Exergy Optimization of a Moving Bed Heat Exchanger

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1. Introduction One of the fundamental problems to be solved for the implementation of biomass gasification processes is the conditioning of the gas yield. When the gas yield is going to be used as syngas or in a integrated gasification and combined cycle (IGCC), a severe gas cleanup (tars, particles and alkalis) is needed [1]. Different equipments have been proposed for hot gas particulate removal, such as electrostatic precipitators, ceramic filters, scrubbers and granular filters. The ceramic filters are the most popular. The interest in moving bed granular filters has increased because of operational problems in ceramic filters due to and increase in pressure drop attributed to poor cleaning, densification of the dust layer and closing of pores by sintering of ash [2]. The same difficulties are found in fixed granular beds [3].
In the pressurized fluidized bed gasification case, filters prevent downstream erosion of the heat exchanger and the turbine [4] whereas in the atmospheric case, filters could fulfil a multipurpose function, removing fine particles (to prevent erosion of the generation equipment) and cooling the gas yield (in IGCC, to reach the compressor design temperature or to recover heat from exhaust gases to preheat the inlet air in the gasification process).
An exergy analysis [5] of the heat exchanger has been performed on a Moving Bed Heat Exchanger (MBHE, hereafter) similar to the one described in [6], to obtain, for a range of incoming fluid flow rates, the operational optimum and the incidence on it of the relevant parameters such as the dimensions of the exchanger, the particle diameter and the gas flow rate. In Figure 1 a schematic illustration of the MBHE is showed.

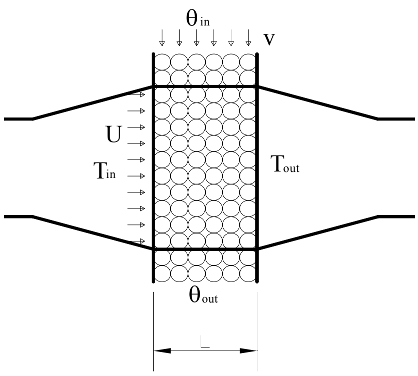


Fig.1 Schematic of the granular bed heat exchanger

The MBHE proposed can be analyzed as a crossflow heat exchanger where one of the phases is a moving granular medium. In the present work the exergy analysis of the MBHE is carried out over operation data of the exchanger obtained in two ways: a numerical simulation of the stationary problem and a simplified analysis. The numerical simulation is carried over the two steady state energy equations (fluid and solid), involving (for the fluid) the convection heat transfer to the solid and the diffusion term in the flow direction, and (for the solid) only the convection heat transfer to the fluid. The simplified analysis followed the well-known e-NTU method, taking the equipment as a crossflow heat exchanger with both fluids unmixed.

2. Modeling heat transfer and pressure drop The equations governing the combined processes of heat transfer and filtration are described in [6]. Both processes are coupled during the transient period as the content of dust particles in the solid phase increases, modifying the heat transfer area and the porosity. In the steady state constant values for heat transfer area and porosity can be assumed and the heat transfer and filtration processes are uncoupled. We will address this second case, as we are interested in the steady state exergy destruction.

The governing equations of the 2-D heat exchanger model, for the gas and solid phases, together with the initial and boundary conditions are:

$$\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}=\frac{k_{eff}}{\rho_g Cp_g}\frac{\partial^2 T}{\partial x^2}+\frac{hS}{\rho_g Cp_g \varepsilon}\left(\theta-T\right); \qquad \frac{\partial \theta}{\partial t}+v\frac{\partial \theta}{\partial y}=\frac{hS}{\rho_s Cp_s\left(1-\varepsilon\right)}\left(T-\theta\right) \tag{1}$$

Initial Conditions: t=0, $T=\theta=\theta_0$

Boundary Conditions: x=0, $T=T_0$; $y=0$, $\theta=\theta_0$

$$\text{x=L}, \quad -k_{ew}\left.\frac{\partial T}{\partial x}\right|_{x=L}=h_w\left(T|_{x=L}-T_w\right)$$

where T is the gas temperature, θ is the solid temperature, u is the interstitial gas velocity, v is the solid velocity, k_{eff} is the effective conductivity of the bed, S is the heat transfer surface divided by the bed volume, L is the bed length along the direction of the gas flow, ε is the porosity and ρ_g, Cp_g, ρ_s, Cp_s are the density and the heat capacity of gas and solid respectively.

