3E Analysis of a Solar Assisted Rotary Type Coal Dryer

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Abstract- In this study, a solar assisted rotary coal dryer is analyzed in terms of exergy economy and environment (3E). In the analysis two different heat sources are utilized. In the first one, a natural gas burner is used for producing the required heat for coal drying. In the second one, a parabolic solar system is employed as a heat source for drying same amount of wet coal. First of all, design parameters of a rotary coal dryer are investigated and economic operation range, the effects of some parameters are determined for the given type of coal. The effects of the dry bulb temperature at the inlet of the dryer are investigated. The second law efficiency of the dryer is calculated. For the parabolic solar system an economic analysis is carried out to estimate the payback period of the investment, which is found to be 6.6 years without heat storage. Also, by applying solar system to the coal dryer, reduced CO_2 emissions are found

Keywords- Rotary dryer, exergy analysis, 3E analysis, coal dryer, parabolic solar

1. Introduction

Coal is a valuable and abundant fuel that is used as raw material for many chemical synthesis processes and combustion systems. The calorific value of coal and its friability increases by drying. Friable coal is suitable for combustion in steam power plants and the presence of moisture in coal causes a reduction in its friability. The final moisture content requirement for coal depends on the combustion process in which it is utilized. In the briquetting process the moisture content of coal should be less than 4%, while in a low temperature carbonization process it should be nearly 0%. In a gasification process it should be in the range of 5 to 15%.

Two types of moisture can be defined for a coal particle, surface and hygroscopic. Surface moisture depends on coal washing or soaking processes. Therefore, it is easy to evaporate the surface moisture, which at a constant drying rate the evaporation of the surface moisture is the first drying period. Hygroscopic moisture depends on the rank of the coal and it decreases with its age [1].

Rotary dryers, pneumatic dryers, fluidized bed dryers etc. are generally used to dry coal particles. Rotary dryers may be classified in two groups, direct and indirect types. Direct type dryers utilize combustion gases and indirect type dryers use steam to dry the coal particles. In direct type rotary dryers combustion gases contact with the particles, while in indirect type, coal particles come in contact with the heated walls or pipes by steam. Also, rotary dryers are classified depending on the flow of combustion gases and coal particles. In a co-current rotary dryer design coal particles and combustion gases enter the dryer shell and flow in the same direction. Generally in coal drying applications co-current flow is preferred to avoid the possibility of self-ignition of coal particles.

In a direct type rotary dryer a combustion chamber produces combustion gases to dry the coal particles. Quench air is supplied to the combustion chamber to control the inlet temperatures of the combustion gases. At the exit of the dryer, cyclone collectors and fans are installed to capture dust of the coal particles. The flights are connected to the shell walls in order to lift and shower the coal particles through the stream. The rotation system controls the residence time of the coal particles spend in the dryer and therefore, the final moisture content of a particle is directly related to the peripheral speed of the shell.

An experimental and numerical study has been carried out by researchers on coal drying and rotary dryer designs. Arruda et al. [2] compared the performance of a rotary dryer to a rotor-aerated dryer. The residence time, the difference between inlet and outlet temperatures of solids and air temperature variations were compared. The residence time in the shell decreased 48% and as a result the process capacity increases when rotor-aerated dryer is preferred. Van Puyvelde [3] modeled the holdup for five different types of lifters. According to the model, simple lifters consisting of a single segment do not hold up as much material as lifters containing a number of segments, such as a hook formation. Further, the more complex lifters are able to distribute the material more evenly across the transverse section of the dryer compared to the simple lifters and this may improve the overall heat and mass transfer of the dryer. Revol [4] also modeled the flights with three segments. According to the results, theoretical equations provide estimates of the flight holdup that were accurate enough to predict the power required to lift the solids. On the other hand, the model is not accurate enough to predict the variations in the flux of solids over the drum cross-section. This shows that the angle of the solids level in a flight can be affected by the flight geometry. Lee and Sheehan [5] developed an unloading model for a generic two section flight. The model was compared to an experimental setup and according to their results the model was developed to accurately reproduce the observed unloading profiles, however the results are reported to be highly sensitive to the material surface angle used in the model. High-speed photography was used to observe the surface of the granular material during unloading and measurements of key properties were made. Observation of these high-speed images shows that the unloading of the flight is discontinuous, and that there are significant fluctuations in the behavior of the discharging solids. Kemp [6] compared the residence time models and offered a new formula which includes all possible particle motions in a rotary dryer. Margono et al. [7] analyzed the effects of feed rate and residence time in rotary dryer using steady-state and unsteady-state plug flow models. Partial differential equations describing heat and mass transfer in the rotary dryer were derived from shell balance. The results show that evaporated feed moisture content in plug flow back mixing model was lower than in plug flow model. Drying system is used in processes to obtain the required moisture content of the feed. Fagernas et al. [8] compared several types of dryers in a biomass drying process. They also investigated the environmental effects of dryers and offered different dryer types with respect to the feed characteristic and feed mass flow rate. Pinacho et al. [9] investigated the utilization of the biodegradable municipal solid wastes as animal feed. According to their results, they offer continuous type of drying systems due to economic operation and suitable microbiological characteristics of the product. Kakaras et al. [10] performed a simulation to investigate the effects of brown coal drying on a thermal power plant. Different types of dryers were compared and the thermal efficiency of the power plant was improved 5% by drying.

