

Dynamic modeling, efficiency analysis, and hierarchical control of industrial airborne pulp dryers

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ABSTRACT: Industrial airborne pulp drying is governed by strongly coupled moisture transport, heat transfer, psychrometric behavior, airflow distribution, pressure dynamics, and fan operation. A nonlinear multi-zone dynamic model was formulated to represent these interactions through coupled mass and energy balances, pressure-driven deck-to-deck airflow, centrifugal-fan dynamics, and through-thickness moisture and temperature transport. Overall and thermal efficiencies, specific energy consumption, and specific moisture evaporation ratio (SMER) were incorporated for energetic assessment. A hierarchical direct synthesis methodology (DSM), comprising moisture-temperature and airflow-fan speed cascade loops with chamber-pressure regulation, was evaluated against proportional-integral-derivative (PID) control and model predictive control (MPC) benchmarks.

The MATLAB simulations of a six-zone dryer predicted progressive moisture reduction toward 0.120 kg H₂O/kg dry pulp, overall efficiency near 0.60, thermal efficiency near 0.61, and SMER of approximately 0.95–0.96 kg H₂O/kWh. The DSM provided the lowest accumulated moisture-tracking error and smooth control action, while PID control gave comparable fast loop regulation. The current MPC implementation exhibited poorer moisture-thermal tracking, indicating the need for further predictive model refinement and systematic tuning. The formulation provides a control-oriented basis for subsequent parameter identification and industrial validation.

Application: The proposed model provides mill engineers and researchers with a control-oriented framework for evaluating moisture regulation, airflow and pressure management, and energy performance in industrial airborne pulp dryers. It can also support controller tuning, operating-point optimization, and future plant data-based model validation and digital twin development.

Drying is among the most energy-intensive operations in industrial processing, accounting for a significant fraction of manufacturing energy consumption [1-2]. This issue is particularly relevant to the pulp and paper industry, where thermal drying requires substantially more energy for water removal than mechanical dewatering [3]. Consequently, improvements in drying efficiency, airflow management, heat recovery, and process control can significantly affect production cost and energy performance.

Pulp drying must remove residual water while maintaining sheet uniformity, fiber quality, brightness, and mechanical integrity. Industrial technologies include steam-heated cylinders, flash and rotary dryers, vacuum systems, through-air drying, impingement drying, and hybrid configurations [4-8]. Airborne pulp dryers differ fundamentally from contact-based systems since the pulp web is supported and dried primarily by high-velocity recirculating hot air [9]. Drying therefore results from strongly coupled heat and mass transfer, moisture evaporation, psychrometric conditions, airflow distribution, chamber pressure, and fan operation.

Mathematical modeling of paper drying has been investigated extensively, particularly for conventional multi-cylinder

sections. Early state-space and dynamic formulations described coupled paper, steam, cylinder-wall, moisture, and ventilation dynamics [10-12]. Subsequent models incorporating mass and energy balances for the web, felt, cylinders, and surrounding air were compared with operating data [10-12]. More recent studies have extended these approaches toward moisture transport, thermodynamic analysis, heat- and mass-transfer prediction, and energy performance assessment [5,13]. These developments demonstrate the maturity of first-principles modeling for conventional paper drying; however, their physical architecture differs substantially from airborne pulp drying, where drying performance is dominated by forced convection, large recirculating airflow rates, aerodynamic pressure losses, and fan-process interactions. This distinction is also supported by published multi-cylinder models, which predominantly formulate the drying problem around steam-heated cylinders and their interaction with the web and surrounding air [14-16].

Dynamic modeling is particularly important from a control perspective since industrial dryers exhibit nonlinear transport phenomena, significant thermal inertia, transport delays, disturbances, and multivariable interactions [17]. Conventional implementations commonly rely on proportional-integral-derivative (PID), cascade, and feedforward

structures [18], whereas more advanced approaches include robust, adaptive, model-predictive, hierarchical, and optimization-based control [19-30]. Such methods can improve moisture regulation and energy utilization, but their effectiveness ultimately depends on the availability of a sufficiently representative dynamic process model [19].

Despite the extensive literature on cylinder, rotary, flash, tissue, and other drying technologies [5,11,31], a comprehensive control-oriented dynamic formulation specifically addressing industrial airborne pulp dryers has not been identified in the reviewed literature. In particular, the simultaneous representation of spatial moisture and temperature evolution, psychrometric behavior, convective heat and mass transfer, deck-to-deck airflow distribution, pressure losses, fan electromechanical dynamics, energy efficiency, and closed-loop control remains insufficiently addressed. This gap is important since these phenomena are intrinsically coupled in commercial airborne drying systems.

Accordingly, this work develops a first-principles, multi-compartment nonlinear dynamic model of an industrial airborne pulp dryer. The dryer is discretized into interconnected drying regions to retain the principal deck-to-deck variations in pulp moisture, air and pulp temperatures, humidity, airflow, and pressure while maintaining a computational structure suitable for control design. The model integrates mass and energy conservation, psychrometric relations, external and internal moisture-transfer limitations, aerodynamic losses, and fan dynamics. An efficiency framework and hierarchical control architecture based on the direct synthesis methodology are subsequently formulated, with conventional PID and predictive control used as comparative benchmarks.

The study is intentionally presented as a proof of concept rather than a validated industrial digital twin. No experimental validation is claimed; model parameters and assumptions therefore represent physically plausible operating conditions and require identification against plant measurements before quantitative industrial deployment. The objective is instead to establish the modeling structure, investigate the dominant coupled dynamics, and assess its suitability for control and energy performance studies.

The remainder of this paper covers the following topics: airborne drying process; development of the nonlinear multi-compartment model and its assumptions and limitations; efficiency formulation; development and benchmarking of the control architecture; analysis of simulation results; and key conclusions.

PROCESS DESCRIPTION

The industrial airborne pulp dryer considered in this work is a multi-deck convective drying system in which a continuous pulp web is transported through successive drying regions while interacting with high-velocity heated air [2,32,33]. Unlike conventional cylinder drying, the domi-

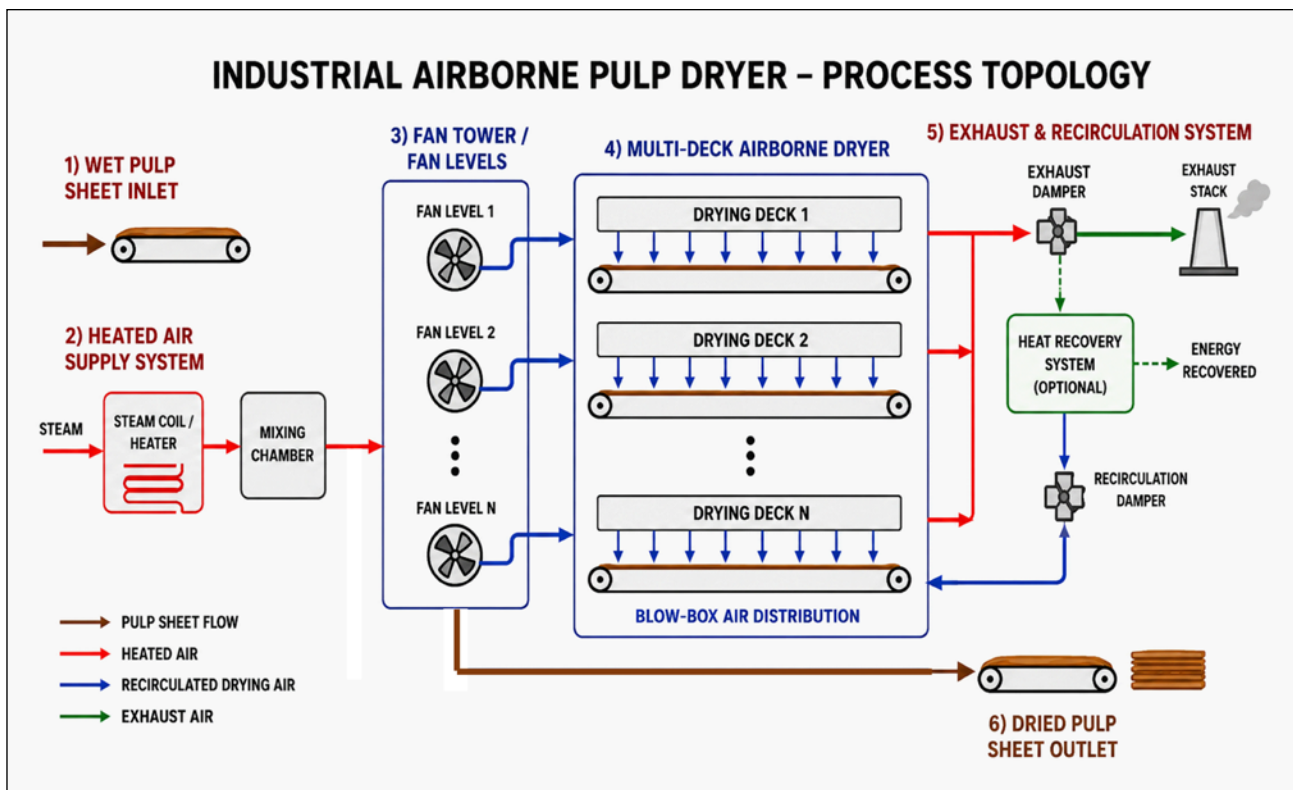
nant heat transfer mechanism is forced convection between the circulating air and the exposed pulp surfaces. Industrial airborne dryers employ modular drying sections, blow boxes, air circulation systems, fan towers, and associated exhaust and heating equipment to provide the thermal and aerodynamic conditions required for moisture removal. Commercial configurations, such as the one from Valmet [34], can incorporate several tens of drying decks, depending on machine capacity and process requirements..

Figure 1 shows the generalized process configuration considered in this study. The wet pulp web enters the dryer at an inlet moisture content $X_{in}(t)$ [kg_{H2O}/kg_{dry pulp}] and pulp temperature $T_{p,in}(t)$ [K], with $m_p(t)$ [kg_{dry pulp}/s] denoting the dry pulp throughput. The web is subsequently transported through the different drying decks by the internal web-handling system. As it progresses through the dryer, moisture is progressively removed until the outlet pulp moisture $X_p(t)$ [kg_{H2O}/kg_{dry pulp}] required for the subsequent cutting, baling, and handling stages is reached [6,32,33].

Within each drying region, heated air is directed toward the pulp surfaces through blow boxes and air-distribution passages. The drying air temperature is denoted by $T_a(t)$ [K], whereas $T_p(t)$ [K] represents the pulp temperature. Heat transferred from the air increases the pulp temperature and supplies the energy required for water evaporation. Simultaneously, moisture migrates through the fibrous porous structure and is transferred from the pulp surface to the surrounding gas phase. Convective paper drying is therefore governed by simultaneous heat and mass transfer, with air temperature, air velocity, humidity, and pressure conditions directly influencing the drying rate [35-38]. This physical behavior is consistent with experimental and numerical studies of convective and through-air paper drying [39,40].

The gas-phase moisture condition in Fig. 1 is characterized by the humidity ratio $Y_a(t)$ [kg_{H2O}/kg_{dry air}], while $Y_s(t)$ [kg_{H2O}/kg_{dry air}] represents the corresponding saturation condition at the pulp surface. Their difference determines the local psychrometric potential available for moisture transfer. The resulting water removal rate is represented by $m_{ev}(t)$ [kg/s]. As drying proceeds, the circulating air becomes progressively more humid while the moisture content of the pulp decreases, producing spatial variations in temperature, humidity, evaporation rate, and pulp condition throughout the dryer [35-38].

Air circulation is generated by the fan system shown in Fig. 1. The electric drive supplies the motor torque $\tau_m(t)$ [N·m], producing the fan angular velocity $\omega_f(t)$ [rad/s]. The fans establish the drying air flow rate $w_a(t)$ [kg_{dry air}/s] and the pressure rise required to overcome the aerodynamic resistance of the internal ducts, plenums, blow boxes, dampers, and other flow passages. The corresponding fan pressure rise and dryer pressure loss are represented by $\Delta P_f(t)$ [Pa] and $\Delta P_d(t)$ [Pa], respectively. Valmet specifically identifies fan towers, blow boxes, and airflow op-



1. Generalized process diagram of an airborne pulp dryer, including the multi-level drying decks, vertical fan columns, airflow recirculation system, heat recovery section, and the principal thermofluid and mechanical interactions involved during pulp drying.

timization as central elements of its airborne dryer architecture, and reports that increased fan-tower airflow can increase evaporation capacity [34].

The thermal subsystem provides the heating duty $q_{in}(t)$ [W] required to maintain the drying air temperature. Part of the circulating air is reheated and returned to the drying sections, whereas moisture-laden air is discharged through the exhaust system. The exhaust damper opening $A_{ex}(t)$ [m²] regulates the extraction path and contributes to maintaining the required chamber pressure $P_s(t)$ [Pa]. The combined action of heating, air recirculation, exhaust extraction, and fan operation therefore establishes the thermofluid environment through which the pulp web travels.

Since the pulp progressively loses moisture along its trajectory, the operating conditions are not spatially uniform. The upstream wetter regions generally sustain a greater evaporation load, whereas the downstream regions process progressively drier pulp. Consequently, $X_p(t)$, $T_p(t)$, $T_a(t)$, $Y_a(t)$, $w_a(t)$, and the associated pressure conditions can vary between drying regions. This physical characteristic motivates the multi-compartment representation introduced in the following section rather than treating the complete dryer as a single perfectly mixed volume. Spatial heat and mass transport is also established in detailed paper-drying models, where moisture, temperature, and phase-change phenomena evolve throughout the drying process.

SYSTEM MODELING

The airborne pulp dryer is represented as a lumped-parameter thermofluid system composed of an equivalent humid air control volume, a moving wet pulp phase, an equivalent airflow network, and a centrifugal-fan subsystem. The physical multi-deck dryer is therefore reduced to spatially averaged mass, thermal, humidity, pressure, and mechanical variables. Lumped and reduced-order formulations based on coupled mass and energy balances have previously been employed for paper machine dryer sections, multi-cylinder dryers, and industrial through-dryers [5,11,12,41-43].

The independent state vector is defined as $\mathbf{x}(t) = [M_v(t), M_p(t), M_w(t), T_a(t), T_p(t), \omega_p(t), P_s(t)]^T$, where symbolically $\mathbf{x}(t) \in \{\mathbb{R}^7\}$. From here, $M_v(t)$ [kg] is the water vapor inventory inside the dryer, $M_p(t)$ [kg] is the dry pulp inventory, and $M_w(t)$ [kg] is the liquid water inventory retained in the pulp.

The dry air inventory $M_a(t)$ [kg] is reconstructed algebraically from the ideal gas relation and is therefore not included as an independent state.

The mathematical formulation is developed under the following assumptions:

1. The humid air phase is represented by a spatially averaged and well-mixed control volume.
2. Dry air and water vapor are treated as ideal gases over the considered operating range.

3. The effective dryer gas volume V_d [m³] remains constant.
4. Thermophysical properties of dry air, water vapor, liquid water, and dry pulp are evaluated at representative operating conditions. When required, temperature-dependent correlations may replace their nominal values.
5. Kinetic- and potential-energy contributions are neglected in the thermal balances.
6. Moisture removal occurs through evaporation; condensation within the modeled drying volume is neglected.
7. Subfiber moisture-transport mechanisms, including capillary migration and bound water diffusion, are not modeled separately; their combined influence is incorporated through the effective diffusivity D_{eff} and mass-transfer coefficient k_m .
8. The wet pulp is represented through spatially averaged dry-fiber mass, liquid water inventory, moisture content, and temperature.
9. Individual deck airflow rates and pressure fields are not independently calculated in the lumped formulation. Their aggregate hydraulic effect is represented through equivalent distributed and local loss coefficients.
10. Multiple industrial fans and airflow branches are reduced to an equivalent fan airflow subsystem.
11. Heat losses through walls, structural elements, openings, and imperfect insulation are represented through an equivalent overall heat transfer coefficient.
12. Each gas-phase drying zone is treated as spatially well mixed; continuous cross machine and local blow box jet distributions are not resolved. Machine-direction variations are represented by six interconnected zones, while through-thickness pulp moisture and temperature variations are represented by a one-dimensional discretization.

Accordingly, the model corresponds to an $N = 6$ lumped representation of the complete drying system. This approximation is consistent with reduced-order paper-drying models in which averaged variables and effective transport coefficients are employed to represent macroscopic dryer behavior [11,12,41,42].

Dry-air mass balance

Conservation of dry air mass inside the effective dryer volume is expressed as

$$\frac{dM_a(t)}{dt} = m_{a,\text{in}}(t) - m_{a,\text{out}}(t), \quad (1)$$

where $m_{a,\text{in}}(t)$ [kg/s] and $m_{a,\text{out}}(t)$ [kg/s] are the inlet and outlet dry air mass flow rates, respectively. Assuming ideal gas behavior [7], the dry air inventory is

$$M_a(t) = \frac{V_d}{R_a} \cdot \frac{P_s(t)}{T_a(t)} \quad (2)$$

where $R_a = 287.05$ [J/kg·K] is the specific gas constant of dry air. The corresponding dry air density is

$$\rho_a(t) = \frac{1}{R_a} \cdot \frac{P_s(t)}{T_a(t)}, \quad (3)$$

in kg/m³. The exhaust mass flow is represented through the equivalent pressure-driven relationship

$$m_{a,\text{out}}(t) = C_d \cdot A_e(t) \cdot \sqrt{2 \rho_a(t) \cdot [P_s(t) - P_{\text{atm}}(t)]_+}, \quad (4)$$

where C_d [-] is the effective discharge coefficient, $A_e(t)$ [m²] is the effective exhaust area, and $P_{\text{atm}}(t)$ [Pa] is atmospheric pressure. The operator $[z]_+ = \max(z, 0)$ prevents a non-physical negative argument in the square-root term.

When exhaust flow is manipulated through a damper, its effective area is represented as

$$A_e(t) = A_{e,\text{max}} \cdot \alpha_e(t), \quad (5)$$

where $A_{e,\text{max}}$ [m²] and $\alpha_e(t)$ [-] are the maximum effective exhaust area and the normalized exhaust-damper position, constrained according to $0 \leq \alpha_e(t) \leq 1$, respectively. Equation 4 represents the outlet flow for relatively small pressure differences and subsonic flow conditions [7].

