Numerical study of waste heat recovery by direct heat exchanger systems

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Abstract. Dependency on fossil fuel and associated carbon footprints have increased the urge to look for methods to increase the thermal efficiency of heating and cooling systems. Exhaust air heat recovery system is one of the promising solutions with viable potential in industrial applications such as mine ventilation and so on. Direct contact heat exchangers can be feasibly used in these applications due to their characteristic advantages such as the ability to exchange at low temperature differences. Given the numerous advantages of such systems, there is a need for thorough understanding of the complex fluid flow and heat transfer performance of these heat exchangers. While water spray is commonly used for capturing the heat from exhaust air in direct heat exchange systems, performance of such system is highly dependent on the various operating parameters such as droplet size distribution, continuous phase temperature, velocity and relative humidity, spray nozzle angle and discrete phase temperature and velocity which is required to be studied in greater depth. Computational Fluid Dynamics can play a key role to investigate the performance of these two-phase flow systems. In this paper, a three-dimensional two-phase model has been presented to study heat recovery from exhaust air by using direct spray water heat recovery systems. Also, an analytical model has been developed using a self-written MATLAB code and compared to the numerical one. The results of the study show that the analytical model can capture the CFD runs outcomes with a high degree of accuracy. Also, the conducted parametric study confirms the dominant impacts of droplet size distribution and air flow rate on the performance of the system.

Keywords: Heat capturing, Direct heat exchanger, Water spray, Energy recovery

1. Introduction
Rising fossil fuel prices and strict carbon emission regulations draw more researchers into finding a sustainable and economically feasible technologies for waste heat recovery (WHR) systems [1]. Over the past decades heat recovery system find applications in a wide range of industries. For example, Shultz [2] investigated the implementation of gas-to-gas heat exchangers for buildings exhaust air heat recovery. Sternlicht [3] studied the utilization of liquid-to-liquid heat exchangers for high-pressure gas compressors heat recovery system. Songhui et al [4] proposed a direct contact heat recovery system in a soy protein powder plant application.
Mining industry is among the most attractive areas for exhaust heat recovery system utilization as amount of the waste heat discarded to environment is significant. However, due to the low temperature of the mine exhaust air, employing an efficient technology for heat recovery has a vital role in making the project economically viable. Compared to other WHR technologies, direct contact heat exchanging system has many desirable attributes but the key one is that it has the ability to exchange heat at low temperature differences which makes it a promising candidate for mine exhaust heat recovery (MEHR) systems.

The Inco’s Creighton mine located in Sudbury Ontario was the first known Canadian mine which utilized the WHR system using a direct contact heat capturing unit at the mine exhaust shaft area. However, the system ceased to operate because of the difficulties regarding the proper operating condition and maintenance issues [5, 6].

Despite its wide applicability, there are a limited number of studies on the MEHR system reported in the existing literature. For example, one of the recent studies was made by Sbarba et al [7, 8] on the feasibility of recovering the heat from mine exhaust air. Following up on Sbarba’s work, Ghoreishi-Madiseh et al. [9, 10] conducted a study to investigate the performance of the MEHR system using two coupled heat exchanging units. However, to the best of the authors’ knowledge there is a lack in the number of studies on the direct contact heat recovery system in the mining literature.

Accordingly, this study aims to bring together two approaches to investigate the performance of direct contact WHR system, the first one is an analytical model using a MATLAB code, and the second one is a numerical CFD model using Fluent software. Throughout this study, the agreement between these two approaches are highlighted and further used for a parametric investigation to see the performance of the system under various operating conditions.

2. Methodology

The schematic of a downward single spray heat exchanger is shown in Fig. 1(a). The falling cold sprayed water is directly contacted with the upward warm exhaust air in order to capture the heat from the air stream. The system has been modeled both analytically and numerically using the same input parameters. For the analytical model, a MATLAB code has been developed and for the numerical CFD simulation Fluent 19.0 was used.

![Fig. 1-a)](image) Schematic diagram of a single spray heat exchanger ![Fig. 1-b)](image) Differential sections of the heat exchanger.

