# THE EFFECT OF THE HEAT RECOVERY ON FUEL CONSUMPTION IN THE STENTER MACHINE

by

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*In this study, the convective drying process on the stenter machine used natural gas as fuel was examined in terms of energy saving. The fuel consumption in the drying process of wool fabric with and without heat recovery from exhaust air of stenter was determined. It was shown that the fuel consumption in stenter can be decreased significantly by the use of heat recovery exchangers. According to the results minimum fuel consumption was observed at inlet drying air temperature of 180 °C and humidity ratio of the exhaust air of 0.08* $kg_v/kg_a$. *In this condition fuel consumption decreased from 205.3* $m^3/h$ *to 177.3* $m^3/h$ *by using heat recovery exchanger. The fuel consumption of drying systems reduced by 13.6%.*

Key words: *stenter, drying, heat recovery system*

## Introduction

Stenter machine is a drying machine which works according to convection drying method. They can be classified based on their air heating systems. The heating of the air can be made by indirect steam, hot oil and direct gas-fired systems. One of the most efficient heating is ensured by direct gas firing [1]. In this drying method hot air is injected directly on the upper and lower surfaces of the fabric by a series of nozzles strategically positioned to remove the moisture by the principle of forced convection. The humid air formed by the drying process, is removed and carried by the drying air towards the exhaust system, which prevents the formation of excess moisture near the fabric. The stenter requires high energy consumption for air heating and subsequent the moisture removes from the fabric as vapor [2]. Therefore, the understanding of drying process is significant to reduce energy consumption.

The numerous studies on the thermodynamic analysis of drying process have recently been undertaken by some researchers. Of these, Johann *et al.* [3] investigated the model for drying process of textile products. A mathematical model was created by using energy and mass balance. The temperature and humidity of the textile product were used instead of the unknown values in the partial differential equation system. The model results were compared with literature data.

Santos *et al.* [2] applied heat and mass transfer in a porous media model using the finite element method to simulate the drying of cotton fabric in a stenter. The results of the simulations show that significant reductions in energy costs can be accomplished when the fabric

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and the drying air are preheated using the exhaust air. The influence of ambient conditions on the performance of the stenter was also investigated. It was found that, in warmer seasons, the stenter can be operated with less fuel consumption.

Dincer and Sahin [4] investigated a new model for thermodynamic analysis, in terms of exergy, of a drying process. Exergy efficiencies are derived as functions of heat and mass transfer parameters. An illustrative example was considered to verify the present model and to illustrate the applicability of the model to actual drying processes at different drying air temperatures, specific exergies of drying air, exergy differences of inlet and outlet products, product weights, moisture contents of drying air, and humidity ratios of drying air. As a result, this work is intended not only to demonstrate the usefulness of exergy analysis in thermodynamic assessments of drying processes, but also to provide insights into their performances and efficiencies.

Xue [5] investigated the carpet manufacturing is an energy-intensive process, therefore, it is vital that the resource is used efficiently in order to reduce costs and keep the carpet industry competitive. Drying is an intermediate step in the dyeing and finishing operations in the textile industry and uses a substantial amount of energy. About one half of the energy used in a typical textile finishing mill is used in the drying processes, due to the increasing fuel cost, it is imperative that the dryers be operated as efficiently as possible. The use of excess air is a major source of energy losses in a dryer operation.

Cay *et al.*[1] revealed energetic and exergetic analysis of a stenter system in a textile finishing factory based on actual operational data. It was concluded that nearly 40% of the input energy (and 13% of the input exergy) is exhausted to the environment by humid exhaust air of the stenter.

Cay *et al.* [6] examined a new model for the exergetic analysis of the convective drying of textiles at stenters. They were examined the variations of exergetic parameters for each chamber of the stenter. It was emphasized that the exergy efficiency of the last chambers decreased drastically due to the lower evaporation rate of the falling rate period of drying. Additionally, the subsystems of the chambers were analyzed. For this purpose, the detailed control volume models were conducted for the stenters. The exhaust air was the major source of waste heat and provided a significant energy saving potential.

Ceylan and Gelir [7] designed a shell and tube heat recovery exchanger to obtain energy saving from exhaust hot air which is released from the stenter machine chimney. In this study, it was determined the pay-back period as 12.84 months in Turkey. Fiaschi *et al*. [8] presented the improvement of low temperature exhaust heat recovery network of an industrial textile-drying machine (Stenter-Rameuse). The overall heat recovered with the five different analysed fin configurations range from 53-97%.

