Heat Transfer and Friction Characteristics of Heat Exchanger Under Lignite Fly-Ash

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Abstract

This research work was to investigate heat transfer and pressure drop characteristics in cross-flow heat exchanger under lignite fly-ash condition. The tube rows of a bank were aligned and the plain tubes were selected for investigation. The result was divided in to 2 items. Firstly, for clean air condition, the longitudinal tube pitch effects on heat exchanger effectiveness, convection coefficient, Nusselt number, and pressure drop. Therefore, at a larger longitudinal tube pitch, the effectiveness, convection coefficient, Nusselt number, and pressure drop values were higher. Finally, for air-fly ash condition, at slightly dust flow rate, the effect of dust is not much and slightly different with clean air condition, but at high dust flow, they were reduced and lower than the values in clean air condition.

Keywords: heat exchanger, lignite fly-ash mixture condition, performance

1. Introduction

Heat exchanger is a device that is used to transfer thermal energy from higher temperature heat source to lower temperature heat sink. There are many types of heat exchanger applicable to recovery of the waste heat such as shell-and-tube heat exchanger, plate-type heat exchanger, and cross-flow heat exchanger. The cross flow type is very popular due to low cost and easy to clean and clear.

In lignite-fired power plants, many of cross-flow heat exchangers were used i.e. primary air heater, economizer, and flue gas desulfurization plant gas-gas reheater etc.

Normally, existing heat exchangers are operated under the high particulate condition which are fly-ash from the combustion process and tend to decrease its performance.

Unfortunately, there is lack of data about the performance decreasing due to this condition. Therefore, the objective of this research work is to investigate the performance of cross-flow heat exchanger under lignite fly-ash condition.

2. Performance data

In this work, the cross-flow heat exchanger, the tube rows of a bank were aligned, and plain tubes were selected for investigation. The tested data in each model were recorded every 10 minutes, 3 hours after completed heat soak. The experiment setup, aligned tube arrangements, dimensions of cross-flow heat exchanger, and fly-ash chemical composition are shown in figure 1, figure 2, table 1, and table 2 respectively.

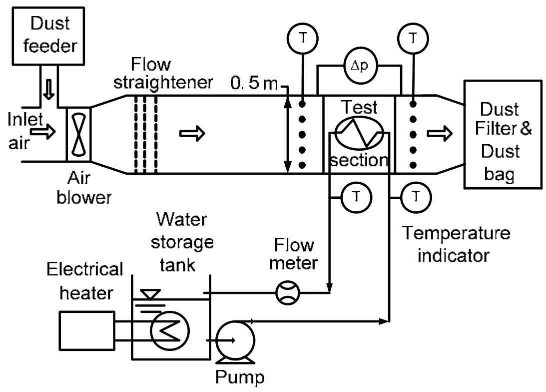

Figure 1 Experiment setup

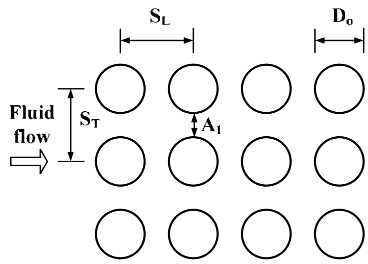

Figure 2 In-line tube arrangements of cross-flow heat exchanger.

Table 1 Dimensions of tested cross-flow heat exchanger.

Categories	**Values**
Tube outside diameter (m)	0.022
Tube inside diameter (m)	0.020
Tube length (m)	0.5
Tube material	Cu
Tube thermal conductivity (W/m K)	388.2
Transverse tube pitch (m)	1.33 D_o
Longitudinal tube pitch (m)	1.33 D_o, 2.66 D_o, 3.99 D_o
Number of tubes	64
Number of tube rows	4

The symbols were listed in nomenclature item.

Table 2 Fly-ash chemical composition.

Categories	**Values (%)**
Na_2O	1.46
MgO	3.41
Al_2O_3	17.86
SiO_2	31.48
P_2O_5	0.19
SO_3	3.05
K_2O	2.44
CaO	21.13
TiO_2	0.36
MnO_2	0.15
Fe_2O_3	18.47

The airflow across the heat exchanger was generated by an electrical air blower with the controllable range of 0.6–1.5 kg/s by using a frequency inverter. The mass flow rate of air stream was measured by a standard flow meter and an inclined manometer with ±0.5 Pa accuracy. The inlet and the outlet temperatures of air stream were also measured by another set of K-type thermocouple mesh. Note that all of thermocouples have been calibrated to ±0.1 ºC accuracy. The pressure drop across the heat exchanger was also measured by the inclined manometer with ±0.5 Pa accuracy. The inline tube arrangements are tested in this study. The effects of air flow and air-fly ash flow on the air side performance are examined accordingly.