2.1. Analytical model

Momentum, mass and heat transfer equations were employed to develop analytical model in a differential section of the heat exchanger height (ΔZ). Using Newton’s second law for a single droplet, following equation can be derived [11].
\[ m_d \frac{d u_d}{dt} = W - F_B - F_D \]  

(1)

where, \( W \), \( F_B \), \( F_D \) and \( m_d \) are the gravitational force, buoyancy force, drag force and mass flow rate of sprayed water, respectively and can be written as follow:

\[ W = \frac{1}{6} \pi d \rho_w g \]  

(2)

\[ F_B = \frac{1}{6} \pi d^2 \rho_a g \]  

(3)

\[ F_D = \frac{1}{2} C_D \rho_a \frac{\pi}{4} d^2 (u_d + u_a)^2 \]  

(4)

\[ m_d = \frac{1}{6} \pi d^3 \rho_w \]  

(5)

\( C_D \) is the drag coefficient which is a function of Reynolds number \([12]\).

\[ C_D = \frac{24}{Re} (1 + 0.15 Re^{0.687}) \quad \text{Re} < 1000 \]  

(6)

\[ C_D = 0.44 \quad \text{Re} > 1000 \]  

(7)

\[ Re = \rho_a (u_d + u_a) d_d \]  

(8)

The heat exchanger overall height (\( H \)) was divided into \( n \) equivalent sections with a differential height of \((\Delta Z)\) as shown in Fig. 1(b). If \( \Delta Z \) is small enough, it can be assumed that:

\[ \frac{dZ}{dt} = u_d, \quad \text{or} \quad dZ = u_d dt \]  

(9)

Substituting Eq. 9 in Newton’s second law yields equation 10:

\[ \frac{du_d}{dZ} = \frac{1}{u_d} \left( g \left( 1 - \frac{\rho_a}{\rho_w} \right) - \frac{3 C_D \rho_a}{4 d_d \rho_w} (u_d + u_a)^2 \right) \]  

(10)

The mass transfer flux of water can be written as follow \([13]\):

\[ N_w = h m_a (y - y_\text{ds}) / M_w \]  

(11)

Here, \( y_\text{ds} \) is the saturated humidity at the droplet surface and it is a function of the temperature \([14]\):

\[ y_\text{ds} = \left( 0.0144 T_w^4 - 0.74617 T_w^3 + 33.6887 T_w^2 + 50.0495 T_w + 4266.6815 \right) \times 10^{-3} \]  

(12)

Using the mass transfer flux of water vapor \((N_w)\), one can get the following equation for humidity:

\[ \frac{dy}{dt} = \frac{N_w M_w A_t}{m_a}, \quad \text{or} \quad \frac{dy}{dZ} = \frac{N_w M_w A_t}{u_d m_a} \]  

(13)

where, \( A_t \) is the mass and heat transfer area per unit time which is the number of droplets released from the nozzle per second multiplied by the surface area of a single droplet:

\[ A_t = \frac{\pi d_d^2}{\rho_w \pi d_d^3 / 6} \]  

(14)

The energy conservation law for a water droplet can be written as follow:

\[ N_w M_w (\pi d_d^2) L_v + h (\pi d_d^2) (T_a - T_w) = \frac{d \left( m_d C_p w T_w \right)}{dt} \]  

(15)

The first term on the left-side of Eq. 15 is for the latent heat and the second term is for the sensible heat transfer. Here, \( m_d \) and \( L_v \) are the mass of a single water droplet and latent heat of vaporization, respectively. Using the Eq. 9, the energy conservation law (Eq. 15) yields the following equation:

\[ \frac{dT_w}{dZ} = \frac{6 h (T_a - T_w) + 6 N_w M_w L_v}{C_p w d_d \rho_w u_d} - \frac{3 T_w d_d}{d_d \rho_w u_d} \]  

(16)

For the water droplet diameter changes inside a differential section, one can get:

\[ \frac{d d_d}{dt} = \frac{2 N_w M_w}{\rho_w}, \quad \text{or} \quad \frac{d d_d}{dZ} = \frac{2 N_w M_w}{\rho_w u_d} \]  