In this paper, it was studied the drying of a woven fabric in a stenter considering the consumption of natural gas. Energy analysis of the stenter system was done based on actual operational data of a textile finishing factory. The fuel saving was determined considering the heat recovery from exhaust air.

## Material and methods

### *Specifications stenter machine and fabric*

The Bruckner model machine shown in fig. 1 has a length of 9 m and 6 chambers. Each chamber has hot air nozzles which distribute the air on the top and bottom of the fabric. Drying process occurs between 130-150 °C in the upper cabins and fixing process occurs between 180-195 °C in the bottom cabins of the stenter machine. The natural gas is burned in the chamber to heat drying air.

The fabric used as the drying material is wool. The values of operating parameters provided by firm for the real process are given in tab. 1. Fabric mass-flow rate was calculated as 0.17 kg/s using the fabric specifications shown in tab. 1.

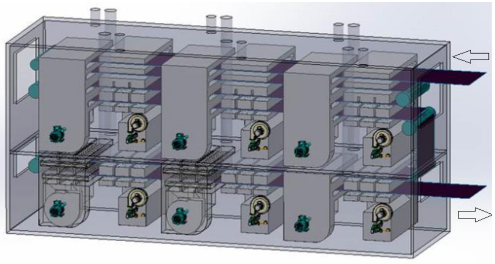

**Figure 1. The internal structure of the cabins in the stenter machine**

**Table 1. The fabric specifications**

| | |
|---|---|
| Fabric specific heat [$Jg^{-1}K^{-1}$] | 1.36 |
| Fabric velocity [$mmin^{-1}$] | 30 |
| Mass per unit area of the fabric [$gm^2$] | 230 |
| Fabric width [m] | 1.5 |
| Fabric thickness [mm] | 1 |
| Initial temperature of the fabric [°C] | 35 |
| Final temperature of the fabric [°C] | 180 |
| Initial moisture content of the fabric [$kg_{H_2O}kg_a^{-1}$] | 0.635 |
| Final moisture content of the fabric [$kg_{H_2O}kg_a^{-1}$] | 0.04 |

*Heat recovery exchanger*

In an earlier paper by the authors [7], a heat recovery exchanger was designed considering the reuse of exhaust air at an average temperature of 150 °C to preheat the fabric and the drying air. The shell and tube heat exchanger was preferred because of that they are much less susceptible to fouling than plate heat exchangers. The tubes were also arranged in staggared configuration considering fouling. The fresh air from the factory environment passes of tube side of the heat exchanger and the hot humid air from the drying machine passes through shell side. The technical drawing of the designed exchanger was shown in fig. 2.

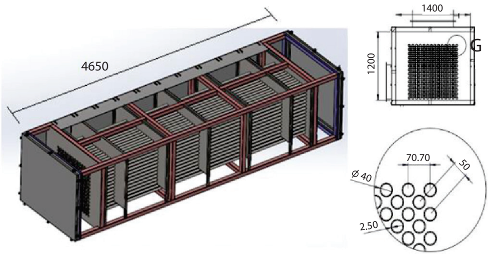


**Figure 2. The drawing of the exchanger**

**Table 2. The technical specifications of the heat exchanger**

| | |
|---|---|
| Pipe inner diameter [m] | 0.036 |
| Pipe outside diameter [m] | 0.04 |
| Pipe thermal conductivity (stainless steel) [$Wm^{-1}K^{-1}$] | 16.2 |
| Number of pipes [piece] | 365 |
| Length of pipe [m] | 4.65 |
| Baffle spacing [m] | 1.16 |
| Exhaust gas inlet temperature [°C] | 150 |
| Exhaust gas outlet temperature [°C] | 91.94 |
| Fresh air inlet temperature [°C] | 30 |
| Fresh air outlet temperature [°C] | 88.28 |
| Air-flow rate [$m^3h^{-1}$] | 15000 |

The logarithmic mean temperature difference method was used to find the air outlet temperatures from the exchanger and the Bell-Deleware method was used to calculate the shell side pressure loss. The baffle spacing which affects both the outlet temperatures and pressure loss was determined by economic analysis. The technical specifications of the designed exchanger are given in tab. 2. Before the fresh air enters the stenter, it is heated up from 30-88.28 °C by exhaust air in heat recovery exchanger. The exhaust gas cools from 150-91.94 °C. There is no condensation in the heat recovery exchanger because of that the exhaust gas outlet temperature of 91.94 °C is higher than its dew point temperature.

