OPTIMIZATION FOR ENERGY CONSUMPTION IN DRYING SECTION OF FLUTING PAPER MACHINE

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Nonlinear programming optimization method was used to optimize total steam and air consumption in the dryer section of multi-cylinder fluting paper machine. Equality constraints of the optimization model were obtained from specified process blocks considering mass and energy balance relationships in drying and heat recovery sections. Inequality constraints correspond to process parameters such as production capacity, operating conditions and other limitations. Using the simulation, the process parameters can be optimized to improve the energy efficiency and heat recovery performance. For a corrugating machine, optimized parameters show the total steam use can be reduced by about 11% due to improvement of the heat recovery performance and optimization of the operating conditions such as inlet web dryness, evaporation rate and exhaust air humidity, accordingly total steam consumption can be decreased from about 1.71 to 1.53 ton steam per ton paper production. The humidity of the exhaust air should be kept as high as possible to optimize the energy performance and avoid condensation in the pocket dryers and hood exhaust air. So the simulation shows the supply air should be increased by about 10% to achieve optimal humidity level which was determined about 0.152 kgH_2O(kg dry air)$^{-1}$.

Key Words: *Paper drying, Multi-cylinder dryers, Optimization, Steam consumption, NLP*

1. Introduction

The pulp and paper industry with about 6% of the total worldwide industrial energy use is the fourth main industrial energy consumer in the world and the drying section is the largest energy consumer in a paper mill [1, 2]. Nowadays, most paper mills have conventional multi-cylinder drying sections with closed hoods [2]. In these dryers, substantial steam energy is required for dryer cylinders and air heating systems. This energy mostly corresponds to the evaporated water and further to the exhaust air. For removal of moisture from wet paper, the hot air is supplied to the pocket dryers which are the space between adjacent cylinders in multi-cylinder drying section [3]. The evaporated water diffuses through the pocket ventilation air and forms the hood exhaust air. The exhaust air has high temperature and moisture content and contains about 90-95% of the total heat used in the drying process [4]. High energy content of the exhaust air makes it potentially suitable for the heat recovery objectives. The heat recovery system of a paper machine is basically a heat exchanger network that transfers energy from the humid exhaust air of the dryer section to various process streams [5]. A typical paper machine heat recovery system setup is

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presented in the fig. 1 [6]. As shown the humid exhaust air from the dryer hood is first directed to the conventional heat recovery (CHR) unit which recovers heat by the inlet supply air to the hood, then heat is recuperated in the heat recovery (AHR) units for more heating of supply air.

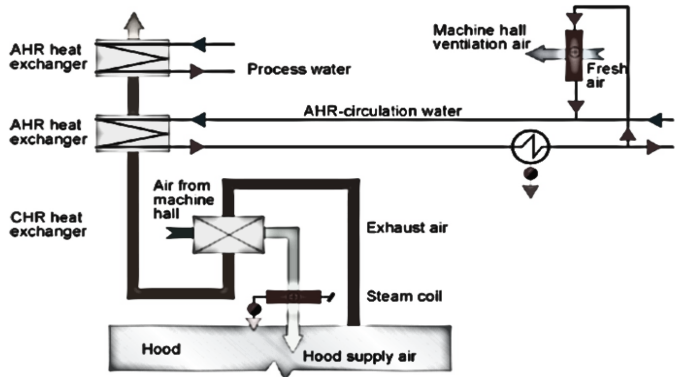

Fig. 1: A typical paper machine heat recovery system [6]

Mathematical programming methods based on the mass and energy balance are the most common approaches for energy optimization of the paper drying processes [7]. The first computerized design application established in the 1980s, used a sequential modular method in order to minimize energy use in heat recovery connecting to thermodynamic models [5]. In 2000 nonlinear programming (NLP) methods were used by Carlssona and Arfvidsson in drying section of a board machine [8]. Kemp used process integration and pinch analysis to optimize the heat consumption and cost optimization in paper dryers in 2005 [9]. Also a modular modeling tool was developed by Lindell and Stenstrom in order to study energy consumption and cost analysis in the pulp and paper industry in 2006 [10]. Pettersson and Soderman in 2007 carried out the economic analysis and studied the impacts of cost factor variations to optimize the heat recovery systems in paper machines [11]. In 2011, Li and his co-workers developed a NLP method for integration of steam and air systems to parameter optimization of the energy usage in a multi-cylinder dryer section of a newsprint and liner board machine. In order to analyze the modeling, the dryer section was divided into different modules based on their functions and the simultaneous modular method was applied to optimize process design and operation [7]. A static energy model for conventional multi-cylinder dryers was developed based on the mass and energy balance relationships for different basic blocks of paper drying by Kong and et al in 2012. Their model was used to evaluate the drying performance and energy efficiency [12].