(17)

Also, the conservation of energy for the air side of the system can be written as follow:

\[ \frac{d(H_a)}{dt} = h A_t (T_a - T_w) + N_w M_w A_t L_v \]  

(18)
The first and second term of the right-hand side of Eq. 17 represent the sensible heat and latent heat, respectively. Also, $H_a$ is the enthalpy of the air which is:

$$H_a = C_{p,a} T_a + y (C_{p,v} T_w + L_v)$$  \hspace{1cm} (19)

One can get the following equation for the air side temperature using the Eqs. 9, 18 and 19.

$$\frac{dT_a}{dZ} = \frac{h A_t}{(C_{p,a} + y C_{p,v}) m_a u_d} (T_a - T_w)$$  \hspace{1cm} (20)

Also, the heat and mass transfer coefficients can be calculated using the Nusselt number and Sherwood number, respectively [13].

$$\frac{h d_d}{k} = Nu = 2 + 0.6 Re^{0.5} Pr^{0.33}$$  \hspace{1cm} (21)

$$\frac{h m d_d}{D} = Sh = 2 + 0.6 Re^{0.5} Sc^{0.33}$$  \hspace{1cm} (22)

Table 1. Mesh grid sensitivity analysis

| Parameters               | Coarse Mesh | Main Mesh | Fine Mesh |
|--------------------------|-------------|-----------|-----------|
| Nodes                    | 327,735     | 622,407   | 1,141,274 |
| Elements                 | 1,064,989   | 2,514,702 | 4,120,814 |
| Air temperature difference | 8.3145 °C  | 8.3838 °C | 8.4024 °C |

Table 2. Direct heat exchanger model parameters

| Parameters           | Value | Unit |
|----------------------|-------|------|
| Air inlet temperature| 300   | K    |
| Water inlet temperature| 278  | K    |
| Air flow rate        | 0.72-1| kg/s |
| Water flow rate      | 0.4   | kg/s |
| Droplet diameter     | 0.6-0.75 | mm |
| Nozzle height        | 2.5   | m    |
| Air inlet velocity   | 1.7-2.3| m/s  |
| Water inlet velocity | 0.4   | m/s  |
Fig. 2. Comparison of the numerical model and analytical model results.

Fig. 3 represents the temperature distribution across the middle plane of the heat exchanger for the case with the droplet diameter of 0.7mm and air velocity of 2.3m/s. As the air moves up toward the outlet interface the temperature drops yields with minimum air temperature being around spray nozzle.

Fig. 4(a) shows variation of the air dry-bulb temperature difference in the various inlet water droplet diameters, predicted by the numerical model. As it can be observed the air temperature difference drops by increasing the inlet droplet diameter. The smaller the droplet size is, the larger the overall heat and mass transfer surface will be. Also, the weight loss increases the droplets contact time with the air stream. Accordingly, the mass and heat transfer overall surface and contact time between the discrete and continuous fluid increases which results in a higher temperature difference between the inlet and outlet air and thus better performance of the heat exchanger. However, it should be pointed out that for any given air velocity the droplet diameter needs to be larger than a certain size, otherwise, it will be carried out by the air stream. Also, Fig. 4(b) illustrates the analytical model results on the variation of the water temperature difference in the various air inlet velocity. As the figure shows, the water temperature difference rises by 15 percent as the air velocity increases from 1.7m/s to 2.3m/s which leads to a better performance of the system. However, it should be noted that a higher number of droplets will be carried out by the air stream as the air velocity increases which has a negative impact on the heat exchanger performance.
4. Conclusion

This study aims to investigate the heat and mass transfer characteristics of a direct contact heat exchanger for WHR purposes both numerically and analytically. The results of both solution techniques have been compared and it is found that the analytical model can capture the results of the CFD runs with an agreement of 3.15% within the range of operating parameters in Table 2. According to the results of the present study, direct contact heat exchangers have a better heat transfer performance with the combination of the smaller droplet size and higher air velocity, however for any given cases there is a limitation on the range of both parameters.

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