Water vapor mass balance and evaporation dynamics

The water vapor inventory satisfies

$$\frac{dM_v(t)}{dt} = m_{v,\text{in}}(t) + m_{\text{ev}}(t) - m_{v,\text{out}}(t). \quad (6)$$

From here, $m_{v,\text{in}}(t)$ [kg_{H2O}/s], $m_{\text{ev}}(t)$ [kg_{H2O}/s], and $m_{v,\text{out}}(t)$ [kg_{H2O}/s] are the incoming water vapor flow rate, the evaporation rate from the wet pulp, and the outgoing water vapor flow rate, respectively. The evaporation rate is represented through an effective mass-transfer formulation [5,41]:

$$m_{\text{ev}}(t) = A_p \cdot k_m(t) \cdot [Y_s(T_p(t), P_s(t)) - Y_a(t)]_+, \quad (7)$$

where $k_m(t)$ [kg_{dry air}/m²·s] is the effective overall mass-transfer conductance per unit area when the moisture-transfer driving force is expressed as a humidity ratio; A_p [m²] is the effective air-pulp interfacial area; and $Y_s(T_p(t), P_s(t))$ [kg_{H2O}/kg_{dry air}] is the saturation humidity ratio corresponding to the pulp surface. The bulk-air humidity ratio is calculated from

$$Y_a(t) = \frac{M_v(t)}{M_a(t)}. \quad (8)$$

The saturation humidity ratio is calculated as [7]

$$Y_s(T_p(t), P_s(t)) = 0.622 \cdot \frac{P_{\text{sat}}(T_p(t))}{P_s(t) - P_{\text{sat}}(T_p(t))}, \quad (9)$$

where $P_{\text{sat}}(T_p(t))$ [Pa] is the saturation vapor pressure of

water evaluated at the pulp temperature $T_p(t)$. The incoming water vapor flow rate is

$$m_{v,in}(t) = Y_{a,in}(t) \cdot m_{a,in}(t), \quad (10)$$

where $Y_{a,in}(t)$ [kg_{H2O}/kg_{dry air}] is the humidity ratio of the incoming drying air. The outgoing vapor flow rate is

$$m_{v,out}(t) = Y_a(t) \cdot m_{a,out}(t). \quad (11)$$

The effective coefficient $k_m(t)$ incorporates the aggregate contribution of external gas-side convective mass transfer and unresolved internal moisture migration through the pulp. Hence, Eq. 6 describes an overall moisture-transfer rate rather than a local fiber-scale diffusion field [5,41]. In its general form, $k_m(t) = f(v_a(t), T_a(t), Y_a(t), X_p(t))$, where $v_a(t)$ [m/s] is the characteristic air velocity and $X_p(t)$ [kg_{H2O}/kg_{dry pulp}] is the dry-basis pulp moisture content.

Dry pulp and liquid water mass balances

The dry pulp inventory satisfies

$$\frac{dM_p(t)}{dt} = m_{p,in}(t) - m_{p,out}(t). \quad (12)$$

From here, $m_{p,in}(t)$ and $m_{p,out}(t)$ are the incoming dry pulp mass flow rate and the outgoing dry pulp mass flow rate, respectively. The pulp moisture content on a dry basis is

$$X_p(t) = \frac{M_w(t)}{M_p(t)}. \quad (13)$$

where $M_w(t)$ [kg_{H2O}] is the liquid water inventory. Conservation of liquid water in the pulp gives

$$\frac{dM_w(t)}{dt} = m_{w,in}(t) - m_{ev}(t) - m_{w,out}(t). \quad (14)$$

From here, $m_{w,in}(t)$ [kg_{H2O}/s] and $m_{w,out}(t)$ [kg_{H2O}/s] are the water flow rate entering with the pulp and the liquid water flow rate leaving with the pulp, respectively. The inlet liquid water flow rate is

$$m_{w,in}(t) = X_{p,in}(t) \cdot m_{p,in}(t), \quad (15)$$

where $X_{p,in}(t)$ [kg_{H2O}/kg_{dry pulp}] is the inlet pulp moisture content. Under the lumped well-mixed approximation,

$$m_{w,out}(t) = X_p(t) \cdot m_{p,out}(t). \quad (16)$$

Differentiating Eq. 13 and combining Eqs. 12–16 yields

$$\frac{dX_p(t)}{dt} = \frac{m_{p,in}(t) \cdot (X_{p,in}(t) - X_p(t)) - m_{ev}(t)}{M_p(t)}. \quad (17)$$

Equation 17 accounts for moisture variation produced by both material replacement through the control volume and evaporation from the pulp [11,12,42].

Drying air energy balance

The effective sensible thermal capacitance of the humid air inventory is

$$C_a(t) = c_{p,a} \cdot M_a(t) + c_{p,v} \cdot M_v(t), \quad (18)$$

where $c_{p,a}$ [J/kg·K] is the specific heat capacity of dry air at constant pressure, and $c_{p,v}$ [J/kg·K] is the specific heat capacity of water vapor at constant pressure. The drying air energy balance is expressed as

$$C_a(t) \cdot \frac{dT_a(t)}{dt} = q_{in}(t) - q_{conv}(t) - q_{loss}(t) + c_{p,a} \cdot m_{a,in}(t) \cdot (T_{a,in}(t) - T_a(t)) + c_{p,v} \cdot m_{v,in}(t) \cdot (T_{v,in}(t) - T_a(t)) + c_{p,v} \cdot m_{ev}(t) \cdot (T_p(t) - T_a(t)). \quad (19)$$

From here, $T_a(t)$ is the dry air temperature; $q_{in}(t)$ [W] is the thermal power supplied to the drying air system; $q_{conv}(t)$ [W] is the convective heat transfer rate from the drying air to the pulp; $q_{loss}(t)$ [W] is the heat loss rate from the dryer to the surroundings; $T_{a,in}(t)$ [K] is the inlet dry air temperature; and $T_{v,in}(t)$ [K] is the temperature associated with the incoming water vapor. The thermal power required for evaporation is

$$q_{ev}(t) = \lambda_v(T_p(t)) \cdot m_{ev}(t), \quad (20)$$

where $\lambda_v(T_p(t))$ [J/kg] is the latent heat of vaporization of water evaluated at the pulp temperature. Heat loss through the dryer enclosure is modeled as

$$q_{loss}(t) = U_d \cdot A_w \cdot (T_a(t) - T_{amb}), \quad (21)$$

and U_d [W/m²·K] is the equivalent overall dryer-to-ambient heat transfer coefficient; A_w [m²] is the effective external heat transfer area; and $T_{amb}(t)$ is the ambient temperature. The latent energy term $q_{ev}(t)$ is included in the pulp energy balance since thermal energy is transferred from the air to the wet pulp through $q_{conv}(t)$ before being consumed by evaporation.

Pulp energy balance

The effective thermal capacitance of the wet pulp is

$$C_p(t) = c_{p,p} \cdot M_p(t) + c_{p,w} \cdot M_w(t), \quad (22)$$

where $c_{p,p}$ [J/kg·K] and $c_{p,w}$ [J/kg·K] are the specific heat capacity of dry pulp and the specific heat capacity of liquid water, respectively. The pulp-phase energy balance is

$$C_p(t) \cdot \frac{dT_p(t)}{dt} = q_{conv}(t) - q_{ev}(t) + m_{p,in}(t) \cdot (c_{p,p} + c_{p,w} \cdot X_{p,in}(t)) \cdot (T_{p,in}(t) - T_p(t)). \quad (23)$$

From here, $T_{p, in}(t)$ [K] is the inlet pulp temperature. The air-to-pulp convective heat transfer rate is

$$q_{conv}(t) = A_p \cdot h_c(t) \cdot (T_a(t) - T_p(t)), \quad (24)$$

where $h_c(t)$ [W/m²·K] is the effective convective heat transfer coefficient. More generally, $bc(t) = f(v_a(t), T_a(t), Y_a(t))$, where the notation $f(\cdot)$ indicates that the effective heat transfer coefficient may depend on air velocity, drying air temperature, and humidity ratio. Effective heat- and mass-transfer coefficients are commonly employed in reduced-order descriptions of paper and through-air drying [11,41,42].

Pressure dynamics and hydraulic losses

The pressure dynamics are obtained by differentiating Eq. 2:

$$\frac{dM_a(t)}{dt} = \frac{V_d}{R_a} \cdot \frac{1}{T_a(t)} \cdot \frac{dP_s(t)}{dt} - \frac{V_d}{R_a} \cdot \frac{P_s(t)}{T_a^2(t)} \cdot \frac{dT_a(t)}{dt}. \quad (25)$$

The first term on the right-hand side of Eq. 25 accounts for pressure variation caused by dry air mass accumulation, while the second term accounts for pressure variation caused by gas heating or cooling. Replacing Eq. 1 in Eq. 25, then solving for $\frac{dP_s(t)}{dt}$, and if

$$\left| \frac{P_s(t)}{T_a(t)} \cdot \frac{dT_a(t)}{dt} \right| \ll \left| \frac{R_a}{V_d} \cdot T_a(t) \cdot (m_{a,in}(t) - m_{a,out}(t)) \right|, \quad (26)$$

the pressure equation can be approximated as

$$\frac{dP_s(t)}{dt} \approx \frac{R_a}{V_d} \cdot T_a(t) \cdot (m_{a,in}(t) - m_{a,out}(t)), \quad (27)$$

which is a quasi-isothermal pressure approximation. The equivalent aerodynamic pressure loss through the airflow network is described by

$$\Delta P_d(t) = \frac{1}{2} \cdot \varrho_a(t) \cdot v_d^2(t) \cdot \left(f_d \cdot \frac{L_d}{D_d} + \sum_{j=1}^{N_K} K_j \right), \quad (28)$$

where $\Delta P_d(t)$ [Pa], $v_d(t)$ [m/s], f_d [-], L_d [m], D_d [m], K_j [-], j [-], and N_K [-] are the total equivalent airflow pressure loss, the representative duct air velocity, the Darcy friction factor, the equivalent duct length, the equivalent hydraulic diameter, the dimensionless local pressure loss coefficient associated with the j -th hydraulic element, the local loss element index, and the total number of equivalent local hydraulic resistances, respectively. Also, Eq. 28 incorporates the summation over all localized hydraulic resistances. For an equivalent airflow cross-sectional area, A_d [m²], the representative velocity is

$$v_d(t) = \frac{1}{A_d} \cdot q_a(t), \quad (29)$$

where $q_a(t)$ [m³/s] is the volumetric airflow rate. Since

$$q_a(t) = \frac{m_{a,in}(t)}{\varrho_a(t)}, \quad (30)$$

the representative velocity can alternatively be expressed as

$$v_d(t) = \frac{1}{A_d} \cdot \frac{m_{a,in}(t)}{\varrho_a(t)}. \quad (31)$$

For approximately constant air density and geometry, $\Delta P_d(t) \propto q_a^2(t)$.

Centrifugal fan dynamics

The rotational mechanical balance of the equivalent fan is

$$J_f \cdot \frac{d\omega_f(t)}{dt} = \tau_m(t) - \tau_a(t) - B_f \omega_f(t) \Rightarrow \frac{d\omega_f(t)}{dt} = \frac{1}{J_f} \cdot (\tau_m(t) - \tau_a(t) - B_f \omega_f(t)). \quad (32)$$

In Eq. 32, J_f [kg·m²] is the combined equivalent moment of inertia of the fan, shaft, and reflected motor inertia; $\tau_m(t)$ [N·m] is the electromagnetic or shaft torque developed by the fan motor; $\tau_a(t)$ [N·m] is the aerodynamic resisting torque, and B_f [N·m·s/rad] is the equivalent viscous friction coefficient.

Within the selected operating region, the aerodynamic torque is approximated by

$$\tau_a(t) = K_a \cdot \omega_f^2(t), \quad (33)$$

where K_a [N·m·s²/rad²] is the equivalent aerodynamic-load coefficient. The fan-generated dry air mass flow rate is represented by

$$m_{a,in}(t) = K_f \omega_f(t) \cdot \sqrt{\frac{P_f(t)}{\varrho_a(t)}}, \quad (34)$$

where $P_f(t)$ [Pa] is the pressure rise generated by the equivalent fan, and K_f [kg·s/m] is the fitted fan flow coefficient corresponding specifically to the functional form adopted in Eq. 34. The equivalent fan pressure flow characteristic is

$$P_f(t) = P_{f0} - K_p \cdot m_{a,in}^2(t), \quad (35)$$

where P_{f0} [Pa] is the extrapolated fan pressure at zero flow, and K_p [Pa·s²/kg²] is the equivalent pressure-flow coefficient. For variable-speed centrifugal-fan operation, the fan affinity relations are $q_a(t) \propto \omega_f(t)$, $P_f(t) \propto \omega_f^2(t)$, and $P_{fan}(t) \propto \omega_f^3(t)$, where $P_{fan}(t)$ [W] is the mechanical power transferred by the fan to the airflow. The instantaneous fan-shaft mechanical power is

$$P_{shaft}(t) = \tau_m(t) \cdot \omega_f(t). \quad (36)$$

The coefficients K_a , K_p , K_p , and P_{i0} describe the equivalent fan characteristic within the selected operating region and can be identified from manufacturer fan curves or measured operating data [35].

Spatial moisture and temperature representation

The variables $T_a(t)$, $T_p(t)$, $Y_a(t)$, $X_p(t)$, and $P_s(t)$ represent spatially averaged quantities within an individual drying zone control volume. Consequently, continuous local variations in the machine direction and across the pulp thickness are not represented by these lumped variables. Machine-direction variations are subsequently represented by the multizone formulation developed in the “Multizone representation of deck-to-deck airflow” section, whereas the internal moisture and temperature fields across the pulp thickness are represented using the spatial coordinate ξ [m]. Spatially resolved paper drying models similarly describe coupled moisture, temperature, and phase-change phenomena within the porous sheet [39,44–46].

$$\frac{\partial X_p(\xi, t)}{\partial t} = \frac{\partial}{\partial \xi} \left(D_{\text{eff}}(X_p(\xi, t), T_p(\xi, t)) \cdot \frac{\partial X_p(\xi, t)}{\partial \xi} \right), \quad (37)$$

where D_{eff} [m²/s] is the effective moisture diffusivity. At the pulp–air interface,

$$\Delta P_d(t) = \frac{1}{2} \cdot \varrho_a(t) \cdot v_d^2(t) \cdot \left(f_d \cdot \frac{L_d}{D_d} + \sum_{j=1}^{N_k} K_j \right). \quad (38)$$

The corresponding through-thickness pulp temperature field is represented by

$$-\varrho_{p,d} \cdot \left(c_{p,p} + c_{p,w} \cdot X_p(\xi, t) \right) \cdot \frac{\partial X_p(\xi, t)}{\partial t} = \frac{\partial}{\partial \xi} \left(k_{p,\text{eff}} \cdot \frac{\partial T_p(\xi, t)}{\partial \xi} \right), \quad (39)$$

where $k_{p,\text{eff}}$ [W/(m·K)] is the effective thermal conductivity of the wet pulp. At the pulp–air interface, the heat flux boundary condition is

$$k_{p,\text{eff}} \cdot \frac{\partial T_p}{\partial \xi} \bigg|_{\xi=\xi_s} = h_c(t) \cdot (T_{p,s}(t) - T_a(t)) + \lambda_v(T_{p,s}) \cdot m''_{\text{ev}}(t), \quad (40)$$

with

$$m''_{\text{ev}}(t) = k_m(t) \cdot (Y_s(t) - Y_a(t)). \quad (41)$$

Equations 37–41 couple internal moisture diffusion, heat conduction, external convection, and surface evaporation. In the present numerical model, these equations are discretized using N_ξ through-thickness nodes and integrated with the multizone thermofluid balances.

Multizone representation of deck-to-deck airflow

Let N [–] denote the total number of interconnected drying zones. For the j -th zone, $j = 1, \dots, N$, the local state vector is defined as $\mathbf{x}_j(t) = [M_{v,j}(t), M_{p,j}(t), M_{w,j}(t), T_{a,j}(t), T_{p,j}(t), P_{s,j}(t)]^T$, for $\mathbf{x}_j(t) \in \{\mathbb{R}^6\}$. The variables $M_{v,j}(t)$ [kg_{H₂O}], $M_{p,j}(t)$ [kg_{dry pulp}], $M_{w,j}(t)$ [kg_{H₂O}], $T_{a,j}(t)$ [K], $T_{p,j}(t)$ [K], and $P_{s,j}(t)$ [Pa] represent, respectively, the water vapor inventory in the gas phase, dry-pulp inventory, liquid water inventory retained in the pulp, mean drying air temperature, mean pulp temperature, and static pressure associated with zone j . Section-wise and modular formulations in which local paper, evaporation, ventilation, and energy variables are evaluated for individual dryer units have been employed for multi-cylinder paper-dryer systems [43,47].

For two adjacent zones j and $j + 1$, define the interzone static pressure difference as

$$\Delta P_{j,j+1}(t) = P_{s,j}(t) - P_{s,j+1}(t). \quad (42)$$

The dry air mass flow rate through the connection between these zones is represented through the pressure-driven airflow relation [7,48,49]

$$m_{a,j \rightarrow j+1}(t) = C_{d,j} A_j \sqrt{2 \cdot \varrho_{a,\text{up},j}(t) |\Delta P_{j,j+1}(t)| \cdot \text{sgn}[\Delta P_{j,j+1}(t)]}, \quad (43)$$

and $C_{d,j}$ [–] is the equivalent discharge coefficient of the interzone connection, A_j [m²] is its effective airflow area, and $\varrho_{a,\text{up},j}(t)$ [kg/m³] is the dry air density evaluated on the upstream side of the connection. Pressure-driven flow between interconnected zones is a standard formulation in multizone airflow network models, in which individual zones possess independent pressures, and airflow is obtained from pressure differences across connecting paths [48,49]. The upstream density is selected according to the instantaneous direction of the pressure gradient:

$$\varrho_{a,\text{up},j}(t) = \begin{cases} \varrho_{a,j}(t), & \Delta P_{j,j+1}(t) \geq 0 \\ \varrho_{a,j+1}(t), & \Delta P_{j,j+1}(t) < 0, \end{cases} \quad (44)$$

where $\varrho_{a,j}(t)$ is

$$\varrho_{a,j}(t) = \frac{1}{R_a} \cdot \frac{P_{s,j}(t)}{T_{a,j}(t)}, \quad (45)$$

which is the mean dry air density in zone j . Therefore, $m_{a,j \rightarrow j+1}(t) > 0$ indicates airflow from zone j toward zone $j + 1$, whereas $m_{a,j \rightarrow j+1}(t) < 0$ indicates reverse airflow.

