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RESEARCH ARTICLE

Modeling of Ventilation's Influence on Energy Consumption in Multi-cylinder Dryer Section Part1: Theoretical Model

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Abstract

Ventilation system affects the water evaporation rate, energy consumption and the paper quality, which is considered as the largest optimization potential part of multi-cylinder dryer section. Energy consumption of drying system is divided into three parts: steam consumption of cylinders, steam consumption of air heaters, power consumption of blowers and exhaust fans. An energy consumption model of drying system integrating steam condensation system and ventilation system is constructed according to the control logic and process flow of the actual ventilation system, by combining the established models of each module and energy consumption indexes. It is the premise to realize the intelligent control of paper drying process. The research results perfects the mechanism of ventilation system for assisted drying, which can promote the improvement of the monitoring, and enable intelligent computing technology to play a greater role in optimizing drying operation and reducing drying energy consumption.

Keywords Paper drying · Ventilation system · Energy saving · Energy consumption model · Simulation and optimization

List of Symbols

DCS	Distributed control system
QCS	Quality control system

Variable

A	Area, m^2
β	Paper width, m
c	Specific heat, kJ/(kg·°C)
D_{wa}	Diffusion coefficient of vapor in air, m^2/s
d	Paper dryness, %
$d.a$	Absolute dry air
$d.f$	Dry fiber
E	Electrical energy, kWh
F	Mass Flow, kg/h
G	Basic weight, kg/m^2
h	Heat transfer coefficient, W/(m^2 °C)
H_a	Absolute humidity of air, kg H_2O/kg d.a.
I	Enthalpy value, kJ/kg
I_0	Latent heat of vaporization at 0 °C, kJ/kg
k	Mass transfer coefficient, m/s
l	Length, m
m	Flow rate, kg/h
M	Molar mass, g/mol
N	Amount
Nu	Nusselt number
p	Electricity consumption, kWh
P	Pressure, Pa
P_0	Atmospheric pressure, Pa
Pr	Prandtl number
Q	Heat flow rate, kJ/s
R	Gas constant, J/(mol °C)
Re	Reynolds number
RH_a	Relative humidity of air, %
Sc	Schmidt number
Sh	Sherwood number
T	Temperature, °C
V	Volumetric flow, m^3/s
V_a	Diffusion volume of air molecule, cm^3/g mol
V_v	Diffusion volume of vapor molecule, cm^3/g mol
v_a	Air flow speed, m/s
v_m	Running speed of drying section, m/min
x_p	Paper moisture ratio, kg H_2O /kg ds
x_a	Relative humidity of air, %
Δh_v	Latent heat of moisture vaporization, kJ/kg

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Δh_s	Absorption Heat of paper, kJ/kg
Δh_{vap}	Latent heat of water vaporization, kJ/kg

Greek Alphabet

β	Air leakage ratio of hood, %
γ	Binding factor of water vapor and air
η_{fan}	Fan efficiency, %
η_{m}	Mechanical efficiency, %
λ_a	Coefficient of themal conductivity, W/(m^2·°C)
μ_a	Air viscosity, Pa·s
ρ_a	Air density, kg/m^3
Φ	The isotherm adsorption line equation

Subscript

ah	Air heater
a-p	Between air and paper
aw	Water vapor in air
c	Dryer
c-a	Air and dryer room
cond	Condensation
cont	Contact drying
conv	Convection drying
c-p	Between the dryer and the sheet
d	Dew point
ex	Exhaust
f	Fiber
fa	Fresh air
fan	Fan
h	Hood
hr	Heat recovery system
a	Air
i	Number of cylinder
j	Number of drying area
la	Air leakage
loss	Loss
p	Paper
pa	Pocket air
pw	Water vapor on paper surface
s	Steam
sa	Supply air
sao	Output of supply air
sai	Input of supply air
sat	Saturation state
ν	Vapor
w	Water

1 Introduction

China will strive to reach the peak of carbon dioxide emissions by 2030 and achieve carbon neutrality by 2060 as the Chinese government proposed at the 75th United Nations General Assembly [1]. In the past 20 years, China's paper industry has been developing rapidly, the yield of paper and paperboard reach to 117 million tons in 2020, accounting for 29.2% of the global yield [2]. As one of the most energy-intensive industries, paper industry consumed 6% of global industrial energy [3]. The study shows that China has the greatest potential for the carbon reduction in the world paper industry, and the total emission reduction potential is approximately 55% [3]. Energy consumption per unit product decreased by 23.6 and 14.9%, respectively, during the 12th and the 13th Five-Year Plan periods by analyzing the statistical data [4], and the 14th Five-Year Plan proposes that the energy consumption per unit GDP should decrease by 13.5% [5], which highlighted the extreme importance, urgency, and sustainability of energy conservation. To achieve the goals of carbon peak and carbon neutralization, paper enterprises need to reduce carbon emissions by optimizing energy structure and reducing energy consumption, in addition to increasing carbon sink throug afforestation and participating in carbon trading. So, as new technologies or upgraded production lines adopted, how to further reduce the energy consumption and improve the energy efficiency is an urgent problem for papermaking workers.

Drying section is the largest energy consumption part in the papermaking process, accounting for more than 67% of the total energy consumption [6, 7], and its energy efficiency has an important influence on the energy consumption of papermaking process. Ventilation system is used to remove the evaporated vapor around the sheet to maintain a high drying rate. Because ventilation system impacts the process of heat and mass transfer, it will affect the water evaporation rate, the energy consumption of paper drying, working environment and even the quality of paper [8, 9]. Theoretical research on the influence of ventilation on paper drying is coupled with the steam condensation system, because ventilation participates in mass and heat transfer. Considering steam condensate system takes the large proportion of energy consumption, it has always been the research hotspot. Many scholars, represented by Nilsson, Gaillemard, Karlsson and others of their universities, have modeled or simulated the paper drying process. The models of paper temperature are deduced based on the law of conservation of mass and energy, the mass transfer equation (paper moisture) based on Fick's law or Stefan equation. It can be seen that the ventilation system controls the key parameters (supply air volume, supply air temperature, exhaust humidity or exhaust dew point, etc.) to affect the air condition in the packet area, which influences the evaporation rate [10–12]. The heat and mass transfer theories above can well explain the evaporation of paper moisture, and have confirmed the influence of ventilation state on drying rate. However, they have not considered the condition of pocket area was affected by the ventilation system. Many scholars have analyzed the influence of ventilation parameters on drying operation and energy consumption. Songok el al. found that the drying rate mainly depends on the heat transfer on the paper surface, and transferring hot air to the drying surface effectively can improve the drying rate [13]. Kong [14] and Nejad [7] studied the effects of different operating parameters (input dryness and temperature, supply air temperature, exhaust humidity, etc.) on drying performance. The above research has confirmed the influence of ventilation parameters on the drying rate, but the mechanism is still unclear and the existing research has not considered the change of ventilation system's operation and energy consumption, which leads to the ventilation system operating far from the reasonable state in the actual production. Although ventilation system only accounts for 14.2% of the total drying energy consumption [15], it is also considered as the part has the largest optimization potential of paper drying process beacuse it directly affects heat and mass transferring [16]. Qin et al. analyzed the influence of exhaust humidity, flow rate and air temperature on heat transfer coefficient, and improve the waste heat recovery efficiency [17]; Gong et al. used orthogonal test method and CFD simulation to analysis air temperature and humidity distribution to solve the condensation problem of hood [18]. Holik found that the higher the exhaust humidity, the greater the hood balance rate and the lower the drying energy consumption on the premise of no condensation [19]. Tanasic et al. determined the optimal heat exchange area of the gas–gas heat exchanger based on the analysis of the potential of waste heat recovery [20]. Those researches focused on a part of ventilation system, although the energy efficiency of the part has been improved and relevant theory has been studied, the influence of ventilation on the steam consumption of steam condensation system has been neglected. Kong et al. found that the drying section had about 4.6% energy saving potential by reasonably adjusting the ventilation system for a corrugated paper machine [21]. Ghodbanan [22] and Li [23] et al. found the total steam consumption can be reduced about 11 and 5.6% by optimizing the operation conditions, respectively; Laurijssen et al. pointed out that heat consumption can be reduced by 32% by increasing the dew point temperature and further recovering the waste heat [24]. But high inlet air temperature will increase the steam consumption of the air heater, Sundqvist points out that increasing the supply air temperature from 80–90 °C to 120–130 °C, the steam consumption of the drying section will increase by 2% [25]. It can be considered that ventilation accounts for the small proportion of energy consumption but plays a very important role in reducing the energy consumption of drying system. The total energy consumption of steam condensate system integrating ventilation system has not been studied, and the improvement of local energy efficiency does not mean the overall energy consumption can be reduced.