This study is aimed to design a direct rotary type coal dryer and perform exergy analysis as a case study. A Fortran code is developed for designing and parametric analysis to investigate the effects of variables. Apart from the literature, an alternative method, solar drying of coal with parabolic trough type collectors, is suggested. The conventional and solar drying system with parabolic trough type collectors are compared with respect to economic analysis and the effects of systems on global warming potential with CO2 emissions

2. Modeling of the dryer

A model for the overall design of rotary dryers is offered by Krokida et al. [11] based on the estimation of the drying kinetics of the material. Drying constant of the material can be determined experimentally as given in Eq. (1). Drying constant, km, depends on the temperature, absolute humidity and gas velocity of the drying air. A simplified diagram of the dryer and burner is given in Fig. 1.

$$k_m = f\left(T_{da}, Y_{da}, u_{da}\right) \tag{1}$$

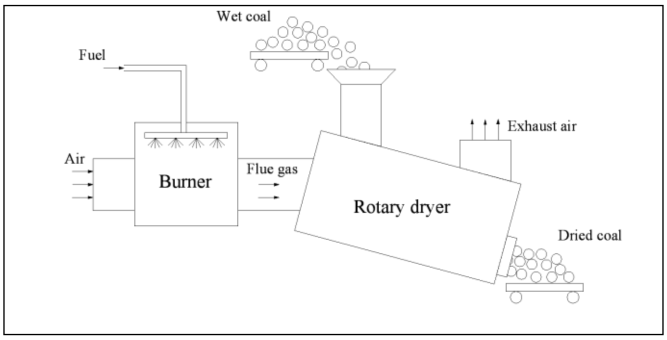

Fig. 1. A simplified diagram of the coal drying unit

In the burner section of the drying unit natural gas is burned to produce combustion gases. According to the combustion reactions, 9 kg of water vapor are produced per kg of combusted hydrogen. Thus, water vapor production can be calculated from Eq. (4), where Pw, mf and αH are the production rate of water vapor, mass flow rate of fuel and hydrogen mass fraction by weight in the fuel, respectively.

$$C + \frac{1}{2}O_2 \rightarrow CO_2 \tag{2}$$

$$H_2 + \frac{1}{2}O_2 \rightarrow H_2O \tag{3}$$

$$P_w = 9\alpha_H \dot{m}_f \tag{4}$$

The total moisture balance for the burner can be written as follows.

$$\dot{m}_g\left(1+Y_g\right) = \dot{m}_{air}\left(1+Y_0\right) + \dot{m}_f \tag{5}$$

$$\dot{m}_g Y_g = \dot{m}_{air} Y_0 + P_w \tag{6}$$

where; mair and mg are the inlet and outlet mass flow rates of gases at the burner and Y0 and Yg are the inlet and outlet humidity of the gases at the burner, respectively. Energy balance over the dryer can be given as follows.

