

Full Length Article

Temperature and humidity decoupling control based on diagonal matrix and information entropy for spinning air-conditioning[☆]

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ABSTRACT

Spray chambers are widely used in spinning air-conditioning systems to humidify air. However, the cooling and humidification air treatment leads to the mutual interference between the temperature and humidity control. Moreover, the traditional proportional-integral (PI) control based on passive set-point tracking tends to cause oscillations in both variables. Therefore, based on mathematical models of the spinning air-conditioning system, decouplers for temperature and humidity control loops are designed using the diagonal matrix decoupling method. This decoupling method incorporates information entropy theory to enhance the performance of the temperature and humidity control. An entropy index-based evaluation method is proposed to assess the control performance of the air-conditioning temperature and humidity. Simulation studies on the decoupling control algorithm, integrated with the proposed automatic control strategy, are conducted using the operational data from a spinning workshop. The results indicate that compared to the conventional proportional-integral control (CPIC) method, the diagonal matrix decoupling control with information entropy (DMDCIE) completely eliminates the coupling of the temperature and humidity. It can keep the workshop temperature consistently stable at the set-point. The entropy indices of the temperature and humidity under DMDCIE are approximately 50 % lower than those under CPIC, indicating that DMDCIE can significantly improve the robustness of the temperature and humidity control.

1. Introduction

Yarn is the raw material for textile products, produced from the cotton through a series of manufacturing processes including cleaning, carding, combing, drawing, roving, spinning, and winding to create the primary product in the form of yarn bobbins. Different types of yarn products require different environmental temperature and humidity conditions during production. The constant temperature and humidity workshop environment is essential for continuous production and product quality stability [1]. Additionally, the hygienic and comfortable working environment is a basic requirement for workers [2,3].

The spinning air-conditioning differs from the comfort air-conditioning used in residential and commercial buildings. The spinning air-conditioning typically employs spray chamber as the cooling source for cooling and humidification. However, the high coupling of the temperature and humidity leads to interfere with control loops [4]. The spinning workshop's large space results in significant thermal inertia and time delay, and the inherent limitations of PI control often cause oscillations of the temperature and humidity [5].

Additionally, changes in the set-points of the temperature and humidity due to different types of yarns, operating rates, external envi-

ronmental temperatures, and other disturbances challenge the disturbance rejection capability of the automatic control system. Ensuring the constant temperature and humidity environment in the workshop is a significant challenge for the process air-conditioning system.

Some modern control systems occasionally incorporate simple feedforward controllers, such as feedforward compensation [6], gain scheduling [7], and cascade control [8], to suppress external disturbances. However, Decoupling control techniques are still required to eliminate the mutual influence between control loops. The decoupling control methods for comfort air-conditioning mainly involve model-based approaches and data-driven intelligent control strategies. In some cases, mathematical models of air-conditioning equipment, control units, and actuators can be established [9], alternatively, system identification techniques are employed to derive models from the input-output data [10,11]. These models facilitate the application of decoupling methods to eliminate coupling effects among controlled variables, thereby achieving the independent control of the temperature, humidity, and other process parameters. Xia et al. [12] proposed the feedforward decoupling control scheme for a direct expansion air-conditioning unit, which effectively eliminated temperature and humidity coupling effects under variable speed conditions and achieved precise simultaneous control of both variables. Shen et al. [13] presented a novel decoupling control system design method for high-dimensional multivariable processes and established a stable, proper and causal ideal-diagonal decoupler. Simulation and test results for the temperature control of a

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Nomenclature

A	windward area of spray chamber (m^2)
C	specific heat capacity ($\text{kJ/kg}\cdot\text{K}$)
C_1, C_2, C_{in}, n	coefficient
D_1	temperature decoupling coefficient
D_2	humidity decoupling coefficient
E	heat exchange efficiency of spray chamber
f	frequency (Hz)
$G_{11}(s), G_{12}(s), G_{21}(s), G_{22}(s)$	transfer function
$H(\tau)$	information entropy
i	enthalpy (kJ/kg)
In	entropy index
L	volume flow (m^3/s)
m	mass flow (kg/s)
M	moisture content (kg/kg dry air)
$N_{11}(s), N_{12}(s), N_{21}(s), N_{22}(s)$	decoupler
$p(\tau)$	relative change rate of relative humidity
Q	cooling load (kW)
s	complex variable
t	temperature ($^{\circ}\text{C}$)
t_1	air temperature of spray chamber inlet ($^{\circ}\text{C}$)
V	volume (m^3)
$\Delta u_t, \Delta u_h$	adjustment output of temperature and humidity control loops
$\Delta t, \Delta \phi$	temperature difference ($^{\circ}\text{C}$) and relative humidity difference (%)
Greek letters	
ρ	air density (kg/m^3)
θ	damper opening
τ	time (sec)
ω	thermal rise factor
Subscripts	
a	air
eq	spinning equipment
F	workshop
Fa	fan
h, h_2	primary return air and the secondary return air
hc	resurgence
lim	lower limit
p	exhaust air
pu	water pump
s_1	wet bulb
sc	spray chamber outlet
w	enclosure
wa	water
x	fresh air

four-room HVAC process showed that the control system performance was satisfactory. Mei et al. [14] proposed a state feedback decoupling approach for the nonlinear control of direct expansion air-conditioning systems to regulate indoor air temperature and humidity. Chen et al. [15] proposed an analytic decoupling proportional-integral-derivative (PID) control method for the coupled loops of supply air temperature and indoor temperature in variable air volume air-conditioning systems, effectively eliminating the coupling effects. However, these decoupling methods require precise mathematical models that are often difficult to establish and inevitably involve simplifications. When simplified model errors are tolerable, more robust approaches like the sliding mode or

internal model decoupling control can be adopted [16,17]. For multi-variable coupled systems, the decoupling performance may degrade due to external disturbances, time-varying dynamics, or model inaccuracies. The adaptive decoupling control with the online parameter identification can effectively address these time-varying system characteristics [18,19]. In contrast, the active disturbance rejection decoupling control does not require a precise mathematical model. This approach demonstrates the strong adaptability to parameter variations, excellent disturbance rejection capability, and superior control performance [20,21]. When accurate mathematical models of complex control systems are difficult to establish, the data-driven intelligent control methods can learn control laws from historical operational data and adaptively adjust control strategies through online learning. These approaches can autonomously accommodate variations in system parameters and coupling relationships, effectively addressing nonlinearity, time-varying dynamics and strong coupling effects in control systems [22–24].

