coefficient:

$$\phi = 1 + \frac{2 \cdot h}{x \cdot d_c}(d_c + h + y) \tag{10}$$

Where d_c is tube diameter (m).

Nusselt's number for air flow in the intercostal space (Sapozhnikov, Mityakov, Gusakov, Seroshtanov, & Subbotina 2019; Tarawneh, Hammad, et al. 2012)):

$$Nu_2 = 0.132 \cdot \left(\frac{s_1 - d_v}{s_2 - d_v}\right)^{0.2} \cdot \left(\frac{d_v}{x}\right)^{-0.54} \cdot \left(\frac{h}{x}\right)^{-0.14} \cdot \mathrm{Re}^{0.73} \tag{11}$$

Where: s_1 - distance between the tubes (width) of the heater (m), s_2 - distance between the pipes along the length of the heater (m). *Re* – Reynolds number.

The coefficient of heat transfer through convection from the ribbed tube to the air:

$$\alpha_2 = \frac{Nu_2 \cdot \lambda_2}{x} \tag{12}$$

Wherein $\lambda_2$coefficient of thermal conductivity of air (W/mK).

The efficiency coefficient of the ribs is a function of the product ·h and the diameter ratio D/d:

$$\beta \cdot h = h \cdot \sqrt{\frac{2 \cdot \alpha_2}{\lambda \cdot y}} \tag{13}$$

Real heat transfer coefficient from the ribs to the heated fluid:

$$\alpha_{2s} = \left(\frac{A_r}{A_c} \cdot E \cdot \psi + \frac{A_{gl}}{A_c}\right) \cdot \alpha_2 \tag{14}$$

Where A_r- surface of ribs per 1 m length of ribbed tube (m^2), A_c-total area of 1 m ribbed pipe (m^2), *E*- coefficient of rib efficiency from the graphs for D/d and product h, – coefficient of heat transfer distribution (0.96).

Pressure drop in the heater was calculated by the following relation:

$$p = 2.7 \cdot n \cdot \left(\frac{l_0}{d_{ek}}\right)^{0.3} \cdot \mathrm{Re}_{lo}^{-0.25} \cdot \frac{\rho_{s2} \cdot w_2^2}{\nu_{s2}} \tag{15}$$

Where *n* - number of rows of pipes, ρ_{s2} - air density for the mean temperature difference of the air in the exchanger (kg/m^3), w_2- air velocity in the intercostal space (m/s), ν_{s2}- kinematic viscosity of air (m^2/s), l_0- the circumference length of the ribbed tube, Re_{lo}- Reynolds number for the narrowest beam cross section.

The circumference length of the ribbed tube is calculated by the following expression:

$$l_0 = \frac{A_{gl}}{A_c} \cdot d_v + \frac{A_r}{A_c}\sqrt{\left(D_r^2 - d_v^2\right) \cdot \frac{\pi}{4}} \tag{16}$$

2.2.2. *Mathematical model of the dryer*

The drying process can be described by mathematical relations that are based on the of heat and mass transfer phenomena. Ambient air is used as a drying agent, so it is also necessary to calculate the parameters of moist air during modeling of the dryer.

Dryer capacity per dry material (kg/h):

$$\overline{m}_{sm} = \frac{\overline{m}_{vm} \cdot (1 + X_1)}{(1 + X_2)} \tag{17}$$

Wherein: X_1, X_2- inlet and outlet moisture content of soda ash ($kg_w/kg_{d.a}$),$\overline{m}_{vm}$- dryer capacity per wet material (kg/h).

The amount of water that needs to be removed, (kg/h):

$$\overline{m}_w = \overline{m}_{vm} - \overline{m}_{sm} \tag{18}$$

The amount of heat required for heating the sodium bicarbonate:

$$Q_m = \overline{m}_{sm} \cdot \left(c_{p2} \cdot t_{m2} - c_{p1} \cdot t_{m1}\right) \tag{19}$$

Where: c_{p1}, c_{p2}- specific heat of material (soda ash) at the inlet and outlet of the dryer (kJ/kg°C), t_{m1}, t_{m2}- the temperature of the material at the inlet and outlet of the dryer (°C).

The heat of vaporization of water from soda ash:

$$Q_w = \overline{m}_w \cdot \left[2499.04 + c_{pw} \cdot (t_2 - t_1)\right] \tag{20}$$

Wherein: c_{pw}- specific heat capacity of water (kJ/kg°C); t_2, t_1- temperatureof the water at the inlet and outlet (°C) and number 2499.04 is latent heat of condensation of water at 0°C (Ražnjević 1989).

The total amount of heat of the dryer (kJ/h):

$$Q_{S=}Q + \overline{m}_a \cdot \left(\hat{h}_3 - \hat{h}_1\right) - \overline{m}_{sm} \cdot c_{p2} \cdot t_{m2} \tag{21}$$

Q- Total heat losses in the dryer (kJ/h), $\hat{h}_3$- specific enthalpy of air at the outlet of the dryer (kJ/kg), $\hat{h}_{outa}$- enthalpy of air at the inlet of the dryer (kJ/kg).