3. Theoretical analysis

In this experiment, cold air flows across the bank of tubes but the hot fluid flowing inside the tubes and transfers heat to the air which placed in the outside of the tubes, and the heat transfer rate ($\dot{Q}$) can be calculated as equation (1) and (2).

$$\dot{Q} = \dot{m}_h c_{p,h} (T_{hi} - T_{ho}) \tag{1}$$

$$\dot{Q} = \dot{m}_c c_{p,c} (T_{co} - T_{ci}) \tag{2}$$

The heat transfer rate can be calculated in the form of log mean temperature difference method as equation (3).

$$\dot{Q} = UA\Delta T_{lm} \tag{3}$$

The overall heat transfer coefficient area of the heat exchanger can be evaluated in the term of thermal resistance as equation (4).

$$\frac{1}{UA} = \frac{1}{h_o A_o} + \frac{\ln(D_o / D_i)}{2\pi k_t L} + \frac{1}{h_i A_i} \tag{4}$$

For the aligned tube arrangement, the maximum velocity occurs at transverse plane A1 on figure 2, can be calculated as equation (5).

$$V_{\max} = \frac{S_T}{S_T - D_o} V \tag{5}$$

Reynolds number across the bank of tubes are defined as equation (6).

$$\mathrm{Re}_{D,o} = \frac{\rho \ V_{\max} \ D_o}{\mu} \tag{6}$$

The tube side heat transfer coefficient can be estimated by Dittus-Boelter equation, [2], in the term of Nusselt number and Reynolds number as equation (7) and (8) respectively.

$$Nu = 0.023 \mathrm{Re}_{D,i}^{0.8} \Pr^n \tag{7}$$

where n =0.4 for heating, n=0.3 for cooling.

$$\mathrm{Re}_{D,i} = \frac{4\dot{m}}{\pi D_i \mu} \tag{8}$$

Note that Nusselt number and Prandtl number in this work are defined as equation (9) and (10) respectively.

$$Nu = \frac{hD}{k} \tag{9}$$

$$\Pr = \frac{c_p \mu}{k} \tag{10}$$

Pressure drop in cross-flow tube arrays, which may be expressed as equation (11), [3].

$$\Delta p = N_L \ \chi \left(\frac{\rho V_{\max}^2}{2} \right) f \tag{11}$$

where friction factor, f and correction factor, χ may be received from figure 7.13 in [3].

4. Results and discussion

The selected cross-flow heat exchanger was tested under clean air and air-dust mixture conditions in the form of inline tube arrangement. For clean air condition, the test was divided into 3 cases such as case 1; $S_L = 1.33 D_o$, case 2; $S_L = 2.66 D_o$, and case 3; $S_L = 3.99 D_o$. Therefore, the test result and discussion were shown as following.

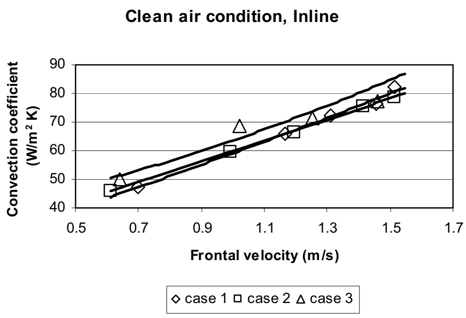

Figure 3 Relationship between convection coefficient and frontal velocity under clean air condition.

Refer to figure 3, shows clearly that convection coefficient values were increased while frontal velocities were increased. This test result has done under clean air condition. It was found that the longitudinal tube pitch effects on convective heat transfer coefficient, at a larger longitudinal tube pitch, the convective heat transfer coefficients were higher. A low flow resistance results in a decreased heat transfer coefficient. Therefore, this figure shows that the heat transfer coefficient decreases as longitudinal tube pitch decreases.