The decoupling control approaches of spinning air-conditioning can benefit from existing studies on the temperature and humidity decoupling in comfort air-conditioning. The distinctive characteristic of spinning air-conditioning is the use of spray chamber as the cooling source. Since no universal mathematical model currently exists for spray chamber, the limited literatures on the temperature and humidity decoupling control exclusively rely on data-driven intelligent methods [25,26]. Spinning air-conditioning automatic control systems typically utilize programmable logic controllers (PLCs), which are constrained by limited memory capacity and restricted mathematical computation capabilities [27]. For spinning air-conditioning systems without the host computer support, implementing complex decoupling control algorithms or intelligent control methods in PLCs is highly challenging. Even simple temperature and humidity decoupling approaches, such as transfer function-based decoupling control algorithms, are difficult to implement on the PLC platforms typically used in spinning air-conditioning. Furthermore, existing studies predominantly focus on the temperature and humidity decoupling alone, without integration with the dedicated automatic control strategy. The stable temperature and humidity control of air-conditioning system requires combination with specific automatic control strategy [28,29].

This paper incorporates spray chamber into the mathematical models of the spinning air-conditioning system, establishing the two-input and two-output relationships between workshop temperature and fresh air damper opening, as well as workshop humidity and spray chamber efficiency, along with two PI control loops. To ensure PLC compatibility, a diagonal matrix decoupling method is adopted to design separate temperature and humidity decouplers. The transfer function-based decoupling algorithm is converted through inverse Laplace transformation into exponential forms suitable for PLC implementation. The temperature and humidity decoupling coefficients are derived from system parameters and integrated with the respective control loop outputs to regulate the damper opening and heat exchange efficiency of spray chamber. To compensate for model simplification errors, an information entropy method is proposed to quantify humidity fluctuation entropy, which is subsequently embedded into the humidity control loop to minimize regulation deviations and enhance control precision. Furthermore, this paper proposes an entropy index to quantitatively evaluate temperature and humidity regulation effectiveness throughout the operational period.

2. Spinning air-conditioning system

The spinning air-conditioning system of a spun yarn workshop is illustrated in Fig. 1. Due to the significant heat generated by spinning equipment and the moisture absorption characteristics of the yarn, the supply air must undergo both cooling and humidification processes. After entering the workshop and participating in heat and moisture exchange with indoor air, a portion of the air is exhausted outdoors through an exhaust damper with a specified opening. To maintain air

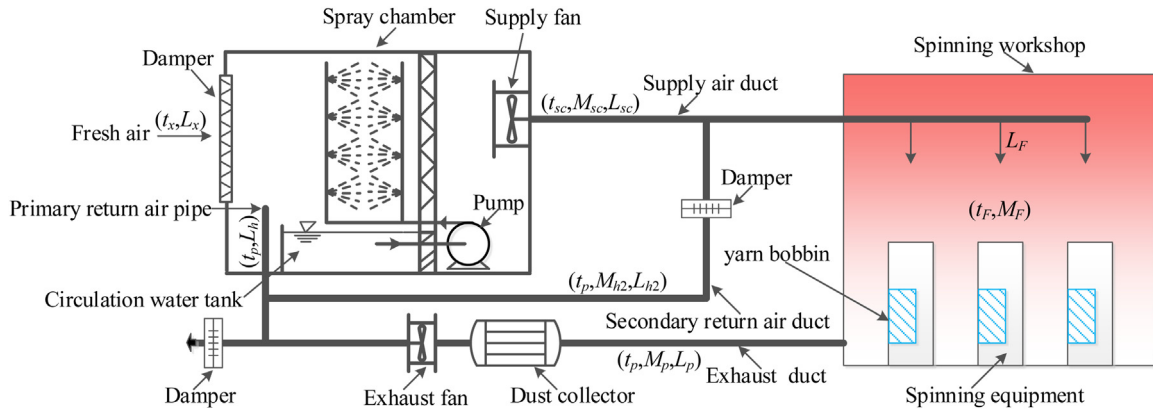


Fig. 1. Schematic diagram of spinning air conditioning system.

volume balance within the workshop, the openings of the fresh air damper and the exhaust damper are set to be equal.

The total supply air volume L_F of the workshop is slightly greater than the exhaust air volume L_p , ensuring a slight positive pressure is maintained within the space. The remaining air is split into two paths, one of which is directly supplied to the workshop as the secondary return air L_{h2} , while the other is directed to spray chamber as the primary return air L_h , where it mixes with fresh air L_x .

In the transitional season, the cooling capacity of the fresh air is fully utilized for temperature control. A high supply air volume, corresponding to an air exchange rate of 20–30 times per hour, is introduced into the spinning workshop. The water in the circulation water tank is atomized through centrifugal nozzles. After undergoing heat and mass exchange between the air and water droplets, the humidified low-temperature air passes through a serpentine baffle to remove entrained water, and is then conveyed to the workshop by a supply fan through ventilation ducts.

3. Mathematical models

Mathematical models of the spinning air-conditioning system are established using a white-box modeling approach, derived from fundamental physical laws. However, the air-conditioned space constitutes a complex thermal system and it is nearly impossible to describe using an exact mathematical model or to obtain precise system parameters. As a result, based on the previously established mathematical models [30], the system is simplified as follows:

- The thermal capacities of doors, windows, and indoor objects are neglected. The entire workshop is treated as a single-volume thermal object, and the individual heat storage capacities of internal objects are ignored.
- Internal airflow within the workshop is not considered.
- The temperature inside the workshop is assumed to be uniformly distributed. The workshop is considered to be well-sealed, with no direct heat exchange occurring between the indoor and outdoor environment.
- The moisture generated by the personnel inside the workshop is neglected.

Other assumptions and constants used in the models are summarized in Table 1.