In the present study a simulation program based on the nonlinear programming (NLP) method was used to optimize the drying parameters and examine the heat recovery performance in the fluting multi-cylinder dryers. In order to analyze the NLP model in detail, similar to the work of Li and his coworkers, the multi-cylinder dryer section was divided into different blocks depending on their functions.

2. Machine specifications and physical properties

The studied paper machine (PM2) is located in the northern Iran which produces about 300 t/day of fluting paper. The dryer section has 35 cylinders. The steam system is divided into three steam groups and the pressure of steam fed to the machine is 6 bars. As shown in the fig. 2, there are two heat recovery structures in the dryer section, one at the wet and the other at the dry end. The supply air is taken from outside and then is heated in the heat exchangers with hood exhaust air and further with fresh steam in the air heating system to the final temperature. The final heated supply air is blown into the drying pockets.

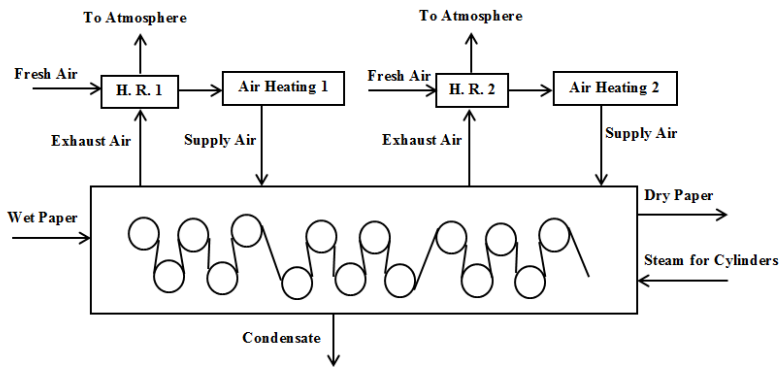

Fig. 2: The PM2 drying section and its heat recovery system

In this study the results of a previously developed model has been applied to determine the required parameters for optimization. The applied model has been developed based on the mass and energy balance relationships in which the heat of sorption and its variations with paper temperature and humidity changes have been taken into account in the falling rate period of the paper drying. Full details of formulation to develop the applied model can be found elsewhere [13, 14]. The operating conditions and paper properties of the machine are tabulated in tab. 1.

Table 1: The operating conditions of the paper machine used in the modeling

Parameter	Value
Paper basis weight (gm^{-2})	127
Inlet paper dryness to drying section (%)	40
Inlet paper temperature (^oC)	45
Final product moisture (%)	9
Paper width in drying section (m)	4.47
Running web speed ($m\ min^{-1}$)	440
Ambient temperature (^oC)	35
Supply air humidity ($gH_2O(kg\ dry\ air)^{-1}$)	16
Average supply air temperature (^oC)	125
Total supply air flow rate ($kg\ dry\ air\ s^{-1}$)	23.5

3. Methodology

The objective function of optimization can be defined as follows:

$$\min f\ (\dot{m}_{st}) = \min(\sum \dot{m}_{st}) = \min\left[(\dot{m}_{sc} + \dot{m}_{s,AH}) / \dot{m}_{p,o}\right] \qquad (1)$$

here $\dot{m}_{st}$ is the total steam mass flow rate in drying section. $\dot{m}_{sc}$ and $\dot{m}_{s,AH}$ are the mass flow of steam inside the cylinders and the steam consumed in the heat recovery heat exchanger, respectively. $\dot{m}_{p,o}$ is the paper machine throughput. The object of the optimization was to minimize the steam consumption in the whole drying section of the paper machine. Process parameters constituted the variables. The constraint functions and the variable bounds were obtained in each module. Various blocks and their input and output streams used in the optimization are shown in the fig. 3.