The dry air inventory within zone j is related to the local static pressure and temperature through

$$M_{a,j}(t) = \frac{V_j}{R_a} \cdot \frac{P_{s,j}(t)}{T_{a,j}(t)}, \quad (46)$$

where V_j [m³] is the effective gas volume of the j -th drying zone. The local humidity ratio is

$$Y_{a,j}(t) = \frac{M_{v,j}(t)}{M_{a,j}(t)}. \quad (47)$$

Since the interzone dry air stream transports water vapor, the corresponding vapor mass flow rate is

$$m_{v,j \rightarrow j+1}(t) = Y_{a,up,j}(t) \cdot m_{a,j \rightarrow j+1}(t), \quad (48)$$

where $Y_{a,up,j}(t)$ [kg_{H₂O}/kg_{dry air}] is the humidity ratio of the upstream zone, defined as

$$Y_{a,up,j}(t) = \begin{cases} Y_{a,j}(t), & \Delta P_{j,j+1}(t) \geq 0 \\ Y_{a,j+1}(t), & \Delta P_{j,j+1}(t) < 0. \end{cases} \quad (49)$$

The use of local airflow and absolute humidity variables in zone-wise paper dryer ventilation balances is consistent with the theoretical multi-cylinder dryer formulation reported in [50]. For each zone, the saturation humidity ratio at the pulp–air interface is calculated as

$$Y_{s,j}(t) = 0.622 \cdot \frac{P_{sat}(T_{p,j}(t))}{P_{s,j}(t) - P_{sat}(T_{p,j}(t))}. \quad (50)$$

From here, $P_{sat}(t)$ is the water saturation vapor pressure evaluated at the local pulp temperature, $T_{p,j}(t)$. The corresponding local evaporation rate is

$$m_{ev,j}(t) = A_{p,j} \cdot k_{m,j}(t) \cdot [Y_{s,j}(t) - Y_{a,j}(t)] +, \quad (51)$$

where $k_{m,j}(t)$ and $A_{p,j}$ are the effective local mass-transfer coefficient and the effective pulp–air interfacial area in zone j , respectively. The local water vapor balance can then be expressed as

$$\frac{dM_{v,j}(t)}{dt} = \sum_{k \in N_j} m_{v,k \rightarrow j}(t) + m_{v,j,ext,in}(t) - m_{v,j,ext,out}(t) + m_{ev,j}(t), \quad (52)$$

and N_j is the set of zones directly connected to zone j ; $m_{v,k \rightarrow j}(t)$ [kg_{H₂O}/s] is the signed interzone vapor flow rate from neighboring zone k to zone j ; $m_{v,j,ext,in}(t)$ [kg_{H₂O}/s] is the external vapor flow rate supplied to zone j ; and $m_{v,j,ext,out}(t)$ [kg_{H₂O}/s] is the external vapor flow rate exhausted from that zone. Similarly, the local dry air balance is

$$\frac{dM_{a,j}(t)}{dt} = \sum_{k \in N_j} m_{a,k \rightarrow j}(t) + m_{a,j,ext,in}(t) - m_{a,j,ext,out}(t), \quad (53)$$

where $m_{a,k \rightarrow j}(t)$ [kg_{dry air}/s] is the signed dry air flow rate between neighboring zone k and zone j ; $m_{a,j,ext,in}(t)$ [kg_{dry air}/s]

is the externally supplied dry air flow rate; and $m_{a,j,ext,out}(t)$ [kg_{dry air}/s] is the dry air flow rate exhausted both from zone j . For a sequentially transported pulp sheet, the dry pulp mass balance in zone j is

$$\frac{dM_{p,j}(t)}{dt} = m_{p,j-1 \rightarrow j}(t) - m_{p,j \rightarrow j+1}(t), \quad (54)$$

where $m_{p,j-1 \rightarrow j}(t)$ [kg_{dry pulp}/s] is the incoming dry pulp mass flow rate and $m_{p,j \rightarrow j+1}(t)$ [kg_{dry pulp}/s] is the dry pulp flow rate leaving zone j . Sequential dryer module formulations, in which the outlet state from one paper-drying module supplies the subsequent module, have been employed for multi-cylinder paper machine dryer sections [47]. The corresponding liquid water balance is

$$\frac{dM_{w,j}(t)}{dt} = m_{w,j-1 \rightarrow j}(t) - m_{w,j \rightarrow j+1}(t) - m_{ev,j}(t), \quad (55)$$

where $m_{w,j-1 \rightarrow j}(t)$ [kg_{H₂O}/s] and $m_{w,j \rightarrow j+1}(t)$ [kg_{H₂O}/s] are the liquid water flow rates transported with the pulp into and out of zone j , respectively. The local dry-basis moisture content is

$$X_{p,j}(t) = \frac{M_{w,j}(t)}{M_{p,j}(t)}. \quad (56)$$

The liquid water flow transported with the outgoing pulp is consequently

$$m_{w,j \rightarrow j+1}(t) = X_{p,j}(t) \cdot m_{p,j \rightarrow j+1}(t). \quad (57)$$

The drying air also transports thermal energy between adjacent zones. The specific sensible enthalpy of dry air is represented as

$$h_a(T(t)) = c_{p,a} \cdot (T(t) - T^*), \quad (58)$$

where $c_{p,a}$ [J/(kg·K)] is the dry air specific heat capacity at constant pressure and T^* [K] is the selected enthalpy reference temperature. The temperature associated with the upstream gas stream is

$$T_{a,up,j}(t) = \begin{cases} T_{a,j}(t), & \Delta P_{j,j+1}(t) \geq 0 \\ T_{a,j+1}(t), & \Delta P_{j,j+1}(t) < 0. \end{cases} \quad (59)$$

The corresponding dry air sensible enthalpy flow rate is

$$H_{a,j \rightarrow j+1}(t) = m_{a,j \rightarrow j+1}(t) \cdot h_a(T_{a,up,j}(t)). \quad (60)$$

The specific enthalpy of the water vapor component is represented as

$$h_v(T(t)) = \lambda_v^* + c_{p,v} \cdot (T(t) - T^*(t)). \quad (61)$$

In this case, λ_v^* [J/kg_{H2O}] is the latent heat of vaporization referred to $T^*(t)$. The water vapor enthalpy transfer between adjacent zones is therefore

$$H_{v,j \rightarrow j+1}(t) = m_{v,j \rightarrow j+1}(t) \cdot h_v(T_{a,up,j}(t)). \quad (62)$$

The total humid air enthalpy flow through an interzone connection is [50]

$$H_{g,j \rightarrow j+1}(t) = H_{a,j \rightarrow j+1}(t) + H_{v,j \rightarrow j+1}(t). \quad (63)$$

For zone j , the effective humid air thermal capacitance is

$$Ca_j(t) = c_{p,a} \cdot M_{a,j}(t) + c_{p,v} \cdot M_{v,j}(t). \quad (64)$$

The local air-to-pulp convective heat transfer rate is

$$q_{conv,j}(t) = A_{p,j} \cdot h_{c,j}(t) \cdot (T_{a,j}(t) - T_{p,j}(t)), \quad (65)$$

where $h_{c,j}(t)$ [W/(m²·K)] is the local effective convective heat transfer coefficient. The thermal loss from zone j to the surroundings is

$$q_{loss,j}(t) = U_{d,j} \cdot A_{w,j} \cdot (T_{a,j}(t) - T_{amb}), \quad (66)$$

where $U_{d,j}$ [W/(m²·K)] is the effective overall heat transfer coefficient, and $A_{w,j}$ [m²] is the effective external heat transfer area associated with zone j . The gas-phase energy balance for zone j is then written as

$$C_{a,j}(t) \cdot \frac{dT_{a,j}(t)}{dt} = q_{in,j}(t) - q_{conv,j}(t) - q_{loss,j}(t) + \sum_{k \in N_j} H_{g,k \rightarrow j}(t) + H_{g,j,ext,in}(t) - H_{g,j,ext,out}(t), \quad (67)$$

where N_f [-] is the number of operating fans; $q_{in,j}(t)$ [W] is the thermal power supplied directly to zone j ; $H_{g,k \rightarrow j}(t)$ [W] is the signed humid air enthalpy flow rate from neighboring zone k ; $H_{g,j,ext,in}(t)$ [W] is the external gas-phase enthalpy inflow, and $H_{g,j,ext,out}(t)$ [W] is the external gas-phase enthalpy outflow. The local wet pulp thermal capacitance is

$$C_{p,j}(t) = c_{p,p} \cdot M_{p,j}(t) + c_{p,w} \cdot M_{w,j}(t). \quad (68)$$

The local latent heat consumption is

$$q_{ev,j}(t) = \lambda_v(T_{p,j}(t)) \cdot m_{ev,j}(t). \quad (69)$$

The local pulp-temperature balance is represented as

$$C_{p,j}(t) \cdot \frac{dT_{p,j}(t)}{dt} = q_{conv,j}(t) - q_{ev,j}(t) + m_{p,j-1 \rightarrow j}(t) \cdot (c_{p,p} + c_{p,w} \cdot X_{p,j-1}(t)) \cdot (T_{p,j-1}(t) - T_{p,j}(t)). \quad (70)$$

Here, $T_{p,j-1}(t)$ [K] is the pulp temperature entering zone j from the preceding drying zone, and $X_{p,j-1}(t)$ [kg_{H2O}/kg_{dry pulp}] is its corresponding moisture content. The pressure dynamics in every zone are obtained by differentiating Eq. 46. Thus,

$$\frac{dP_{s,j}(t)}{dt} = \frac{R_a}{V_j} \cdot T_{a,j}(t) \cdot \frac{dM_{a,j}(t)}{dt} + \frac{P_{s,j}(t)}{T_{a,j}(t)} \cdot \frac{dT_{a,j}(t)}{dt}. \quad (71)$$

Substituting the local dry air balance in Eq. 53 into Eq. 71 gives

$$\frac{dP_{s,j}(t)}{dt} = \frac{R_a}{V_j} \cdot T_{a,j}(t) \cdot \left(\sum_{k \in N_j} m_{a,k \rightarrow j}(t) + m_{a,j,ext,in}(t) - m_{a,j,ext,out}(t) \right) + \frac{P_{s,j}(t)}{T_{a,j}(t)} \cdot \frac{dT_{a,j}(t)}{dt}. \quad (72)$$

Therefore, the static pressure differences entering the pressure-driven interzone airflow relation in Eq. 43 evolve dynamically as functions of the interzone mass-transfer network and local thermal states.

EFFICIENCY EXPRESSION

For the adopted dryer boundary, the energetic input is composed of the thermal power supplied to the heating subsystem and the electrical power required by the airflow generation system. When auxiliary electrical loads are included within the selected system boundary, their contribution is incorporated separately. The useful energetic term is defined as the latent power associated with the water evaporated from the pulp [2,5,7,41,51-53].

The instantaneous total input power is defined as

$$p_{in,tot}(t) = q_{in}(t) + p_{f,el}(t) + p_{aux}(t), \quad (73)$$

where $p_{f,el}(t)$ [W] is the total electrical power consumed by the fan drive system, and $p_{aux}(t)$ [W] is the electrical power consumed by auxiliary equipment included within the system boundary. If auxiliary loads are excluded from the present model, i.e., $p_{aux}(t) = 0$, Eq. 73 reduces to

$$p_{in,tot}(t) = q_{in}(t) + p_{f,el}(t). \quad (74)$$

For a system containing N_f individual fans,

$$p_{f,el}(t) = \sum_{i=1}^{N_f} p_{f,el,i}(t), \quad (75)$$

where $p_{f,el,i}(t)$ [W] is the electrical power consumed by the i -th fan-drive assembly. The overall instantaneous energetic efficiency is defined as

$$\eta_{tot}(t) = \frac{q_{useful}(t)}{p_{in,tot}(t)}, \quad (76)$$

where $\eta_{\text{tot}}(t)$ [-] is the overall dryer efficiency and $q_{\text{useful}}(t)$ [W] is the useful thermal power associated with moisture evaporation. When the auxiliary consumption is neglected, Eq. 76 becomes

$$\eta_{\text{tot}}(t) = \frac{q_{\text{useful}}(t)}{q_{\text{in}}(t) + p_{\text{f,el}}(t)}. \quad (77)$$

Equation 77 includes both thermal and fan electrical consumption within a common energetic boundary. Therefore, $p_{\text{f,el}}(t)$, rather than only the mechanical shaft power, must be used when the objective is to evaluate the total energy consumed by the dryer. Since the present model is dynamic, the instantaneous ratio in Eq. 77 can temporarily exceed unity during startup or other transients if previously stored thermal energy is released from the pulp, humid air, or dryer structure. Therefore, the interval-averaged efficiency provides a more physically appropriate performance indicator for comparison between operating conditions. Over the interval $t_0 \leq t \leq t_1$,

$$\bar{\eta}_{\text{tot}}(t) = \frac{\int_{t_0}^{t_1} q_{\text{useful}}(t) \cdot dt}{\int_{t_0}^{t_1} p_{\text{in,tot}}(t) \cdot dt}. \quad (78)$$

From here, $\bar{\eta}_{\text{tot}}(t)$ [-] is the time-integrated overall efficiency, t_0 [s] is the initial evaluation time, and t_1 [s] is the final evaluation time. When the relevant input and useful powers remain constant over the evaluation interval, including steady-state operation, the interval-based definition becomes identical to the corresponding instantaneous power ratio.

Useful drying energy

The useful thermal power is defined as the latent-energy rate associated with moisture transferred from the liquid phase retained in the pulp to the water vapor phase [2,5,41,51]

$$q_{\text{useful}}(t) = \lambda_v(T_p(t)) \cdot m_{\text{ev}}(t). \quad (79)$$

Substitution of Eq. 79 into Eq. 76 gives

$$\eta_{\text{tot}}(t) = \frac{\lambda_v(T_p(t)) \cdot m_{\text{ev}}(t)}{q_{\text{in}}(t) + p_{\text{f,el}}(t) + p_{\text{aux}}(t)}. \quad (80)$$

For the system boundary used in the present simulations, for which $p_{\text{aux}}(t) = 0$,

$$\eta_{\text{tot}}(t) = \frac{\lambda_v(T_p(t)) \cdot m_{\text{ev}}(t)}{q_{\text{in}}(t) + p_{\text{f,el}}(t)}. \quad (81)$$

Equation 81 is a latent-energy efficiency. It therefore assigns useful energy exclusively to moisture phase change. Sensible heating of the incoming pulp, retained water, and circulating gas is treated as an energy require-

ment rather than useful evaporation energy. This definition must be maintained consistently when comparing the proposed dryer with alternative drying systems. For the N -zone representation developed previously, the total evaporation rate is given by

$$m_{\text{ev,tot}}(t) = \sum_{j=1}^N m_{\text{ev},j}(t). \quad (82)$$

The corresponding useful drying power becomes

$$q_{\text{useful}}(t) = \sum_{j=1}^N \lambda_v(T_{p,j}(t)) \cdot m_{\text{ev},j}(t). \quad (83)$$

Thus, the energetic formulation remains applicable when the lumped model is extended to resolve individual drying regions.

Thermal efficiency

The thermal efficiency is defined independently of the electrical fan demand as

$$\eta_{\text{th}}(t) = \frac{q_{\text{useful}}(t)}{q_{\text{in}}(t)} \Rightarrow \eta_{\text{th}}(t) = \frac{\lambda_v(T_p(t)) \cdot m_{\text{ev}}(t)}{q_{\text{in}}(t)}. \quad (84)$$

Here, $\eta_{\text{th}}(t)$ [-] is the instantaneous thermal efficiency and $q_{\text{in}}(t)$ [W] is the net thermal power transferred across the dryer-system boundary. For steam-based heating,

$$q_{\text{in}}(t) = m_s(t) \cdot (h_{s,\text{in}}(t) - h_{c,\text{out}}(t)), \quad (85)$$

where $m_s(t)$ [kg/s] is the steam mass flow rate; $h_{s,\text{in}}(t)$ [J/kg] is the specific enthalpy of the supplied steam; and $h_{c,\text{out}}(t)$ [J/kg] is the specific enthalpy of the condensate leaving the heating subsystem. When thermal power is evaluated from the drying air heater rather than the steam side,

$$q_{\text{in}}(t) = m_{g,h}(t) \cdot (h_{g,h,\text{out}}(t) - h_{g,h,\text{in}}(t)), \quad (86)$$

where $m_{g,h}(t)$ [kg/s] is the humid air mass flow rate through the heater, and $h_{g,h,\text{in}}(t)$ and $h_{g,h,\text{out}}(t)$ [J/kg] are the corresponding inlet and outlet specific enthalpies. For the thermal control volume, the supplied heat can be decomposed as

$$q_{\text{in}}(t) = q_{\text{useful}}(t) + q_{\text{loss}}(t) + q_{\text{sens}}(t) + q_{\text{exh}}(t) + \frac{dU_{\text{st}}(t)}{dt}, \quad (87)$$

where $q_{\text{loss}}(t)$ [W] is the heat transferred from the dryer enclosure to the surroundings, $q_{\text{sens}}(t)$ [W] is the net sensible heating requirement of the pulp, retained water, and process air; $q_{\text{exh}}(t)$ [W] is the net unrecovered enthalpy transported out of the selected system boundary by the exhaust stream; and $\frac{dU_{\text{st}}(t)}{dt} = 0$ is the rate of thermal en-

ergy accumulation within the modeled dryer inventory. The enclosure heat loss term is defined in Eq. 21, and using Eq. 87, the instantaneous thermal efficiency can equivalently be expressed as

$$\eta_{th}(t) = 1 - \frac{q_{loss}(t) + q_{sens}(t) + q_{exh}(t) + \frac{dU(t)}{dt}}{q_{in}(t)} \quad (88)$$

Equation 88 supersedes the simplified expression containing only $q_{loss}(t)$ and $q_{sens}(t)$, since exhaust enthalpy and transient thermal accumulation are also required for a complete dynamic energy balance [7,51-53]. At steady state, $\frac{dU(t)}{dt}$ and Eq. 88 reduces to

$$\eta_{th}^{ss} = 1 - \frac{q_{loss}^{ss} + q_{sens}^{ss} + q_{exh}^{ss}}{q_{in}^{ss}}, \quad (89)$$

where the superscript “ss” denotes steady-state quantities. For transient performance comparisons, the interval-based thermal efficiency is defined as

$$\bar{\eta}_{th}(t) = \frac{\int_{t_0}^{t_1} \lambda_v(T_p(t)) m_{ev}(t) dt}{\int_{t_0}^{t_1} q_{in}(t) dt} \quad (90)$$

The interval definition in Eq. 90 prevents startup thermal storage from being interpreted as sustained conversion efficiency and is therefore the appropriate quantity for comparing simulations, operating campaigns, or industrial production periods.