Based on the principle of energy conservation, the temperature balance equation is formulated as Eq. (1):

$$\rho_F C_F V_F \frac{dt_F}{d\tau} + \frac{1}{\omega} \rho_w C_w V_w \frac{dt_F}{d\tau} = \rho_{sc} C_{sc} L_{sc} t_{sc} + Q_w + Q_{eq} + \rho_{h2} C_{h2} L_{h2} t_p - \rho_p C_p L_p t_p \quad (1)$$

Table 1

Assumptions and constants.

Assumptions	constants
$C_F = C_{sc} = C_{h2} = C_p$	1.0 kJ/(kg·K)
$\rho_F = \rho_{sc} = \rho_{h2} = \rho_p$	1.2 kg/m ³
$L_F = L_p$	/

The relationship between t_p and t_F can be approximated as $t_p \approx t_F + 4$ °C. Based on this approximation, Eq. (1) can be reformulated as:

$$(1.2V_F + \frac{1}{\omega} \rho_w C_w V_w) \frac{dt_F}{d\tau} + 1.2(L_p - L_{h2})t_F = 1.2L_{sc}t_{sc} + Q_w + Q_{eq} + 4.8L_{h2} - 4.8L_p \quad (2)$$

By incorporating the heat exchange efficiency formula of spray chamber, $E = \frac{t_1 - t_{sc}}{t_1 - t_{s1}}$, the following conclusion can be drawn.

$$t_{sc} = (1 - E)t_1 + Et_{s1} \quad (3)$$

By considering the pre-mixing of fresh air and the primary return air before spray chamber, and applying the energy equation under the assumptions of the constant specific heat capacity and air density, the following expression can be derived.

$$L_x t_x + L_h(t_F + 4) = (L_x + L_h)t_1 \quad (4)$$

$$t_1 = \frac{L_h}{L_x + L_h} t_F + \frac{L_x t_x + 4L_h}{L_x + L_h} \quad (5)$$

By combining with Eq. (3), the following expression can be derived.

$$t_{sc} = (1 - E) \frac{L_h}{L_x + L_h} t_F + (1 - E) \frac{L_x t_x + 4L_h}{L_x + L_h} + Et_{s1} \quad (6)$$

Assuming a linear relationship between the airflow volume and the damper opening, it follows that:

$$L_{h2} = 0.6L_F \theta_{h2} \quad (7)$$

According to the air volume balance, the total air volume L_F supplied to the workshop minus the secondary return air volume equals the air volume entering the spray chamber, as expressed by:

$$L_x + L_h = L_F - 0.6L_F \theta_{h2} = (1 - 0.6\theta_{h2})L_F \quad (8)$$

Let $\beta = 1 - 0.6\theta_{h2}$, Thus, the following expression is derived.

$$t_{sc} = (1 - E) \frac{\beta L_F - L_x}{\beta L_F} t_F + (1 - E) \frac{L_x t_x + 4\beta L_F - 4L_x}{\beta L_F} + Et_{s1} \quad (9)$$

By substituting Eq. (9) into Eq. (2), the following expression can be derived.

$$(1.2V_F + \frac{1}{\omega}\rho_w C_w V_w) \frac{dt_F}{d\tau} + 1.2\beta L_F t_F = 1.2\beta L_F(1-E)t_F - 1.2(1-E)L_x t_F + 1.2L_x t_x(1-E) + 4.8\beta L_F(1-E) - 4L_x(1-E) + 1.2L_{sc} E t_{s1} + Q_w + Q_{eq} + 4.8L_{h2} - 4.8L_{sc} \quad (10)$$

Reformatted as:

$$\frac{(1.2V_F + \frac{1}{\omega}\rho_w C_w V_w)}{1.2\beta L_F E} \frac{dt_F}{d\tau} + t_F = \frac{(1-E)(t_x - t_F - 4)}{\beta L_F E} L_x + \frac{4(1-E)}{E} + t_{s1} + \frac{Q_w + Q_{eq} + 4.8L_{h2} - 4.8L_F}{1.2\beta L_F E} \quad (11)$$

Let $L_x = \beta L_F \theta$, Eq. (11) can be derived as:

$$\frac{(V_F + \frac{1}{1.2\omega}\rho_w C_w V_w)}{\beta L_F E} \frac{dt_F}{d\tau} + t_F = \frac{(1-E)(t_x - t_F - 4)}{E} \theta + \frac{4(1-E)}{E} + t_{s1} + \frac{Q_w + Q_{eq} + 4.8L_{h2} - 4.8L_F}{1.2\beta L_F E} \quad (12)$$

The transfer function G_{11} of t_F with respect to θ can be expressed as:

$$G_{11}(s) = \frac{K_{11}}{T_{11}s + 1} \quad (13)$$

Where, $K_{11} = \frac{(1-E)(t_x - t_F - 4)}{E}$, $T_{11} = \frac{V_F + \frac{1}{1.2\omega}\rho_w C_w V_w}{\beta L_F E}$.

By transforming the temperature balance equation Eq. (10), the relationship between t_F and E can be derived.

$$\frac{V_F + \frac{1}{1.2\omega}\rho_w C_w V_w}{L_x} \frac{dt_F}{d\tau} + t_F = \frac{L_x(t_F - t_x + 4) + \beta L_F(t_{s1} - t_F - 4)}{L_x} E + t_x - 4 + \frac{4(\beta L_F + L_{h2} - L_F)}{L_x} + \frac{Q_w + Q_{eq}}{1.2L_x} \quad (14)$$

Thus, the transfer function of t_F with respect to E is expressed as:

$$G_{12}(s) = \frac{K_{12}}{T_{12}s + 1} \quad (15)$$

Where, $K_{12} = t_F + 4 - t_x + \frac{1}{\theta}(t_{s1} - t_F - 4)$, $T_{12} = \frac{V_F + \frac{1}{1.2\omega}\rho_w C_w V_w}{L_x}$.