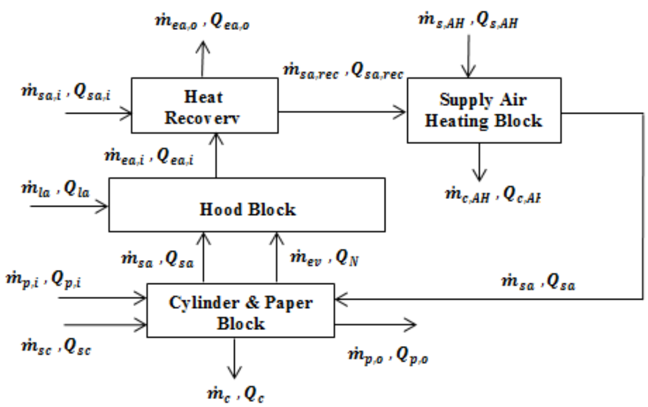

Fig. 3: The PM2 drying section and separate process blocks

3.1. Cylinders and Paper Block

Cylinders and paper block are the main modules of the drying system. The supply air can be distributed through the ventilator into the pocket dryers and the total mass flow of the humid air from the pocket dryers can be determined as follows:

$$\dot{m}_{pa} = \sum_{pd=1}^{35} (\dot{m}_{\sup,pd} + \dot{m}_{ev,pd}) = \dot{m}_{sa}(1+X_{sa}) + \dot{m}_{ev} \qquad (2)$$

here $\dot{m}_{pa}$ is the total output mass flow of humid air from the pocket dryers. $\dot{m}_{sa}$ is the mass flow of the dry supply air and X_{sa} represents the absolute humidity of supply air. $\dot{m}_{\sup}$ is the humid supply air and *pd* represents the pocket dryer No. $\dot{m}_{ev}$ denotes the amount of evaporation rate and can be obtained from the equation of mass balance or based on the initial and final web dryness:

$$\dot{m}_{ev} = \dot{m}_{p,o}\left(\frac{D_{p,o} - D_{p,i}}{D_{p,i}}\right) \qquad (3)$$

The available range for the inlet web dryness can be expressed as:

$$40 \le D_{p,i} \le 43 \qquad (4)$$

Also conventional standard range for moisture target of product is:

$$89 \le D_{p,o} \le 93 \tag{5}$$

Consequently for given paper grammage, web speed and machine width, the upper and lower bounds of evaporation rate for drying section can be obtained as follows:

$$4.1 \le \dot{m}_{ev} \le 5.5 \tag{6}$$

Also as shown in the fig. 4, the energy balance for the cylinder and paper block can be written as:

$$Q_{sc} + Q_{p,i} = Q_c + Q_N + Q_{p,o} \tag{7}$$

Q_{sc} is the energy of steam inside the cylinders and subscripts *i* and *o* point to input and output. Q_p is the paper energy in the drying section and can be calculated by the following correlation:

$$Q_p = (\dot{m}_f C_f + \dot{m}_w C_w) \tag{8}$$

where C_f and C_w represent the heat capacity of fiber and water, respectively. $\dot{m}_f$ is the fiber mass rate which is constant throughout the drying process and can be determined as follows:

$$\dot{m}_f = \dot{m}_{p,o} D_{p,o} \tag{9}$$

here D_p represents the web dryness and $\dot{m}_{p,o}$ is the final product out of the dryer which is calculated in terms of the paper basis weight (grammage), web speed and machine width in the drying section:

$$\dot{m}_{p,o} = GVW \tag{10}$$

here *G*, V and W denote paper basis weight, web speed and web width, respectively. $\dot{m}_w$ also is the mass flow of water content in the paper through drying section and can be calculated as follows:

$$\dot{m}_w = \dot{m}_f \left(\frac{100 - D_p}{D_p}\right) \tag{11}$$

Q_N represents the required energy for evaporation and can be determined as follows:

$$Q_N = \dot{m}_{ev} Q_{ev} = \dot{m}_{ev} (\lambda + Q_s) \tag{12}$$

here Q_{ev} is the total required heat for the evaporation of moisture from paper which is the sum of the latent heat (λ) and the heat of sorption (Q_s).