Some research analyzed or optimized the drying process based on the intelligent computational technology. Due to the lack of mechanism model and online monitoring of some key variables, the optimization results can not involve the key influence variables. In view of this, to realize the final optimization of the drying process, this paper focuses on the derivation of the mechanism model to discover their mechanism of ventilation action and grasp the key indicators that are not monitored. To master the influence mechanism of key ventilation parameters on energy consumption of each part and overall energy consumption of drying system, the energy transfer model integrating steam condensation system and ventilation system should be established based on energy consumption indexes in the actual situation. It is the key to the integration the control of ventilation system and steam condensate system, and it is also the premise to realize the intelligent control of paper drying process, intelligent fault diagnosis with uncertainties and intelligent real-time industrial process monitoring and control. This research will perfect the theory of drying system, which can be used to guide the design and operation of dryer section, provide theoretical and technical basis for improving energy efficiency. Meanwhile, this study is the key to integrate the control of ventilation system and steam condensate system, and the premise to realize the intelligent control and the operation optimization of the whole drying process.

2 Methodologies

Take multi-dryer section as an example, considering it accounts for 85–90% of the total drying type [25]. The main works include the division and modeling of functional modules for key operation unit in drying system, the construction of energy consumption model for multi-cylinder dryer section, and the solution of complex models. Then, the influences of key ventilation parameters on each part and overall energy consumption of drying system are analyzed based on case study. Considering the energy consumption models of drying system integrating steam condensation system and ventilation system involve many variables, it is difficult to directly solve the complex models, so this study solved complex systems numerically with the help of intelligent computational technology. A Runge Kutta optimizer by searching the optimal point based on the concept of computational gradient search is used to solve the paper temperature and paper moisture ratio, which has the characteristics of strong optimization ability and fast convergence speed.

2.1 System Configuration of a Typical Ventilation System

Generally, from the perspective of energy utilization, drying section can be divided into steam condensate system and ventilation system [7], as shown in Fig. 1.

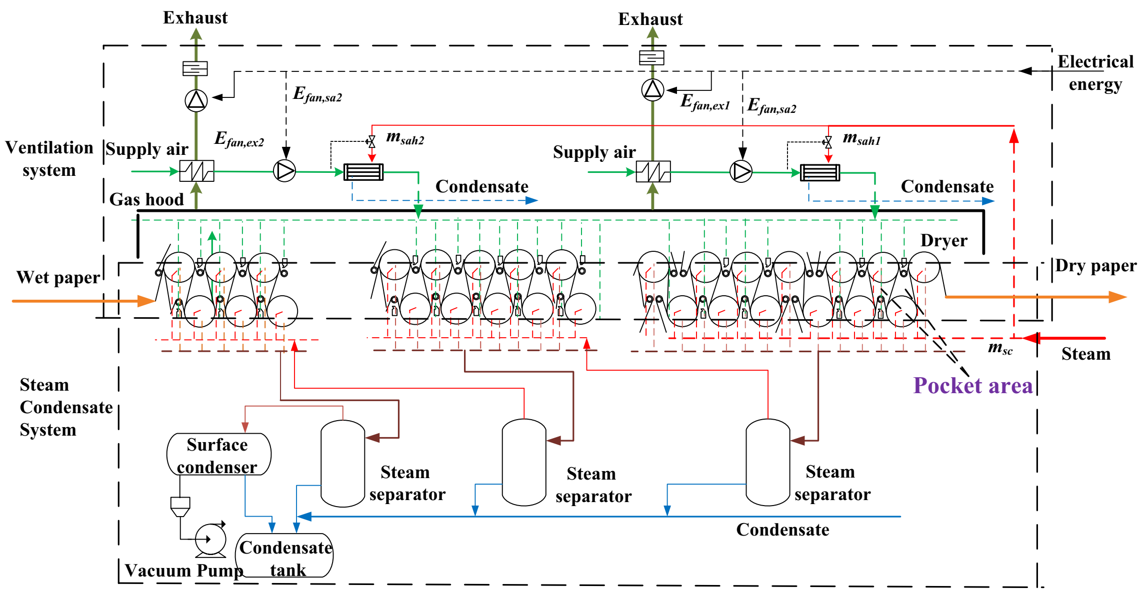


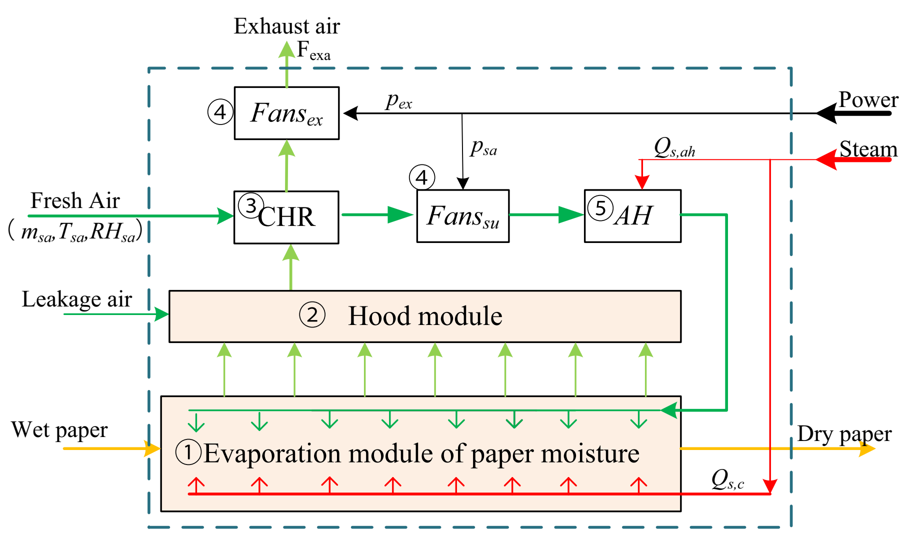
Fig. 1 Schematic diagram of a typical ventilation system

For the steam condensate system, the paper absorbs the latent heat released by the steam from cylinders to evaporate the paper moisture. Condensate of the cylinder is discharged into the steam-water separators to generate secondary steam for the next stage. Meanwhile, vapor generated by evaporation is diffused into the pocket area. For the ventilation system, supply air preheated by exhaust air and heated to the setting temperature by the steam heater, which is used to absorb the moisture evaporated in the pocket area. In addition to preheat supply air, exhaust air is also used to heat process water or ventilation of workshop, and finally discharged into the outside. Ventilation system affects the air condition in pocket area, the evaporation rate, and the steam consumption of cylinders. As an energy carrier and conveying material, exhaust air influences the steam consumption of air heater and power consumption of fans in the ventilation system.