The moisture balance equation of the spinning workshop can be formulated as:

$$\rho_F V_F \frac{dM_F}{d\tau} = \rho_{sc} L_{sc} M_{sc} + \rho_{h2} L_{h2} M_{h2} - \rho_p L_p M_p - M_{hc} \quad (16)$$

The air treatment in spray room is an adiabatic humidification process and the enthalpy remains constant before and after spraying. The enthalpy can be expressed by the following formula:

$$i = \frac{L_x i_x + L_h i_h}{(L_x + L_h)} = 1.01t_{sc} + (2500 + 1.84t_{sc})M_{sc} \quad (17)$$

Thus, the calculation formula for the moisture content of the supply air is derived as:

$$M_{sc} = \frac{\frac{L_x i_x + (\beta L_F - L_x) i_h}{\beta L_F} - 1.01t_{sc}}{2500 + 1.84t_{sc}} = \frac{L_x i_x + (\beta L_F - L_x) i_h}{2537\beta L_F} - \frac{t_{sc}}{2537} \quad (18)$$

Where, $1.84t_{sc} \approx 1.84 \times 20 = 37$.

By incorporating Eq. (18), the moisture balance equation Eq. (16) can be reformulated as:

$$V_F \frac{dM_F}{d\tau} + \beta L_F M_F = \frac{L_x}{2537} [i_x - i_h + (1-E)t_F - (1-E)t_x + 4(1-E)] + \frac{\beta L_F [i_h - (1-E)t_F - 4(1-E) - Et_{s1}]}{2537} - \frac{M_{hc}}{1.2} \quad (19)$$

$$\frac{1}{2537} [i_h - Et_{s1} - (1-E)(t_F + 4)] - \frac{M_{hc}}{1.2} \quad (20)$$

Table 2

Detailed parameters of the spinning workshop.

Parameters	Value
V_F (m ³)	14742
V_w (m ³)	201.6
ρ_w (kg/m ³)	2500
C_w (kJ/kg·K)	0.93
ω	40
Q_{eq} (kW)	270-300
Q_w (kW)	$3(t_x - t_F)$
M_{hc} (kg)	0.05

The transfer function G_{21} of M_F with respect to θ can be expressed as:

$$G_{21}(s) = \frac{K_{21}}{T_{21}s + 1} \quad (21)$$

Where, $K_{21} = \frac{(1-E)(t_F + 4 - t_x) + i_x - i_h}{2537}$, $T_{21} = \frac{V_F}{\beta L_F}$.

The moisture balance equation can be transformed into an equation of M_F with respect to E .

$$\frac{V_F}{\beta L_F} \frac{dM_F}{dt} + M_F = \frac{E}{2537} [(t_F + 4 - t_{s1}) + \theta(t_x - t_F - 4)] + \frac{1}{2537} (i_h - t_F - 4) + \frac{\theta}{2537} (i_x - i_h + t_F - t_x + 4) - \frac{M_{hc}}{1.2\beta L_{sc}}$$

The transfer function G_{22} of M_F with respect to E is given by:

$$G_{22}(s) = \frac{K_{22}}{T_{22}s + 1} \quad (23)$$

Where, $K_{22} = \frac{t_F + 4 - t_{s1} + \theta(t_x - t_F - 4)}{2537}$, $T_{22} = \frac{V_F}{\beta L_F}$.

The parameters of the spinning workshop in the transfer function are listed in Table 2.

4. Decoupling method

Small PLCs, such as the S7-200SMART, are widely used in the automatic control systems of spinning air-conditioning units, while upper computers are rarely employed. As a result, the decoupling method that rely on complex algorithm is not well suited for spinning air-conditioning system that only utilize an industrial control computer. Based on the established mathematical models, the simple diagonal matrix decoupling control approach is adopted, as illustrated in Fig. 2.

According to the requirements for the diagonal matrix decoupling, it follows that:

$$\begin{bmatrix} G_{11}(s) & G_{12}(s) \\ G_{21}(s) & G_{22}(s) \end{bmatrix} \begin{bmatrix} N_{11}(s) & N_{12}(s) \\ N_{21}(s) & N_{22}(s) \end{bmatrix} = \begin{bmatrix} G_{11}(s) & 0 \\ 0 & G_{22}(s) \end{bmatrix} \quad (24)$$

Thus, the decouplers can be expressed as:

$$\begin{bmatrix} N_{11}(s) & N_{12}(s) \\ N_{21}(s) & N_{22}(s) \end{bmatrix} = \begin{bmatrix} G_{11}(s) & G_{12}(s) \\ G_{21}(s) & G_{22}(s) \end{bmatrix}^{-1} \begin{bmatrix} G_{11}(s) & 0 \\ 0 & G_{22}(s) \end{bmatrix} \quad (25)$$

$$= \begin{bmatrix} \frac{G_{22}(s)G_{11}}{C_{in}} & \frac{-G_{12}(s)G_{22}}{C_{in}} \\ \frac{-G_{21}(s)G_{11}}{C_{in}} & \frac{G_{11}(s)G_{22}}{C_{in}} \end{bmatrix} \quad (25)$$

Where, $C_{in} = G_{11}(s)G_{22}(s) - G_{12}(s)G_{21}(s)$. Therefore, the temperature decoupling coefficient is $D_1 = G_{21}(s)N_{12}(s) + G_{22}(s)N_{22}(s)$, and the humidity decoupling coefficient is $D_2 = G_{11}(s)N_{11}(s) + G_{12}(s)N_{21}(s)$. The transfer functions are subjected to the inverse Laplace transform to facilitate their integration with industrial PLCs, that is:

$$G_{11} = \frac{K_{11}}{T_{11}} e^{-1/T_{11}} \quad (26)$$

$$G_{12} = \frac{K_{12}}{T_{12}} e^{-1/T_{12}} \quad (27)$$

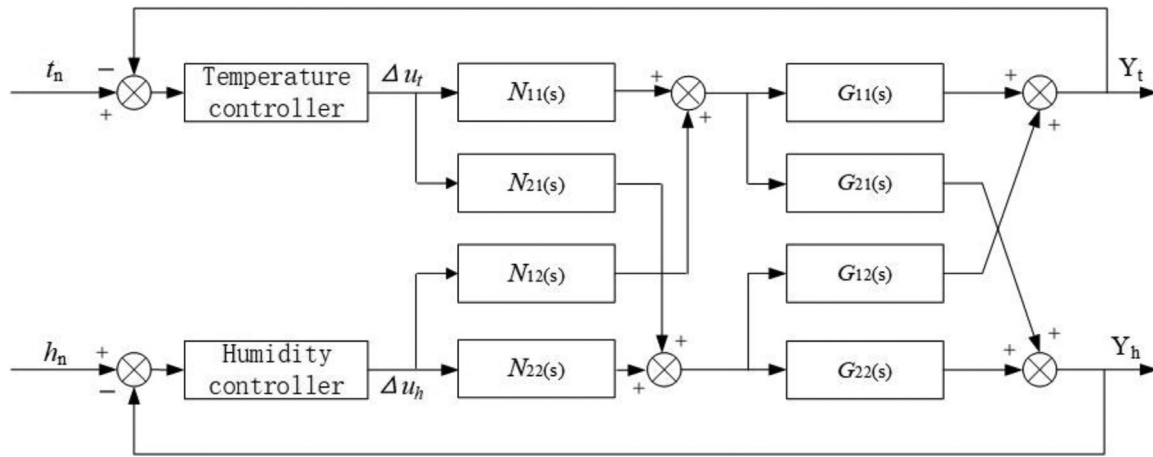


Fig. 2. Schematic diagram of the diagonal matrix decoupling control.

$$G_{21} = \frac{K_{21}}{T_{21}} e^{-1/T_{21}} \quad (28)$$

$$G_{22} = \frac{K_{22}}{T_{22}} e^{-1/T_{22}} \quad (29)$$

Where, $K_{11} = \frac{(1-E)(t_x-30)}{E}$, $T_{11} = \frac{14742 + \frac{2500 \times 0.93 \times 201.6}{1.2 \times 40}}{33E} = \frac{742.64}{E}$,
 $K_{12} = \frac{1}{\theta}(t_{s1} - 30) + (30 - t_x)$, $T_{12} = \frac{14742 + \frac{2500 \times 0.93 \times 201.6}{1.2 \times 40}}{L_x} = \frac{24507}{L_x}$,
 $K_{21} = \frac{(t_x - t_h) + (1-E)(4+t_F - t_x)}{2537}$, $T_{21} = T_{22} = \frac{V_F}{L_s} = \frac{14742}{33} = 446.73$, $K_{22} = \frac{(t_x - 4 - t_F)\theta + (4 + t_F - t_{s1})}{2537}$.

The variables in the equations are derived from real-time temperature and humidity values, feedback signals of the fresh air damper opening, and intermediate system quantities, all of which exhibit time-varying characteristics. The adjustment output from the temperature control loop, denoted as Δu_t , is combined with D_1 (i.e., $\Delta u_t + D_1$) to regulate the fresh air damper opening. Similarly, the adjustment output from the humidity control loop, Δu_h , is combined with D_2 (i.e., $\Delta u_h + D_2$) to regulate the heat exchange efficiency of spray chamber.

Based on the empirical formula $E = 0.75(\frac{m_a}{A})^{C_1}(\frac{m_{wa}}{m_a})^{C_2}$, the pump flow rate m_{wa} is calculated. Considering that the spray chamber uses a dual-row counter-spray method with Y1-shaped centrifugal nozzles, the coefficients C_1 and C_2 are set to 0.15 and 0.29 [31], respectively. On the basis of the water flow rate, the operating frequency of the pump can be calculated.

5. Entropy regulation coefficient

The temperature and humidity control of spinning air-conditioning is subject to various stochastic disturbances, including meteorological variation, operational load fluctuation of spinning equipment, and moisture content of cotton fibers. These random factors can cause deviations from the predetermined set-points. As a passive control strategy with inherent coupling between temperature and humidity, conventional PID control often leads to unstable regulation and oscillatory behavior in both temperature and humidity under such disturbances.

Given the inherent uncertainty in temperature and humidity variations, larger deviation from the set-point results in reduced control stability. To address this issue, information entropy can be employed as a quantitative measure to characterize the control stability of temperature and humidity. The information entropy is mathematically defined as follows:

$$H(\tau) = -|p(\tau)| \cdot \log |p(\tau)| \quad (30)$$

During the regulation of the temperature and humidity in the spinning workshop, the relative humidity tends to exhibit the oscillatory

behavior, making it more challenging to control precisely. In contrast, the workshop temperature remains relatively stable and is less responsive to control actions. Due to the simplification of the spinning air-conditioning models, the complete decoupling between the temperature and humidity cannot be achieved. As a result, information entropy is calculated exclusively for the relative humidity and is integrated with the diagonal matrix decoupling control to optimize its regulation. Accordingly, the control output for the humidity regulation can be expressed as:

$$S_\varphi = \Delta u_\varphi + D_2 + nH(\tau) \quad (31)$$

Where, n is a system-dependent coefficient that varies depending on the specific spinning air-conditioning system. It is introduced to normalize the order of magnitude of the information entropy value so that it can align with that of the decoupler. In the simulation presented in this study, the value of n is determined as follows:

- When the absolute deviation between the relative humidity and its set-point exceeds 2 %, $n = 1/30$.
- When the deviation falls between 1 % and 2 %, $n = 1/200$.
- When the deviation is less than 1 %, $n = 1/500$.

6. Control strategy

The PI control algorithm is implemented to establish two independent control loops of the temperature and humidity. The temperature control is achieved by adjusting the opening of the fresh air damper, which changes the flow rate of fresh air with cooling capacity to offset the incremental heat load within the workshop. This approach enables the automatic temperature control based on the deviation between the set-point and the measured workshop temperature.

In the yarn production process, the humidity control is critical and follows a principle that prioritizes precise humidity regulation while maintaining energy efficiency. When the relative humidity in the workshop exceeds the set-point (i.e., $\Delta\varphi < 0$), a sequence control strategy is applied under energy conservation principles. This involves first reducing the power of high-energy-consuming equipment (e.g., fan), followed by less energy-intensive equipment (e.g., pump). Specifically, the control sequences are as follows:

- (1) Decrease the frequency of the fan to enhance the spray chamber's heat exchange efficiency.
- (2) Reduce the frequency of the water pump to lower the spray chamber's heat exchange efficiency.
- (3) Increase the opening of the secondary return air damper, as illustrated in Fig. 3.