Q_c represents the energy of condensate out of the dryer cylinders:

$$Q_c = (1-bts)\, \dot{m}_{sc}\, h_c + bts\, \dot{m}_{sc}\, h_s = \dot{m}_{sc}\, [h_c + bts\,(h_s - h_c)] \tag{13}$$

where h_s and h_c show the enthalpy of steam and condensate inside the cylinders, respectively. Parameter bts is ‘blow-through steam’ and it is defined as the percentage of total steam entering the dryer cylinders which is never condensed and leaves the dryer mixed with condensate as two-phase flow [3]. It depends on the type and size of siphon and differential pressure required to obtain continuous evacuation of condensate through the siphon [15]. The machine has rotary siphon and the inner diameter of siphon pipes is about 1 inch. The average differential pressure between the steam and condensate was about 0.5 bar, so the average blow through steam of the cylinders was assumed to about 12%.

3.2. Hood Block

As shown in the fig. 4, the humid air from the paper drying block and leakage air from the machine room mix together and form the hood exhaust air to the heat recovery block.

$$\dot{m}_{exh} = \dot{m}_{\sup} + \dot{m}_{ev} + \dot{m}_{leak} = \dot{m}_{pa} + \dot{m}_{leak} \tag{14}$$

here $\dot{m}_{exh}$, $\dot{m}_{\sup}$ and $\dot{m}_{leak}$ are the mass flow of humid exhaust, supply and leakage air, respectively.

The dry air mass balance in the drying section can be written as:

$$\dot{m}_{ea} = \dot{m}_{sa} + \dot{m}_{la} \tag{15}$$

here $\dot{m}_{sa}$, $\dot{m}_{ea}$ and $\dot{m}_{la}$ are the dry supply air, dry exhaust air and dry leakage air mass flow, respectively. The heat loss due to radiation and natural convection to the machine room can be considered as output energy. It is assumed that the hood has good conditions, proper insulation and ventilation. Moreover calculations show that the heat loss of hood is negligible [13], and this is not taken into account in the modeling. Then the energy balance for heat block can be written as follows:

$$Q_{ea} = \sum_{pd=1}^{35} (Q_{sa,pd} + Q_{N,pd}) + Q_{la} = Q_{pa} + Q_{la} \tag{16}$$

Q_{pa} is the energy of humid air from the pocket dryers and Q_N represents required energy for evaporation. In general, energy amount of air having the certain humidity can be computed as:

$$Q_a = \dot{m}_a\,[(C_a + X_a C_v)T_a + \lambda X_a] \tag{17}$$

here X_a is the air absolute humidity and C_v denotes the water vapor heat capacity. To maintain a good energy saving in the dryer section and to obtain proper operating conditions in the machine room, it is necessary that the amount of hood exhaust and supply air to be in a correct ratio. The hood balance is the ratio of the supply air to the exhaust air [3]. It can be defined as:

$$HB = \frac{\dot{m}_{sa}}{\dot{m}_{ea}} \tag{18}$$

HB represents the hood air balance. The hood balance is typically about 60–70% in the older hoods and as much as 80% in the newer structures for closed hoods [6]. In the considered machine the upper and lower bounds for hood balance can be defined as:

$$0.7 \le HB \le 0.8 \tag{19}$$

The leakage coefficient which is used to keep the hood air pressure in a proper zero level can be expressed as:

$$LC = \frac{\dot{m}_{la}}{\dot{m}_{sa}} \tag{20}$$

For a closed hood [7]:

$$0.25 \le LC \le 0.35 \tag{21}$$

The hood balance and leakage coefficient of the studied machine are 0.76 and 0.31, respectively. The poor in tightness or condition of the hood causes condensation, a phenomenon which paper producers want to avoid. In this case reducing the air humidity level due to increasing the amount of exhaust air prevents condensation inside the hood. Meanwhile excessive amount of exhaust air and lower exhaust air humidity unnecessarily increases the energy consumption. The evaporated water increases the hood humidity; consequently the humidity of exhaust air can be calculated as follows:

$$X_{ea} = \frac{\dot{m}_{ev}}{\dot{m}_{ea}} + X_{sa} = \frac{\dot{m}_{ev} \times HB}{\dot{m}_{sa}} + X_{sa} \tag{22}$$

here X_{sa} is the supply air absolute humidity and X_{ea} represents the absolute humidity of the exhaust air and it can be computed as follows, too [16]:

$$X_{ea} = \frac{\sum_{pd=1}^{35} (X_{pd}\, \dot{m}_{sa}) + X_{sa}(\dot{m}_{sa} + \dot{m}_{la})}{\dot{m}_{ea}} \tag{23}$$

where X_{pd} is the humidity level of the surrounding air in the pocket drying zone. So the highest allowable level of the exhaust air humidity can be estimated by following relation [16]:

$$X_{ea,\max} = HB \times X_{pd,\max} + X_{sa} \tag{24}$$

$X_{pd,\max}$ is the maximum humidity of the pocket drying air that it should not exceed 250 $gH_2O(kg\ dry\ air)^{-1}$ in double-felted areas [6]. Ambient temperature and relative humidity variation for Sari city was evaluated during a year. The annual environment temperature ranged from -4 to 40 ^oC and its average was 18 ^oC. The maximum, minimum and average values of the relative humidity were 60, 94 and 78%, respectively [13]. So the lower and upper bounds of absolute humidity can be determined as follows:

$$0.06 \le X_{sa,i} \le 0.038 \tag{25}$$

So the maximum level of the exhaust air humidity can be determined to be about 0.175 $kgH_2O(kg\ dry\ air)^{-1}$. According to the Eq. 6, the lower bound of the evaporation rate is 4.1 kgs^{-1}. Moreover as represented in the design documents, the minimum humidity of the hood air for original evaporation level (4.17 kgs^{-1}) has been set to about 0.136 $kgH_2O(kg\ dry\ air)^{-1}$ which can be defined as minimum humidity of the exhaust air in the present study. then:

$$0.136 \le X_{ea} \le 0.175 \tag{26}$$

Also temperature of hood exhaust air can be calculated as:

$$T_{ea} = \left(\frac{Q_{ev}}{\dot{m}_{ea}} - \lambda X_{sa}\right) / (C_{ea} + X_{ea} C_v) \tag{27}$$

where T_{ea} is the hood exhaust air temperature. The supply air mass flow also can be obtained according to the Eq. 22 and based on the limits of X_{ea}, HB, $\dot{m}_{ev}$ and X_{sa}.

Other important variable in the paper drying process is the dew point. At temperatures above the dew point, no condensation occurs within the hood. In higher dew point temperatures, the drying air can contain more water vapor and less drying air is required to remove evaporated water, accordingly more energy can be recovered from the exhaust air [2]. However regarding to the limitation of the evaporation rate and the maximum allowable range of the exhaust air humidity, unnecessary rising the exhaust air temperature increases the energy consumption in the drying section and as a result the optimized performance of heat recovery system cannot be attained. Thus the hood air temperature should be set as high as to obtain maximum heat recovery and avoid condensation in the pocket drying and hood exhaust air. For a typical condition of the drying section the bounds for hood air temperature are:

$$T_d < T_{ea} \le 95 \tag{28}$$

where T_d and T_{ea} represent dew point and exhaust air temperatures, respectively.

3.3. Heat Recovery Block

The energy of hood exhaust air can be recovered in order to preheat supply air which can reduce the steam use in the air heating. The heat balance for the heat recovery block is presented as follows:

$$Q_{sa,i} + Q_{ea,i} = Q_{ea,o} + Q_{sa,rec} \quad (29)$$

where $Q_{sa,i}$ and $Q_{ea,i}$ are the energy of fresh supply air and the energy of exhaust air out of the hood block, respectively. $Q_{sa,rec}$ and $Q_{ea,o}$ represent the energy of preheated supply air out of the primary heat recovery system and the energy of exhaust air to the atmosphere, respectively. The supply air temperature after heat recovery $(T_{sa,rec})$ can be determined by the following equation:

$$T_{sa,rec} = \left(\frac{Q_{sa,rec}}{\dot{m}_{sa}} - \lambda X_{sa,rec}\right) / (C_{sa,rec} + X_{sa,rec} C_v) \quad (30)$$