$$\dot{m}_g\left(1+Y_g\right)Cp_g\left(T_g - T_0\right) = \dot{m}_f \, \Delta H_f \tag{7}$$

where; Cpg is the specific heat of the gases and ΔHf is the heat of combustion. Mass and energy balance at dryer section is given in Eq. (8) and (9).

$$\dot{m}_g\left(Y_{ge}-Y_g\right)=\dot{m}_{ds}\left(X_i-X_e\right) \tag{8}$$

$$m_g Cp_g\left(1-Y_g\right)\left(T_g-T_{ge}\right)+m_{ds}\Delta H_v\left(X_i-X_e\right) \tag{9}$$

where; ΔH_v is the latent heat of vaporization of water and Tge is the mean air-vapor temperature at the dryer outlet. In a drying process heat is supplied to the material to be dried, and as a result of mass transfer the moisture content of the material decreases. Therefore, drying process is the combination of heat and mass transfer. The amount of the heat transfer in direct type rotary dryer is expressed in Eq. (10).

$$Q=U_v aV(\Delta T)_m \tag{10}$$

where; Q, is the rate of heat transfer [W], $U_v a$, is the volumetric heat transfer coefficient [W/m3K], V, is the dryer volume [m3] and ΔT_m is the mean temperature difference between the hot gases and the material. ΔT_m can be found by wet bulb depressions of the drying air at the inlet and outlet of the dryer. Wet bulb depression is the difference of dry bulb gas temperature and wet bulb gas temperature, as given in Eq. (11).

$$T_{gdb}-T_{gwb}=\frac{K\Delta H_v}{h_c+h_R}\left(Y_W-Y_G\right) \tag{11}$$

where; K is the mass transfer coefficient, ΔHv is the latent heat of vaporization at T_{wb}, h_c is the convective heat transfer coefficient, hR is the radiation heat transfer coefficient and Y is the humidity. Radiative heat transfer coefficient can be ignored and hc/K can be assumed as 29.08 J/mol°C for pure air.

In the case of economic operation of rotary dryers a dimensionless number, number of heat transfer units, is used in Eq. (12).

$$N_t=\frac{\left(T_1-T_2\right)}{\Delta T_m} \tag{12}$$

where; T_1 is the inlet gas temperature [K] and T_2 is the outlet gas temperature [K]. The most economical operation of rotary dryers can be achieved for Nt in the range of 1.5-2.5. Another important ratio is the length to diameter ratio, which is in the range of 4 to 10 for industrial dryers.

$$4<L/D<10 \tag{13}$$

During the operation the total volume of the dryer should be filled with the material in the range of 10 to 15% of total volume. Lower filling, causes uneconomic operation and greater filling causes higher moisture content in the product. As a result, the quality of the product material decreases. The power required to drive a rotary dryer may be calculated by Eq.(14).

$$P=\frac{\omega\left(4.75D.W_m+0.1925D^{'}W_t+0.33W_t\right)}{100000} \tag{14}$$

$$D^{'}=D+2 \tag{15}$$

where; P is the power required [brake horse power], ω is the rotational speed [r/min], D is the shell diameter [ft], Wm is the material weight [lb], Wt is the total weight [lb] and D' is the riding ring diameter [ft].

According to the expressions above a direct type rotary dryer for anthracite coal is to be designed. Counter current flow type direct rotary dryer is selected to dry the anthracite coal. Coal feed rate to the dryer is 12 t/h and initial humidity ratio of coal is given to be 20%. The humidity ratio in the product is to be 1%. In Table 1 inlet and outlet temperatures of air and coal are given. A direct type rotary dryer design is carried out according to the given parameters in Table 1. A parametric study is performed and the effects of variables are provided.