2.2 Modeling of Energy Consumption in Multi-cylinder Dryer Section

To understand the influence of the ventilation system on the energy consumption of each part in drying system, the subsystems or equipment affected by the ventilation system are divided into five modules in view of different functions, including evaporating module of paper moisture, hood module, waste heat recovery module; fan module and air heating module, as shown in Fig. 2. Modelings of each functional

Fig. 2 Function modules of a typical ventilation system



module are the basis to analyse the influence of the ventilation system on energy consumption systematically. So, the above five functional modules will be theoretically analyzed and established according to the principle of heat and mass transfer as follows.

2.2.1 Energy Consumption Model of Paper Moisture Evaporation

As the heat energy for evaporating comes from steam, whether fresh steam or secondary steam will eventually turn into the condensate after releasing its latent heat. Although the final exhaust discharged from the dryer is discharged together with the non-condensable gas through the condensation tank or vacuum pump, and part of the exhaust steam may be used to heat the process water, it has no impact on the energy consumption because the recovered heat is not returned to the drying system. So, the steam consumption of dryers ($m_{s\text{-}c}$, kg/kg) can be calculated by Eq. (1).

$$m_{s-c} = \frac{Q_{s-c}}{h_s - h_{cond}} \tag{1}$$

Here, $Q_{s\text{-}c}$ is the heat transferred from steam to cylinders, kJ/kg paper; h_s and h_{cond} are the enthalpies of inlet steam and outlet condensate in drying system, kJ/kg. Modeling process of energy consumption model can be simplified as shown in Fig. 3.

Analysing from the paper drying system shown in Fig. 3a, the sheet evaporates the moisture by absorbing heat (including heat transferred by steam and surrounding air). Steam condensate system can be considered as the heat source of evaporation, so the cylinder is taken as the research boundary in this study, and the influence of steam condensate system variation on the ventilation system can be ignored. The evaporation of paper moisture can be simplified as the module shown in Fig. 3b. The inputs of the paper moisture evaporation module include wet sheet and supply air, and the output includes exhaust air and dry paper. According to the laws of material balance and energy conservation, the Eq. (2) can be obtained.

$$\begin{cases} m_{a,s} = m_{a,ex} \\ m_{a,s}H_{a,s} + m_f\left(\frac{1}{d_{i,p}} - \frac{1}{d_{o,p}}\right) = m_{a,ex}H_{a,ex} \\ Q_{s-c} + m_{a,s}I_{a,s} + m_f(c_f + c_w x_{i,p})T_{i,p} = m_{a,ex}I_{a,ex} + m_f(c_f + c_w x_{o,p})T_{o,p} \end{cases} \tag{2}$$

where, $m_{a,s}$ and $m_{a,ex}$ are the flow rate of supply and exhaust air, respectively, kg/h; m_f is the flow rate of fiber, which can be calculated according to the machine speed v, basic weight G, paper width B and paper dryness $d_{o,p}$, kg/h. $d_{i,p}$ and $d_{i,p}$ are the inlet and outlet dryness of the sheet, %; $H_{a,s}$ and $H_{a,ex}$ are the absolute humidities of supply and exhaust air, respectively, kg H_2O/kg d.a.; $I_{a,s}$ and $I_{a,ex}$ are the enthalpies of supply and exhaust air, kJ/kg; c_f and c_w are the specific heat of dry fiber and water, respectively, c_f = 1.423 kJ/(kg·°C) and c_w = 4.1868 kJ/(kg·°C); $x_{i,p}$ and $x_{o,p}$ are the paper moisture ratio inlet and outlet the drying section, respectively, kgH_2O/kg d.f; $T_{i,p}$ and $T_{o,p}$ are the paper temperatures inlet and outlet the drying section, respectively, °C. Among them, $H_{a,s}$, m_f, $d_{o,p}$, $d_{i,p}$, c_f, c_w and $T_{i,p}$ are known variables, and $m_{a,ex}$ can be obtained through the hood module. $I_{a,s}$ can be calculated by Eq. (3) if supply air humidity $H_{a,s}$ and the temperature $T_{a,s}$ are known, and the paper moisture ratio ($x_{i,p}$, $x_{o,p}$) can be calculated according to Eq. (4) [26] if $d_{o,p}$ and $d_{i,p}$ are known.

$$I_{a,s} = (c_g + c_v H_{a,s})T_{a,s} + I_0 H_{a,s} \tag{3}$$

$$x_p = \frac{1}{d_p} - 1 \tag{4}$$

Here, c_g and c_v are the special heat of dry air and vapor, respectively, c_g = 1.01 kJ/(kg·°C), c_v = 1.88 kJ/(kg·°C); I_0 is the latent heat of water vaporization at 0 °C, I_0 = 2501 kJ/kg.

Therefore, Eqs. (2) consist of 3 equations with 16 variables, of which 11 variables are known or can be calculated. Two additional variables ($Q_{s\text{-}c}$ and $T_{o,p}$) need to be solved to solve the equations. As shown in Fig. 3b, each dryer can be regarded as a drying unit, the whole drying section can be considered as the superposition of N_c drying units, Eq. (5)–(7) can be obtained.