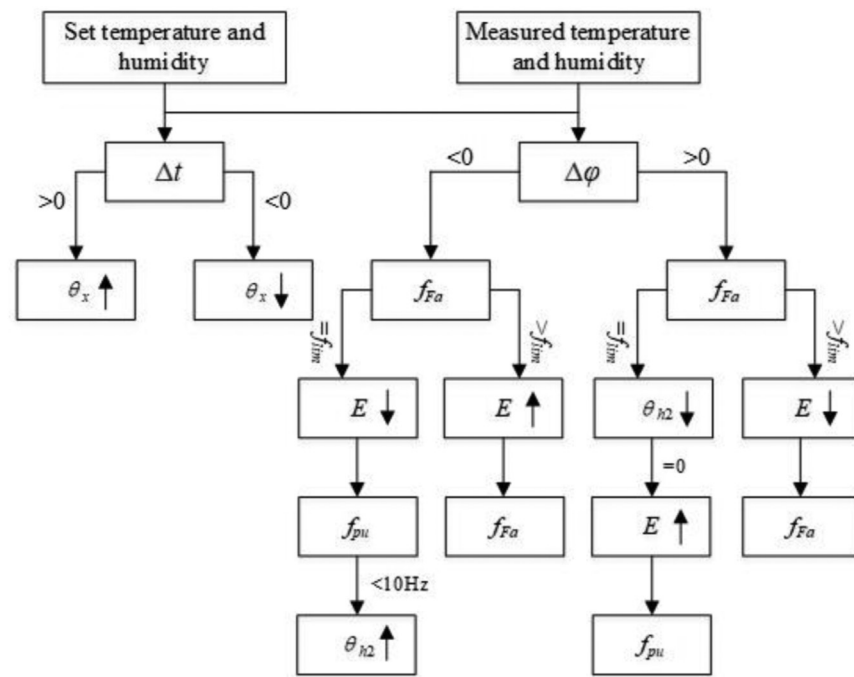


Fig. 3. Schematic diagram of the automatic control strategy.

Conversely, when the relative humidity falls below the set point (i.e., $\Delta\phi > 0$), adjustments are made in ascending the order of energy consumption:

- (1) Decrease the opening of the secondary return air damper.
- (2) Increase the pump frequency to adjust the spray chamber's heat exchange efficiency.
- (3) Raise the fan frequency to further optimize the spray chamber's heat exchange efficiency.

This control strategy ensures precise regulation of both temperature and humidity while minimizing energy consumption, thereby maintaining optimal operational conditions of spinning air-conditioning.

7. Results

7.1. Control performance

The temperature and humidity decoupling control was simulated using MATLAB, based on the incremental PI control algorithm. The initial condition parameters are listed in Table 3. The temperature and humidity response curves of spinning air-conditioning under CPIC, diagonal matrix decoupling control (DMDC), and DMDCIE are shown in Fig. 4. Under CPIC, the strong coupling effect between the temperature and humidity is clearly observed, with both response curves exhibiting simultaneous fluctuations. Although the temperature fluctuates within a relatively narrow range, the relative humidity shows significantly larger oscillations. Notably, during the transition phase from the initial conditions to the target set-points and particularly when heat release disturbances increased from 270 kW to 300 kW between 6000 s and 12000 s,

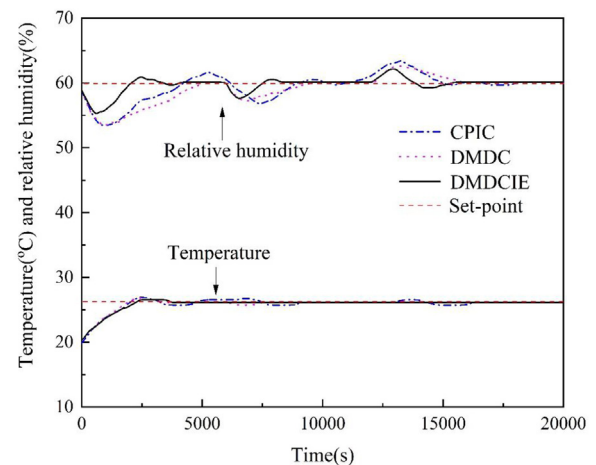


Fig. 4. Temperature and humidity response curves under CPIC, DMDC, and DMDCIE.

dramatic fluctuations of relative humidity are observed. The relative humidity exhibits an overshoot of 10.8 %, with the longest settling time recorded as 2.4 h.

The application of DMDC effectively weakens the coupling between the temperature and humidity. Although minor fluctuations in the relative humidity are still observed during the transition phase, the temperature control remains robust and unaffected by humidity variations. Furthermore, under heat disturbance conditions, the system demonstrates the improved dynamic performance, with a notable reduction in the relative humidity overshoot and a shortened settling time of 1.3 h.

The DMDCIE method demonstrates the highly steady performance compared to both DMDC and CPIC in regulating the temperature and humidity. It not only significantly reduces the fluctuation amplitude of the relative humidity but also eliminates the coupling between the temperature and humidity. After transition from the initial state to the set-point, the workshop temperature remains consistently stable at the set-point. The relative humidity exhibits an 8.2 % overshoot with 0.54 h

Table 3

Initial conditions.

Parameters	Value
t_F (°C)	20
M_F (kg/kg dry air)	0.0086
t_x (°C)	15
M_x (kg/kg dry air)	0.0063
Q_{eq} (kW)	270

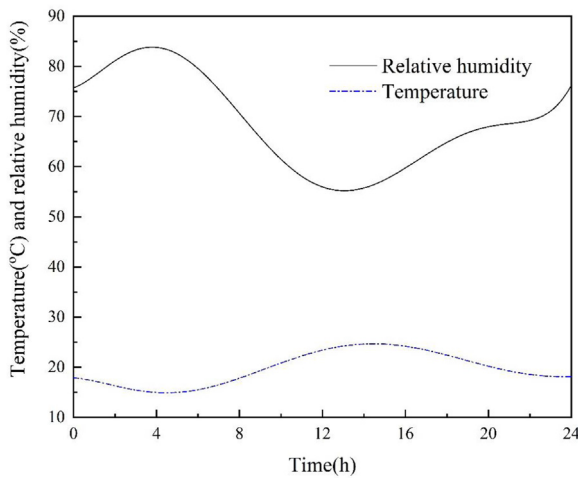


Fig. 5. Temperature and humidity variations of fresh air.