Table 1. Thermodynamic properties of air and coal in the different points

Characteristic	Value
Inlet temperature of air to dryer [°C]	200
Outlet temperature of air from dryer [°C]	90
Inlet air wet bulb temperature [°C]	77
Coal inlet temperature [°C]	25
Coal outlet temperature [°C]	75
Air mass velocity [$kgm^{-1}s^{-2}$]	3.7
Residence time [h]	0.3

3. Exergy analysis

The exergy analysis of a coal dryer is an important point to determine the dimensions and operational processes. The exergy of a system is defined as the maximum available work that can be done by the system-environment combination. Higher value of exergy means the higher value of the obtainable work. The exergy analysis is the combination of the first and second laws of thermodynamics. In this analysis heat does not have the same value as work, and exergy loss represents real loss of work. When analyzing novel and complex thermal systems, experience needs to be supplemented by more rigorous quantitative analytical tools. Exergy analysis provides those tools and it helps in locating weak spots in a process. This analysis provides a quantitative measure of the quality of the energy in terms of its ability to perform work and leads to a more rational use of energy [12].

Conservation of mass, conservation of energy and exergy destruction equations for any control volume at steady state with ignorable kinetic and potential energy changes can be written as in Eqs. 16-18. by ignoring the potential and kinetic exergy changes of the system, Equation (18) defines total exergy rate. In Eq. 18, subscript "0" indicates the reference conditions, which are, in this study, ambient temperature as 25ºC and atmospheric pressure as 101.32 kPa, respectively.

$$\sum_i \dot{m}_i=\sum_e \dot{m}_e \tag{16}$$

$$\dot{Q}_j - \left(\dot{W}_{cv}\right) = \sum_e \dot{m}_e h_e - \sum_i \dot{m}_i h_i \tag{17}$$

$$\dot{E}_D = \sum_j \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_j - \left(\dot{W}_{cv}\right) + \sum_i \dot{E}_i - \sum_e \dot{E}_e \tag{18}$$

$$\dot{E} = \dot{m}\left[h - h_0 - T_0\left(s - s_0\right) + \bar{e}_{CH}\right] \tag{19}$$

Another parameter in exergy analysis of a system is determining the exergy efficiency, which is the percentage of the fuel exergy provided to a system over exergy of the product. Moreover, the difference between 100% and the actual value of the exergy efficiency, expressed as a percent, is the percentage of the fuel exergy wasted in this system as exergy destruction and exergy loss [12].

$$\eta_{II} = \frac{\dot{E}_P}{\dot{E}_F} \tag{20}$$

In the exergy analysis of a dryer the physical expressions should be modified to dryer and also psychometric calculations have to be carried out. In the following equations these modified expressions are given.

$$ex_1 = [(C_P)_a + \omega_1 (C_P)_v](T_1 - T_0) - T_0\left\{[(C_P)_a + \omega_1 (C_P)_v]\ln\left(\frac{T_1}{T_0}\right) - (R_a + \omega_1 (R_v)\ln\left(\frac{P_1}{P_0}\right)\right\}$$
$$+ T_0\left\{(R_P)_a + \omega_1 (R_v)\ln\left(\frac{1+1.6078\omega^0}{1+1.6078\omega_1}\right) + 1.6078\omega_1 R_a \ln\left(\frac{\omega_1}{\omega^0}\right)\right\} \tag{21}$$

$$ex_3 = [(C_P)_a + \omega_3 (C_P)_v](T_3 - T_0) - T_0\left\{[(C_P)_a + \omega_3 (C_P)_v]\ln\left(\frac{T_3}{T_0}\right) - (R_a + \omega_3 (R_v)\ln\left(\frac{P_{31}}{P_0}\right)\right\}$$
$$+ T_0\left\{(R_P)_a + \omega_3 (R_v)\ln\left(\frac{1+1.6078\omega^0}{1+1.6078\omega_3}\right) + 1.6078\omega_3 R_a \ln\left(\frac{\omega_3}{\omega^0}\right)\right\} \tag{22}$$

$$ex_p = \left[h_p(T,P) - h_p(T_0,P_0)\right] - T_0\left[s_p(T,P) - s_p(T_0,P_0)\right] \tag{23}$$

$$ex_W = \left[h_W(T) - h_W(T_0)\right] - T_0\left[s_W(T) - s_W(T_0)\right] \tag{24}$$

$$\dot{E}x_q = \dot{m}_a ex_q = \dot{m}_a (1 - \frac{T_0}{T_{av}})q_1 = (1 - \frac{T_0}{T_{av}})Q_1 \tag{25}$$