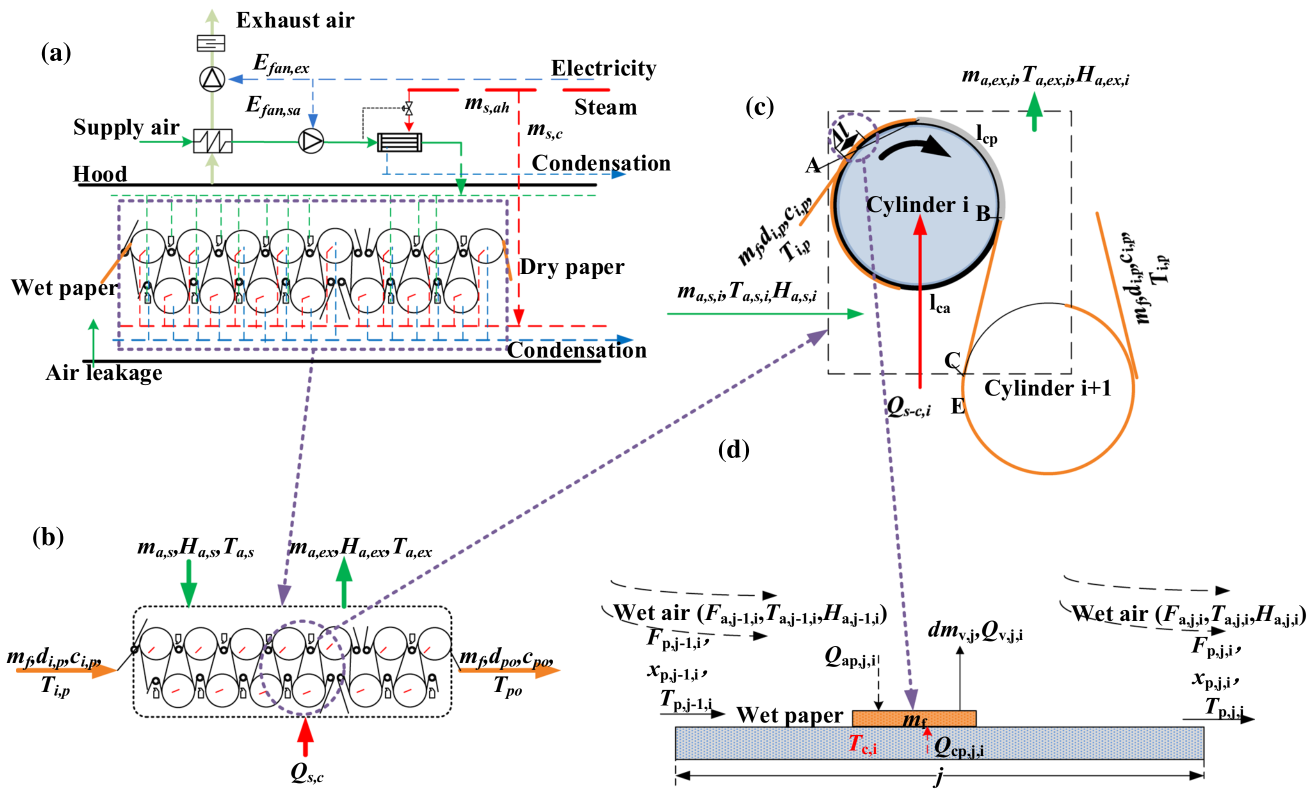


Fig. 3 Modeling of paper moisture evaporation in multi-cylinders dryer section

$$Q_{s-c}=\sum_{i=1}^{N_c}Q_{s-c,i} \tag{5}$$

$$m_{a,s}=\sum_{i=1}^{N_c}m_{a,s,i}=\sum_{i=1}^{N_c}\theta_i m_{a,s,0} \tag{6}$$

$$m_{a,ex}H_{a,ex}=\sum_{i=1}^{N_c}m_{a,ex,i}H_{a,ex,i} \tag{7}$$

where, $Q_{s\text{-}c,i}$ is the heat transferred from steam to the i-th cylinder clylinder, kJ/kg paper; $m_{a,s,i}$ and $m_{a,ex,i}$ are the flows of supply and exhaust air in the i-th dryer, respectively, kg/h; $H_{a,ex,i}$ is the absolute humidity of exhaust air in the i-th dryer. The i-th drying unit was selected for analysis, as shown in Fig. 3c. According to mass conservation and energy balance, Eq. (8) can be obtained.

$$\begin{cases} m_{a,ex,i}=m_{a,s,i} \\ m_{a,ex,i}H_{a,ex,i}=m_{a,s,i}H_{a,s,i}+dm_{v,i} \\ m_{a,s,i}I_{a,s,i}+Q_{s-c,i}=dQ_{p,i}+m_{a,ex,i}I_{a,ex,i} \end{cases} \tag{8}$$

where, $H_{a,s,i}$ is the absolute humidity of supply air in the i-th dryer; $dm_{v,i}$ is the evaporation flow of the i-th cylinder; $dQ_{p,i}$ is the increased heat of the sheet in the i-th cylinder. Here, $H_{a,s,i}=H_{a,s}$, $I_{a,s,i}=I_{a,s}$, $m_{a,ex,i}$ is can be obtained from the hood module. Therefore, three variables in Eq. (8) are known and can be solved only after $Q_{s\text{-}c,i}$, $dm_{v,i}$ and $dQ_{p,i}$ are obtained.

As shown in Fig. 3d, if the wet paper in the i-th dryer is equally divided by length Δl, the paper in the i-th dryer can be divided into $(l_{cont}+l_{conv})/\Delta l$ sections. Here, l_{cont} and l_{conv} are the length of contact drying stage and convective drying stage, respectively. An element with the mass of m_p and the temperature of $T_{p,j\text{-}1,i}$ is analyzed, which passes through the j-section of the i-th dryer at the speed of v, as the ambient air temperature is $T_{a,j\text{-}1,i}$. As long as $T_{p,j\text{-}1,i}$ is not equal to $T_{a,j\text{-}1,i}$, heat transfer will occur. According to the previous research, heat and mass transfer of paper moisture can be expressed by Eq. (9) and (10) [6]:

$$\frac{dT_p}{dl} = \frac{h_{c-p,j,i}(T_{c,i} - T_{p,j,i})A_{cp} + h_{a-p,j,i}(T_{a,j,i} - T_{p,j,i})A_{a-p} + v\Delta h_{v,j,i}dx_p/dl}{vm_f(c_f + c_w x_{p,j,i})} \tag{9}$$

$$\frac{dx_p}{dl} = -\frac{k_\rho A_{a-p}M_v}{vm_f R}\left(\frac{p_{pv,j-1,i}}{T_{p,j-1,i}} - \frac{p_{av,j-1,i}}{T_{a,j-1,i}}\right) \tag{10}$$

Here, $h_{c\text{-}p,j,i}$ is the average heat transfer coefficient between cylinder and the sheet, W/(m^2 °C); $h_{a\text{-}p,j,i}$ is the average heat transfer coefficient between the sheet and surrounding air, W/(m^2 °C). $A_{c\text{-}p}$ is the contact area between the cylinder and the sheet, m^2; $A_{a\text{-}p}$ is the contact area between the sheet and surrounding air, m^2; $\Delta h_{v,j,i}$ is the demand heat of vaporization, which is composed of the vaporization heat and the adsorption heat of paper. $P_{pv,j\text{-}1,i}$ is the vapor partial pressure corresponding to the paper temperature $T_{p,j\text{-}1,i}$. $P_{av,j\text{-}1,i}$ is the vapor partial pressure of air at temperature $T_{a,j\text{-}1,i}$ and humidity $H_{a,j\text{-}1,i}$; k_ρ is mass transfer coefficient, R is general gas constant; M_v is the molecular weight of water ($M_v = 18$ g/mol). $h_{cp,j,i}$ is difficult to obtain by experimental method. Here, it is determined by computer fitting based on the measured data. $h_{a\text{-}p,j,i}$ meets the Nusselt number calculation formulas [27, 28], which can be obtained by solving Eq. (11).