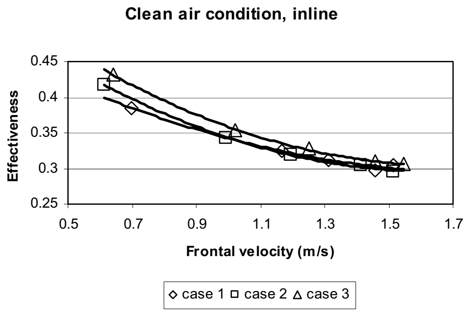

Figure 4 Relationship between effectiveness and frontal velocity under clean air condition.

Figure 4, shows clearly that the heat exchanger effectiveness values were reduced while frontal velocities were increased. This test result has done under clean air condition. It was found that the longitudinal tube pitch effects on effectiveness, at a larger longitudinal tube pitch, the effectiveness values were higher.

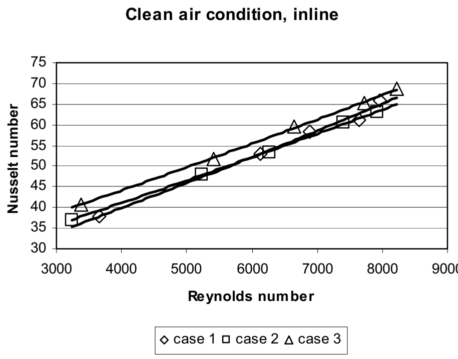

Figure 5 Relationship between Nusselt number and Reynolds number under clean air condition.

Refer to figure 5, shows clearly that Nusselt number values increases as Reynolds increases. It was found that at a larger longitudinal tube pitch, the Nusselt number values were higher. The factors governing resistance to flow also determine heat transfer. Therefore, this factors affecting the Nusselt number too.

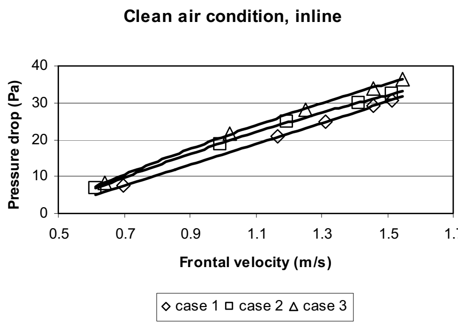

Figure 6 Relationship between pressure drop and frontal velocity under clean air condition.

Figure 6, shows clearly that pressure drop values across the bank of tubes increases as frontal velocity increases. It was found that at a larger longitudinal tube pitch, the pressure drop values were higher. The pressure drop decreases as the tubes are brought closer together. This result was occurred because the reducing the area of turbulence between the tubes has a remarkable effect on the friction factor and the pressure loss.

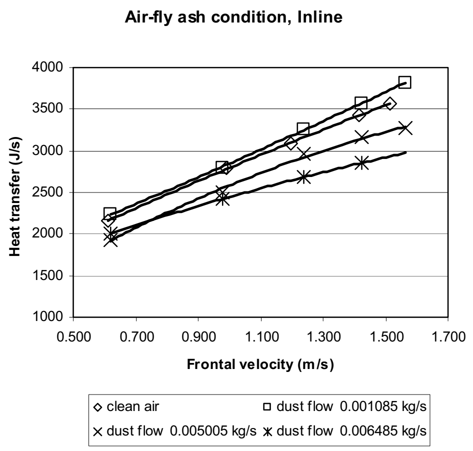

Figure 7 Relationship between heat transfer and frontal velocity at longitudinal tube pitch equal to 2.66 D_o under air fly-ash mixture condition.

Figure 7 and 8, In the longitudinal tube pitch equal to 2.66 D_o and under air fly-ash mixture condition, the heat transfer values and convection coefficient values were increased while the frontal velocities were increased. At slightly dust flow rate, the effect of dust is not much and slightly different than the values in clean air condition, but at high dust flow rate, the heat transfer values and convection coefficient values were lower than the values in clean air condition respectively.

Refer to figure 9, shows relationship between effectiveness and frontal velocity under air-fly ash mixture condition at longitudinal tube pitch equal to 2.66 D_o, the effectiveness values were reduced while frontal velocities were increased. At slightly dust flow rate, the effect of dust is not much and slightly different with clean air condition, but at high flow rate of dust, the effectiveness values were lower than the values in clean air condition.