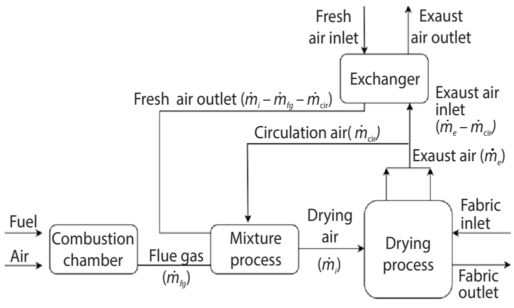


**Figure 3. The schematic of drying system**

### *Thermodynamic analysis*

The heat that is transferred from the drying air to the product in stenter is used to raise the temperature of the product as well as to evaporate moisture from it. It is necessary to quantify this heat determined the air-flow rate and fuel consumption. Therefore, we applied mass and energy balance equations based on the First law of thermodynamics for combustion chamber, mixture process and drying process shown in fig. 3.

### *Combustion chamber*

The content of natural gas that is burned to heat the fabric and the drying air was determined based on flue gas analyzer of the firm. Combustion equation with a 10% excess air was defined:

$$\begin{aligned}&0.86\mathrm{CH_4}+0.05\mathrm{C_2H_6}+0.02\mathrm{C_3H_8}+0.015\mathrm{C_4H_{10}}+0.03\mathrm{N_2}+0.025\mathrm{CO_2}+\\&+1.1\left[2.092\left(\mathrm{O_2}+3.76\mathrm{N_2}+0.093\mathrm{H_2O}\right)\right]\rightarrow 1.105\mathrm{CO_2}+2.239\mathrm{H_2O}+8.68\mathrm{N_2}+0.21\mathrm{O_2}\end{aligned} \tag{1}$$

Moisture content the flue gases is evaluated:

$$w_{\mathrm{fg}}=\frac{\dot{m}_{\mathrm{H_2O}}}{\dot{m}_{\mathrm{d_{fg}}}} \tag{2}$$

where $\dot{m}_{\mathrm{dfg}}$ [kg(kmol$_{\mathrm{fuel}}$)$^{-1}$] is total mass-flow rate of dry flue gases:

$$\dot{m}_{\mathrm{d_{fg}}}=\sum \dot{n}_{\mathrm{i}} M_{\mathrm{i}} \tag{3}$$

Enthalpy of the flue gases on a dry basis is evaluated:

$$H_{\mathrm{d_{fg}}}=\sum n_{\mathrm{i}} \bar{h}_{\mathrm{i}} \tag{4}$$

$$\bar{h}_{\mathrm{i}}=\bar{h}_{T,\mathrm{fg}}-\bar{h}_{273} \tag{5}$$

The values of the combustion analysis were shown in tab. 3.

**Table 3. The values of the combustion analysis**

| Flue gas temperature [K] | 1900 |
|---|---|
| Enthalpy of flue gas on dry base [kJkg$^{-1}$] | 1884 |
| Total mass of dry flue gas [kgkmolfuel$^{-1}$] | 298.4 |

Adiabatic flame temperature was determined as 2160 K considering that combustion air temperature is 30 °C, relative humidity is 46% and pressure is 100 kPa. But it was evaluated as 1900 °C based on heat loss from combustion chamber.

### *Adiabatic mixing process*

A major portion of the exhaust air is recirculated for preheating the drying air as well as the fabric entering the dryer. In adiabatic mixing process, the combustion gas from the burner is mixed with recirculating air and fresh air. The mixed gas as jetted over the fabric is evaporated water from it. Drying process occurs between 130-150 °C in the upper cabins and fixing process occurs between 180-195 °C in the bottom cabins of the stenter machine. It was assumed as average value for simplicity.

The mass-flow rate and absolute humidity of the drying air supplied to the system on a dry basis are evaluated:

$$\dot{m}_\mathrm{i} = \dot{m}_{\mathrm{d}_\mathrm{fg}} + \dot{m}_\mathrm{cir} + \dot{m}_\mathrm{fa} \tag{6}$$

$$w_\mathrm{i} = zw_\mathrm{fg} + (1 - p - z)w_\mathrm{fa} + pw_\mathrm{cir} \tag{7}$$