Fan efficiency

The aerodynamic power transferred by the fan to the drying air is

$$p_{air}(t) = \Delta p_f(t) \cdot q_a(t), \quad (91)$$

where $p_{air}(t)$ [W] is the aerodynamic air power, $\Delta p_f(t)$ [Pa] is the total pressure rise across the fan, and $q_a(t)$ [m³/s] is the volumetric airflow rate [35,36]. The pressure quantity in Eq. 91 is the fan pressure rise $\Delta p_f(t)$, rather than the dryer pressure loss term $\Delta p_d(t)$. Under quasi-steady operation,

$$\Delta p_f(t) \approx \Delta p_d(t) + \Delta p_{other}(t), \quad (92)$$

where $\Delta p_{other}(t)$ [Pa] accounts for any additional pressure requirements associated with filters, heaters, heat recovery equipment, dampers, inlet/outlet losses, or static pressure differences across the complete airflow circuit. Only when these contributions are negligible may $\Delta p_f(t) \approx \Delta p_d(t)$ be assumed. The mechanical shaft power is

$$P_{shaft}(t) = \tau_m(t) \cdot \omega_f(t), \quad (93)$$

and the aerodynamic fan efficiency is consequently

$$\eta_{f,aero}(t) = \frac{p_{air}(t)}{p_{shaft}(t)} \Rightarrow \eta_{f,aero}(t) = \frac{\Delta p_f(t) \cdot q_a(t)}{\tau_m(t) \cdot \omega_f(t)}. \quad (94)$$

If the electrical energy consumed by the drive is considered, the electrical fan power is related to shaft power by

$$p_{f,el}(t) = \frac{p_{shaft}(t)}{\eta_m(t) \cdot \eta_{drv}(t)}, \quad (95)$$

where $\eta_m(t)$ [-] is the motor efficiency, and $\eta_{drv}(t)$ [-] is the efficiency of the variable frequency drive and any additional transmission elements included in the boundary. The corresponding wire-to-air efficiency is

$$\eta_{f,el}(t) = \frac{p_{air}(t)}{p_{f,el}(t)} \Rightarrow \eta_{f,el}(t) = \eta_{drv}(t) \cdot \eta_m(t) \cdot \eta_{f,aero}(t). \quad (96)$$

The electrical fan power $p_{f,el}(t)$ enters the total input power defined in Eq. 73, and consequently, the overall efficiency expressions in Eqs. 76–78, whereas Eq. 94 evaluates only the aerodynamic performance of the fan. This distinction prevents motor/drive losses from being omitted from the plant-level energy calculation. For N_f operating fans, the overall airflow system efficiency can be calculated from aggregate powers rather than by directly averaging individual efficiencies:

$$\eta_{f,sys}(t) = \frac{\sum_{i=1}^{N_f} \Delta p_{f,i}(t) \cdot q_{a,i}(t)}{\sum_{i=1}^{N_f} p_{f,el,i}(t)}. \quad (97)$$

Here, $\Delta p_{f,i}(t)$ [Pa], $q_{a,i}(t)$ [m³/s], and $p_{f,el,i}(t)$ [W] are the fan pressure rise, volumetric airflow, and electrical input power associated with fan i , respectively. The fan affinity relationships $q_a(t) \propto \omega_f(t)$, $\Delta p_f(t) \propto \omega_f^2(t)$, and $p_f(t) \propto \omega_f^3(t)$ indicate that electrical power demand can increase approximately with the cube of rotational speed for geometrically similar operating conditions [35,36]. Therefore, variations in airflow that produce relatively small improvements in evaporation rate can produce substantially larger increases in fan power.

Specific moisture evaporation ratio and specific energy consumption

The SMER is defined as the mass of water evaporated per unit of total supplied energy [1,2,5]. Using SI power units, $SMER(t)$ can be defined as

$$SMER(t) = 3.6 \cdot 10^6 \cdot \frac{m_{ev}(t)}{p_{in,tot}(t)}, \quad (98)$$

where $SMER(t)$ [kg_{H2O}/kWh]. The factor $3.6 \cdot 10^6$ [J/kWh] converts joules to kilowatt-hours. When $p_{in, tot}(t)$ is expressed directly in kW and the evaporation rate is expressed in kg/h, Eq. 98 becomes

$$SMER(t) = \frac{m_{ev, h}(t)}{p_{in, tot, kW}(t)}, \quad (99)$$

where $m_{ev, h}(t)$ [kg/h] is the hourly evaporation rate, and $p_{in, tot}$ kW is the corresponding total input power. For dynamic simulations, the interval-based $SMER(t)$ is

$$\overline{SMER} = 3.6 \cdot 10^6 \cdot \frac{\int_{t_0}^{t_1} m_{ev}(t) \cdot dt}{\int_{t_0}^{t_1} p_{in, tot}(t) \cdot dt}. \quad (100)$$

The corresponding specific energy consumption (SEC) is

$$SEC(t) = \frac{p_{in, tot}(t)}{m_{ev}(t)}. \quad (101)$$

When expressed in MJ/kg_{H2O},

$$SEC_{MJ/kg}(t) = 10^{-6} \cdot \frac{p_{in, tot}(t)}{m_{ev}(t)}. \quad (102)$$

The corresponding relationship between $SMER(t)$ and specific energy consumption is

$$SMER(t) = 3.6 \cdot \frac{1}{SEC_{MJ/kg}(t)}, \quad (103)$$

where $SMER(t)$ is expressed in kg/kWh and $SEC(t)$ in MJ/kg_{H2O}. For a finite operating interval, SEC is more rigorously evaluated from accumulated energy and evaporated water [5]:

$$\overline{SEC}(t) = \frac{\int_{t_0}^{t_1} p_{in, tot}(t) \cdot dt}{\int_{t_0}^{t_1} m_{ev}(t) \cdot dt}. \quad (104)$$

CONTROL SYSTEM ANALYSIS AND DESIGN

Control-oriented linearization and input-output structure

Controller synthesis is performed around a nominal operating point $(\bar{x}, \bar{u}, \bar{d})$, where $\bar{x}$, $\bar{u}$, and $\bar{d}$ denote the nominal state, manipulated-variable, and disturbance vectors, respectively. Small perturbations around this operating point are defined as

$$\begin{cases} \delta x(t) = x(t) - \bar{x} \\ \delta u(t) = u(t) - \bar{u} \\ \delta d(t) = d(t) - \bar{d} \end{cases} \quad (105)$$

Linearization of the nonlinear dryer model gives

$$\begin{cases} \frac{d\delta x(t)}{dt} = \mathbf{A} \cdot \delta x(t) + \mathbf{B} \cdot \delta u(t) + \mathbf{E} \cdot \delta d(t) \\ \delta y(t) = \mathbf{C} \cdot \delta x(t) + \mathbf{D} \cdot \delta u(t), \end{cases} \quad (106)$$

where $\mathbf{A}$ is the state matrix, $\mathbf{B}$ is the manipulated-input matrix, $\mathbf{E}$ is the disturbance matrix, $\mathbf{C}$ is the output matrix, and $\mathbf{D}$ is the direct feedthrough matrix [18,54,55]. The corresponding multivariable transfer-function matrix, in s -domain is

$$G(s) = \mathbf{C} \cdot (s \cdot \mathbf{I} - \mathbf{A})^{-1} \cdot \mathbf{B} + \mathbf{D}, \quad (107)$$

where s [s⁻¹] is the Laplace variable and $\mathbf{I}$ is the identity matrix. The principal regulatory controlled variables are selected as $\mathbf{y}_r(t) = [X_p(t), q_a(t), P_g(t)]^T$ where $\mathbf{y}_r(t) \in \{\mathbb{R}^3\}$. In addition,

$$P_g(t) = P_s(t) - P_{atm} \quad (108)$$

is the chamber gauge pressure [Pa]. The corresponding physical manipulated variables are $\mathbf{u}(t) = [q_{in}(t), \tau_m(t), A_{ex}(t)]^T$ for $\mathbf{u}(t) \in \{\mathbb{R}^3\}$. The variables $T_a(t)$ [K] and $\omega_f(t)$ [rad/s] are treated as secondary controlled variables within the corresponding cascade loops rather than as independent primary outputs competing for the same manipulated variables.

The nominal steady-state process matrix is $\mathbf{K} = \mathbf{G}(0)$. Multivariable interaction is evaluated using the relative gain array (RGA) [55]: $\mathbf{\Lambda} = \mathbf{K} \circ (\mathbf{K}^{-T})$, where $\circ$ denotes the Hadamard element-by-element product. For a decentralized control configuration, pairings are selected such that the corresponding diagonal elements satisfy $\lambda_{ii} > 0$ and preferably $\lambda_{ii} \approx 1$. The proposed regulatory pairings are $X_p(t) \leftrightarrow q_{in}(t)$, $q_a(t) \leftrightarrow \tau_m(t)$, and $P_g(t) \leftrightarrow A_{ex}(t)$.

The first two pairings are implemented through cascade structures using $T_a(t)$ and $\omega_f(t)$ as secondary controlled variables. If the RGA or dynamic interaction analysis indicates strong off-diagonal coupling, static decoupling or a multivariable model predictive control (MPC) formulation is required rather than assuming independent loops [19,20,26,55,56].

The numerical RGA obtained from the linearized dryer model is listed in **Table I**. The complete hierarchical control architecture adopted in this work is shown in **Fig. 2**.

Regulatory control goals and constraints

The outlet moisture tracking error is defined as

$$e_X(t) = X_p^*(t) - X_p(t), \quad (109)$$

where $X_p^*(t)$ [kg_{H2O}/kg_{drypulp}] is the required outlet moisture reference. The drying air temperature error is

$$e_T(t) = T_a^*(t) - T_a(t), \quad (110)$$

where $T_a^*(t)$ [K] is the temperature reference supplied to the inner thermal loop. The airflow tracking error is

Controlled Variable/ Manipulated Variable	q_{in}	τ_m	A_{ex}
X_p	$\lambda_{11} = 0.9936$	$\lambda_{12} = 0.0015$	$\lambda_{13} = 0.0049$
q_a	$\lambda_{21} = 0.0070$	$\lambda_{22} = 0.7572$	$\lambda_{23} = 0.2358$
P_g	$\lambda_{31} = -0.0006$	$\lambda_{32} = 0.2413$	$\lambda_{33} = 0.7593$

I. Relative gain array (RGA) at the nominal operating point, supporting the selected decentralized input–output pairings.

$$e_q(t) = q_a^*(t) - q_a(t), \quad (111)$$

where $q_a^*(t)$ [m³/s] is the airflow reference. The fan speed tracking error is

$$e_\omega(t) = \omega_f^*(t) - \omega_f(t), \quad (112)$$

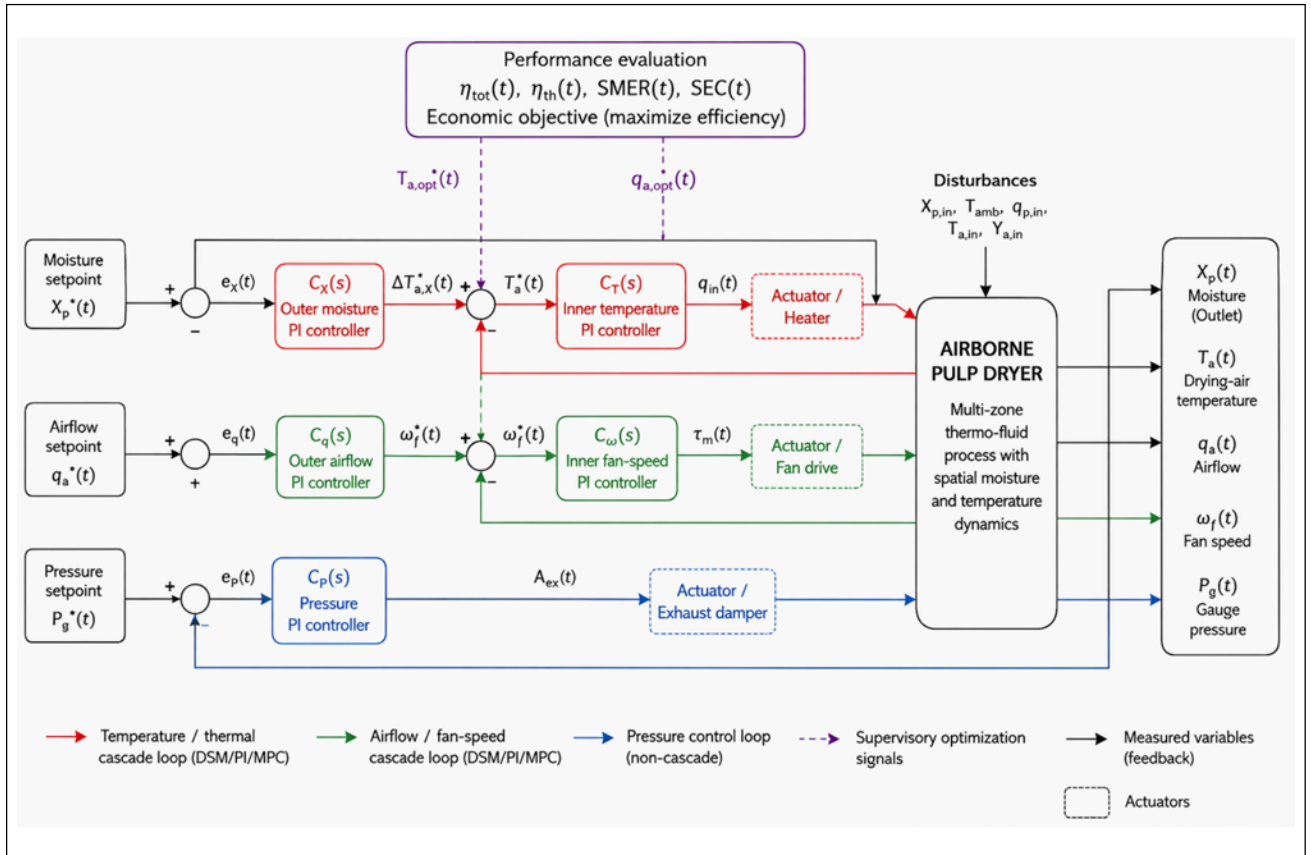
where $\omega_f^*(t)$ [rad/s] is the reference generated by the outer airflow controller. The pressure control error is

$$e_p(t) = P_g^*(t) - P_g(t), \quad (113)$$

where $P_g^*(t)$ [Pa] is the required chamber gauge-pressure reference. The physical actuators are subject to $q_{in,min} \leq q_{in}(t) \leq q_{in,max}$, $\tau_{m,min} \leq \tau_m(t) \leq \tau_{m,max}$, and $A_{ex,min} \leq A_{ex}(t) \leq A_{ex,max}$. Rate limitations are represented as, $\left| \frac{dq_m(j)}{dt} \right| \leq \left| \frac{dq_m(j)}{dt} \right|_{a, \max}$, $\left| \frac{d\tau_m(j)}{dt} \right| \leq \left| \frac{d\tau_m(j)}{dt} \right|_{\tau_{m,max}}$, and $\left| \frac{dA_{ex}(j)}{dt} \right| \leq \left| \frac{dA_{ex}(j)}{dt} \right|_{A_{ex,max}}$. Integral anti-windup is therefore included in the PI implementations whenever any actuator reaches its magnitude or rate constraint [17,54].

Direct-synthesis controller design

For each paired control channel, the local process dynam-



2. Proposed hierarchical control architecture for the airborne pulp dryer, combining direct synthesis methodology (DSM)-based cascade moisture–temperature and airflow–fan speed control loops, a non-cascade pressure controller, and a supervisory efficiency optimizer that maximizes drying performance while minimizing energy consumption.

ics are approximated by a first-order-plus-dead-time (FOPDT) model,

$$G_p(s) = K_p \cdot \frac{e^{-\theta_p s}}{\tau_p s + 1}, \quad (114)$$

where K_p is the steady-state process gain, τ_p [s] is the dominant process time constant, and θ_p [s] is the effective dead time. The parameters K_p and θ_p are obtained from local step response identification or from reduction of the linearized model in Eqs. 107 and 108 [18,54]. The desired closed-loop response is

$$G_{cl}(s) = \frac{e^{-\theta_p s}}{\lambda_p s + 1}, \quad (115)$$

where λ_p [s] is the selected closed-loop time constant. Direct synthesis gives Eq. 116,

$$G_c(s) = \frac{G_{cl}(s)}{G_p(s) \cdot (1 - G_{cl}(s))}. \quad (116)$$

Using the standard direct-synthesis internal-model-control approximation for an FOPDT process, the PI controller can be written as follows [56], [57]:

$$\begin{cases} G_c(s) = K_c \cdot \left(1 + \frac{1}{\tau_I s}\right) \\ K_c = -\frac{\tau_p}{K_p(\lambda_p + \theta_p)} \\ \tau_I = \tau_p. \end{cases} \quad (117)$$

The sign of K_c follows directly from K_p . Therefore, a process channel having a negative steady-state gain produces a negative controller gain. This is particularly relevant to moisture-temperature and pressure-damper channels.

The parameter λ_p controls the nominal speed-robustness compromise. Smaller λ_p produces faster closed-loop dynamics but larger actuator movement and reduced robustness, whereas larger λ_p produces slower but less sensitive control [54-57]. For every loop,

$$L(s) = G_p(s) \cdot G_c(s) \quad (118)$$

defines the open-loop transfer function. The sensitivity and complementary-sensitivity functions are

$$\begin{cases} S(s) = \frac{1}{1 + L(s)} \\ T(s) = \frac{L(s)}{1 + L(s)}. \end{cases} \quad (119)$$

The maximum sensitivity is

$$M_s = \|S(s)\|_\infty, \quad (120)$$

where $\|\cdot\|_\infty$ denotes the H_∞ . The value of λ_p is increased

when necessary to maintain acceptable values of M_s , gain margin, phase margin, and actuator excursion [17,54,55,57].

Direct synthesis methodology (DSM) is an analytical controller design and tuning technique used within a feedback control system. A DSM formulation was selected in the current study since it uses the identified process gain, dominant time constant, and dead time explicitly and allows the desired closed-loop time scale to be prescribed directly through λ_p [54,56,57]. This provides a less aggressive tuning mechanism than classical ultimate-cycle or Ziegler-Nichols-type rules when substantial transport delay is present. The resulting controller retains the PI form commonly available in industrial DCS/PLC platforms [17,54].

Relative to MPC, DSM does not intrinsically optimize multiple coupled variables or enforce multivariable constraints; however, it requires substantially lower online computation and substantially fewer model parameters. Therefore, MPC was retained as a benchmark for quantifying the performance penalty associated with the decentralized DSM architecture [19-24,26,27].

Cascade moisture-temperature control

Inner drying air temperature loop

The local thermal response from supplied heating power to drying air temperature is represented as

$$G_T(s) = \frac{\Delta T_a(s)}{\Delta q_{in}(s)} \Rightarrow G_T(s) = K_T \cdot \frac{e^{-\theta_T s}}{\tau_T s + 1}, \quad (121)$$

where K_T [K/W] is the thermal process gain, τ_T [s] is the dominant thermal time constant, and θ_T [s] is the associated effective delay. The DSM temperature controller is

$$\begin{cases} G_{c,T}(s) = K_{c,T} \cdot \left(1 + \frac{1}{\tau_{I,T} s}\right) \\ K_{c,T} = \frac{\tau_T}{K_T(\lambda_T + \theta_T)} \\ \tau_{I,T} = \tau_T. \end{cases} \quad (122)$$

The controller output is

$$q_{in}(t) = \text{sat} \left[q_{in,0} + K_{c,T} \cdot \left(e_T(t) + \frac{1}{\tau_{I,T}} \int e_T(t) \cdot dt \right) \right], \quad (123)$$

where $q_{in,0}$ [W] is the nominal heating duty.