$$\begin{cases} h_{ap,j,i} = Nu\dfrac{\lambda_a}{l_{pa}} \\ Nu = 0.064\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}, \mathrm{Re} < 5\times10^5; \\ Nu = \dfrac{0.037\mathrm{Re}^{4/5}\,\mathrm{Pr}}{1 + 2.443\mathrm{Re}^{-0.1}(\mathrm{Pr}^{2/3} - 1)}, \mathrm{Re} \geq 5\times10^5 \\ \mathrm{Re} = \dfrac{\rho_a v_a l_{pa}}{\mu_a} \\ \mathrm{Pr} = \dfrac{c_a\mu_a}{\lambda_a} \end{cases} \tag{11}$$

where, l_{pa} is the characteristic length of air flowing through the paper surface. For the contact drying stage, the characteristic length is the length l_{cont}. For the convective drying stage, the characteristic length is the length l_{conv}. ρ_a is the density of pocket air, kg/m^3; v_a is the volume flow rate of air; μ_a is the viscosity of air, Pa·s; λ_a is the thermal conductivity of air, W/(m^2·°C). The physical parameters of pocket air change as it absorbs the moisture evaporated from the paper, which can be calculated by the following equations [29, 30].

$$\begin{cases} \rho_a = \rho_{da}(1 + x_a) = \rho_0\dfrac{273.15}{273.15 + \mathrm{T_a}}\dfrac{\mathrm{p_a}}{\mathrm{p_0}}(1 + x_a) \\ H_a = 0.622\dfrac{x_a P_{sat}(T_a)}{P_0 - x_a P_{sat}(T_a)} \\ \mu_a = \dfrac{\mu_{da}M_v}{M_v + H_a M_{da}\gamma_{da,v}} + \dfrac{\mu_v H_a M_{da}}{H_a M_{da} + M_v\gamma_{v,da}} \\ c_a = \dfrac{c_{da}}{1 + H_a}(1 + \dfrac{H_a c_v}{c_{da}}) \\ \lambda_a = \dfrac{\lambda_{da}}{1 + H_a(\frac{M_{da}}{M_v})\gamma_{da,v}} + \dfrac{H_a\lambda_v}{H_a + \frac{M_v}{M_{da}}\gamma_{v,da}}) \\ \gamma_{da,v} = \dfrac{[1 + (\mu_{da}/\mu_v)^{1/2}(M_v/M_{da})^{1/4}]^2}{[8(1 + M_{da}/M_v)]^{1/2}} \\ \gamma_{v,da} = \dfrac{\mu_v M_{da}\gamma_{da,v}}{\mu_{da}M_v} \end{cases} \tag{12}$$

Here ρ_0 is the dry air density at 0 °C and 1 atm, $\rho_0 = 1.293$ kg/m^3; ρ_{da} is dry air density, kg/m^3; μ_{da}, μ_v, c_{da}, c_v, λ_{da} and λ_v are the dynamic viscosity, specific heat capacity and thermal conductivity of dry air and vapor respectively, which can be obtained from the table of thermophysical properties of dry air and vapor or calculated by fitting; M_{da} and M_v are the relative molecular weights of dry air and vapor, respectively, $M_{da} = 28.96$ g/mol and $M_v = 18.016$ g/mol.

The mass transfer coefficient between paper and air ($k_{ap,j,i}$) can be obtained according to Lewis relation, as expressed by Eq. (13) [27, 31, 32]:

$$\begin{cases} k_{ap,j,i} = \dfrac{h_{ap,j,i}}{\rho_a c_a}Le^{-2/3} \\ Le = \dfrac{Sc}{\mathrm{Pr}} \\ Sc = \dfrac{\mu_a}{\rho_a\frac{10^{-7}(T_{a,j,i}+273.15)^{1.75}}{P_0(V_a^{1/3}+V_v^{1/3})^2}\sqrt{\frac{1}{M_{da}} + \frac{1}{M_v}}} \end{cases} \tag{13}$$

Here, $P_0 = 1$ atm, $V_\mathrm{a} = 20.1$ cm^3/mol, $V_\mathrm{v} = 12.7$ cm^3/mol.

The partial pressure of vapor on the paper surface ($P_{pv,j\text{-}1,i}$) is the function of the paper moisture. Refer to the Antoine equation, $P_{pv,j\text{-}1,i}$ can be calculated by Eq. (14) [33, 34].

$$P_{pv,j-1,i} = \varphi P_{sat,pv} = 133.322(1 - e^{-(47.58x_{p,j-1,i}^{1.877} + 0.10085T_{p,j-1,i}x_p^{1.0585})})\exp^{(18.3036 - \frac{3816.44}{T_{p,j-1,i}+227.03})} \tag{14}$$

where, Φ is the isotherm adsorption line equation, considered as the correction factor of vapor partial pressure on the paper surface; $P_{pv,j\text{-}1,i}$ is the saturated vapor partial pressure corresponding to the temperature of the sheet at $T_{p,j\text{-}1,i}$. The vapor partial pressure of surrounding air ($P_{av,j-1,i}$) is the function of air relative humidity (RH_a) and air temperature, as expressed by Eq. (15) [25, 34].

$$P_{av,j-1,i} = 133.322RH_a \exp^{(18.3036-\frac{3816.44}{T_{a,j-1,i}+227.03})} \tag{15}$$

Evaporation heat of paper moisture ($\Delta h_{v,j,i}$) needs additional adsorption heat, in addition to the latent heat of vaporization. It satisfies the following equation [25, 27].

$$\Delta h_{v,j,i} = 10^3\left(2501 - 2.3237T_{p,j,i}\right) + \frac{R}{M_v}0.10085x_p^{1.0585}(T_{p,j,i} + 273.15)^2\frac{1-\varphi}{\varphi} \tag{16}$$

Moreover, $dQ_{p,i}$ can be calculated based on the sensible heat change of the sheet in i-th drying unit, as shown in Eq. (17). From Eq. (9), the paper moisture ratio entering and leaving the i-th drying unit can be obtained. According to the mass conservation, the evaporation water of the drying unit ($dm_{v,i}$) can be obtained by Eq. (18).

$$dQ_{p,i} = \sum_{j=0}^{(l_{cont}+l_{conv})/\Delta l} dQ_{p,j,i} = \sum_{j=0}^{(l_{cont}+l_{conv})/\Delta l} m_f(c_f + c_w x_{p,j,i})dT_{p,j,i} \tag{17}$$

$$dm_{v,i} = \sum_{j=0}^{(l_{cont}+l_{conv})/\Delta l} dm_{v,j,i} = \sum_{j=0}^{(l_{cont}+l_{conv})/\Delta l} -m_f dx_{p,j,i} \tag{18}$$

$Q_{s\text{-}c,i}$ is the sum of the heat transferred to the paper and the heat dissipation. According to Newton's cooling law and the law of mass conservation, $Q_{s\text{-}c,i}$ can be calculated by Eq. (19).

$$Q_{s-c,i} = \sum_{j=0}^{(l_{cont}+l_{conv})/\Delta l} h_{c-a,j,i}(T_{c,i} - T_{a,j,i})(Bl_{ca} + \pi D_c^2/2)/(GBv) + \sum_{j}^{l_{cont}/\Delta l} h_{c-p,j,i}(T_{c,i} - T_{p,j,i})A_p dl/(Gv) \tag{19}$$

Based on the calculation results of heat and mass transfer models (9) and (10), $Q_{s\text{-}c,i}$, $dm_{v,i}$ and $dQ_{p,i}$ can be obtained. $m_{sa,i}$, $H_{ex,i}$ and $I_{ex,i}$ can be obtained by substituting them into Eqs. (8). Then, $T_{ex,i}$ can be obtained according to Eqs. (2). According to the calculation results of Eqs. (5) and $Q_{s\text{-}c,i}$, the steam consumption of cylinders ($m_{s\text{-}c}$) can be calculated by Eq. (1). The obtained exhaust parameters ($H_{ex,p}$, $I_{ex,p}$ and $T_{ex,p}$) can be used for subsequent calculation of hood module.