2.2.3. *Additional relations for the dryer calculations*

Since this is a process of drying the material in a fluidized bed, it is necessary to report the relations related to the calculation of the fluidized bed, such as porosity, velocity of the drying air, pressure drop, etc.

The porosity of the layer can be calculated from the following relation:

$$\varepsilon = \frac{V_3}{V_3 + \frac{\overline{m}_{sm}}{\rho_3}} \tag{22}$$

Wherein V_3- airflow at the the cyclone inlet (m^3/h) and ρ_3- outlet air density of the dryer (kg/m^3).

Mass ratio of the material to be dried and the air in the dryer:

$$\mu_{mo} = \frac{\overline{m}_{sm}}{V_3 \cdot \rho_3} \tag{23}$$

Density of the mixture in the dryer:

$$\rho_{sm} = \frac{(1-\varepsilon)\cdot\rho_m}{\varepsilon\cdot\rho_3} \tag{24}$$

where ρ_m density of soda ash (kg/m^3).

For the calculation of the mean particle diameter, the Archimedes (Ar) expression was used:

$$Ar = \frac{g\cdot d_e^3\cdot(\rho_m-\rho_3)}{\rho_3\cdot\nu^2} \tag{25}$$

Wherein: g gravitational acceleration (m/s^2), d_e- equivalent particle diameter at the sieve residue R=25 %(μm), ν- kinematic viscosity of the air at the outlet of the dryer (m^2/s).

The speed of the air in the dryer can be calculated based on the diameter of the dryer and the volume flow of air in the dryer, as follows from the relation:

$$w_o = \frac{4\cdot V_2}{3600\cdot\pi\cdot D_s^2} \tag{26}$$

D_s- he average diameter of the dryer (m), V_2- air flow at the inlet to the dryer (m^3/h).

The velocity of medium-diameter particles for a compressed flow can be calculated from the relation:

$$w_m = w_L\left[1-\left(\frac{d_e}{D}\right)^2\right] \tag{27}$$

Wherein: w_{L}- the limiting velocity of a single grain (m/s), D - cyclone diameter (m).

The expression for the velocity of particles of medium diameterd_e represents a modified Newton correction factor (1-c)$^{2.4}$=1 (the value of c-concentration of solid particles in the mixture) for normal industrial conditions.In relation to the free movement of particles, many forces affect particles in compressed conditions. The ratio of these forces is expressed through the deposition rate, and in conditions along the vertical cylindrical vessel, due to the finite boundaries of space, this influence is not negligible. During the drying process, it is necessary to check the critical speed or the Reynolds number in the case of pneumatic drying, i.e., the transport of particles in the dryer. Also, it is necessary to calculate the value of the Nusselt criterion as well as the pressure drop in the dryer.

The Reynolds number for the critical particle velocity (Re$_{\mathrm{cr}}$) can be calculated from the following expression:

$$\mathrm{Re}_{cr} = \frac{Ar\,(1-\nu_m)^{4,75}}{18+0,6\cdot\sqrt{Ar\cdot(1-\nu_m)^{4,75}}} \tag{28}$$

Where ν_{m} volume rate of soda ash in the dryer (m^3/s).

The Nusselt number is typically calculated from one of the many correlations existing in the literature. In the case of fluidized bed, Gunn (1978) proposes the correlations of equation (29), which can be used for a range of bed porosity of 0.35—1.0 and the particle Reynolds number up to 10^5:

$$Nu_{kr} = 0,6\cdot\sqrt{\mathrm{Re}_{kr}} \tag{29}$$

The surface area of a single particle:$A_m = \frac{\overline{m}_{sm}}{600\cdot d_e\cdot\rho_m}$ (30)

Heat transfer between a single particle and the gas phase can be defined by the conventional equation of heat transfer:

$$Q_{con} = A_m\cdot\alpha\cdot\left(t_2{}'' - t_p\right) \tag{31}$$

Where:t_p- is the temperature of the particle (°C), -heat transfer coefficient (W/m^{2}K).

The time required for drying can be calculated from the total amount of heat supplied in the dryer from the following relation:

$$\tau = \frac{1000\cdot Q_s}{\alpha\cdot A_m\cdot t_{\log S}} \tag{32}$$

$t_{\log S}$- mean logarithmic temperature difference in the dryer.

The mean logarithmic temperature difference in the dryer can be calculated by the expression:

$$t_{\log s} = \frac{\left(t_2{}''-t_{m1}\right)-\left(t_3-t_{m2}\right)}{\ln\frac{t_2{}''-t_{m1}}{t_3-t_{m2}}} \tag{32}$$

The pressure drop in the dryer can be calculated from the relation obtained from Bernoulli's equation for the flow of real fluids and the particle momentum equation, i.e., the integral of the differential equation of motion of the mixture in a limited space, i.e., as the sum of the pressure drop due to the friction of the mixture against the walls, the pressure drop for lifting the material and the transport fluid (air) and the pressure drop for the acceleration of the particles or the transport fluid:

$$p = \rho_s \cdot \frac{w_o^2}{2} \cdot \frac{L}{D_s} \cdot \left(1 + \lambda_t + k \cdot \mu_{mo} + 2 \cdot \frac{w_m}{w_o} \cdot \mu_{mo}\right) \quad (31)$$

w_0 - air velocity in the dryer (m/s), L- total length of dryer (m), $_t$ - dryer friction coefficient, k - coefficient of tangential stress due to material movement, w_m- mean particle velocity for compressed flow (m/s).