The energy balance equation based on the first law of thermodynamics can be written for the adiabatic mixing process:

$$\dot{m}_\mathrm{i}h_\mathrm{i} = \dot{m}_\mathrm{cir}h_\mathrm{cir} + \dot{m}_\mathrm{fg}h_\mathrm{fg} + \dot{m}_\mathrm{fa}h_\mathrm{fa} \tag{8}$$

$$h_\mathrm{i} = ph_\mathrm{cir} + zh_\mathrm{fg} + \left(1 - p - z\right)h_\mathrm{fa} \tag{9}$$

Enthalpies of moisture air ($h_\mathrm{fa}$, $h_\mathrm{cir}$, and $h_\mathrm{i}$) are defined:

$$h = h_\mathrm{a} + wh_\mathrm{v} = c_pT + wh_g \tag{10}$$

where $z$, $p$ and $(1 - p - z)$ are the ratio of the mass-flow rate of flue gas, re-circulation air, and fresh air to the mass-flow rate of supply air based on dry basis, $\dot{m}$.

### *Drying process*

The hot air entered to stenter is contact with fabric. It gives heat energy to fabric and taken out of water vapor from fabric. A major portion of the exhaust air that contains much amount of humidity is recirculated while the remaining small portion is exhausted out.

In order to write the mass balance equations for the drying process shown in fig. 4, three components such as product itself, dry air and the water contained in the drying air and product were considered. Therefore, the mass balance equations are written for these three elements:

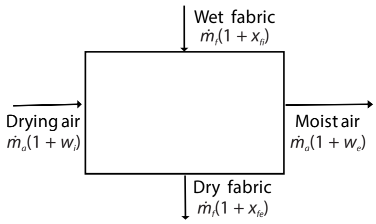


**Figure 4. The schematic of drying process**

$$(\dot{m}_f)_\mathrm{wet} = (\dot{m}_f)_\mathrm{dry} = \dot{m}_f \text{ (fabric)} \tag{11}$$

$$(\dot{m}_\mathrm{a})_\mathrm{dry} = (\dot{m}_\mathrm{a})_\mathrm{moist} = \dot{m}_\mathrm{a} \text{ (air)} \tag{12}$$

$$w_\mathrm{i}\dot{m}_\mathrm{a} + (\dot{m}_w)_\mathrm{wet} = w_\mathrm{e}\dot{m}_\mathrm{a} + (\dot{m}_w)_\mathrm{dry} \text{ (water)} \tag{13}$$

For drying process, the energy balance equation based on the First law of thermodynamics:

$$\begin{aligned}\dot{m}_f[c_{p,f}(T_\mathrm{fe} - T_\mathrm{fi}) + x_\mathrm{fe}(h_{f,\mathrm{fe}} - h_{f,\mathrm{fi}}) + \dot{m}_\mathrm{v}(h_{g,\mathrm{e}} - h_{g,\mathrm{fi}}) = \\ = \dot{m}_\mathrm{a}[c_{p,\mathrm{a}}(T_\mathrm{i} - T_\mathrm{e}) + w_\mathrm{i}(h_{g,i} - h_{g,\mathrm{e}})]\end{aligned} \tag{14}$$

The first term on the left hand side represents the sensible heat associated with heating the fabric from its inlet temperature to the outlet temperature. The second term is the sensible heat associated with heating the water inside wet fabric to its exit temperature. The third and last term in the aforementioned equation represents the latent heat associated with evaporating the water into the gas phase. In the equation, heat loss to the environment through the walls of the dryer was not considered. The heat loss from the dryer was calculated as 0.0027 kW/m$^2$ using the wall temperature of machine measured as average 50 °C.

Enthalpy of fabric is defined:

$$h_f = c_{p,f} T_f + x_f h_g \tag{15}$$

Using the knowledge of the amount of water removed in the process, $\dot{m}_\mathrm{v}$, the required amount of drying air, $\dot{m}_\mathrm{a}$, was calculated by mass balance:

$$\dot{m}_\mathrm{v} = \dot{m}_f (x_\mathrm{fi} - x_\mathrm{fe}) = \dot{m}_\mathrm{a} (w_\mathrm{i} - w_\mathrm{e}) \tag{16}$$

The air temperatures in the adiabatic mixing process were shown in tab. 4.