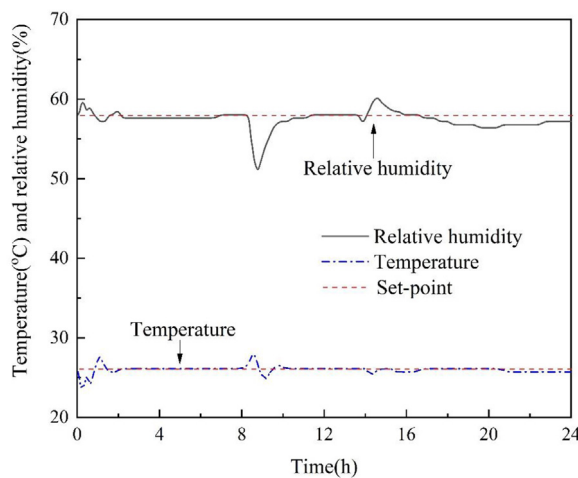


Fig. 6. Temperature and humidity variations under CPIC.

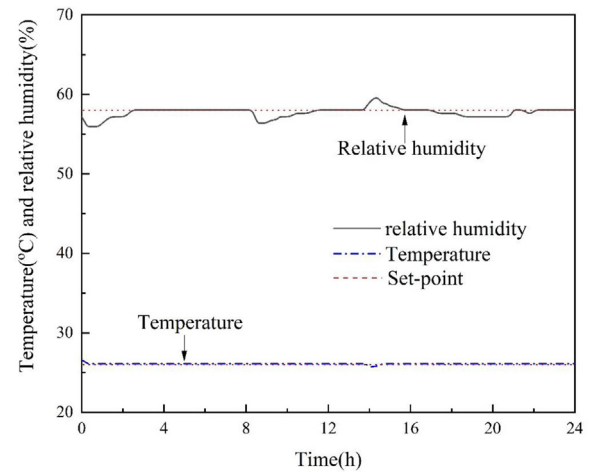


Fig. 7. Temperature and humidity response curves under DMDCIE.

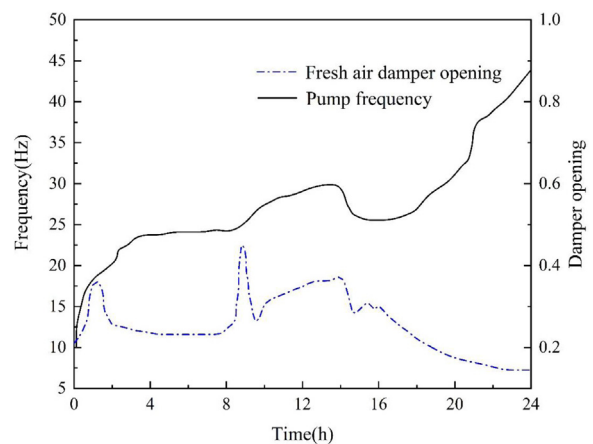


Fig. 8. Response curves of fresh air damper opening and pump frequency under CPIC.

settling time. Compared to CPIC and DMDC, the overshoot of the relative humidity is reduced by 24.1 %. It indicates that DMDC can effectively weaken or even eliminate the coupling between the temperature and humidity. Furthermore, the incorporation of information entropy into DMDC (i.e., DMDCIE) not only achieves the complete decoupling of the two variables but also significantly improves the accuracy and stability of the relative humidity control.

7.2. Temperature and relative humidity of 24-hour

To further evaluate the performance of DMDCIE in the temperature and humidity regulation, a simulation was conducted under conditions involving machine load heat release disturbances ranging from 280 to 300 kW and hourly variations in fresh air temperature and humidity over a 24-hour period. The initial workshop temperature and humidity are 26 °C and 58 %, respectively. The variations of fresh air are depicted in Fig. 5, while the response curves of the temperature and relative humidity under CPIC and DMDCIE are presented in Figs. 6 and 7, respectively.

As shown in Fig. 6, despite the hourly fluctuations in the fresh air temperature and humidity, the relative humidity under CPIC undergoes significant adjustments in response to machine load heat release disturbances. Specifically, when the machine load increases from 280 kW to 300 kW, the relative humidity drops to 51.0 %, accompanied by a rise in temperature. Eventually, the relative humidity stabilizes at the

set-point. Conversely, when the machine load decreases from 300 kW to 280 kW, the relative humidity initially increases and then declines. However, due to the influence of fresh air variations, the workshop relative humidity fails to reach the set-point within the simulation time period, and the workshop temperature fluctuates in tandem with the changes of the relative humidity. These observations highlight the limitations of CPIC in maintaining stable workshop conditions under dynamic external and internal disturbances.

When DMDCIE is applied, the workshop temperature remains consistently stable at the set-point of 26 °C, demonstrating the complete elimination of the coupling between the temperature and humidity. The fluctuation range of the relative humidity is significantly reduced, with the overshoot of only 3.4 %, representing the 71.9 % improvement compared to CPIC. Moreover, under heat disturbances caused by the increased machine load, the system achieves the transition time of just 2.1 h, which is the 67.2 % reduction compared to that under CPIC. These results clearly demonstrate the superior performance of DMDCIE in maintaining the stable temperature and humidity control under dynamic operating conditions.