Smoldier's expression for the coefficient for vertical transport of particles at the ratio $Fr^2/a<0.5$, was calculated by the expression:

$$k = 0.17 \cdot \sqrt{\frac{\alpha}{F_r}} \quad (35)$$

α- Heat transfer coefficient from air to soda (W/m^{2}K), Fr - Frud's number for the mean diameter of the dryer.

Also, it is necessary to calculate the concentration of particles at the entrance in to the cyclone, or at the exit from the dryer, kg/kg:

$$c_c = \frac{1000 \cdot \overline{m}_{sm}}{V_3 \cdot \rho_3} \quad (36)$$

2.2.4. *Equations for cyclone calculation*

As the calculation of the cyclone is based on theoretical assumptions and the calculation of geometrically similar models, a geometrically similar cyclone collector was found in the literature (NIOGAS CN 34), on the basis of which the calculation was performed to check the pressure drop in the cyclone. The pressure drop significantly affects the performance parameters of a cyclone and the presence of a solid will increase the pressure drop (Pelegrina & Crapiste 2001).

A widely used empirical pressure drop equation is as follows:

$$p = \xi_c \cdot \frac{\rho \cdot w_D^2}{2} \quad (37)$$

Where :ξ_c - the factor of hydraulic resistance of the cyclone, w_D- speed of the gas-particle flow in the cyclone (m/s), ρ- air density at the inlet of the cyclone (kg/m^3).

2.3. INPUT DATA FOR SIMULATION

For the numerical simulation of the drying process, data from the soda drying plant were used, which include thermo physical data of soda, energy sources, and geometric characteristics of the process units, as well as literature (tabular) data. To perform numerical simulations based on the material and energy balance of processes in individual process units, a special sheet in MS Excel was created, as well as a mathematical model implemented in the MATLAB software package. To solve the mathematical model, solvers for solving linear and nonlinear algebraic equations within MATLAB were used. The presented mathematical model will be used to test the operating parameters in individual process units, to reach the optimal values. The input data used for the dryer calculation are shown in Table 2. The air temperature at the inlet of the dryer was 160°C, and the moist air parameters were calculated for air temperature and different values of relative humidity. Simulations were performed for different values of initial and final moisture contents in the material under various values of industrial plant.

Based on the input air parameters (temperature and relative humidity), other moist air parameters were calculated (Mujumdar & Devahastin 2000) which were used in the simulations.

Thermo-physical air data were calculated for the mean temperature value. The model was solved for different values of dryer capacity of the wet material: 4300kg/h and 7500 kg/h (industrial values), noting that the results are given for both extreme cases of air temperature at the inlet to the dryer for which they have performed dryer calculations, as follows:

t_2=160°C, t_1=25°C and $_1$=90%,
t_2=130°C, t_1=25°C and $_1$=90%.

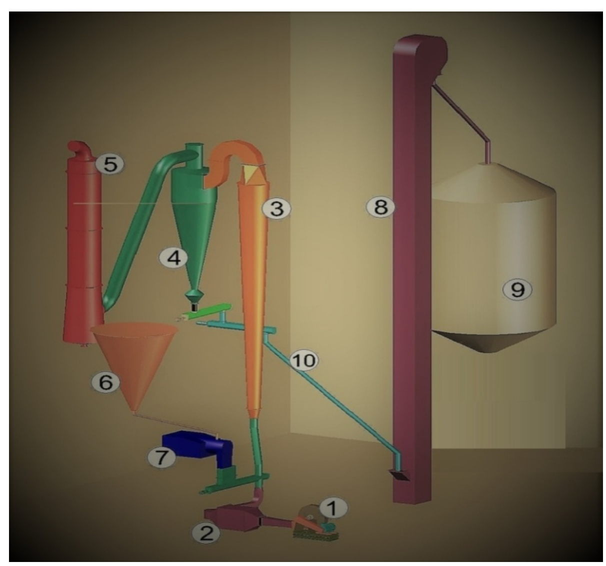

Figure 1. Scheme of the drying process of baking soda (1. air fan, 2. air heater, 3. dryer, 4. cyclone, 5. wet separator, 6. decanter, 7. centrifuge, 8. elevator, 9. silo).

As for other equipment, it was necessary to check the heater to determine whether the heater reaches the recommended capacity and what the optimal parameters concerning steam consumption and air temperature at the outlet of the dryer are. Also, the model vas solved for different values of air at the inlet and outlet of the heater.

Table 2. Input data of sodium bicarbonate used for simulation.**factory of Soda Lukavac).**

	Unit	Value
Inlet material temperature	°C	62
Outlet material temperature	°C	75
Specific heat capacity at the inlet	kJ/kgK	1.133
Specific heat capacity at the outlet	kJ/kgK	1.065
Density of soda ash	kg/m^3	2.200

Table 3. Literature data related to the heating medium (Ražnjević, 1989).