2.2.2 Energy Transfer Model of Hood

The hood is mainly used to collect the hot air absorbed the evaporated water and remove it in time. Ignoring the influence of the leakage air on the evaporation process, the supply air sent into the pocket area absorbs the evaporated vapor and evenly mixes with the leakage air in the hood. The hood module can be considered as a mixer of the wet air and the leakage air, as shown in Fig. 4.

The heat source of the hood module includes the heat carried by the leaked air and wet air. According to the conservation of mass and energy, the Eqs. (20) can be obtained.

$$\begin{cases} m_{ex,h} = m_{ex,p} + m_{la} \\ m_{la} = \varepsilon m_{ex,h} \\ m_{ex,h}H_{ex,h} = m_{ex,p}H_{ex,p} + m_{la}H_{la} \\ m_{ex,h}I_{ex,h} + Q_{loss,h} = m_{ex,p}I_{ex,p} + m_{la}I_{la} \\ Q_{loss,h} = k_hA_h(T_{o,h} - T_{i,h}) + k_bA_b(T_{o,b} - T_{i,b}) \\ m_{ex,h} = \dfrac{m_f(d_{po} - d_{pi})}{d_{po}d_{pi}(H_{ex,h} - H_{sa})} \end{cases} \tag{20}$$

where, H_{la} and I_{la} are the absolute humidity and enthalpy of leakage air, respectively, which can be obtained by testing the parameters of indoor air. Generally, the leakage coefficient ε of the hood is 20–30% [7]. k_h and k_b are the heat dissipation coefficients of hood body and foundation, respectively, taking $k_h = 3$ W/(m^2 °C) and $k_b = 8$ W/(m^2·°C) [25], respectively; A_h and A_b are the heat dissipation areas of hood

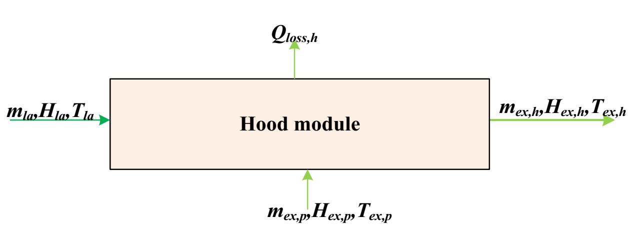


Fig. 4 Modular diagram of the hood

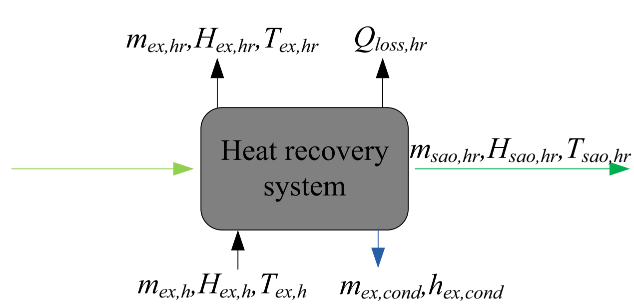


Fig. 5 Modular diagram of heat recovery system

body and foundation, respectively, m^2; $T_{o,h}$, $T_{i,h}$, $T_{o,b}$ and $T_{i,b}$ are the temperature inside and outside hood body and foundation, respectively, °C.

Therefore, as the input air parameters ($H_{ex,p}$, $I_{ex,p}$) can be obtained from the evaporation module above, the hood exhaust enthalpy $I_{ex,h}$ can be obtained by solving the Eq. (20) combined with the known parameters and determined coefficients. Then, the hood exhaust temperature $T_{ex,h}$ can be obtained through Eq. (3).

2.2.3 Energy Transfer Model of Heat Recovery System

Heat recovery system recovers the waste heat of exhaust air to improve the energy utilization efficiency of drying system. Generally, heat recovery systems include gas–gas heat exchangers and gas–liquid heat exchangers, which are mainly used to preheat the supply air and the process water or workshop ventilation. Since heat recovery of heating process water or workshop ventilation has no direct impact on the energy consumption of drying system. Only the gas–gas heat exchange system that uses exhaust to preheat the supply air needs to be considered, its structural diagram can be shown as Fig. 5.

Heat source of gas–gas heat recovery system is the wet air exhausted from the hood module, and heat sink is the pocket air extracted from the outdoor or indoor. The output is the cooled exhaust and the supply air after preheating. As the temperature of exhaust air decreases, condensation may occur if the temperature declines below the dew point [35]. According to the mass and energy balance, energy transfer model of heat recovery system can be expressed by Eqs. (21).

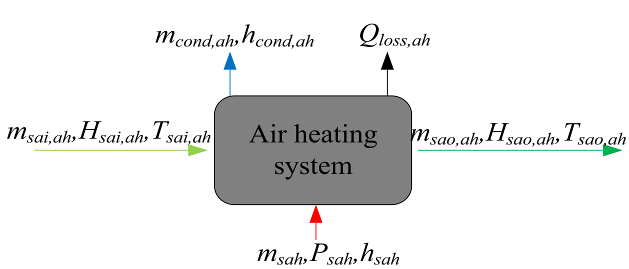


Fig. 6 Modular diagram of the air heater

$$\begin{cases} m_{sai,hr} = m_{sao,hr} \\ m_{exi,hr} = m_{exo,hr} \\ m_{sai,hr}H_{sai,hr} = m_{sao,hr}H_{sao,hr} \\ m_{exi,hr}H_{exi,hr} = m_{exo,hr}H_{exo,hr} + m_{cond,hr} \\ m_{sao,hr}I_{sao,hr} - m_{sai,hr}I_{sai,hr} = k_{hr}A_{hr}\Delta T dt \\ m_{sao,hr}I_{sao,hr} + m_{exo,hr}I_{exo,hr} \\ + \int_{T_{exo,hr}}^{T_{ex,d}} m_{cond,hr}c_w dT_a + Q_{loss,hr} = m_{sai,hr}I_{sai,hr} + m_{exi,hr}I_{exi,hr} \\ I_{sai,hr} = (c_a + c_v H_{sai,hr})T_{sai,hr} + I_0 H_{sai,hr} \\ I_{sao,hr} = (c_a + c_v H_{sao,hr})T_{sao,hr} + I_0 H_{sao,hr} \\ I_{exo,hr} = (c_a + c_v H_{exo,hr})T_{exo,hr} + I_0 H_{exo,hr} \\ T_{exi,hr} = \dfrac{I_{exi,hr} - I_0 H_{exi,hr}}{(c_a + c_v H_{exi,hr})} \\ \Delta T = \left| \dfrac{(T_{exo,hr} - T_{sai,hr}) - (T_{exi,hr} - T_{sao,hr})}{\ln(\frac{T_{exo,hr} - T_{sai,hr}}{T_{exi,hr} - T_{sao,hr}})} \right| \\ dt = \dfrac{m_p}{vBG} \end{cases} \tag{21}$$

where: $T_{sai,hr}$ and $H_{sai,hr}$ are the temperature and absolute humidity of supply air sent to the heat recovery system, respectively; $m_{sao,hr}$ is the flow rate of supply air outlet the heat exchanger, which equals to the flow rate of supply air and can be obtained through the paper moisture evaporation module; k_{hr} and A_{hr} are the heat transfer coefficient and area of the heat exchangers, respectively. The heat dissipation loss $Q_{loss,hr}$ of the heat exchanger also can be calculated by similar equation in (20), and the dew point temperature $T_{ex,d}$ is calculated by Eq. (22). So, if the parameters of heat source (exhaust air) are known, the parameters of supply and exhaust air after heat exchanging can be obtained.