3.4. Supply Air Heating Block

The temperature of the supply air after the heat recovery is not sufficiently high for blowing into the pocket dryers; hence the steam is used to rise the supply air temperature to the required value in the air heating system. The pressure and temperature of steam for air heating are the same as that used inside the cylinder dryers. The expressions for supply air heating block are obtained by the following equation:

$$Q_{sa,rec} + Q_{s,AH} = Q_{c,AH} + Q_{sa,o} \quad (31)$$

where $Q_{s,AH}$ and $Q_{c,AH}$ are the energy of input steam and output condensate for air heating system. To keep the supply air temperature in the required level, during the arrival in the drying pockets, the optimal temperature range for the supply air after the steam heater can be set to 100-120 °C. Higher temperatures do not give more gain [6, 17]. So the bounds for temperature of supply air out of steam heater $(T_{sa,o})$ are:

$$100 \le T_{sa,o} \le 120 \quad (32)$$

It is assumed that the steam used in the air heating system condenses totally; hence the mass flow rate of the steam $(\dot{m}_{s,AH})$ can be calculated by:

$$\dot{m}_{s,AH} = (Q_{sa,o} - Q_{sa,rec}) / (h_s - h_c) \quad (33)$$

h_s and h_c are the enthalpy of steam and condensate, respectively.

The quality parameters and physical properties of dry paper i.e. moisture, grammage, caliper, … were measured by standard methods. Operational parameters of paper machine such as machine speed, steam pressure, total steam consumption and supply air temperature were determined by local instruments and DCS. The temperature of condensate is assumed to be constant. Measuring the temperature of the paper sheet was carried out by IR-Thermometer. The dry, wet and dew temperatures of air streams were measured using the appropriate thermometers. All measuring positions are at the front side, approximately 50 cm from the machine frames towards the middle of the machine.

Finally the separate blocks and specified mathematical models as constraint functions were simultaneously applied to optimize the process parameters. To solve this model, the *Fmincon* function for constrained nonlinear programming minimization of MATLAB R2010a was used. *Fmincon* finds a constrained minimum of a scalar function of several variables starting at an initial estimate. This is generally referred to as constrained nonlinear optimization or nonlinear programming. In this optimization the following function has been used:

$$[x,fval]=fmincon(@fun,x0,Aineq,bineq,Aeq,beq) \tag{34}$$

here *fun.* shows the objective function and *x0* is the initial point for *x* which is the optimization variable. *Aineq* and *Aeq* represent the matrix for linear inequality constraints and matrix for linear equality constraints, respectively which can be defined as:

$$Aineq\ .\ x = bineq \qquad \text{and} \qquad Aeq\ .\ x \le beq \tag{35}$$

where *bineq* and *beq* represent the vector for linear inequality constraints and vector for linear equality constraints, respectively. *fval* is the function value too.

4. Results and discussion:

The modeling results of the considered drying in given conditions based on the developed model before optimization are represented in tab. 2. Also the optimized results of parameters of paper are shown in tab. 3.

Table 2: The modeling results of the considered drying section before optimization

Parameter	Value	Parameter	Value
Web inlet dryness (%)	40	Hood air humidity (kgkg^{-1})	0.192
Steam for cylinders (kgs^{-1})	6.47	Hood air temperature (^{o}C)	94
Energy for cylinders (kJs^{-1})	17889	Hood air dew point (^{o}C)	63
Supply air (kg dry air s^{-1})	23.5	Total air (kgs^{-1})	30.9
Supply air temperature after recovery (^{o}C)	64.5	Total hood energy (kJs^{-1})	17224
Supply air temperature after steam heating (^{o}C)	125	Exhaust air temperature	72
Supply air energy (kJs^{-1})	3914	Evaporation rate (kgs^{-1})	5.34
Steam for air heating (kgs^{-1})	0.7	Total steam per evaporated water (kgkg^{-1})	1.34
Energy for air heating (kJs^{-1})	1936	Total steam per production (kgkg^{-1})	1.71

Table 3: The optimized results of parameters

Parameter	Value	Parameter	Value
Web inlet dryness (%)	43	Hood air humidity (kgkg^{-1})	0.152
Steam for cylinders (kgs^{-1})	5.55	Hood air temperature (^{o}C)	82
Energy for cylinders (kJs^{-1})	15311	Hood air dew point (^{o}C)	59
Supply air (kg dry air s^{-1})	26	Total air (kgs^{-1})	34.3
Supply air temperature after recovery (^{o}C)	59	Total hood energy (kJs^{-1})	15706
Supply air temperature after steam heating (^{o}C)	100	Exhaust air temperature	68
Supply air energy (kJs^{-1})	3665	Evaporation rate (kgs^{-1})	4.68
Steam for air heating (kgs^{-1})	0.85	Total steam per evaporated water (kgkg^{-1})	1.37
Energy for air heating (kJs^{-1})	2339	Total steam per production (kgkg^{-1})	1.53