The system of differential equations (1), for the steady state and considering spherical particles, can be written in non-dimensional form in the following way

$$\frac{\partial \tilde{T}}{\partial \tilde{x}}=\frac{1}{Pe_g}\frac{\partial^2 \tilde{T}}{\partial \tilde{x}^2}+\frac{Nu}{Pe_g}\frac{6(1-\varepsilon)}{\varepsilon}\left(\tilde{\theta}-\tilde{T}\right); \qquad \frac{\partial \tilde{\theta}}{\partial \tilde{y}}=\frac{Nu}{Pe_g}\frac{C''_g}{C''_s}\frac{6(1-\varepsilon)}{\varepsilon}\left(\tilde{T}-\tilde{\theta}\right) \tag{2}$$

Boundary Conditions: $\tilde{x}=0$, $\tilde{T}=1$; $\tilde{y}=0$, $\tilde{\theta}=0$

$$\tilde{x}=\frac{L}{d_p}, \quad \left.\frac{\partial \tilde{T}}{\partial \tilde{x}}\right|_{\tilde{x}=L/d_p}=-Nu_w\left(\tilde{T}|_{\tilde{x}=L/d_p}-\tilde{T}_w\right)$$

where $\tilde{T}=\left(T-\theta_{in}\right)/\left(T_{in}-\theta_{in}\right)$, $\tilde{\theta}=\left(\theta-\theta_{in}\right)/\left(T_{in}-\theta_{in}\right)$, $\tilde{x}=x/d_p$, $\tilde{y}=y/d_p$, Nu is the Nusselt number, Pe_g is the Peclet number and C''_g and C''_s are the heat capacity rates per unit of area of the gas and solid respectively.

The dimensionless system (2) was solved numerically using a finite differences technique, with up-wind differences for the first derivatives and central differences in the second derivative. The spatial step was 1 mm for both directions (d_p order). The two algebraic equations obtained for each point from the discretization of the differential system (2) were combined, resulting in a tridiagonal system that can be solved to obtain $\tilde{T}_{i,j}$ for every vertical coordinate *j* and for a given horizontal coordinate *i*. Once $\tilde{T}_{i,j}$ was obtained $\forall j$, the nondimensional solid temperature $\tilde{\theta}_{i,j}$ was computed for each point. Then we repeated the process for the following horizontal coordinate *i+1*.

The convection heat transfer coefficient *h* was obtained using the correlation of Gnielinski [7] and the effective thermal conductivities inside the bed k_{eff} and at the wall k_w were evaluated using the expressions proposed by [8].

We assume in our study that the gas velocity *u* is much higher than the solid velocity *v*. In consequence the pressure drop of the gas crossing the bed can be obtained using Ergun equation [9], which was originally obtained for a fixed bed. Thus, the pressure drop is

$$\Delta P=\left[150\frac{\left(1-\epsilon\right)^2}{\epsilon^3}\frac{\mu_g U}{d_p^2}+1.75\frac{1-\epsilon}{\epsilon^3}\frac{\rho_g U^2}{d_p^2}\right]L \tag{3}$$

where μ_g is the dynamic viscosity of the gas, U is the superficial gas velocity and ΔP the pressure drop due to the friction.
The exergy balance was performed in the control volume defined by the boundaries of the MBEH. The exergy destruction per unit of time during the steady operation of the MBHE is produced due to the entropy generated by three mechanisms: heat transfer between the solid and the gas (which is the dominant term), gas pressure drop and the decrease of potential energy of the particles. The expression for the destroyed exergy is:

$$\dot{A}_d = T_0 \left\{ \dot{m}_g Cp_g \left[\ln\left(\frac{\overline{T}_{out}}{\overline{T}_{in}}\right) + \frac{\gamma-1}{\gamma} \ln\left(\frac{P_{in}-\Delta P}{P_{in}}\right) \right] + \dot{m}_s Cp_s \ln\left(\frac{\overline{\theta}_{out}}{\overline{\theta}_{in}}\right) \right\} \quad (4)$$

where T_0 is ambient temperature and H the height of the moving bed along the particle flow direction. This expression was minimized in order to find the optimum design point.

3. Simplified analysis Comparing the two mechanism of heat transfer (convection and conduction), we obtain

$$Nu\frac{6(1-\varepsilon)}{\varepsilon} \sim 10\frac{1}{10^{-1}} = 10^2 \gg 1$$

Therefore, the dominant mechanism in the heat transfer is the convection between the solid and the gas phases. In consequence, in first approximation we can neglect the conduction heat transfer in the gas equation. Then, we can study the heat transfer problem as a heat exchanger in cross-flow with both streams unmixed. Using the well known e-NTU method, the efficiency of the heat exchanger [10] is expressed as

$$e = 1 - \exp\left(NTU^{0.22} \frac{\exp(-0.78 \cdot C \cdot NTU) - 1}{C} \right) \quad (5)$$

where $NTU=(h \cdot A)/(m \cdot Cp)_{min}$ and $C=(m \cdot Cp)_{min}/(m \cdot Cp)_{max}$.
NTU and e were calculated for given parameters (bed dimensions, particle diameter and incoming temperatures and flow rates), obtaining the outlet temperatures on both streams. The pressure drop and the destroyed exergy were then calculated using equations (3) and (4).