$$T_{ex,d} = 99.64 + 329.64\frac{\ln(p_{aw})}{11.78 - \ln(p_{aw})} \tag{22}$$

2.2.4 Power Consumption Model of Fans

Although conveying materials and functions are different, both blowers and exhaust fans are used to pump the air by consuming electric energy. Therefore, energy consumption models can be expressed by the similar equations [35, 36], as follows.

$$E_{fan,sa} = \frac{m_{sa}P_{fan,sa}}{\rho_{sa}\eta_m\eta_{fan}} \tag{23}$$

$$E_{fan,ex} = \frac{m_{ex}P_{fan,ex}}{\rho_{ex}\eta_m\eta_{fan}} \tag{24}$$

where, m_{sa} and m_{ex} are the flow rate of blowers and exhaust fans, respectively. Generally, the blowers are installed between the air heaters and heat recovery exchangers, $m_{sai,ah} = m_{sa} = m_{sao,hr}$; the exhaust fans are installed between hood and heat recovery exchangers, $m_{ex,h} = m_{ex} = m_{exi,hr}$. ρ_{sa} and ρ_{ex} are the densities of supply air and exhaust under actual working conditions, kg/m^3; $P_{fan,sa}$ and $P_{fan,ex}$ are the total pressure of blowers and exhaust fans, Pa. By referring to the instructions of the equipment manufacturers, η_{fan} is the fan efficiency (usually among 70%–80%), η_m is the transmission efficiency, taken as 95% as the pulley selected.

2.2.5 Energy Consumption Model of Air Heater

The function of the air heaters is to heat the supply air preheated by the heat recovery system to the setting temperature of pocket air. The module can be simplified by Fig. 6. The inputs of the air heater include steam and the preheated supply air, and the outputs are condensate water and hot air.

According to the mass and energy balance, energy consumption model of air heater can be expressed by Eq. (25).

$$\begin{cases} m_{sai,ah} = m_{sao,ah} \\ m_{s,ah} = m_{cond,ah} \\ m_{sai,ah}H_{sai,ah} = m_{sao,ah}H_{sao,ah} \\ m_{sao,ah}I_{sao,ah} - m_{sai,ah}I_{sai,ah} = k_{ah}A_{ah}(T_{s,ah} - \dfrac{T_{sao,ah} + T_{sai,ah}}{2})dt \\ m_{sao,ah}I_{sao,ah} + m_{cond,ah}h_{cond,ah} + Q_{loss,ah} = m_{sai,ah}I_{sai,ah} + m_{s,ah}h_{s,ah} \\ I_{sao,ah} = (c_a + c_v H_{sao,ah})T_{sao,ah} + I_0 H_{sao,ah} \\ dt = \dfrac{m_p}{vBG} \end{cases} \tag{25}$$

Generally, the enthalpies of steam ($h_{s,ah}$) and condensate ($h_{cond,ah}$) can be obtained by checking up the table of steam properties. The temperature of supply air should be raised to the setting temperature $T_{sao,ah}$ (i.e.$T_{sa,i}$, T_{sa}). TAPPI recommends that the supply air temperature should be between 82 and 93 °C [8]; k_{ah} and A_{ah} refer to the heat transfer coefficient and heat exchange area of the air heater, respectively. The heat dissipation loss of the heater ($Q_{loss,ah}$) usually adopts the empirical coefficient of 10–20% [37]. As the parameters of supply air pumped into the air heater are known, the steam consumption of the air heater can be calculated.

2.2.6 Energy Consumption Model of Drying System

As the analysis above, energy consumption of drying system includes three parts, including steam consumption of cylinders, power consumption of blowers and exhaust fans, and steam consumption of air heaters. To maintain the consistency of energy consumption evaluation with the actual production, energy consumption indexes per unit product are adopted to evaluate the energy consumption of each module or system, as expressed by the following equations.

$$\begin{cases} \Delta m_{sc} = \dfrac{\sum_{n=1}^{N1} m_{sc,n}}{m_p} \\ \Delta m_{sah} = \dfrac{\sum_{n=1}^{N2} m_{sah,n}}{m_p} \\ \Delta E = \Delta E_{fan,sa} + \Delta E_{fan,ex} = \dfrac{\sum_{n=1}^{N3} E_{fan,sa,n}}{m_p} + \dfrac{\sum_{n=1}^{N4} E_{fan,ex,n}}{m_p} \end{cases} \tag{26}$$

where, Δm_{sc} and Δm_{sah} are the steam consumption of cylinders and air heaters per unit product respectively, kg steam/kg paper. $\Delta E_{fan,sa}$ and $\Delta E_{fan,ex}$ are the electrical energy consumption of blowers and exhaust fans per unit product respectively, kWh/kg paper. $N1$, $N2$, $N3$ and $N4$ represent the numbers of cylinders, air heaters, blowers, and exhaust fans, respectively. The whole energy consumption can be divided into steam and power, which can be expressed as follows.

$$\begin{cases} \Delta m_s = \Delta m_{sc} + \Delta m_{sah} \\ \Delta E_{fan} = \Delta E_{fan,sa} + \Delta E_{fan,ex} \end{cases} \tag{27}$$

where, Δm_s is the steam consumption per unit product in drying system, kg steam/kg paper; ΔE_{fan} is the electrical energy consumption per unit product, kWh/kg paper.

According to the relationship of the modules shown in Fig. 2, the energy consumption model of drying system can be constructed by combining the energy consumption models of each module established above, as shown in Fig. 7. The inputs of the model include the physical parameters of the paper (the inlet dryness and outlet of multi-dryer section, and so on), operation parameters (yield, basic weight, machine speed, parameters of steam and condensate, opening degree of ventilation valves, surface temperature of each cylinder, etc.), basic information of paper machine (paper width, geometric parameters, number and diameter of cylinders, performance parameters of heat exchangers and air heaters, etc.). The outputs are the energy consumptions per unit product (including steam and electrical energy consumption per kg paper).