$$\dot{E}x_d = \dot{m}_a ex_1 + \dot{m}_P (ex_P)_2 + (\dot{m}_W)_2 (ex_W)_2 - \dot{m}_a ex_3 - \dot{m}_P (ex_P)_4 - (\dot{m}_W)_4 (ex_W)_4 - \dot{E}x_q \tag{26}$$

$$\Psi = \frac{(\dot{m}_W)_{ev}[(ex_W)_3 - (ex_W)_2]}{\dot{m}_a ex_1} \tag{27}$$

Table 2 shows the thermodynamic properties of inlet and outlet streams. Stream 1 indicates the flue gas originated from natural gas combustion.

Table 2. Thermodynamic properties of inlet and outlet streams of the dryer

State	T(°C)	P(kpa)	m(kg/s)	ω (kg-water/kg-air)	M_W(kg/s)
Flue gas (1)	200	92	21.658	0.014	0.3
Wet coal (2)	25	92	3.33	0	0.66
Exhaust air (3)	90	92	22.92	0.042	0.932
Dried coal (4)	75	91.500	2.69	0	0.033

4. Solar drying of coal

Solar energy might be chosen as an alternative instead of burning natural gas or other fossil fuels in drying applications. Flat plate solar collectors are widely used in drying processes. Parabolic trough type solar collectors provide higher working fluid temperatures than flat plate collectors. This useful heat can be used to produce electricity, heating and cooling applications or industrial processes. The main obstacle in this area is the discontinuity of solar energy. However, in recent years a lot of projects are performed to store the solar energy. Therefore, it should be noted that solar energy might be an alternative or a support in any kind of process. In this study, same amount of energy, provided from sun, is fed to the dryer for the same coal properties. At 890 m elevation, the direct irradiance is estimated to be 867 W/m2 in June at 12:00 and in day time the average direct irradiance is 651.7 W/m2. The design of parabolic collectors has an important effect on solar system efficiency. Table 3 summarizes the parabolic collector properties to provide the same amount of energy into the dryer.

Table 3. Parabolic solar collector properties

Characteristic	Value
Receiver tube outside diameter [mm]	70
Receiver tube wall thickness [mm]	3
Reflector aperture width [m]	5.6
Reflector geometric concentration ratio	80
Reflector rim angle	80
Reflector focal length [m]	1.668
Reflector cleanliness factor	0.95
Receiver tube emissivity	0.15
Receiver glass envelope emissivity	0.9
Convective heat transfer coefficient outside glass envelope [W/m^2°C]	56.78

5. Results

5.1. Result of the coal dryer with NG burner

The total heat supplied by the hot air is used for five different operations in a dryer.

a) To evaporate water from the coal (Q_1)

b) To heat the vapor from the initial wet bulb temperature of the air to the exit air temperature (Q_2)

c) To heat the water from its initial temperature to the inlet wet bulb temperature of the air (Q_3)

d) To heat the dry solid from its inlet temperature to outlet temperature (Q_4)

e) To heat the water which remains in the product from inlet to exit temperature (Q_5)

In the calculations of the total heat requirement, the specific heat of vapor, water and solid are taken as 1.98, 4.18 and 1.26 kJkg^{-1}K^{-1} respectively. A factor is used to increase the total heat requirement due to heat losses from the dryer which is taken as 7.5-10% of the total heat requirement. In the estimation of the dryer diameter two critical points have to be taken into account, air mass velocity and free area of air pass. Higher values of air mass velocity cause entrainment of the product. Free area of air pass, dryer holdup and the bulk density of the coal is taken as 85%, 7% of the total dryer volume and 570 kg/m respectively. Table 4 provides a summary of the results.

According to the thermal design results 69% of the total heat is used for the evaporation of water from the coal. Economic operation ratio is found in the range of 1.5 to 2.5, and therefore the designed dryer can operate economically. Also L/D ratio is calculated in the range of 4 to 10.