	Unit	Value
Steam density	kg/m^3	4.878
Kinematic viscosity of vapor	m^2/s	$3.960 \cdot 10^{-6}$
Specific heat capacity	J/kg°K	2286
Vapor conductivity coefficient	W/m^{2}K	0.0423
Enthalpy of steam at the inlet to the heater	kJ/kg	2970.6

The model was tested for the case of superheated and saturated steam of pressure 12 bar.

3. RESULTS AND DISCUSSION

To determine the energy required for heating the ambient air, the model was solved by varying the temperature and humidity at the inlet to the heater. Also, the model was solved to provide the required amount of air, with the appropriate parameters at the inlet to the dryer, or output parameters of the final product (temperature and moisture content of soda). The model for the pneumatic dryer was solved numerically to obtain data for drying of soda ash with hot air under different situations. When solving the model, it is necessary to take care that the concentration of the product at the inlet of cyclones (after drying) is within the limits for cyclones that work with medium-coarse dust. The change in the capacity of the ambient air fan and the concentration of baking soda at the inlet to the cyclone, depending on the temperature and relative humidity of the ambient air and the air temperature at the inlet to the dryer is given in Figure 2.

Figure 2 shows that at lower air temperatures there is a larger amount of air at the inlet to the dryer. Given the other limitations in the process, as well as the heating power of the heat exchanger, it can be concluded that the process is more efficient when the air temperature at the inlet of the dryer is 130°C. Further, it is necessary to check the restrictions with the consumption of steam and the particle size after the drying process. The change in the specific consumption of superheated steam depending on the temperature and relative humidity of the ambient air and the air temperature at the inlet to the dryer is shown in Figure 3. (Fyhr & Rasmuson 1997) investigated superheated steam properties in a model for a pneumatic-conveying dryer. They also investigated the effects of steam and material properties on the drying with different design parameters. In the case of a temperature value of 130°C at the inlet to the dryer, the specific steam consumption ranges from 0.15 to 0.23 t/t of dried soda. These values are much higher when the value of air temperature in the dryer is 160 °C, regardless of relative humidity. In the continuation of the research, numerical simulations were performed which showed the justification of specific steam consumption, i.e., air temperature at the inlet 160 °C dryer as is the case in an industrial plant. Similar results were obtained by Rajan, Srivastava, Pitchumani, and Mohanty (2006). To verify the model, numerical solutions were performed for values of dryer capacity with the wet material of 4300 kg /h, moisture content of baking soda at the inlet to the dryer 6%, moisture content at the outlet of the dryer 0.1%, and the already stated optimal inlet air values to the heater. The data obtained by solving the model for the heat exchanger are given in Table 4, while Table 5 presents the model results for the dryer process unit. Based on the previously obtained data, the parameters of the moist air were the inlet air temperature 10 °C and the relative humidity 70%. To verify the dimensional values of industrial equipment, listed in section 2, for operation in new industrial conditions, numerical solutions were obtained for the following parameter values:

- dryer capacity per wet material 7500 kg/h;
- moisture content of soda at the inlet 2.5 %.

Moisture content of wet soda at the outlet of the dryer is 0.1%. The values of the moist air parameters at the inlet to the system are as follows:

- air temperature at the inlet to the dryer 160°C, for ambient air parameters 25°C,$_1$=90%;

Table 4. Data obtained by solving the model for the heater.

Variables	Unit	Value	
		Dryer capacity of wet soda[1]	Dryer capacity of wet soda[2]
Air velocity in cross section in front of the heater section (w_g)	m/s	3.37	4.11
Coefficient of reduction of the bright surface of the heater section	-	0.46253	0.462
Cross-sectional area of the air passage heater (A_{sg})	m^2	1.1520	0.8234
Air velocity in the intercostal space (w_2)	m/s	7.289	8.89
The surface ribs on the 1 m long finned tubes (A_r)	m^2/m	1.224	1.223
The surface 1 m of the ribbed tube (A_{gl})	m^2/m	0.0510	0.0508
Total area of 1 m ribbed pipe (A_c)	m^2/m	1.274	1.270
Pipe finning coefficient (ϕ)	-	19.60	19.60
Total area of 1 ribbed heater pipe (A_H)	m^2	2.131	1.84
Reynolds number for intercostal space flow (Re_2)	-	827.05	1044.90
Nusselt's number for flow in the intercostal space (Nu_2)	-	4.31	5.12
Coefficient of transition from finned tube to heated fluid (α_2)	W/m^2K	51.658	60.30
Value βh	-	0.75	0.81
Real heat transfer coefficient from rib to heated fluid (α_{2s})	W/m^2K	45.95	48.10
Mass air flow	kg/s	8.07	7.29
The amount of heat transferred to the air (Q_h)	kW	1230	1000
Real fluid velocity in the supply line (wo)	m/s	13.56	18.59
Heating fluid velocity in the distribution pipeline (wd)	m/s	3.84	3.78
Reynolds number for heating medium (Re1)	-	4216	4216
Prandtlnumber for heating medium (Pr1)	-	1.04	1.04
Nusselt number for heating medium (Nu1)	-	15.90	15.66
Heat transfer coefficient of condensing steam (a1)	W/m^2K	3989	4034
Heat transfer coefficient of finned tube, k	W/mK	34.62	35.92
Mass flow of heating fluid	kg/s	0.60	0.48
Heating fluid supply pipe diameter	m	0.102	0.09

* Dryer capacity 4300 kg/h and moisture content of 6 to 0.1%.