7.3. Fresh air damper and pump

The response curves of the fresh air damper opening and pump frequency under the two control methods are shown in Figs. 8 and 9. Since the fan frequency (25 Hz) and the secondary return air damper opening (0°) remained constant throughout the 24-hour simulation, their corre-

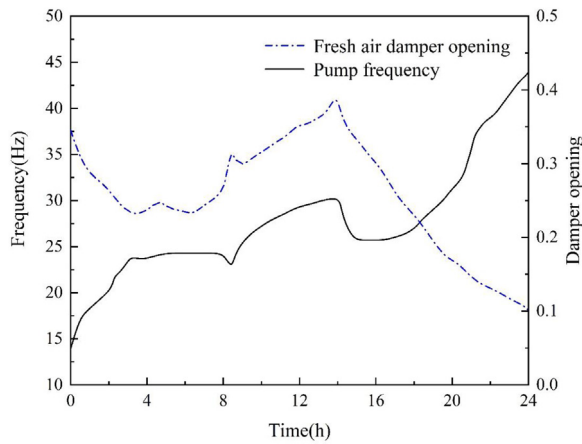


Fig. 9. Response curves of fresh air damper opening and pump frequency under DMDCIE.

sponding curves are not included in the figures. As can be seen from the plots, the trends of the pump frequency response curves are generally similar under both control strategies. However, notable differences are observed in the response curves of the fresh air damper opening under the same operating conditions. Under DMDCIE, the fresh air damper opening is significantly larger compared to that under CPIC. This difference can be attributed to the integrated temperature regulation mechanism in DMDCIE, which actively adjusts the fresh air damper opening to compensate for humidity fluctuations, thereby influencing the humidity control. In other words, DMDCIE employs a proactive control strategy that takes into account the dynamic interaction between the damper opening and outdoor air conditions. This approach not only assesses their combined impacts on the temperature and humidity but also actively compensates for humidity fluctuations resulting from damper adjustments.

In contrast, CPIC functions as a passive control method that does not consider the coupling between the temperature and humidity regulation, leading to the relatively lower control accuracy compared to DMDCIE. By dynamically adjusting the fresh air damper opening to balance humidity levels, DMDCIE maintains a higher damper opening than CPIC, further demonstrating its superior capability in achieving precise and stable environmental control.

The energy consumption of the pump over a 24-hour period is calculated for both control methods by integrating their respective pump frequency response curves. Under the CPIC method, the energy consumption is 137.5 kWh, whereas under DMDCIE, it is 139.1 kWh. This indicates that the overall energy consumption of the two approaches is nearly identical, with DMDCIE exhibiting only a slightly higher consumption.

7.4. Control performance assessment

The control performance of the temperature and humidity in spinning air-conditioning is evaluated using an entropy index, with a 24-hour operational period serving as the assessment window. Temperature and humidity levels within the workshop are subject to various disturbances, such as random fluctuations in outdoor fresh air conditions, variations in machine load, and changes of temperature and humidity set-points. As a result, the behavior of these environmental parameters inherently contains a degree of uncertainty.

The primary objective of the automatic control system is to regulate the temperature and humidity toward their respective set-points. If the system maintains high precision and stability over a 24-hour period, the resulting entropy index will be small. This indicates that the regulation of the temperature and humidity is more deterministic, reflecting the effective and consistent control performance.

Conversely, a larger entropy index suggests poor control stability, indicating significant deviations of the temperature and humidity from their set-points or frequent fluctuations around them.

An entropy index, denoted as In , is employed to characterize this concept:

$$In = - \sum_{\tau=0}^{24} |p(\tau)| \cdot \log |p(\tau)| \quad (32)$$

A smaller value of In indicates better control performance, characterized by more stable and precise regulation of the temperature and humidity at their target set-points. Conversely, a larger In reflects poor control effectiveness, marked by reduced stability and greater deviations from the desired levels. To evaluate the control performance of the temperature and humidity under both the CPIC and DMDCIE methods over a 24-hour period, the following entropy indices were obtained: $In_h(DMDCIE)=4.8$, $In_t(DMDCIE)=2.3$, $In_h(CPIC)=10.6$, and $In_t(CPIC)=7.5$, where, In_h and In_t represent the entropy indices for the relative humidity and temperature, respectively. It is evident that the entropy indices under CPIC are more than twice those observed under DMDCIE. This indicates that DMDCIE demonstrates superior performance in regulating the temperature and humidity, achieving the higher control precision and enhanced stability. In summary, this evaluation approach quantifies the uncertainty in the temperature and humidity control over a 24-hour period using the entropy index. It enables a comprehensive assessment of the automatic control system's effectiveness in managing these environmental parameters, providing valuable insights into the overall control performance.

8. Conclusions

The temperature and humidity decoupling control method based on diagonal matrix and information entropy is proposed in this study. The temperature and humidity control performance under DMDCIE and CPIC is compared through simulation experiments implementing the proposed automatic control strategy. The entropy index is employed to quantitatively evaluate the performance of both control approaches.

The results show that the coupling between the temperature and humidity of spinning air-conditioning is completely eliminated under DMDCIE. The workshop temperature remains stably controlled at the set-point, unaffected by disturbances from fresh air and machine load variations, as well as fluctuations in relative humidity. Compared to CPIC, the fluctuation range of the relative humidity in the workshop is reduced, and the overshoot is decreased by 71.9 %, with the transition time shortened by 67.2 % under DMDCIE.

The entropy index is proposed to evaluate the control effectiveness of the temperature and relative humidity. The findings indicate that under DMDCIE, the daily entropy indices for the humidity and temperature are 4.8 and 2.3, respectively, which are about one half of those under CPIC. This shows that DMDCIE significantly enhances the precision and stability of the temperature and humidity control.

The proposed decoupling control method for the temperature and humidity is primarily focused on simulation studies. In the future, the energy consumption associated with fan and pump operation will be considered. This will involve long-term simulations under various operating conditions to investigate the impact of the decoupling control strategy on energy efficiency of spinning air-conditioning. Furthermore, the proposed decoupling control approach will be applied to the air-conditioning control system of an actual spinning plant. It is necessary to implement the simple decoupling algorithm and control strategy in the PLC. The necessary variables and parameters should be acquired online from sensor outputs and equipment feedback signals. The actual control performance can be assessed by monitoring the variations of the temperature and humidity tracking the set-points through the human-machine interface of the automation system. Additionally, the calculated entropy index will serve as a quantitative indicator for evaluating control stability.

Declaration of competing interest

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

CRediT authorship contribution statement

Yunlong Han: Writing – review & editing, Validation, Methodology, Conceptualization. **Biao Lu:** Writing – review & editing, Writing – original draft, Software, Methodology, Investigation. **Yongmei Hu:** Writing – original draft, Software. **Demin Chen:** Supervision, Resources, Conceptualization.

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