**Table 4. The air temperatures in the adiabatic mixing process**

| | |
|---|---|
| Exit temperature of drying cabins [°C] | $T_\mathrm{e} = 150$ |
| Inlet temperature of drying cabins [°C] | $T_\mathrm{i} = 160\text{-}180$ |
| Circulation temperature of drying cabins [°C] | $T_\mathrm{cir} = T_\mathrm{e}^{-5}$ |

### *Fuel consumption*

Fuel (natural gas) volumetric flow rate was determined with ideal gas equation at 100 kPa pressure and 25 °C temperature, R = 8.315 kJ/kgK:

$$\dot{V}_\mathrm{fuel} = \dot{n}_\mathrm{fuel} \frac{\mathrm{R}T}{P} \ [m^3 h^{-1}] \tag{17}$$

where

$$\dot{n}_\mathrm{fuel} = \frac{\dot{m}_{\mathrm{d}_\mathrm{fg}}}{m_{\mathrm{d}_\mathrm{fg}}} \tag{18}$$

$$\dot{m}_{\mathrm{d}_\mathrm{fg}} = z \dot{m}_s \ [\mathrm{kg}_{\mathrm{d}_\mathrm{fg}} \mathrm{h}^{-1}] \tag{19}$$

$$m_{\mathrm{d}_\mathrm{fg}} = 293.44 \frac{\mathrm{kg}_{\mathrm{d}_\mathrm{fg}}}{\mathrm{kmol}_\mathrm{fuel}} \tag{20}$$

## Results and discussion

There are many parameters affecting the fuel consumptıon in stenter such as drying air temperature, the drying air mass-flow rate, the humidity ratio of the exhaust air.

Figures 5 and 6 show the change of drying air mass-flow rate and fuel consumption, respectively, with exhaust moisture content range between 0.05 and 0.08 kg$_\mathrm{v}$/kg$_\mathrm{a}$ for different inlet air temperature. The exhaust moisture of 0.08 kg$_\mathrm{v}$/kg$_\mathrm{a}$ is the appropriate value used by the company. T1, T2, and T3 are inlet drying air temperatures and their values are 160 °C, 170 °C, and 180 °C, respectively. Exhaust air temperature is 150 °C.

As shown in fig. 5, drying air mass-flow rate decreases slightly with increasing the exhaust moisture while decreases significantly with increasing air temperature. It is seen that the drying air temperature is an effective factor in the drying. In practice, the temperature increase of the drying air is not always feasible because of that the reactions involved in the heat setting process may be affected by temperature changes, resulting in a product of undesirable quality.

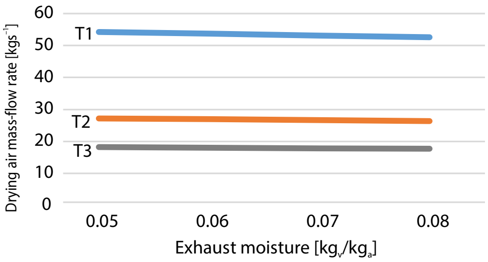


**Figure 5. The variation of drying air mass-flow rate with exhaust moisture at different drying air temperature**

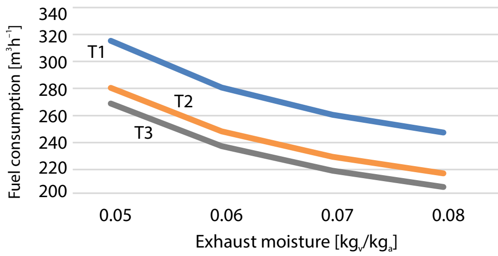


**Figure 6. The variation of fuel consumption with exhaust moisture at different inlet drying air temperature**

It is seen from fig. 6 that the increasing the exhaust air humidity ratio decreased the fuel consumption due to decreasing the mass-flow rate of drying air. Initially, the fuel consumption decreases rapidly, but later this rate declines. Because the moisture content difference between the inlet and the outlet of the drying air becomes large. The increase in the exhaust air humidity ratio to the possible highest value is a very important factor for textile drying. Moreover, the fuel consumption decreases considerably with increasing the inlet drying air temperature.

The most of the input energy is exhausted to the environment by humid exhaust air of the stenter. The heat of the humid exhaust air can be recovered using a heat exchanger to heat drying air. The preheated air by heat recovery can use to form make up air and hence energy required to heat fresh air is saved.

Figure 7 shows the variation of fuel consumption with exhaust moisture for different drying air temperature in with/without heat recovery. The behavior of the curves is similar to those shown in fig. 5. The fuel consumption decreases with increasing the inlet drying air temperature and humidity ratio of the exhaust air. TT1, TT2, and TT3 are inlet drying air temperatures at situation the heat recovery takes place and their values are 160 °C, 170 °C, and 180 °C, respectively.