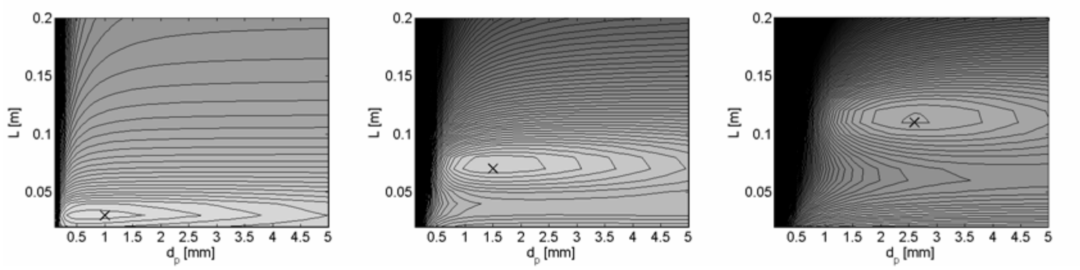
4. Results and discussion The steady state temperatures of the two outgoing flows and the pressure drop were obtained for different superficial gas velocities ranging between 0.2 and 1.1 m/s, varying, for each velocity, the length of the bed (L) and the particle diameter (d_p) between 2 and 20 cm and 0.5 and 5 mm respectively. For each fixed value of U, the optimum combination of d_p and L, that is, the pair of values that minimized the destroyed exergy during the steady state operation of the MBHE, was obtained. The destroyed exergy was computed from equation (4).
The inlet temperatures, particle velocity and the rest of the fixed parameters used in the numerical simulation are summarized in table 1. Note that the minimum value of gas velocity is 0.2 m/s, which is two orders of magnitude higher than the velocity of the solid phase. So the Ergun's equation is applicable as we supposed initially in section 2.

Bed Height	0.50 m	Gas viscosity	10^{-5} (N·s)/m^2
Bed Thickness	0.50 m	Gas conductivity	$30 \cdot 10^{-3}$ W/(m·K)
Cross sectional area of the gas	0.25 m^2	Solid conductivity	15 W/(m·K)
Inlet gas temperature	100 ºC	Porosity	0.4
Inlet solid temperature	25 ºC	Wall porosity	0.5
Solid velocity	15 cm/min	Inlet gas pressure	101325 Pa
Cp_g	1.005 J/(kg·K)	Solid density	7800 kg/m^3
Cp_s	544 J/(kg·K)	Gas density	1 kg/m^3

Table.1 Input parameters for the numerical simulation

Figure 2 shows the curves of constant exergy destruction obtained using the numerical method explained in section 2. The optimum point moves to higher particle diameters and higher lengths of the bed as the superficial gas velocity increases. Similar figures were obtained using the e-NTU method.

Figure 3 compares the optimum particle diameter and optimum length *L* of the bed obtained using both methods. Its shows that the e-NTU method seems to underpredict the value of the optimum particle diameter, although it predicts properly the tendency of increasing d_p with U. The results obtained for the optimum length show similar results with both methods.



(a) (b) (c)

Fig.2 Exergy destruction maps for different superficial gas velocities. The cross indicates the optimum point. (a) U=0.2 m/s, (b) U=0.5 m/s, (c) U=0.8 m/s

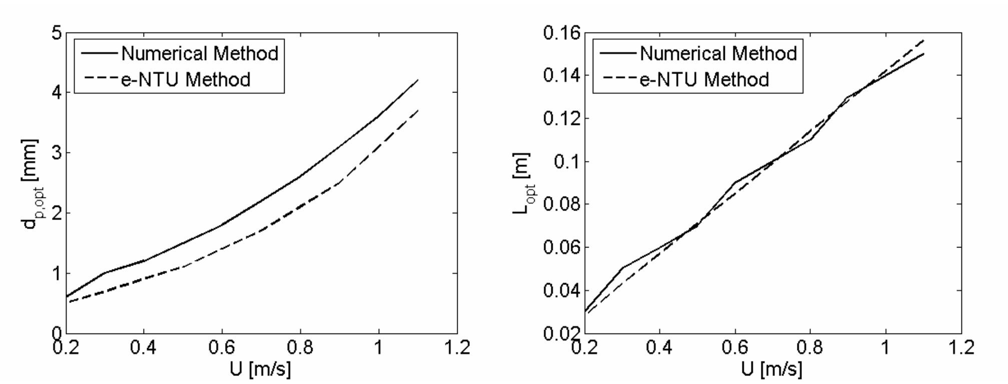


Fig.3 Optimum particle diameter and length of the bed obtained using the numerical and the e-NTU methods for different superficial gas velocities

5. Conclusions The main conclusions of this work are:

- The exergy analysis of a MBHE has been presented using the general heat transfer equation
- An alternative simplified method, applicable when the heat transfer due to convection dominates over the conduction and based on the e-NTU method employed in heat exchangers, has been presented
- The optimum particle diameter and bed length points displace to higher values as the superficial gas velocity increases.

6. Acknowledgements This work has been partially supported by the National Energy Program of the Spanish Department of Science and Education under the project number ENE2006-01401.

7. References

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