According to Sect. 2.2.1, based on the combination of heat and mass transfer models and model coefficient

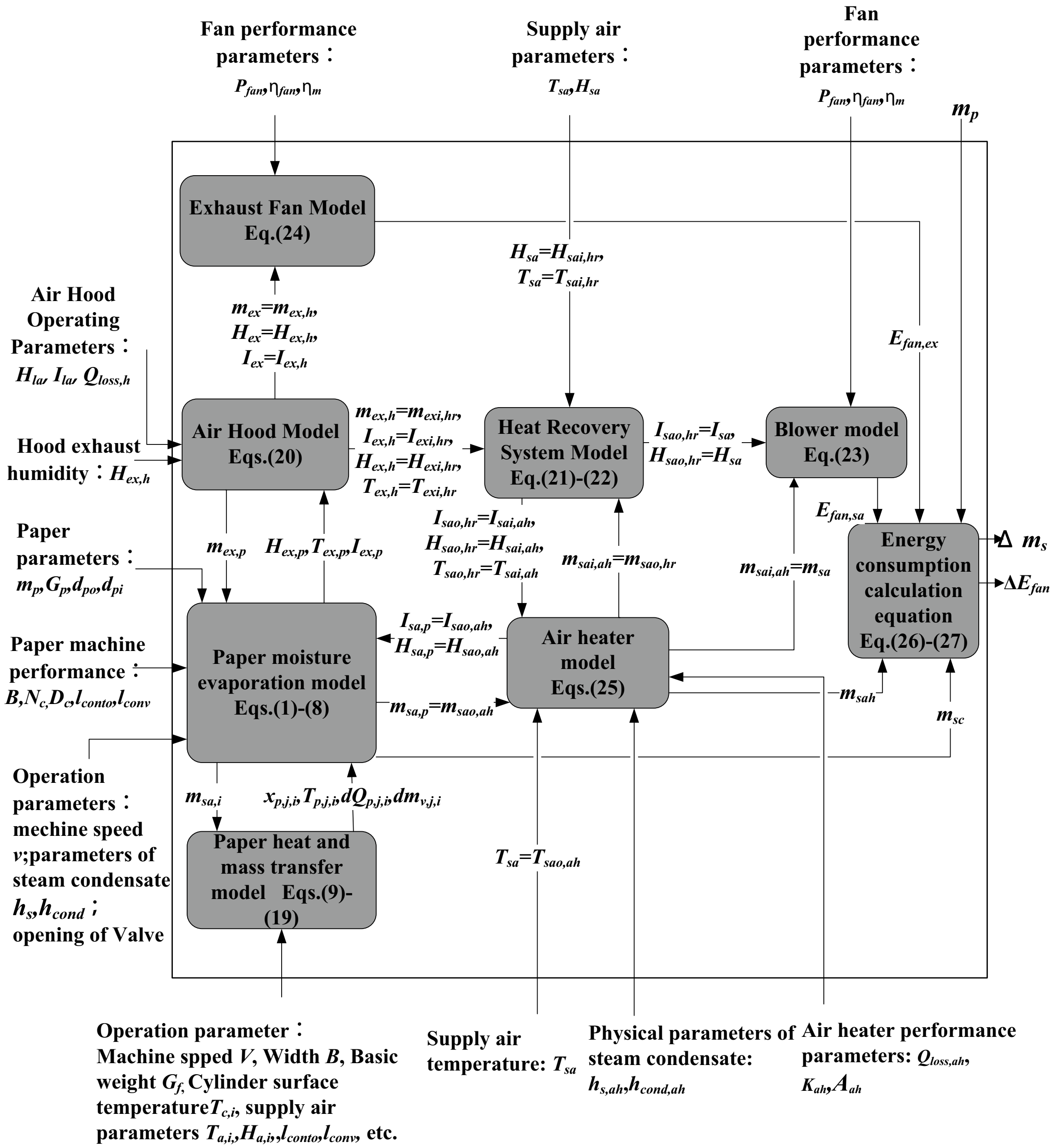


Fig. 7 Energy consumption model of drying system

Eqs. (11)–(19), $Q_{s\text{-}c,i}$, $dm_{v,i}$ and $dQ_{p,i}$ of each dryer can be obtained. Substituting them into the Eqs. (1)–(8), we can get the amount of supply air ($m_{sa,i}$) and the exhaust parameters ($H_{ex,i}$, $I_{ex,i}$, $T_{ex,i}$) in each cylinder, the total amount of supply air ($m_{sa,p}$), the exhaust parameters of hood ($H_{ex,p}$, $I_{ex,p}$, $T_{ex,p}$). According to Eqs. (1)–(8), the steam consumption ($m_{s\text{-}c}$) of cylinders and the parameters of pocket air can be solved. The flow rate of exhaust air ($m_{ex,h}$) can be obtained based on the exhaust dew point or exhaust humidity in the hood as shown in Fig. 7. Combined with the parameters of pocket air calculated by the paper moisture evaporation module, the parameters of exhaust air can be calculated. Combined with the known parameters and coefficients, the parameters of the supply and exhaust air after heat exchangers can be calculated. Power consumption of supply and exhaust air can be calculated by Eqs. (23) and (24), respectively. Combined with the known and obtained parameters, the steam consumption of the air heater (m_{sah}) can be obtained by Eq. (25).

Finally, the above models are combined with the energy consumption indexes models (Eqs.26 and 27) to form the energy consumption model of drying system. The inputs of the model include three main categories. The first one is the performance parameters of the paper machine or related equipment, which is mainly obtained from the design data of the paper machine or related equipment; such as paper width, the diameter of cylinders, the length of contact drying and convection drying section, heat transfer coefficients and areas of heat exchangers and air heaters, the performance parameters of fans and the heat dissipation performance of each device or system. Second, the setting parameters are mainly obtained from distributed control system (DCS), such as exhaust dew point or exhaust humidity, supply air temperature, etc. Third, the operation parameters: mainly obtained through testing or DCS system and quality control system (QCS) or calculated through relevant parameters, such as the machine speed, enthalpies of steam and condensate, opening status of valve, and so on. The outputs of the model are the steam consumption of cylinders and air heates, power consumption of fans.

3 Conclusion and Future Work

By analyzing the influence of ventilation system on energy consumption in drying system, energy consumption of drying system is divided into three parts: steam consumption of cylinders, steam consumption of air heaters, power consumption of blowers and exhaust fans. An energy consumption model of drying system integrating steam condensation system and ventilation system is constructed according to the control logic and process flow of the actual ventilation system, by combining the established models of each module and energy consumption indexes, which perfects the mechanism of ventilation system for assisted drying. The research of the models is the key to the integration the control of ventilation system and steam condensate system. As the theoretical model of the integrating system is obtained, the more optimal operating parameters and a greater degree of energy-saving effect can be realized by intelligent computing technology. It is also the premise to realize the intelligent control of paper drying process, intelligent fault diagnosis with uncertainties and intelligent real-time industrial process monitoring and control.

Considering the models involve too many equations and variables to get the analytical solution, moreover, some parameters of the models need to be fitted according to the specific object operation, so the application of models to simulate the energy consumption of drying system and analyzing the influence of various operating parameters on the total energy consumption systematically will be discussed in Part 2.

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Availability of Data and Material The data are given in the paper.

Declarations

Conflict of 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.

Ethics Approval Not applicable.

Consent to Participate Not applicable.

Consent for Publication Not applicable.

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