* Dryer capacity 7500 kg/h and moisture content of 2.5 to 0.1%.

- air temperature at the inlet to the dryer 130°C, for ambient air parameters 25°C, $_1$=90%.

These parameters present the extreme values of ambient air parameters and temperature at the inlet to the dryer. The goal is to determine the optimal air temperature at the outlet of the air heater with the existing operating characteristics of ambient air fan, which provide soda ash concentrations at the cyclone inlet which do not exceed the maximum recommended value for this type of cyclone (max. 400 g/m^3). The results of heater with the existing operating characteristics of ambient air fan, which provide soda ash concentrations at the cyclone inlet which do not exceed the maximum recommended value for this type of cyclone (max. 400 g/m^3). The results of numerical solving of the model for the new process conditions are also given in Tables ?? and 6. Analysis of the calculated dryer parameters for different parameters of the air at the inlet to the dryer shows that the minimum energy consumption is for the value of the inlet air temperature of 25°C and relative humidity of 90%. In the case of 160°C at the inlet of the dryer, the concentration of soda ash at the inlet of cyclone is 585.62 g/kg which is above the recommended value. At the air temperature at the inlet to the dryer of 130°C, the specific steam consumption is 150 kg/t of dry soda, but the concentration of particles at the inlet to the cyclone decreases to ≈300 g/kg. The model shows that it is possible to increase the capacity in an industrial plant by adjusting the ambient air parameters and regulating the air temperature at the inlet of the dryer. From the industrial plant data of the baking

Table 5. Data obtained by the model for pneumatic dryer.

Variables	Unit	Values	
		Dryer capacity of wet soda[1]	Dryer capacity of wet soda[2]
The amount of water that needs to be removed	kg/s	0.0705	0.05005
Total heat required (Qs)	kJ/h	4418956	1633858
Heat required to dry the material (Qm)	kJ/h	4349514	542891
The amount of superheated steam	kg/h	2032.5	751.5
Air flow at the fan inlet in front of the heater	m^3/h	20920.8	10242
Air flow at the inlet to the dryer	m^3/h	32003.7	14880
Particle concentration at the entrance to the cyclone (cc)	g/kg	156.28	581.25
Bulk concentration of baking soda in the dryer (vm)	m^3/s	0.1479	0.5364
Porosity of the layer in the dryer (□)	-	0.99993	0.99975
Archimedes number for mean grain diameter (Ar)	-	44.1907	49.5567
Reynolds number for mean grain diameter (Re)	-	2.00972	2.22989
The limiting velocity of a single grain (wL)	m/s	0.4994	0.50414
Mean particle velocity for compressed flow (wm)	m/s	0.4994	0.50414
Reynolds number for the final particle velocity (Rekr)	-	0.99665	0.06884
The final particle velocity of baking soda (wkr)	m/s	0.24766	0.01556
Nuselt number for final speed (Nukr)	-	0.59899	0.15743
Coefficient of heat transfer from air to soda (a)	W/m^2K	182.32	46.92
Particle area per unit time (Am)	m^2/s	28.36	51.50
Mean logarithmic temperature in the dryer (□tlog)	^oC	74.64	72.71
Frud's number for the mean diameter of the dryer (Fr)	-	5.25238	2.34790
Reynolds number for air (Re)	-	518275	254325
Dryer friction coefficient (□t)	-	0.0123	0.0123

* Dryer capacity 4300 kg/h and moisture content of 6 to 0.1%.

* Dryer capacity 7500 kg/h and moisture content of 2.5 to 0.1%.

soda drying line, the values of the operating parameters of the cold air fan was as follows: air flow 16000 m^3/h and allowable pressure drop 4900 Pa. This value refers to the pressure drop for the fan, i.e., the heat exchanger and the dryer. The airflow value for optimal conditions is 21023 m^3/h obtained by themodel, which indicates that for these conditions it is not possible to achieve optimum capacity of wet soda, unlike the obtained value of pressure drop of 2157.5 Pa, which meets the limitations of the drying line. Pelegrina and Crapiste (2001) have similar conclusionsby testing the model for different air velocity values. Air velocity, regarding airflow, must exceed a certain minimum to ensure the stability of pneumatic transport. Air velocity values in the range 10-20 $m \cdot s^{-1}$ are usually recommended. The values obtained by this model, are in accordance with industrial data.