Outer outlet-moisture loop

With the inner temperature loop closed, the effective outer process from drying air temperature reference to outlet pulp moisture is approximated as

$$G_X(s) = \frac{\Delta X_p(s)}{\Delta T_a(s)} \Rightarrow G_X(s) = K_X \cdot \frac{e^{-\theta_X s}}{\tau_X s + 1}, \quad (124)$$

where K_X [(kg_{H2O}/kg_{dry} pulp)/K] is the effective moisture-temperature gain, τ_X [s] is the dominant moisture time constant, and θ_X [s] is the moisture transport and measurement

delay. For normal dryer operation, $K_X < 0$, since an increase in drying air temperature reduces outlet pulp moisture. The DSM formulation automatically retains this sign through $K_{c,X}$. The outer moisture controller is

$$\begin{cases} G_{c,X}(s) = K_{c,X} \cdot \left(1 + \frac{1}{\tau_{1,X} \cdot s}\right) \\ K_{c,X} = \frac{\tau_X}{K_X \cdot (\lambda_X + \theta_X)} \\ \tau_{1,X} = \tau_X \end{cases} \quad (125)$$

The moisture controller generates a corrective temperature trim,

$$\Delta T_{a,X}^*(t) = G_{c,X}(s) \cdot e_X(t). \quad (126)$$

The actual inner-loop temperature reference is defined as

$$T_a^*(t) = \text{sat}[T_{a,\text{opt}}^*(t) + \Delta T_{a,X}^*(t)], \quad (127)$$

where $T_{a,\text{opt}}^*(t)$ [K] is the nominal economically optimal temperature supplied by the supervisory layer. Equation 125 removes the setpoint conflict that occurs if both the moisture controller and the supervisory optimizer independently assign $T_a^*(t)$. The supervisory layer determines the nominal operating point, whereas the moisture controller provides the faster regulatory correction required to satisfy product quality specifications.

For a proper cascade separation, $\lambda_X \geq 3 \cdot \lambda_T$ is imposed initially. A stronger separation, $\lambda_X \approx (4 - 5) \cdot \lambda_T$, is employed when the inner thermal dynamics permit it [17,54].

When the moisture channel dead time is significant relative to its dominant time constant, a Smith predictor can be incorporated [58]. A practical criterion is $\frac{\theta_X}{\tau_X}$, after which dead time compensation is evaluated against the non-compensated PI loop under model mismatch before implementation.

Cascade airflow–fan speed control

Inner fan speed loop

The linearized fan speed dynamics are represented as

$$G_{\omega_f}(s) = \frac{\Delta \omega_f(s)}{\Delta \tau_m(s)} \Rightarrow G_{\omega_f}(s) = K_{\omega} \cdot \frac{e^{-\theta_{\omega} \cdot s}}{\tau_{\omega} \cdot s + 1}, \quad (128)$$

where K_{ω} [(rad/s)/(N·m)] is the torque-to-speed process gain, τ_{ω} [s] is the fan electromechanical time constant, and θ_{ω} [s] represents any drive or measurement delay.

The DSM speed controller is

$$\begin{cases} G_{c,\omega}(s) = K_{c,\omega} \cdot \left(1 + \frac{1}{\tau_{1,\omega} \cdot s}\right) \\ K_{c,\omega} = \frac{\tau_{\omega}}{K_{\omega} \cdot (\lambda_{\omega} + \theta_{\omega})} \\ \tau_{1,\omega} = \tau_{\omega} \end{cases} \quad (129)$$

The resulting motor-torque command is

$$\tau_m(t) = \text{sat}[\tau_{m,0} + G_{c,\omega}(s) \cdot e_{\omega}(t)], \quad (130)$$

where $\tau_{m,0}$ [N·m] is the nominal fan torque.

Outer airflow loop

After closing the fan speed loop, the effective airflow dynamics are represented as

$$G_q(s) = \frac{\Delta q_a(s)}{\Delta \omega_f^*(s)} \Rightarrow G_q(s) = K_q \cdot \frac{e^{-\theta_q \cdot s}}{\tau_q \cdot s + 1}, \quad (131)$$

where K_q [(m³/s)/(rad/s)] is the effective airflow gain, τ_q [s] is the dominant airflow time constant, and θ_q [s] is the airflow transport delay. The outer airflow controller is

$$\begin{cases} G_{c,q}(s) = K_{c,q} \cdot \left(1 + \frac{1}{\tau_{1,q} \cdot s}\right) \\ K_{c,q} = \frac{\tau_q}{K_q \cdot (\lambda_q + \theta_q)} \\ \tau_{1,q} = \tau_q \end{cases} \quad (132)$$

The resulting fan speed reference is

$$\omega_m^*(t) = \text{sat}[\omega_{m,0} + G_{c,q}(s) \cdot e_q(t)], \quad (133)$$

where $\omega_{f,0}$ [rad/s] is the nominal fan speed. The cascade bandwidth is selected such that $\lambda_q \geq 3 \cdot \lambda_{\omega}$ with $\lambda_q \approx (4 - 5) \cdot \lambda_{\omega}$ preferred when actuator bandwidth and fan dynamics permit.

Chamber pressure control

Pressure regulation is based on chamber gauge pressure, i.e., Eq. 109. The identified pressure response to exhaust-area variation is represented as Eq. 115. For the usual configuration in which increasing the exhaust area decreases chamber pressure,

$$\begin{cases} G_{c,p}(s) = K_{c,p} \cdot \left(1 + \frac{1}{\tau_{1,p} \cdot s}\right) \\ K_{c,p} = \frac{\tau_p}{K_p \cdot (\lambda_p + \theta_p)} \\ \tau_{1,p} = \tau_p \end{cases} \quad (134)$$

The exhaust damper command is

$$A_{\text{ex}}(t) = \text{sat}[A_{\text{exp},0} + G_{c,p}(s) \cdot e_p(t)], \quad (135)$$

where $A_{\text{ex},0}$ [m²] is the nominal exhaust area.

Since A_{ex} also affects total airflow, the assumption of an independent pressure loop is accepted only if the RGA and closed-loop interaction analysis indicate sufficiently weak coupling between the pressure and airflow channels. If this condition is not satisfied, a decoupling matrix or multivariable MPC formulation is used [19,20,26,55].

Efficiency-based supervisory control

The supervisory variables were selected as $\mathbf{r}_{\text{sup}}(t) = [T_{a,\text{opt}}^*(t), q_{a,\text{opt}}^*(t)]^T$ for $\mathbf{r}_{\text{sup}}(t) \in \{\mathbb{R}^2\}$, where $T_{a,\text{opt}}^*(t)$ [K] is the economically optimal nominal drying air temperature and $q_{a,\text{opt}}^*(t)$ [m³/s] is the corresponding airflow target. The final airflow setpoint was $q_a^*(t) = q_{a,\text{opt}}^*(t)$, whereas the final temperature setpoint was obtained from Eq. 127, subject to the temperature constraints.

To avoid dimensional inconsistency between moisture error, efficiency, and $SMER(t)$, normalized variables are given by Eq. 136, where $X_{p,b}$, $\eta_{\text{tot},b}$, $\eta_{\text{th},b}$, and $SMER_b$ are positive normalization constants. The supervisory objective is formulated as Eq. 137, where $w_i \geq 0$, $i \in \{1, \dots, 6\}$, are dimensionless weighting factors, while ΔT_b [K] and Δq_b [m³/s] are normalization scales:

$$\begin{cases} \tilde{e}_X(t) = \frac{X_p(t) - X_p^*(t)}{X_{p,b}} \\ \tilde{\eta}_{\text{tot}}(t) = \frac{\eta_{\text{tot}}(t)}{\eta_{\text{tot},b}} \\ \tilde{\eta}_{\text{th}}(t) = \frac{\eta_{\text{th}}(t)}{\eta_{\text{th},b}} \\ \widetilde{SMER}(t) = \frac{SMER(t)}{SMER_b} \end{cases} \quad (136)$$

$$J_{\text{sup}}(t) = w_1 \tilde{e}_X^2(t) - w_2 \tilde{\eta}_{\text{tot}}(t) - w_3 \tilde{\eta}_{\text{th}}(t) - w_4 \widetilde{SMER}(t) + w_5 \left(\frac{\Delta T_{a,\text{opt}}^*(t)}{\Delta T_b(t)} \right)^2 + w_6 \left(\frac{\Delta q_{a,\text{opt}}^*(t)}{\Delta q_b(t)} \right)^2 \quad (137)$$

The last two terms penalize excessive supervisory setpoint movement and prevent the optimizer from producing rapidly varying temperature and airflow references. The optimization problem is

Controller / Loop	Parameter	Value
DSM: $T_a \leftarrow q_{\text{in}}$	Closed-loop parameter	$\lambda_T = \max(0.90 \cdot \tau_p, 6 \text{ [s]})$
	Integral time	$\tau_{i,T} = \max(0.55 \tau_p, 45 \text{ [s]})$
DSM: $X_p \leftarrow T_a^*$	Minimum dead time	$\theta_X \geq 156 \text{ [s]}$
	Integral time	$\tau_{i,X} = 190 \text{ [s]}$
	Controller gain limit	$-110 \leq K_{c,X} \leq -35$
	Reference-filter time constant	35 [s]
DSM: $\omega_f \leftarrow \tau_m$	Closed-loop parameter	$\lambda_{\omega} = \max(\tau_{\omega}, 5 \text{ [s]})$
	Integral time	$\tau_{i,\omega} = \tau_{\omega}$
DSM: $q_a \leftarrow \omega_f^*$	Closed-loop parameter	$\lambda_q = \max(4 \cdot \tau_{\omega}, 1.30 \cdot \tau_{qf}, 4 \text{ [s]})$
	Integral time	$\tau_{i,q} = \tau_q$
DSM: $P_g \leftarrow A_{\text{ex}}$	Closed-loop parameter	$\lambda_P = \max(\tau_P, 6 \text{ [s]})$
	Integral time	$\tau_{i,P} = \tau_P$
PID: $T_a \leftarrow q_{\text{in}}$	Robustness factor	$f_T = 1.10$
	Integral-time lower bound	$\tau_{i,T} \geq 55 \text{ [s]}$
	Derivative time	$\tau_{D,T} = 0$
PID: $X_p \leftarrow T_a^*$	Robustness factor	$f_X = 1.35$
	Minimum dead time	$\theta_X \geq 156 \text{ s}$
	Integral time	$\tau_{i,X} = 240 \text{ s}$
	Controller gain limit	$-90 \leq K_{c,X} \leq -30$
	Derivative time	$\tau_{D,X} = 0$
	Reference-filter time constant	45 [s]
PID: $\omega_f \leftarrow \tau_m$	Robustness factor	$f_{\omega} = 1.20$
	K_c	$\frac{\tau_{\omega} + 0.5 \cdot \theta_{\omega}}{K_{\omega} \cdot (\lambda_{\omega} + 0.5 \cdot \theta_{\omega})}$
	τ_i	$\tau_{\omega} + 0.5 \cdot \theta_{\omega}$
	τ_D	$\frac{\tau_{\omega} \cdot \theta_{\omega}}{2 \cdot \tau_{\omega} + \theta_{\omega}}$

II. Control parameters and numerical settings used in the direct synthesis methodology (DSM), proportional-integral-derivative (PID), and model predictive control (MPC) benchmarks.

$$T_{a,opt}^*(j), q_{a,opt}^*(j) J_{sup}, \quad (138)$$

subject to

$$\begin{cases} X_{p,min} \leq X_p(j) \leq X_{p,max} \\ T_{a,min} \leq T_a^*(j) \leq T_{a,max} \\ q_{a,min} \leq q_a^*(j) \leq q_{a,max} \\ P_{g,min} \leq P_g(j) \leq P_{g,max} \end{cases} \quad (139)$$

together with the actuator constraints in this paper's subsection "Regulatory control goals and constraints." The supervisory update interval T_{sup} is selected slower than the regulatory dynamics. A practical separation is $T_{sup} \geq 5 \cdot \tau_{cl, slow}$, where $\tau_{cl, slow}$ [s] is the largest dominant closed-loop time constant among the regulatory loops.

The optimization formulation is consistent with hierarchical and economic control structures in which regula-

tory loops operate at the faster time scale while optimization updates economically favorable references at a slower level [19,26,27,29]. Energy-related quantities entering Eq. 137 retain the thermodynamic definitions established in the efficiency analysis [51-53].

PID benchmark controller

A conventional PID structure is retained as the first benchmark required for comparative evaluation. For each control channel,

$$G_{c,PID}(s) = K_{c,PID} \left(1 + \frac{1}{\tau_{I,PID} \cdot s} + \frac{\tau_{D,PID} \cdot s}{1 + \frac{\tau_{D,PID} \cdot s}{N_D}} \right), \quad (140)$$

where $K_{c,PID}$ is the proportional gain, $\tau_{I,PID}$ [s] is the integral time, $\tau_{D,PID}$ [s] is the derivative time, and N_D [-] is the derivative-filter coefficient [17,54]. For first-order channels in which derivative action provides no measurable improvement, $\tau_{D,PID} = 0$, and Eq. 140 reduces to a PI controller.

Controller / Loop	Parameter	Value
PID: $q_a \leftarrow \omega_f^*$	Robustness factor	$f_q = 1.35$
PID: $P_g \leftarrow A_{ex}$	Robustness factor	$f_p = 1.20$
PID derivative filtering	Filter coefficient	$\alpha_D = 0.10$
MPC	Controlled variables	$[X_{p, out}, T_a]^T$
	Manipulated variable	q_{in}
	Prediction horizon, N_p	54
	Control horizon, N_u	6
	MPC execution period	10 [s]
	Prediction horizon in time	540 [s] (9 [min])
	Control horizon in time	60 [s] (1 [min])
	Q	$\text{diag}(2.2400 \cdot 10^5, 1.3889 \cdot 10^{-2})$
	R	$4.0000 \cdot 10^{-11}$
	First-difference penalty	$R_d = 0.65,$ $R = 2.6000 \cdot 10^{-11}$
	Second-difference penalty	$R_{dd} = 0.35,$ $R = 1.4000 \cdot 10^{-11}$
	Thermal duty limits	$0 \leq q_{in} \leq 20$ [MW]
	Applied first-move factor	0.45
	Output bias filter coefficient	0.97
	State filter coefficient	0.92
	Integral-servo gains	$[1/220, 1/180]^T$ [s ⁻¹]
	Integral-servo limits	$[0.003, 0.80]^T$
	Feedforward blending factor	0.35
	Delay representation	First-order Padé
	State reconstruction	Pseudoinverse ($C^\dagger$)
	Explicit energy weight, w_E	0

II. (Cont.)

The PID benchmark is tuned using an established industrial tuning procedure and is evaluated using exactly the same process model, operating point, actuator saturation, measurement filtering, disturbances, and reference trajectories as the DSM. Comparison against a different disturbance set or different actuator restrictions is not permitted.

To avoid an artificially favorable comparison, the nominal PID closed-loop speed is selected to be comparable to that of the corresponding DSM loop before robustness and energy performance metrics are evaluated [11,54,57].

MPC benchmark controller

The second benchmark is based on the discrete-time linear model

$$\begin{cases} \delta \mathbf{x}_{k+1} = \mathbf{A}_d \delta \mathbf{x}_k + \mathbf{B}_d \delta \mathbf{u}_k + \mathbf{E}_d \delta \mathbf{d}_k \\ \delta \mathbf{y}_k = \mathbf{C}_d \delta \mathbf{x}_k + \mathbf{D}_d \delta \mathbf{u}_k, \end{cases} \quad (141)$$

where k is the discrete sampling index and $\mathbf{A}_d$, $\mathbf{B}_d$, $\mathbf{E}_d$, $\mathbf{C}_d$, and $\mathbf{D}_d$ are the discrete-time model matrices. For a prediction horizon N_p and control horizon N_u , the MPC cost function is defined as [19-24]

$$J_{\text{MPC}} = \sum_{i=1}^{N_p} \left\| \hat{\mathbf{y}}_{k+i|k} - \mathbf{y}_{k+i}^* \right\|_{\mathbf{Q}}^2 + \sum_{i=0}^{N_u-1} \left\| \Delta \mathbf{u}_{k+i|k} \right\|_{\mathbf{R}}^2 + w_E \cdot \sum_{i=1}^{N_p} \frac{p_{\text{in,tot},k+i|k}}{p_b}, \quad (142)$$

where $\hat{\mathbf{y}}_{k+i|k}$ is the predicted controlled-variable vector; $\mathbf{y}_{k+i}$ is the reference vector; $\Delta \mathbf{u}_{k+i|k}$ is the predicted manipulated-variable increment; $\mathbf{Q} \geq 0$ is the output-error weighting matrix; $\mathbf{R} \geq 0$ is the control movement weighting matrix; $w_E \geq 0$ is the energy weighting coefficient; and p_b [W] is the normalization power. The MPC optimization is subject to

$$\begin{cases} u_{\min} \leq u_{k+i|k} \leq u_{\max} \\ \Delta u_{\min} \leq \Delta u_{k+i|k} \leq \Delta u_{\max} \\ y_{\min} \leq \hat{y}_{k+i|k} \leq y_{\max}. \end{cases} \quad (143)$$

The MPC therefore provides the benchmark for the achievable improvement obtained by explicitly considering multivariable interactions, future predicted behavior, actuator constraints, and energy consumption [19,20,23,26,27]. Hierarchical MPC has also been reported for energy regulation in industrial drying processes [29].