It can be seen that steam consumption and air using are reduced under the optimized conditions. As shown the steam use inside the cylinder dryers is reduced from the operating 6.47 kgs^{-1} to an average value of 5.55 kgs^{-1}, but use of air is increased from 23.5 kgs^{-1} to 26 kgs^{-1}. The air heating steam consumption increases with raising the amount of used air. This is mainly because the supply air should be heated by steam before entering the pocket dryers, accordingly the required steam in the air heating system, after optimization was raised from 0.7 kgs^{-1} to 0.85 kgs^{-1}. Therefore total steam consumed in the drying section can be reduced from 25.8 ton h^{-1} to around 23 $tonh^{-1}$. In general, total steam consumption per paper production can be decreased from about 1.71 to 1.53 ton steam per ton produced paper due to the optimization. The humidity of the exhaust air was set as high as possible to optimize the performance of heat recovery system as pocket drying air humidity were kept under the allowable level to improve the drying performance. For the studied corrugating machine with an annual production of 100,000 tons, the amount of saturated steam with pressure 6.5 bars that can be saved is about 11% or 18,000 ton per year. The cost of steam generation in Mazandaran Wood and Paper Industries (MWPI) in 2015 was about 4.2 $ per ton. Therefor for the studied machine, total saving in the dryer section is around 75,000 dollars each year. However with an increasing in air use the electricity consumption of centrifugal blower will be increased which corresponds to utmost about 6,500 dollars per year.

5. Conclusions

The purpose of heat recovery in the dryer section is to reduce the energy consumption of a paper machine as economically as possible. In this paper by using simulation program, the effect of different conditions on the heat recovery performance of multi-cylinder dryer section was investigated and the optimized model was applied for fluting machine dryer section. The dryer section was divided into different function blocks. The nonlinear programming (NLP) problem with constraints including the mass and energy balance of each block was developed to optimize process parameters and reduce energy consumption rate. As the results show, total steam consumption can be decreased from about 1.71 to 1.53 ton steam per ton produced paper due to optimization which can save about 11% steam. This corresponds to about 75,000 dollars each year. However the supply air should be increased to attain optimal humidity level of pocket dryers which can lead to improvement of drying performance, though it causes about 6,500 dollars rising in cost of extra electricity consumption annually.

6. Nomenclature

bts	*- blow through steam, [%]*
C	*-heat capacity* [$kJkg^{-1}K^{-1}$]
D	*- web dryness, [%]*
G	*- grammage or basis weight,* [gm^{-2}]
h	*- enthalpy,* [$kJkg^{-1}$]
HB	*- hood balance, [%]*
LC	*- leakage coefficient, [%]*
$\dot{m}_{st}$	*- mass flow rate,* [kgs^{-1}]
$\dot{m}_{ev}$	*-evaporation rate* [kgs^{-1}]
Q	*- energy, [kW]*

Subscripts

a	*- air*
c	*-condensate*
ea	*-exhaust air*
ev	*evaporation*
f	*-fiber*
i	*-input*
la	*-leakage air*
o	*-output*
p	*-paper*

Q_{ev}	- *total required heat for evaporation [kW]*	*pa*	*-pocket drying air*
Q_N	- *evaporation heat, [kW]*	*pd*	*-pocket dryer*
Q_s	- *sorption heat, [kW]*	*rec*	*-recovery*
T	- *temperature, [C]*	*sa*	*-supply air*
T_d	- *dew temperature, [C]*	*s,AH*	*-steam for air heater*
V	- *web speed,* $[ms^{-1}]$	*sc*	*-steam for cylinder*
W	- *web width, [m]*	*st*	*-steam*
X	- *moisture ratio,* $[kgH_2O(kg\ dry\ air)^{-1}]$	*w*	*-water*

Greek letters

λ - *latent heat,* $[kJkg^{-1}]$

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