The thermal power of the heat exchanger is 1230 kW and the calculated value of the superheated steam heat is 1294.74 kW. As a compromise, the optimal air temperature at the inlet to the dryer of 145°C was determined, at which the ambient air fan operates at the allowed installed capacity, at a real extremely low ambient temperature of 10°C and relative humidity of 70%, with the concentration of particles at the entrance to the cyclone approaching the values that the existing cyclone can accept at a technically acceptable total degree of separation. The velocity incrases with solid loading (Pelegrina & Crapiste 2001) which is the case in this paper.The obtained value for the inlet airflow is 14880 m^3/h and the particle concentration is 290.84 g/kg, which is below the maximum allowed value (400 g/kg). The pressure drop, in this case, does not exceed the maximum allowed value of 4900 Pa and is 3804.11 Pa. The calculated value of outlet temperature of the dryer is 124.12°C. The total installed length of the dryer is 12 m. The values obtained by the simulations ranged approximately between 8 and 10 m, which means that the existing system could be additionally used for new conditions, with the installation of an additional

Table 6. Calculated values for different inlet temperatures at the dryer.

		Model data	
	Unit	tin=160°C	tin=130°C
Dryer capacity per dry material	kg/h	7319.82	7319.82
Air temperature at the outlet of the dryer	°C	130.3	108.6
Air velocity in the dryer	m/s	15.6	13.9
Drying time	s	1.1	0.51
Total length of the dryer	m	10.85	8.71
Particle concentration at the entrance to the cyclone(cc)	g/kg	581.2	299.7
Pressure drop for mass transport	Pa	4492	4185

fan and heat exchanger, in order to achieve the capacity in relation to the air passing through the system. Baeyens, van Gauwbergen, and Vinckier (1995) investigated the drying path.One of the key parameters for modeling a pneumatic dryer is the ratio of the amount of wet material to the air flow. The amount of water can change the granule coalescence and consolidation and increasing the amount of water affects granule hardness, porosity, and size distribution (Meng et al. 2016). Excessive amounts of water lead to dense and uncontrolled granule growth (Levy & Borde 1999). Results from the presented verified model can serve as the basis for the development of a more complex mathematical model for drying process. A one-dimensional steady-state mathematical model for dilute phase flow in a pneumatic dryer was presented by Mortier et al. (2011). The prediction of the numerical simulation was compared with experimental data of drying with air in a large-scale pneumatic dryer. One of the major advantages of using the models is that they are more suitable for computing many scenarios as opposed to expensive experiments (Thapa, Tripathi, & Jeong 2019). Modelling approaches based on principles and experimental data are important in designing, optimizing, and controlling critical quality attributes of pharmaceutical (Tarawneh et al. 2012) and other industrial processes.

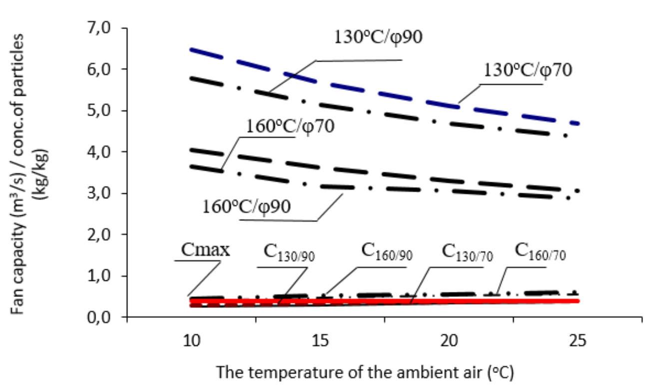

Figure 2. Changes of the ratio of the ambient air fan capacity and the concentration of soda at the inlet to the cyclone, depending on the temperature and relative humidity of the ambient air and the air temperature at the inlet to the dryer.

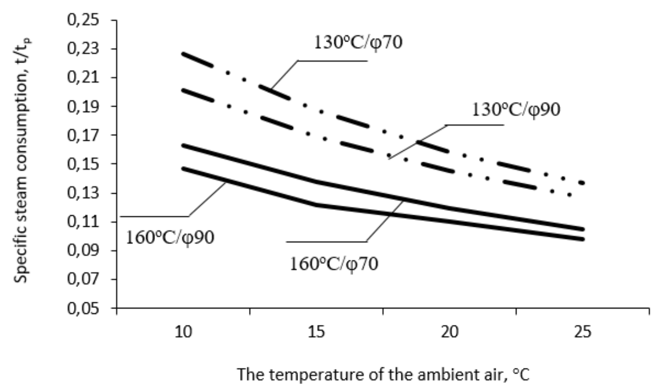

Figure 3. The effect of air temperature and relative air humidity on the superheated steam consumption (p=12 bar; t=270°C).