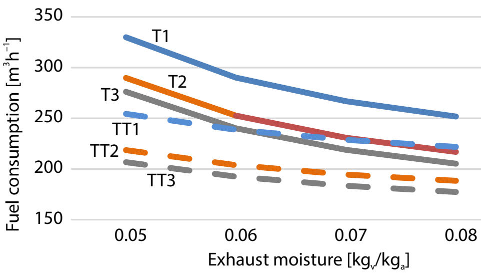


**Figure 7. The variation of fuel consumption with exhaust moisture at different drying air temperature with/without heat recovery**

The air mass-flow rate did not change after heat recovery for same condition because there was no change in the humidity of fresh air which is taken from the factory environment. As shown in fig. 7, the fuel consumption is reduced only due to heat recovery. Minimum fuel consumption was seen at inlet drying air temperature of 180 °C and exhaust moisture of 0.08 $kg_v/kg_a$. In this condition, fuel consumption was decreased from 205.3-177.3 $m^3$/h by using heat recovery exchanger and was reduced by 13.6%.

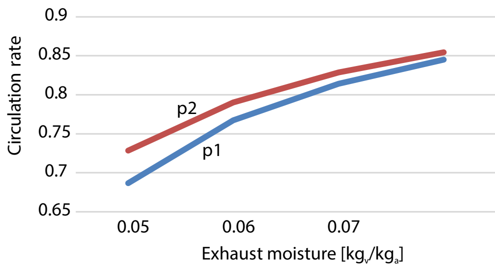


**Figure 8. The circulation rate *vs.* exhaust moisture before and after heat recovery**

Exhaust air from stenters contains a high quantity of energy in the form of heat. To improve efficiency of drying, recycling the exhausted air released away as waste energy is proposed. The recycling of outflow air is also proposed by taking into account the final product quality.

Figure 8 shows the change of circulation rate, $p$, with the exhaust moisture for inlet drying air temperature of 180 °C without/with heat recovery. The $p_1$ and $p_2$ are circulation rate without and with heat recovery, respectively. As shown in fig. 8, the circulation rate and exhaust moisture directly proportional. If the exhaust moisture increases, the air circulation rate increases. This increase slightly decreased after a certain value of the exhaust moisture.

After heat recovery, the circulation rate was higher than the situation without heat recovery at same exhaust humidities. In high humidity values, the difference between them was smaller.

## Conclusions

Energy is one of the main cost factors in the textile industry. This paper presented an analysis of the drying process in stenter in terms of fuel consumption or energy saving. The energy analysis is based on the First law of thermodynamics, which expressed the principle of the conservation of energy.

Results show that the air mass-flow rate and humidity ratio of the exhaust air were inversely proportional. The air mass-flow rate and fuel consumption were directly proportional. The effect of drying air temperature on mass-flow rate of drying air was predominant than the effect of humidity ratio of the exhaust air.

In this study, the heat recovery exchanger was applied on the stenter machine. The effect of drying air temperature, the drying air mass-flow rate and the humidity ratio of the exhaust air on change of fuel consumption were examined for the situation with and without heat recovery exchanger. It is shown that the fuel consumption of stenter can be decreased significantly by the use of heat-recovery heat exchangers. Minimum fuel consumption was observed at inlet drying air temperature of 180 °C and humidity ratio of the exhaust air of 0.08 $kg_v/kg_a$. In this condition, fuel consumption decreased from 205.3-177.3 $m^3$/h by using heat recovery exchanger and reduced by 13.6%. The air circulation rate and the exhaust air humidity were directly proportional. After heat recovery the circulation rate increased at same exit humidity.

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## Nomenclature

$H$ – enthalpy, [kJ]
$h$ – specific enthalpy, [kJkg$^{-1}$]
$h_f$ – saturated liquid specific enthalpy, [kJkg$^{-1}$]
$h_g$ – saturated vapour specific enthalpy, [kJkg$^{-1}$]
$\dot{m}$ – mass-flow rate, [kgs$^{-1}$]
$n$ – number of mole, [kmol]
$T$ – temperature, [°C]
$P$ – pressure, [kPa]
$p$ – re-circulation rate
$w$ – absolute humidity, [$kg_v(kg_a)^{-1}$]
$z$ – flue gas rate
$x$ – fabric moisture

*Subscripts*

a – air
cir – circulation
$d_{fg}$ – dry flue gas
e – exit
fa – fresh air
fg – flue gas
i – inlet
v – vapour

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