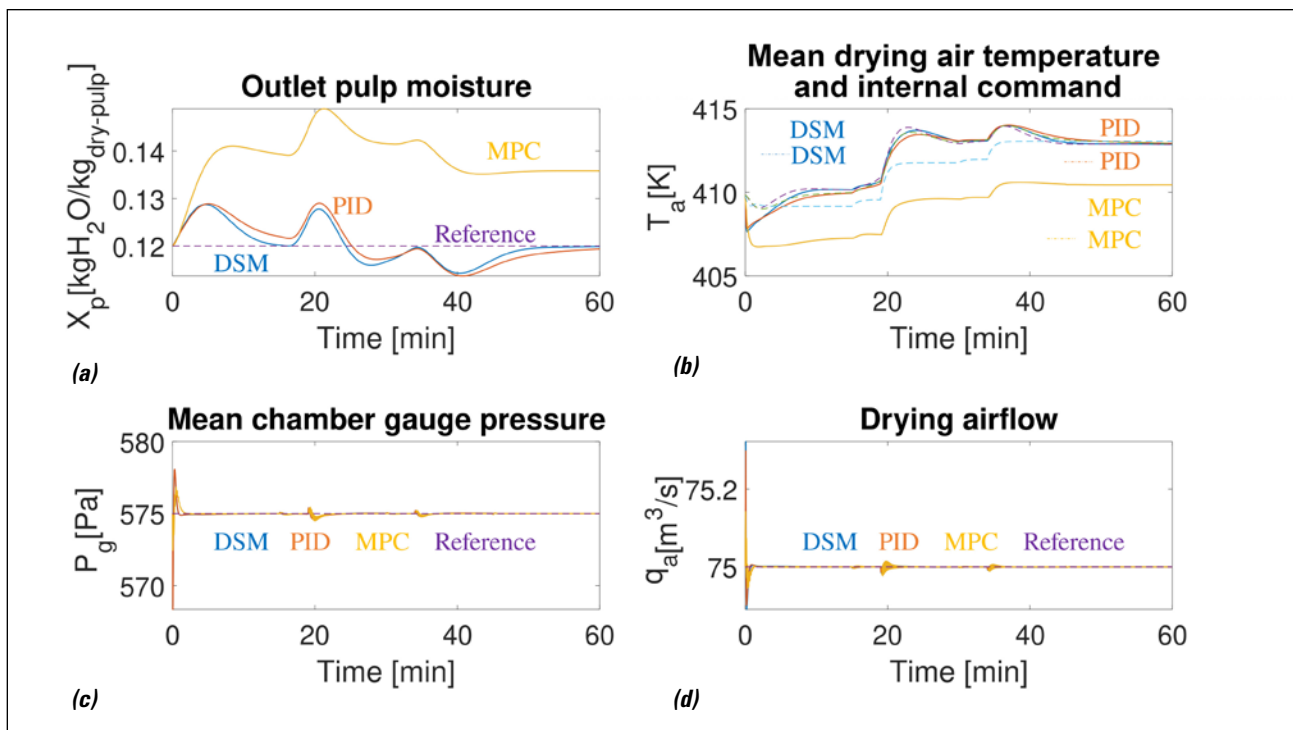
Finally, the controller parameters used in the numerical benchmark are summarized in **Table II**.

SIMULATION RESULTS

All simulations were implemented and executed in MATLAB (MathWorks; Natick, MA, USA) using the complete nonlinear multizone thermofluid model; the through-thickness moisture and temperature representation; the fan-pressure subsystem; and the DSM, PID control, and MPC control algorithms. The continuous nonlinear states

Parameter	Value
Simulation horizon	3600 [s]
Drying zones, N_z	6
Through-thickness nodes, N_{ξ}	5
Controller sampling time, T_s	1 [s]
RK4 integration step	0.20 [s]
Nominal pulp residence time, τ_{res}	300 [s]
Inlet pulp moisture, $X_{p,\text{in}}(0)$	0.600 [kg H_2O /kg dry pulp]
Outlet pulp moisture, $X_{p,\text{out}}(0)$	0.120 [kg H_2O /kg dry pulp]
Dry pulp flow rate, $m_p(0)$	8.0 [kg/s]
Mean drying air temperature, $T_a(0)$	≈ 410 [K]
Mean chamber gauge pressure, $P_g(0)$	575 [Pa]
Drying airflow, $q_a(0)$	75 [m ³ /s]
Thermal duty, $q_{\text{in}}(0)$	≈ 14.3 [MW]
Fan motor torque, $\tau_m(0)$	≈ 1.18 [kN·m]
Effective exhaust area, $A_{\text{ex}}(0)$	≈ 3.77 [m ²]
Ambient temperature, $T_{\text{amb}}(0)$	295 [K]
MPC execution period	10 [s]
MPC prediction/control horizons	$N_p = 54$, $N_u = 6$
First disturbance	Inlet moisture step
Second disturbance	Dry pulp-throughput step
Third disturbance	Ambient temperature step
Inlet moisture disturbance	Applied at $t = 15$ min
Dry pulp throughput disturbance	Applied at $t = 30$
Ambient temperature disturbance	Applied at $t = 45$
RK4: fourth-order Runge-Kutta method; MPC: model predictive control.	

III. Main simulation parameters.



3. Closed-loop dynamic responses for the direct synthesis methodology (DSM), proportional-integral-derivative (PID), and model predictive control (MPC) strategies: (a) outlet pulp moisture; (b) mean drying air temperature and internal temperature commands; (c) mean chamber gauge pressure; and (d) drying airflow. Dashed lines denote the corresponding reference trajectories.

were integrated using a fixed-step fourth-order Runge–Kutta scheme. Fixed-step integration is appropriate when a prescribed numerical sampling structure is required for dynamic system simulation and controller execution.

Before applying any disturbance or controller performance test, the dryer was initialized at its calculated non-zero steady-state operating point. Therefore, the initial state and manipulated input vectors satisfied, within the numerical convergence tolerance $\dot{\mathbf{x}}(0) = \mathbf{0}$, therefore, that the initial transient did not originate from an artificial mismatch between the initial conditions and the nominal operating point. This is consistent with standard dynamic simulation practice, in which the initial states and inputs are selected from an equilibrium operating point before subsequent transient analysis. The principal simulation conditions are summarized in **Table III**.

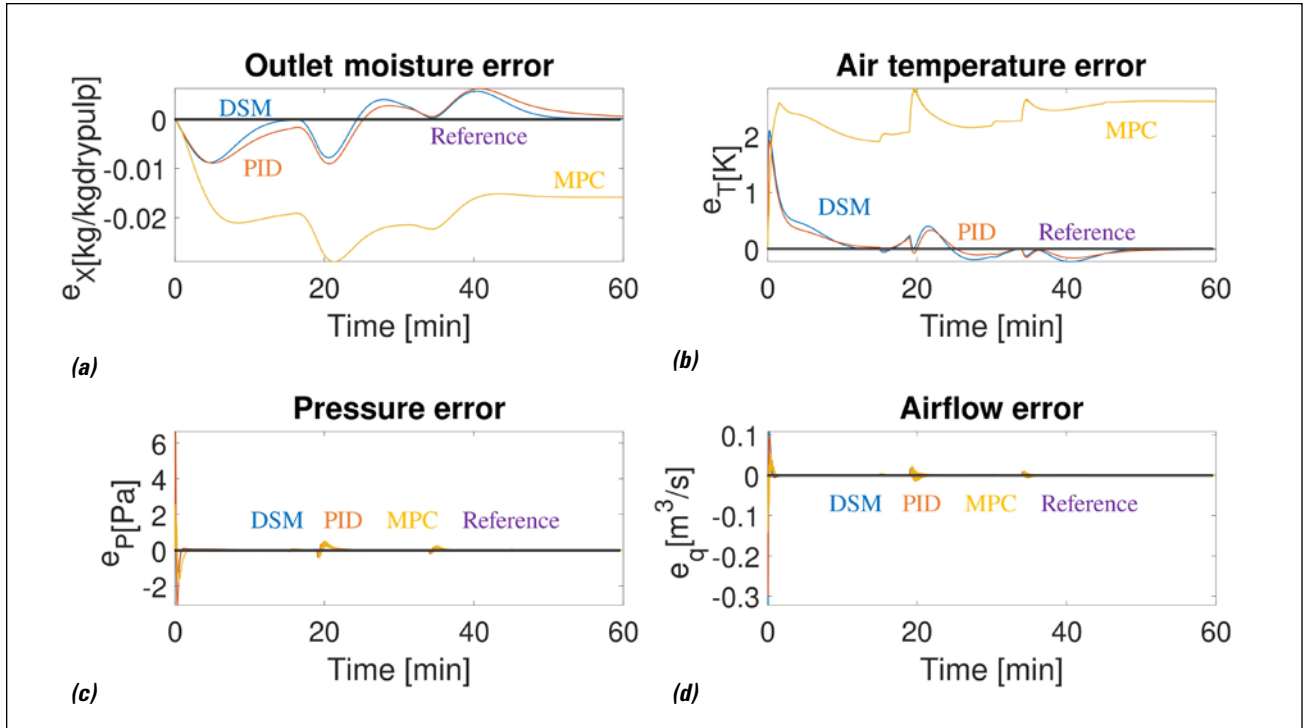
The steady-state values of $q_{in}(0)$, $\tau_m(0)$, and $A_{ex}(0)$ correspond to the equilibrium manipulated variables required to maintain simultaneously $X_{p,out}(0) = 0.120$, $T_a(0) \approx 410$ K, $P_g(0) = 575$ Pa, and $q_a(0) = 75$ m³/s. Thus, the controller comparisons begin from the same physically consistent operating condition, rather than from different controller-specific initial states.

Figure 3 compares the main controlled variables: (a) outlet pulp moisture, (b) mean drying air temperature, (c) chamber gauge pressure, and (d) drying airflow. The PID control and DSM maintain the outlet moisture substantially closer to $X_p^* = 0.120$ kg_{H₂O}/kg_{dry-pulp} than MPC alone.

The DSM generally exhibits the smallest moisture excursions, while PID control provides a closely comparable but slightly slower response. The thermal trajectories in Fig. 3b remain smooth for PID control and DSM, whereas MPC develops considerably larger deviations. In contrast, the faster pressure and airflow dynamics in Figs. 3c–3d are tightly regulated around $P_g^* = 575$ Pa, $q_a^* = 75$ m³/s, particularly for PID control and DSM.

Figure 4 shows the associated errors: (a) moisture error e_x , (b) temperature error e_T , (c) pressure error e_p , and (d) airflow error e_q . The DSM presents the lowest overall moisture error accumulation, followed by PID control, whereas MPC exhibits both larger error amplitudes and slower attenuation. The pressure and airflow errors for PID control and DSM remain practically zero after short transients, confirming effective dynamic separation between the slow moisture–thermal subsystem and the faster pneumatic dynamics.

The corresponding manipulated variables are shown in **Fig. 5**: (a) thermal duty q_{in} , (b) fan motor torque τ_m , and (c) effective exhaust area A_{ex} . The PID control and DSM generate smooth and bounded control actions, with q_{in} increasing from approximately 14.0 MW toward 15 MW as drying demand increases. Fan torque remains close to 1.18 kN·m and A_{ex} remains near its nominal operating range. The MPC generates larger low-frequency variations, consistent with its poorer moisture–thermal response.



4. Closed-loop tracking errors obtained for the DSM, PID, and MPC strategies: (a) outlet pulp moisture error; (b) drying air temperature error; (c) mean chamber pressure error; and (d) drying airflow error. The zero-error reference is indicated by the dashed line.

Figure 6 presents (a) overall efficiency η_{tot} , (b) thermal efficiency η_{th} , (c) wire-to-air fan efficiency η_f , and (d) *SMER*. The PID control and DSM produce similar energetic behavior, with representative values around $\eta_{tot} \approx 0.60$, $\eta_{th} \approx 0.61$, and *SMER* ≈ 0.95 – 0.96 kg_{H₂O}/kWh.

The fan efficiency remains approximately 0.777, indicating operation within a narrow aerodynamic region. Although MPC occasionally presents favorable instantaneous energetic values, these cannot be interpreted independently of its larger outlet moisture deviation.

Figure 7 reports the multizone states: (a) drying air temperatures, (b) zone gauge pressures, (c) humidity ratios, and (d) zone-average pulp moisture. The six zones retain an ordered temperature and pressure distribution. Simultaneously, the gas humidity increases progressively through the dryer as evaporated moisture is transferred to the air, while the pulp moisture decreases from approximately 0.54 kg/kg in the first zone toward approximately 0.12 kg/kg at the outlet.

Figure 8 presents (a) the pressure-driven deck-to-deck dry air flow rates and (b) local evaporation rates. The five interzone airflow trajectories remain closely grouped around 64–65 kg/s, indicating consistent hydraulic continuity. Local evaporation remains approximately within 0.56–0.73 kg/s and changes smoothly with the operating condition.

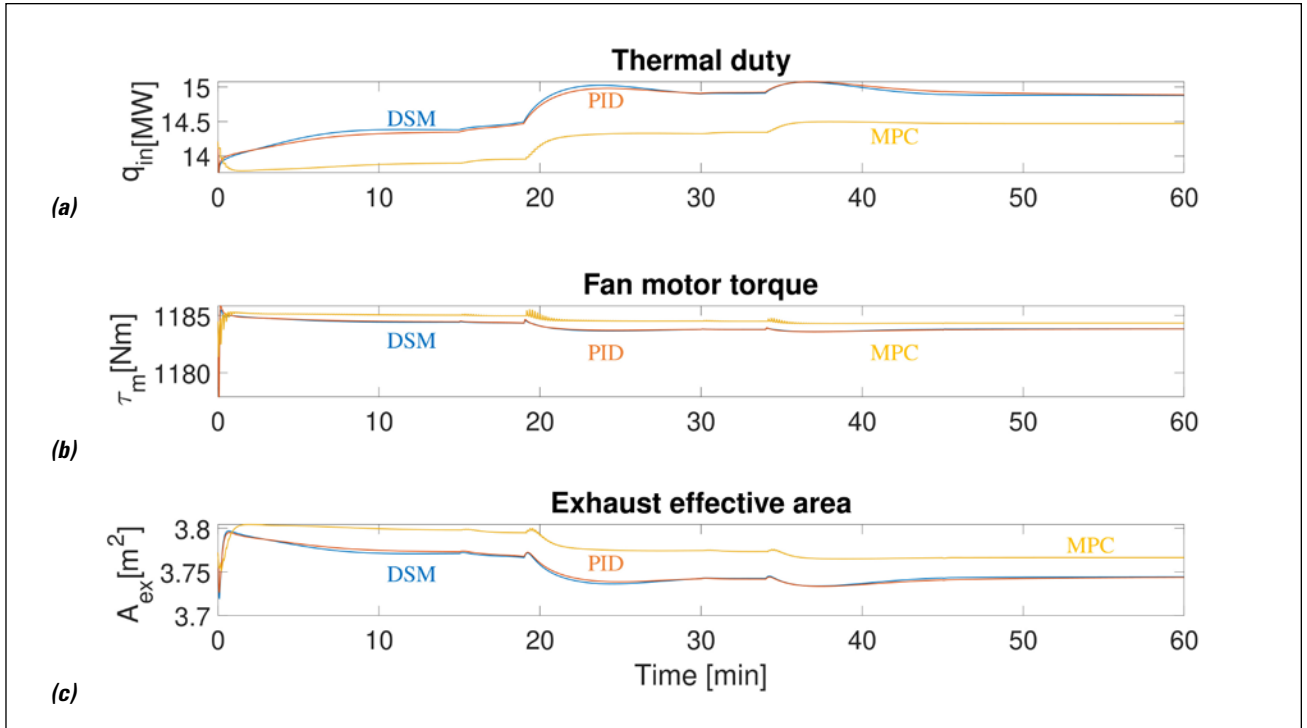
The spatial drying behavior is presented in **Fig. 9**, where (a) shows the final through-thickness moisture field $X_p(\zeta, t)$ and (b) the corresponding pulp-temperature field

$T_p(\zeta, t)$. A strong longitudinal reduction of moisture is accompanied by finite through-thickness gradients, demonstrating that the model does not impose spatially uniform pulp states. Experimental studies of through-air paper drying likewise report spatially nonuniform moisture fields and significant thickness-direction transport [39].

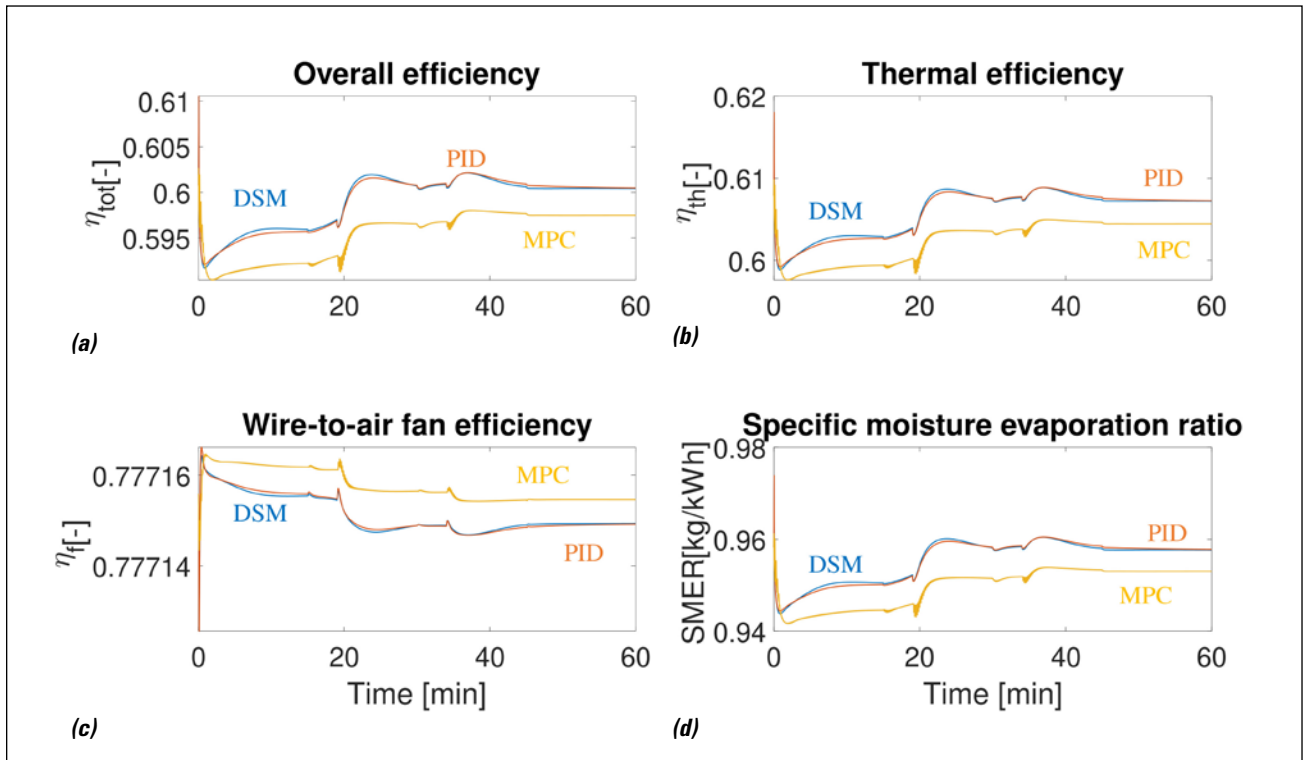
Figure 10 gives the outlet zone distributions: (a) moisture profiles $X_p(\zeta, t)$, and (b) temperature profiles $T_p(\zeta, t)$. Moisture remains higher near the pulp mid-plane and lower at the exposed surfaces, while temperature exhibits the complementary curvature. This behavior is consistent with the interaction between surface evaporation and finite internal moisture diffusion reported for through-air drying of fibrous sheets [39,40].