4. CONCLUSIONS

A one-dimensional mathematical model for pneumatic drying of baking soda with mechanical concepts was presented. The presented model shows a good agreement between numerical data and the data obtained from the industrial plant. Based on these results, simulations were performed for different capacity values which showed that is possible to perform optimization in the system in terms of increasing the capacity of the dryer and the airflow through the system. Simulations have shown that the main limitation is the concentration of particles at the outlet of the dryer. Following the above, the optimal regime conditions obtained by the model are air temperature at the inlet to the dryer of 145°C, at an inlet air temperature of 10°C and relative humidity of 70%. These conditions are necessary to achieve a dryer capacity of 7500 kg/h of wet soda. The presented model shows that the efficiency of the mentioned industrial dryer is about 55%, which leaves the possibility for optimization of the system and the model under the other capacities of the factory. For most of the compared values, the model shows good data agreement, but the values of some constants are taken from the literature, which could be calculated in the next step of the modeling using data obtained from an industrial plant. The level of detail embedded in a model depends on the goal of the model. Since a large number of limitations are built into the mathematical model – e.g., air values are calculated for the mean temperature, –future research should be directed at incorporating more detailed calculations related to the calculation of air parameters, steam and geometric characteristics, all to obtain the better agreement of models with industrial data that could serve as a basis for optimization of the industrial plant to save energy, i.e., to increase the energy efficiency of the process. The mathematical correlations should be adapted to the specific equipment to obtain reliable results. The model with mechanical concepts can be used to simulate different scenarios that would give results related to the length of the dryer, the drying time, as well as the capacities concerning the material and the drying air.

CONFLICT OF INTEREST

No potential conflict of interest was reported by the authors.

ACKNOWLEDGEMENTS

The authors would like to thank management of SISECAM Soda Lukavac, Bosnia and Herzegovina, for providing process data. This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors.

5. NOMENCLATURE

Ar – Archimedes number for the mean particle diameter,

C_z – coefficient of the number of rows in the longitudinal section of the beam,

c_{pw} – specific heat capacity of moisture (kJ/kg°C),

c_2 – specific heat capacity of the material at the outlet of the dryer (kJ/kg°C),

D_s– the average diameter of the dryer (m),

de – equivalent particle diameter at the sieve residue R=25% (m),

d_0 – inner diameter of the pipe (m),

dc – pipe diameter (m)

d_v – outer diameter of steel pipe (m),

d_u – inner diameter of steel pipe (m),

D_s – the average diameter of the dryer (m),

Fr – Frud's number for the mean diameter of the dryer,

f_c – total area of 1 m ribbed pipe (m^2),

f_r – surface of ribs per 1 m length of ribbed tube (m^2),

h – rib height (mm),

$\hat{h}$ – specific enthalpy (kJ/kg),

L – Total length of dryer (m),

$\overline{m}$– Mass flow (kg/h),

Nu – Nusselt's number,

Q – the amount of heat transferred (kJ/h),

Re – Reynolds number,

s_1– the distance between the pipes by the width of the heater (m),

s_2 – the distance between the pipes along the length of the heater beam (m),

w_g - Air velocity in cross section in front of the heater section (m/s),

w_L – limiting velocity of a single grain (m/s),

w_2 – air velocity in the intercostal space (m/s),

X – Moisture content in soda ash (%).

α – heat transfer coefficient (W/m^2K),

λ – conductivity coefficient ($W/m^2{}^\circ C$),

μ_{mo} – mass ratio of particles and air in the dryer,

ν – kinematic viscosity (m^2/s),

ρ – density (kg/m^3),

φ – Relative humidity (%),

ξ – the factor of hydraulic resistance of the cyclone

REFERENCES

Arsenijević, Z. (2006). *Sušenje rastvora i suspenzija u pokretnom sloju inertnih čestica, doktorska disertacija*. Tehnološko-metalurški fakultet, Univerzitet u Beogradu.

Baeyens, J., van Gauwbergen, D., & Vinckier, I. (1995). Pneumatic drying: the use of large-scale experimental data in a design procedure. *Powder Technology*, *83*(2), 139–148. https://doi.org/10.1016/0032-5910(94)02945-k

Chapuis, A., Precoppe, M., Méot, J. M., Sriroth, K., & Tran, T. (2016). Pneumatic drying of cassava starch: Numerical analysis and guidelines for the design of efficient small-scale dryers. *Drying Technology*, *35*(4), 393–408. https://doi.org/10.1080/07373937.2016.1177537

El-Behery, S. M., El-Askary, W., Hamed, M. H., & Ibrahim, K. (2012). Numerical simulation of heat and mass transfer in pneumatic conveying dryer. *Computers &amp Fluids*, *68*, 159–167. https://doi.org/10.1016/j.compfluid.2012.08.006

Fyhr, C., & Rasmuson, A. (1997). Mathematical model of a pneumatic conveying dryer. *AIChE Journal*, *43*(11), 2889–2902. https://doi.org/10.1002/aic.690431102

Górnicki, K., & Kaleta, A. (2011). Some problems related to mathematical modelling of mass transfer exemplified of convection drying of biological materials. In *Heat and mass transfer–modeling and simulation*. InTech.

Gunn, D. (1978). Transfer of heat or mass to particles in fixed and fluidised beds. *International Journal of Heat and Mass Transfer*, *21*(4), 467–476. https://doi.org/10.1016/0017-9310(78)90080-7

Hopmans, J. W. (2006). Principles of soil and plant water relations. *Vadose Zone Journal*, *5*(1), 506–506. https://doi.org/10.2136/vzj2005.0100br

Jamaleddine, T. J., & Ray, M. B. (2011). The drying of sludge in a cyclone dryer using computational fluid dynamics. *Drying Technology*, *29*(12), 1365–1377. https://doi.org/10.1080/07373937.2011.567946

Josimovic, M., Prvulovic, S., Tolmac, J., & Mihajlovic, V. (2020). Determination of the parameters of the drying process of sodium bicarbonate in a pneumatic dryer. *Machines. Technologies. Materials.*, *14*(5), 199–200.