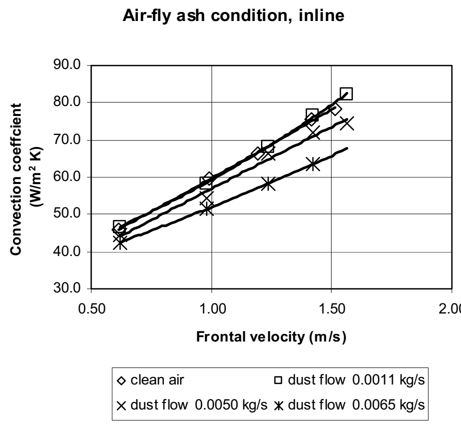

Figure 8 Relationship between convection coefficient and frontal velocity under air-fly ash mixture condition.

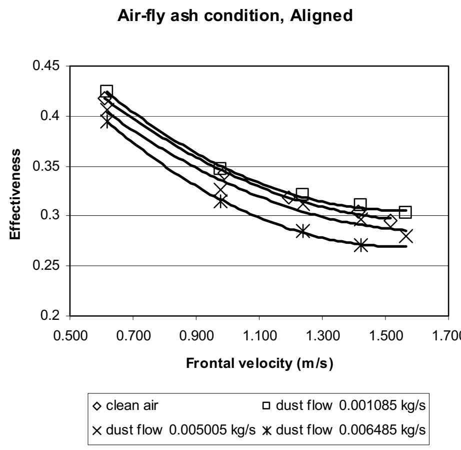

Figure 9 Relationship between heat exchanger effectiveness and frontal velocity at longitudinal tube pitch equal to 2.66 D_o under air fly-ash mixture condition.

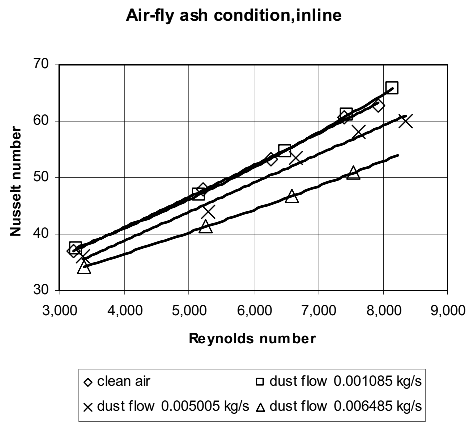

Figure 10 Relationship between Nusselt number and Reynolds number at longitudinal tube pitch equal to 2.66 D_o under air fly-ash mixture condition.

Figure 10, In the longitudinal tube pitch equal to 2.66 D_o and under air fly-ash mixture condition, the Nusselt numbers were increased while the Reynolds numbers were increased. At slightly dust flow rate, the effect of dust is not much and slightly different with clean air condition, but at high dust flow rate, the Nusselt numbers were lower than the values in clean air condition.

5. Conclusion

It can be concluded as following.

5.1 For clean air condition, the longitudinal tube pitch effects on heat exchanger effectiveness, convection coefficient, Nusselt number, and pressure drop. Therefore, at a larger longitudinal tube pitch, the effectiveness, convection coefficient, Nusselt number, and pressure drop values were higher.

5.2 For air-fly ash condition, at slightly dust flow rate, the effect of dust is not much and slightly different with clean air condition, but at high dust flow, they were reduced and lower than the values in clean air condition.

Acknownledgments

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Nomenclature

A	area (m^2)
c_p	specific heat at constant pressure (J/kg K)
$c_{p,c}$	specific heat of cold medium at constant pressure (J/kg K)
$c_{p,h}$	specific heat of hot medium at constant pressure (J/kg K)
D	diameter (m)
f	friction factor
h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
p	pressure (Pa)
Pr	Prandtl number
$\dot{Q}$	heat transfer rate (W)
Re_D	Reynolds number
S_T	transverse tube pitch (m)
S_L	longitudinal tube pitch (m)
T	temperature (°C)
U	overall heat transfer coefficient (W/m^2K)
V	velocity (m/s)

Greek symbol

μ	dynamic viscosity (Pa s)
ρ	density (kg/m^3)
χ	correction factor

Subscript

a	air side
c	cold
h	hot
i	inner or inlet
max	maximum
o	outer
t	tube