Figure 11 provides the quantitative controller benchmark through (a) moisture integrated absolute error, (b) total supplied energy, (c) specific energy consumption, and (d) *SMER*. The principal distinction occurs in product moisture regulation, for which $IAE_X^{DSM} < IAE_X^{PID} < IAE_X^{MPC}$. The DSM therefore provides the best closed-loop quality regulation under the present conditions, followed by PID. The differences in total supplied energy, SEC, and *SMER* are considerably smaller, showing that energetic performance must be interpreted together with fulfillment of the outlet moisture requirement.

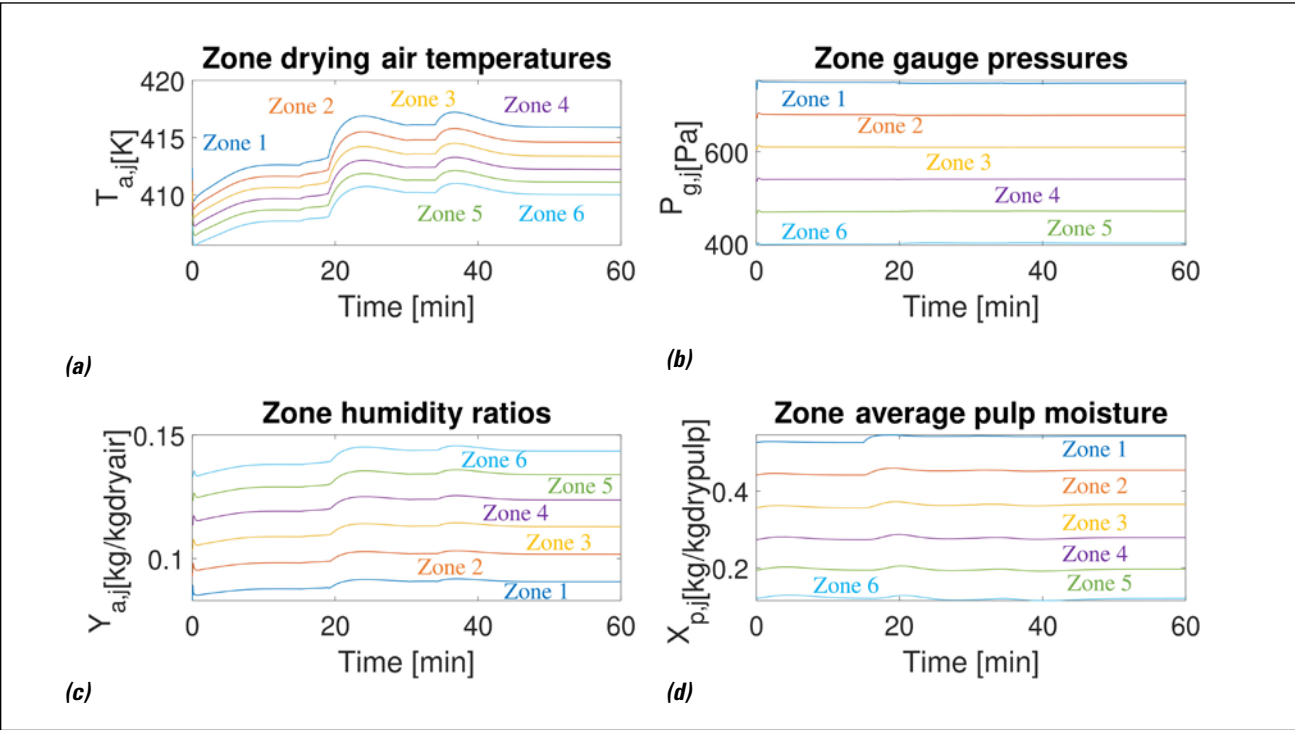
The quantitative closed-loop performance is further summarized in **Table IV**. Settling time is evaluated as the time required for the response to enter and remain within



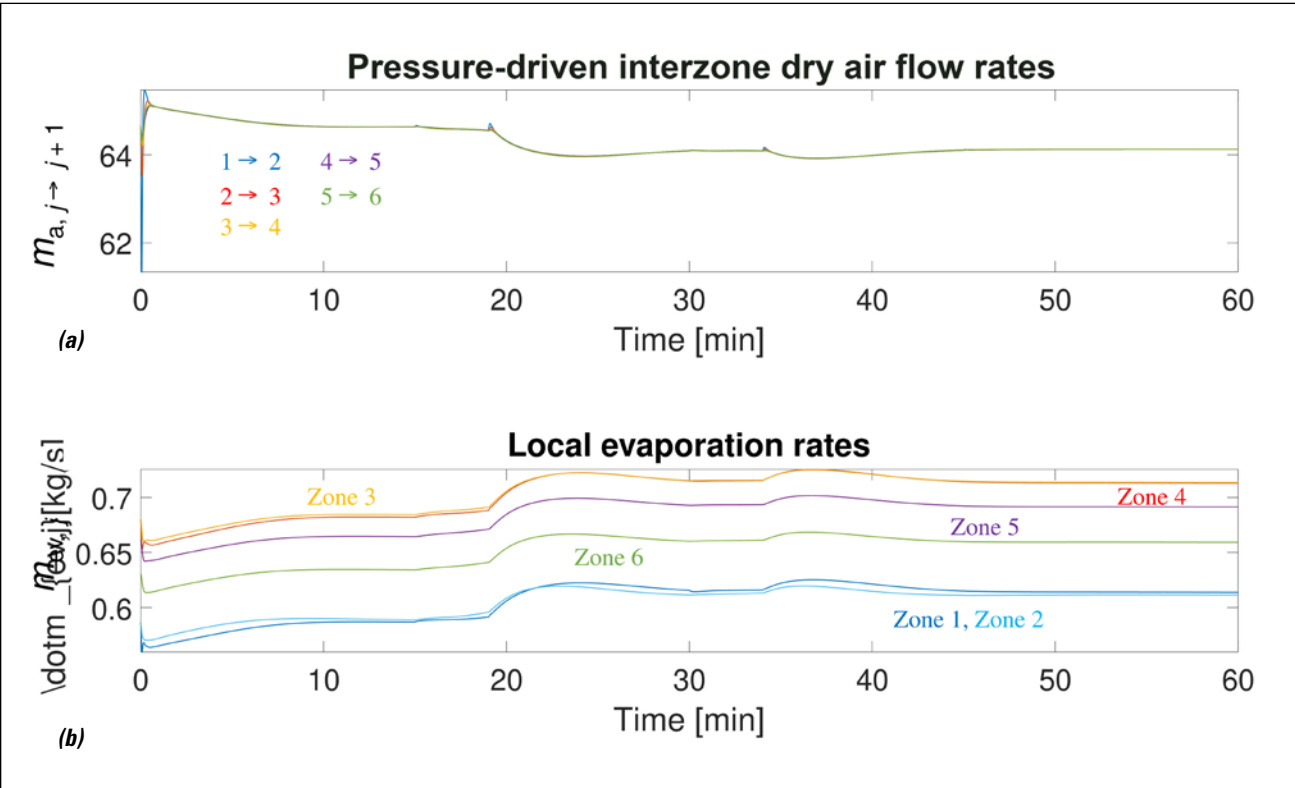
5. Manipulated variable responses obtained for the DSM, PID, and MPC strategies: (a) thermal duty command $q_{in}(t)$; (b) fan motor torque $\tau_m(t)$; and (c) effective exhaust area $A_{ex}(t)$, showing the control effort required to regulate the thermofluid dryer dynamics.



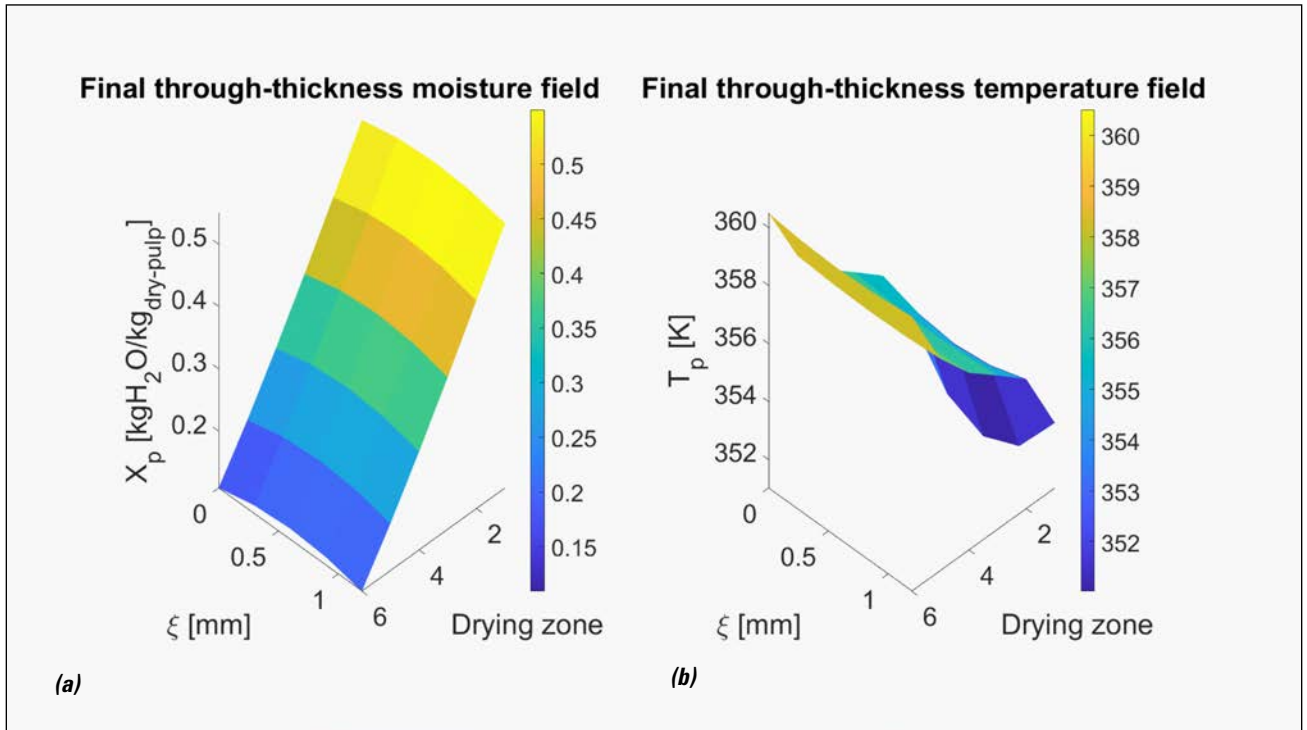
6. Energetic performance responses obtained for the DSM, PID, and MPC strategies: (a) overall dryer efficiency $\eta_{tot}(t)$; (b) thermal efficiency $\eta_{th}(t)$; (c) wire-to-air fan efficiency $\eta_f(t)$; and (d) specific moisture evaporation ratio (SMER(t)), expressed in kg_{H_2O}/kWh .



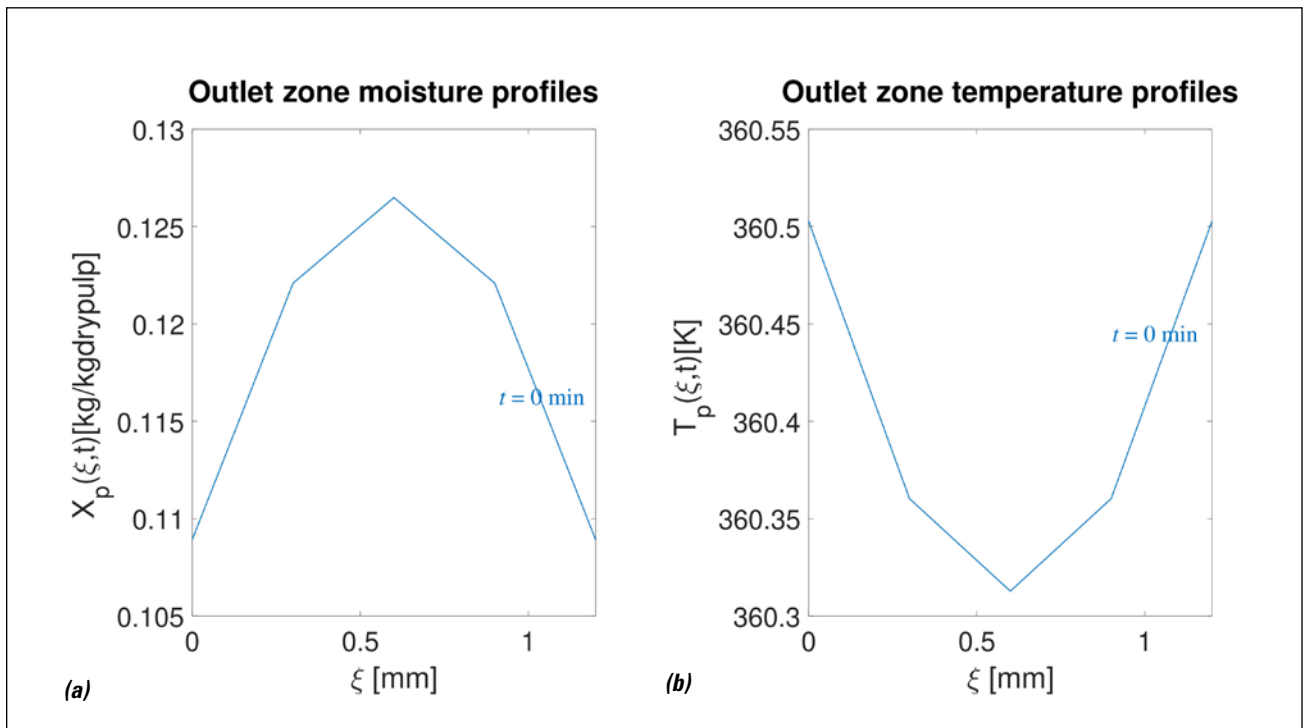
7. Multizone thermofluid responses of the airborne pulp dryer: (a) drying air temperature in each zone; (b) zone gauge pressure distribution; (c) zone humidity ratio evolution; and (d) zone-average pulp moisture content, illustrating the progressive thermal, vapor, pressure, and drying profiles along the six-zone system.



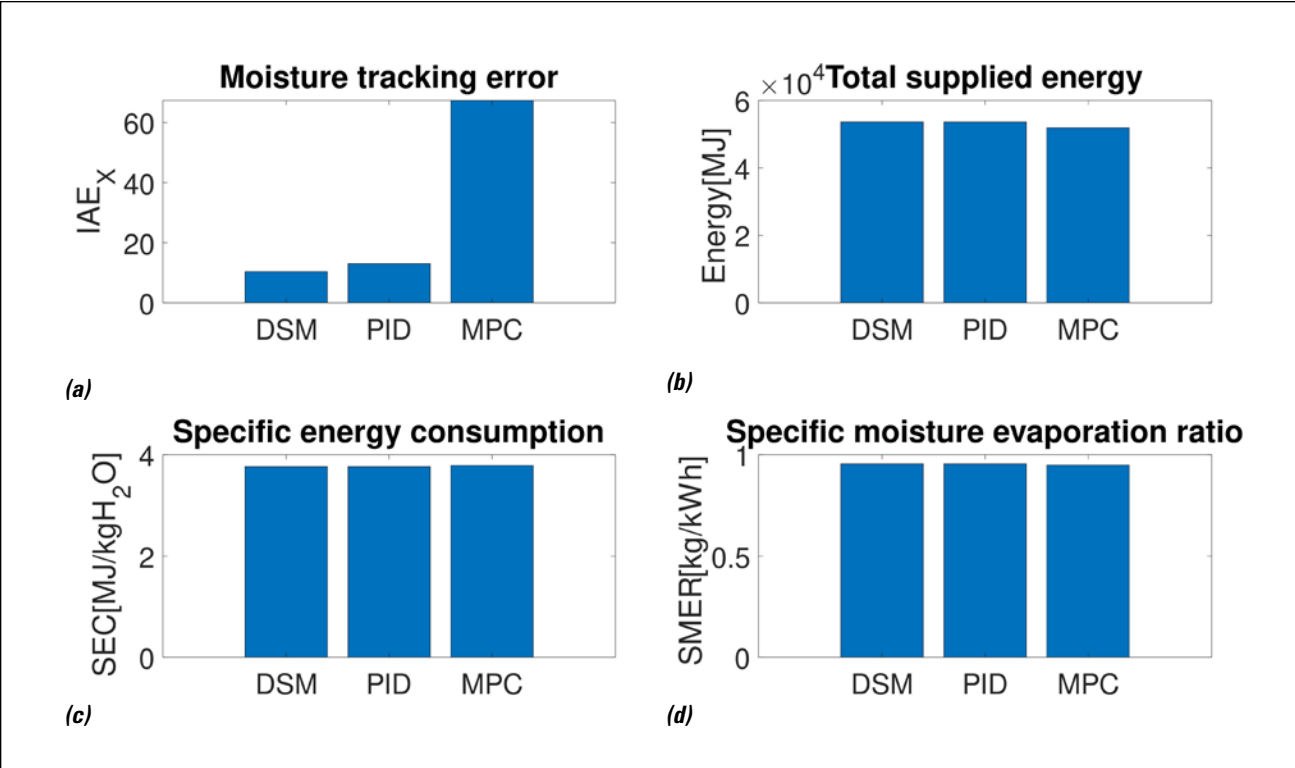
8. Interzone drying dynamics: (a) pressure-driven dry air mass flow rates between adjacent zones, and (b) local evaporation rates in the individual drying zones.



9. Final spatial distributions predicted by the multi-zone drying model: (a) through-thickness pulp moisture field $X_p(\zeta, j)$, and (b) through-thickness pulp temperature field $T_p(\zeta, j)$, showing the coupled longitudinal and internal gradients developed across the six drying zones.



10. Outlet zone through-thickness distributions at the steady-state startup condition $t = 0$: (a) pulp moisture profile $X_p(\zeta, 0)$, and (b) pulp temperature profile $T_p(\zeta, 0)$, showing a wetter, slightly cooler sheet core and drier, warmer exposed surfaces.



11. Quantitative performance comparison of the DSM, PID, and MPC controllers: (a) integrated absolute outlet moisture tracking error $IAE_x(t)$; (b) total supplied energy; (c) specific energy consumption (SEC(t)); and (d) specific moisture evaporation ratio (SMER(t)). Higher SMER indicates greater moisture removal per unit of supplied energy.

the prescribed tolerance band; a 2% criterion is adopted for outlet moisture.