Korn, O. (2001). Cyclone dryer: A pneumatic dryer with increased solid residence time. *Drying Technology*, *19*(8), 1925–1937.

Levy, A., & Borde, I. (1999). Steady state one dimensional flow model for a pneumatic dryer. *Chemical Engineering and Processing: Process Intensification*, *38*(2), 121–130. https://doi.org/10.1016/s0255-2701(98)00079-8

Li, J., & Mason, D. (2000). A computational investigation of transient heat transfer in pneumatic transport of granular particles. *Powder Technology*, *112*(3), 273–282. https://doi.org/10.1016/s0032-5910(00)00302-8

Luna, F., & Martínez, J. (1994). Selectivity during convective drying of a ternary mixture. In *9th international drying symposium (ids'94* (p. 969–976). Gold Coast, Australia: the University of Queensland.

Meng, W., Kotamarthy, L., Panikar, S., Sen, M., Pradhan, S., Marc, M., … Ramachandran, R. (2016). Statistical analysis and comparison of a continuous high shear granulator with a twin screw granulator: Effect of process parameters on critical granule attributes and granulation mechanisms. *International Journal of Pharmaceutics*, *513*(1-2), 357–375. https://doi.org/10.1016/j.ijpharm.2016.09.041

Mortier, S. T. F., Beer, T. D., Gernaey, K. V., Remon, J. P., Vervaet, C., & Nopens, I. (2011). Mechanistic modelling of fluidized bed drying processes of wet porous granules: A review. *European Journal of Pharmaceutics and Biopharmaceutics*, *79*(2), 205–225. https://doi.org/10.1016/j.ejpb.2011.05.013

Mujumdar, A. S., & Devahastin, S. (2000). *Fundamental principles of drying* (Vol. 1). Brossard, Canada: Exergex.

Narimatsu, C. P., Ferreira, M. C., & Freire, J. T. (2007). Drying of coarse particles in a vertical pneumatic conveyor. *Drying Technology*, *25*(2), 291–302. https://doi.org/10.1080/07373930601119599

Neugebauer, C., Palis, S., Bück, A., Tsotsas, E., Heinrich, S., & Kienle, A. (2017). A dynamic two-zone model of continuous fluidized bed layering granulation with internal product classification. *Particuology*, *31*, 8–14. https://doi.org/10.1016/j.partic.2016.07.001

Patro, P., Patro, B., & Murugan, S. (2014). Prediction of two-phase heat transfer and pressure drop in dilute gas–solid flows: A numerical investigation. *Drying Technology*, *32*(10), 1167–1178. https://doi.org/10.1080/07373937.2014.887095

Pelegrina, A., & Crapiste, G. (2001). Modelling the pneumatic drying of food particles. *Journal of Food Engineering*, *48*(4), 301–310. https://doi.org/10.1016/s0260-8774(00)00170-9

Rajan, K., Srivastava, S., Pitchumani, B., & Mohanty, B. (2006). Simulation of gas–solid heat transfer during pneumatic conveying: Use of multiple gas inlets along the duct. *International Communications in Heat and Mass Transfer*, *33*(10), 1234–1242. https://doi.org/10.1016/j.icheatmasstransfer.2006.06.011

Rajan, K., Srivastava, S., Pitchumani, B., & Surendiran, V. (2010). Thermal conductance of pneumatic conveying preheater for air–gypsum and air–sand heat transfer. *International Journal of Thermal Sciences*, *49*(1), 182–186. https://doi.org/10.1016/j.ijthermalsci.2009.07.001

Ražnjević, K. (1989). *Termodinamičke tablice*. Narodna tehnika Hrvatske.

Sapozhnikov, S. Z., Mityakov, V. Y., Gusakov, A. A., Seroshtanov, V. V., & Subbotina, V. V. (2019). Experimental determination of circular fin effectiveness. *Journal of Physics: Conference Series*, *1382*(1), 012036. https://doi.org/10.1088/1742-6596/1382/1/012036

Satpati, B., Koley, C., & Datta, S. (2019). Nonlinear model of pneumatic conveying dryer for economic process control. *Drying Technology*, *38*(11), 1516–1537. https://doi.org/10.1080/07373937.2019.1648286

Tarawneh, M., Hammad, M., et al. (2012). Dimensional analysis and empirical correlations for heat transfer and pressure drop in condensation and evaporation processes of flow inside micropipes: case study with carbon dioxide (co2). *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, *34*(1), 89–96.

Thapa, P., Tripathi, J., & Jeong, S. H. (2019). Recent trends and future perspective of pharmaceutical wet granulation for better process understanding and product development. *Powder Technology*, *344*, 864–882. https://doi.org/10.1016/j.powtec.2018.12.080