Table 4. Thermal and mechanical design results

Thermal Results		
Q1	Evaporation heat [kW]	1443.87
Q2	Vapor heating [kW]	16.47
Q3	Water heating [kW]	139.05
Q4	Solid heating [kW]	169.70
Q5	Water heating to exit solid temperature [kW]	140.74
Qt	Total heat [kW]	1909.83
Qt2	Increased total heat requirement [kW]	2100.81
β	Evaporation to increased total heat ratio[-]	0.69
G	Air flow rate [m^3/h]	103394.78
Mechanical Results		
D	Dryer diameter [m]	2.77
L	Dryer length [m]	12.13
Nt	Economic operation ratio [-]	2.25
Nf	Number of flights	27
Vp	Operational volume of dryer [m^3]	7.29
L/D	Length diameter ratio [-]	4.38
bhp	Total power to rotate dryer [kW]	8.70
n	Dryer speed [r/min]	2.43
Nv	Specific drying per volume	31.59

Figure 2 shows the effects of the dry bulb temperature variation at the inlet of the dryer. The dryer diameter and economic operation ratio change slightly. However, as dryer length and L/D ratio increase, the first investment cost is directly affected.

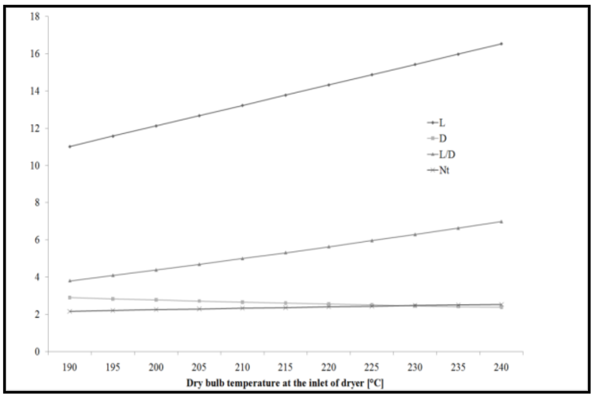

Fig 2. The effects of dry bulb temperature at the inlet of dryer to the dimensions

Figure 3 shows the temperature variation of inlet air with respect to operational parameters. As can be seen in the figure, by increasing the inlet air temperature, required power for rotating the dryer increases. On the other hand, rotation of the dryer remains nearly constant and air mass flow rate decreases. In fact, decreasing air mass flow rate reduces the power requirement of the fans, whereas, rotation power increases. Therefore, in designing a rotary type dryer, an optimum point has to be determined.

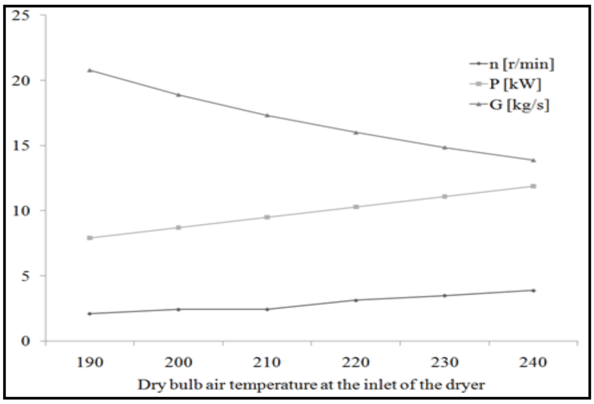

Fig. 3. The effects of dry bulb temperature at the inlet of the dryer to the operational parameters

Product residence time also affects the operational and first investment costs of a dryer. Fig. 4 shows the uneconomic operation field of the selected coal type.

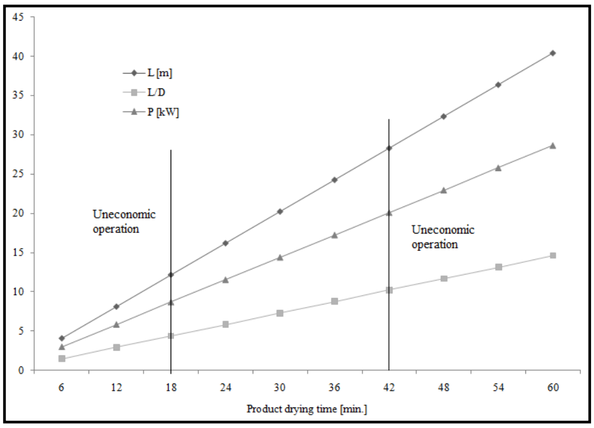

Fig. 4. The effects of residence time variation

5.2. Results of the exergy analysis

Table 5 shows the total exergy destruction for accepted design conditions. The exergetic efficiency of the system is calculated to be 17%. As a result, the performance of the dryer could be improved in terms of energy efficiency. The highest rate of exergy destruction is found at Stream 1. Natural gas has higher exergy potential compared to alternative energy sources. Other alternative sources have to be taken into account when energy efficiency is the main purpose. In this regard, solar drying with parabolic trough type collectors is investigated.