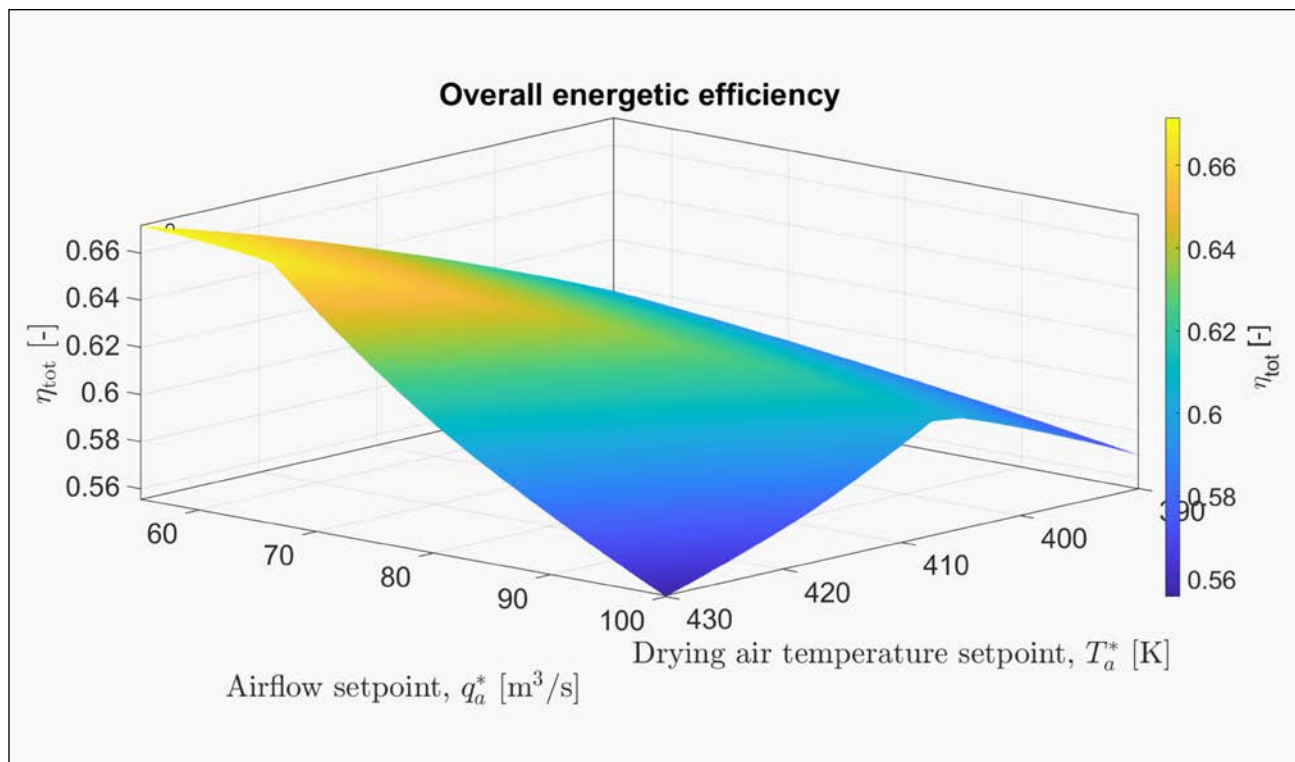
The comparatively poor MPC response should not be interpreted as an intrinsic limitation of MPC. The present controller relies on a reduced prediction model for a strongly nonlinear, distributed and delay-dominated process. The MPC behavior is particularly sensitive to prediction and

control horizons, output and input weighting, move suppression, state estimation, and model–plant mismatch; systematic tuning of these quantities is essential for achieving the desired closed-loop dynamics. Consequently, optimization of N_p , N_u , Q , and R improved transport delay representation, and enhanced disturbance/state estimation, multi-operating-point modeling, and nonlinear or adaptive MPC

Performance Metric	DSM	PID	MPC
Moisture IAE_x	9.6	12.4	67.1
Moisture settling time, $t_{s,x}$ [min]	≈ 1.0	≈ 4.1	Not settled
Temperature settling time, $t_{s,T}$ [min]	≈ 5.0	≈ 5.0	Not settled
Total supplied energy [MJ]	$5.34 \cdot 10^4$	$5.34 \cdot 10^4$	$5.15 \cdot 10^4$
SEC [MJ/kg H_2O]	≈ 3.74	≈ 3.74	≈ 3.78
SMER [kg H_2O /kWh]	≈ 0.96	≈ 0.96	≈ 0.95
Final moisture reference recovery	Yes	Yes	No
Stability under nominal disturbance sequence	Stable	Stable	Stable, but with offset

SEC: specific energy consumption; SMER: specific moisture evaporation ratio; DSM: direct synthesis methodology; PID: proportional-integral-derivative; MPC: model predictive control.

IV. Closed-loop control and energetic performance benchmark.



12. Three-dimensional overall energetic efficiency surface as a function of drying air temperature and airflow setpoints. The marker indicates the maximum feasible operating point.

are retained as future work. The present MPC results should therefore be regarded as a benchmark of the current tuning rather than evidence of an inherent disadvantage of predictive control.

Figure 12 shows that overall efficiency increases toward the high-temperature, low-airflow region, reaching $\eta_{\text{tot}} = 0.6679$ at $T_a^* = 427.5$ K and $q_a^* = 55.0$ m³/s. This trend reflects the energetic penalty of excessive airflow and the benefit of higher drying temperature within the outlet moisture constraint.

Figure 13 shows the corresponding thermal efficiency surface, and confirms that thermal efficiency depends jointly on the drying temperature and airflow because both variables modify evaporation, sensible energy transport, and thermal losses.

Figure 14 presents the *SMER* surface, with the maximum region approaching approximately 1.05–1.07 kg_{H₂O}/kWh. The trend is consistent with the efficiency surfaces, with the highest specific moisture removal occurring at elevated temperature and comparatively moderate airflow.

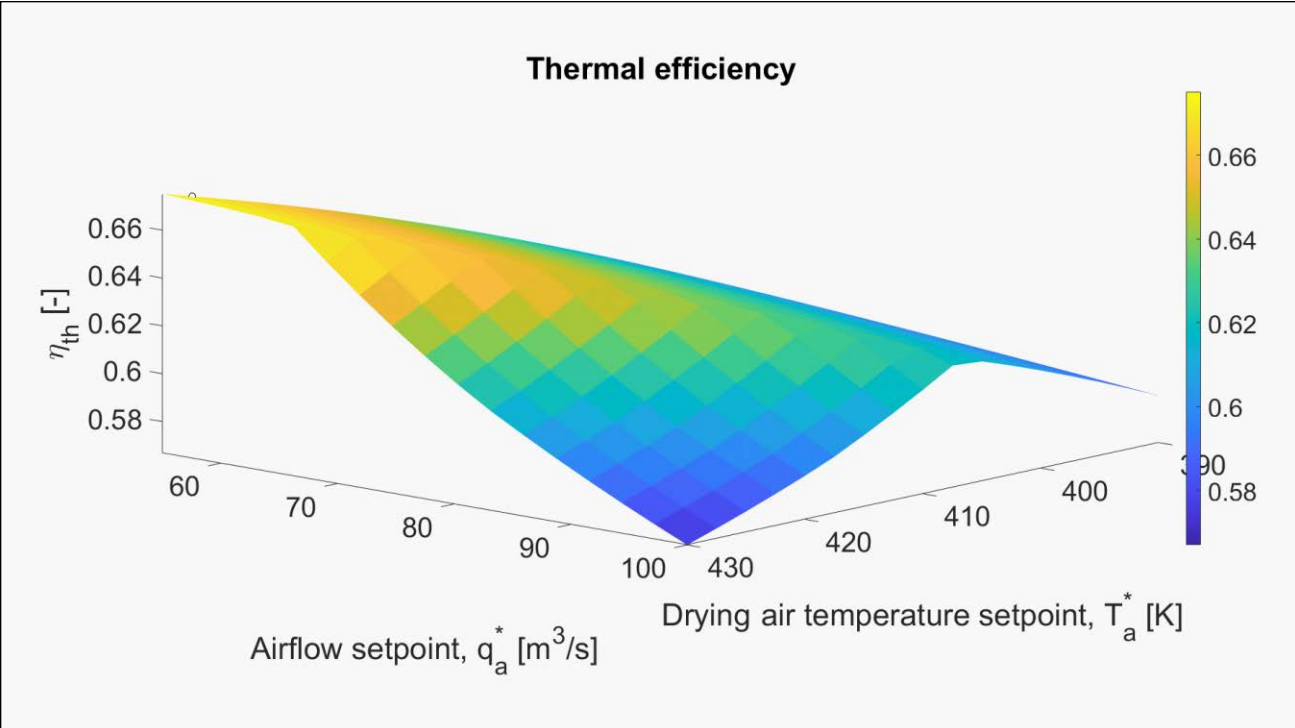
Finally, **Fig. 15** summarizes the steady-state operating domain through (a) overall efficiency, (b) thermal efficiency, (c) *SMER*, and (d) outlet pulp moisture. Figs. 15a–15c identify energetically favorable regions, while Fig. 15d demonstrates that some of these operating points would produce excessive drying. Thus, optimization must be constrained by $X_{p, \text{out}} \approx 0.120$ kg_{H₂O}/kg_{dry-pulp} rather than by maximizing an energy metric independently.

Overall, the MATLAB simulations represented by Figs. 3–15 indicate that DSM provides the best current compromise between outlet-moisture tracking, smooth actuator action, pneumatic regulation, and energetic performance. The PID control remains closely competitive, whereas the current MPC implementation requires additional model refinement and systematic tuning to achieve comparable moisture–thermal regulation.

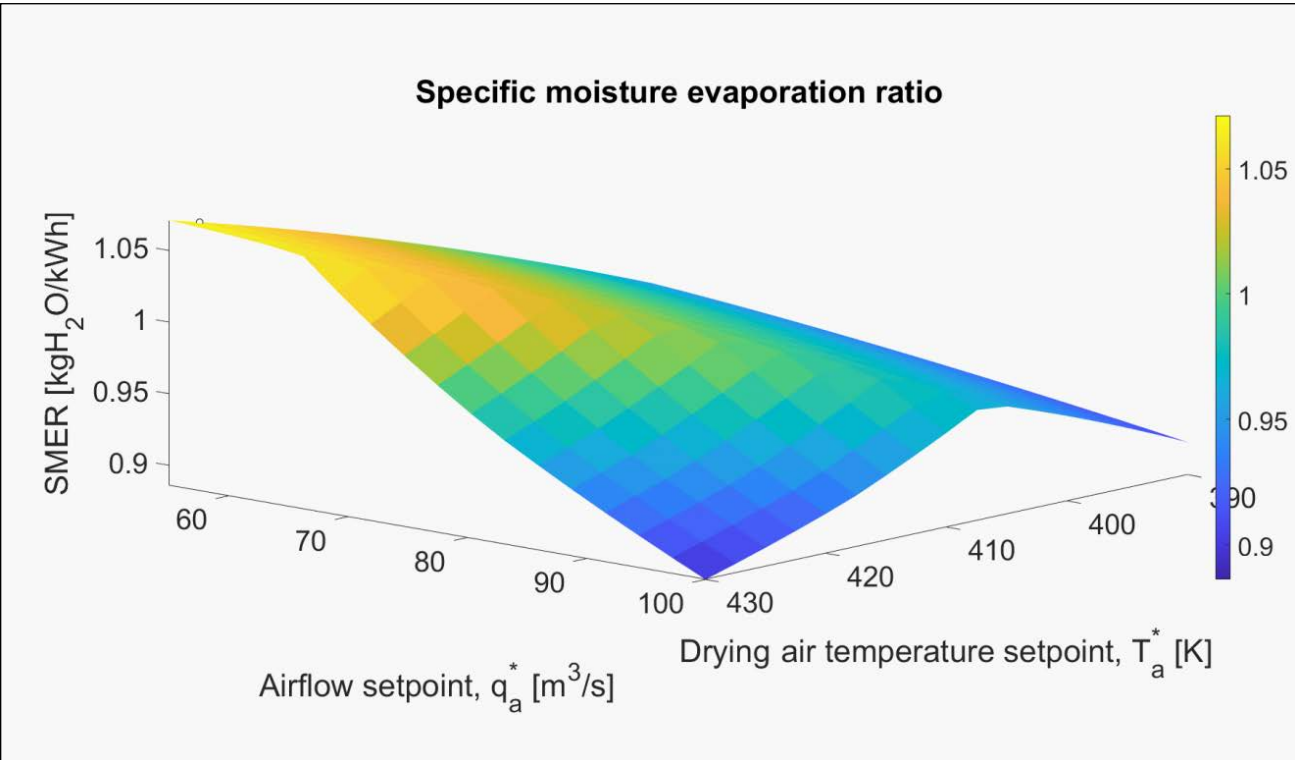
CONCLUSIONS

A nonlinear control-oriented model of an industrial airborne pulp dryer was developed by coupling multizone mass and energy balances, psychrometric moisture transfer, pressure-driven airflow, fan dynamics, and through-thickness moisture and temperature transport. MATLAB simulations reproduced the expected progressive drying behavior across six zones while maintaining consistent pressure, humidity, airflow, evaporation, and spatial pulp state distributions. The energetic formulation predicted representative values of $\eta_{\text{tot}}(t) \approx 0.60$, $\eta_{\text{th}}(t) \approx 0.61$, and $\text{SMER}(t) \approx 0.95$ kg_{H₂O}/kWh, while the operating maps demonstrated the required compromise between energy performance and the outlet-moisture constraint.

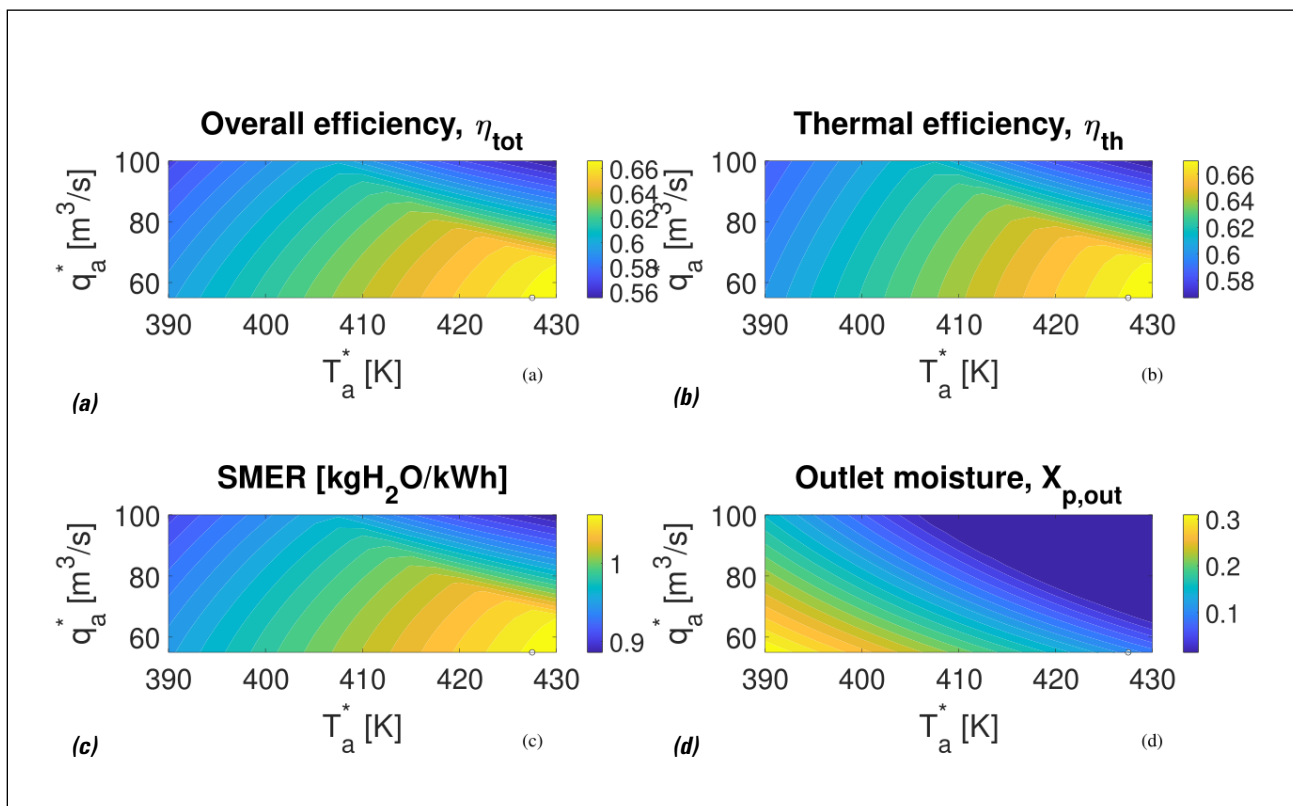
Among the evaluated control methods, DSM provided the lowest accumulated moisture tracking error, smooth manipulated variable trajectories, and tight pressure and airflow regulation; PID control produced closely comparable fast-loop behavior but moderately higher moisture



13. Three-dimensional thermal efficiency surface, $\eta_{th}(T_a^*, q_a^*)$, showing the combined influence of drying air temperature and airflow setpoints on the thermal performance of the airborne pulp dryer.



14. Three-dimensional specific moisture evaporation ratio surface, $SMER(T_a^*, q_a^*)$, showing the combined effect of drying air temperature and airflow setpoints on the amount of moisture removed per unit of supplied energy.



15. Two-dimensional operating maps as functions of drying air temperature and airflow setpoints: (a) overall efficiency η_{tot} ; (b) thermal efficiency η_{th} ; (c) specific moisture evaporation ratio (SMER(t)); and (d) outlet pulp moisture $X_{p,out}$ (t). The maps illustrate the energetic–quality trade-off used to identify admissible operating regions.

error. The present MPC implementation produced the poorest moisture–thermal regulation. This result is attributed to the reduced predictive representation and current selection of prediction horizon, control horizon, weighting matrices, and disturbance/delay treatment rather than to an intrinsic limitation of MPC; these parameters are well known to strongly influence MPC performance. Future work will therefore focus on plant data parameter identification and experimental validation, systematic MPC retuning, improved transport-delay and state estimation, multi-operating point models, and non-linear or adaptive MPC. Further refinement of the internal moisture transport description is also appropriate given the recognized importance of spatial transport phenomena in paper drying.

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We selected airborne pulp drying for study because moisture control, airflow, pressure, and energy use are strongly coupled and directly affect dryer productivity and operating efficiency. This research extends previous paper drying models by combining multi-zone thermofluid dynamics, through-thickness moisture and temperature transport, energy-efficiency analysis, and direct synthesis methodology (DSM)/model predictive control (MPC)/proportional-integral-derivative (PID) control comparison within one framework.

The main difficulty in this study was representing strongly coupled thermal, moisture, airflow, pressure, and spatial dynamics while retaining a model suitable for control design, which was addressed using reduced-order zone balances combined with spatial discretization.

The study showed that good outlet moisture regulation requires coordinated control of thermal input, airflow, and chamber pressure rather than treating these variables independently. The most interesting result was that the relatively simple DSM architecture achieved better moisture tracking than the implemented MPC benchmark while maintaining similar energy performance.

Mills could use the studied framework to support controller tuning, energy performance assessment, operating point optimization, and future dryer digital-twin development. The next step in this research is to identify and validate the model using industrial dryer data and then improve and retune the MPC using a higher-fidelity prediction model.



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