Table 5. Exergy destruction of inlet and outlet streams

State	ex (kJ/kg)	Ex (kW)
Flue gas (1)	40.06	856
Wet coal (2)	10.56	232.3
Exhaust air (3)	0	0
Dried coal (4)	15.91	0.52

5.3. Result of the coal dryer with parabolic solar trough collector

In solar collector area, a pump and a heat exchanger is considered. The heat transfer fluid is chosen as Therminol VP-1. Inlet and outlet temperatures of the working fluid are taken as 200°C and 300°C respectively with a pressure of 5 bars. In the air side of the heat exchanger, ambient air enters the system via a fan. The outlet temperature of the air is taken as 200°C same as the first design. The hot air then enters the direct type fuel dryer where the temperature of the coal at the inlet is taken as 25°C.

After the calculations, dried fuel mass flow rate is found to be 2.693 kg/s with a temperature of 104°C. The humidity of the dried coal is found as %1 and the total thermal power requirement of this system is 1920 kW. This result is in good agreement with the result in design section.

In the heat exchanger the mass flow rate of the working fluid is found to be 17.92 kg/s. The total heat transfer between the heat transfer fluid and air is 3872 kW, where the heat loss to the environment is found to be 39.11 kW. In the collector field, heat transfer to the working fluid is found to be 3910 kW. Inlet and outlet velocities of the working fluid are found to be 0.46 m/s and 0.514 m/s respectively. The required collector field area is calculated to be 24376 m2, where total effective aperture is 6983 m2 with given design conditions in Table 4.

In the first design, where heat input to the dryer is supplied by a natural gas burner, the natural gas consumption of the process is found to be 318.85 m3/h. According to the natural gas prices 2011 in Turkey (0.5 $/m3), daily energy cost of coal drying is found to be 2,551 $/day with 16 hours operation in a day. CO2 emissions originated from natural gas combustion are calculated to be 576 kg/h.

The estimated total first investment cost of the solar collector field with labor costs is assumed to be 2,510,000 $. Yearly total operation hours of the solar collector system can be taken as 2000 hours/year. By using the simple payback method, the payback period of the solar collector field is found to be 6.6 years when carbon credit is taken as 50$ per tones of CO2. One way to decrease the payback time is thermal energy storage. When 12 hours energy storage is considered for the given design the total first investment cost increases up to 5,100,00$ and the payback time decreases to 5.46 years. However, the payback time of the application is still high for energy efficiency applications. However, when looking at in terms of global warming this type of renewable and alternative energy projects has to be supported.

6. Conclusion

In this study design and exergy analysis of a direct type rotary coal dryer is carried out. The economic operation range and the effects of some parameters are investigated for a given type of coal. Dry bulb temperature is found to directly affect the design and it should be kept in certain limits for an economic operation. In the exergy analysis, the second law efficiency of the dryer is found to be 17% and an improvement potential can be achieved for this type of systems. Waste heat utilization or solar energy systems are recommended be integrated to dryers instead of burning natural gas which has higher exergy potential. Solar collector field is considered with parabolic trough type collectors. An economic analysis is carried out to estimate the payback period of the investment and 6.6 years is found without heat storage. Heat storage increases the first investment cost but, yearly working hours of the dryer increases. The payback time is calculated as 5.4 years when 12 hours heat storage system is added to the system. Carbon taxes and carbon credits are also taken into account with a price of 50$ per tones of CO2. For energy conversion systems the payback period time of less than 4 years might be acceptable. However, this type of renewable and alternative energy projects has to be supported to avoid our future negative effects of global warming

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