Design and test data analysis of a heating system for a wellhead room in a coal mine

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Abstract. This paper introduces the design and the test data analysis of a heating system for a wellhead room in a coal mine. Combined with the system design, the actual operational data of the water jet spray heat exchange device was analysed to calculate some coefficients, the thermal efficiency and the spray heat resistance. For demonstrating the practical application of this heating system, a field test was conducted where the outdoor environment temperature was -7.4°C. The air outlet temperature of the air heating unit was 27.8°C and the air inlet temperature of a wellhead room after mixing the outdoor air and the hot air supplied from the air heating unit was 11.5°C. This performance data demonstrated that this mine return air heat exchange system well satisfies the heating requirement for the wellhead facilities in coal mines located in cold areas and shows significant advances in the designing and operation of the heating system.

1. Introduction
According to the "BP Statistical Yearbook of World Energy" (2018 edition), the global energy demand increased by 2.2% in 2017, and the coal consumption rebounded for the first time since 2013[1]. In this year, the Chinese energy consumption increased by 3.1%, ranking first in the energy growth during recent 17 consecutive years. After experiencing low or zero energy growth from 2014 to 2016, the carbon emission caused by the energy consumption increased by 1.6% in 2017. Coal accounts for 60.4% of the Chinese energy consumption. It can be seen that coal resources still occupy a pivotal position in Chinese energy supply.

In order to effectively alleviate the contradiction between the resource development and the ecological environment protection, it is urgent to realize clean heating for coal mining companies. Especially in severe cold areas, most of the mine heating sources are low-tonnage small-scale coal-fired boilers, which have been inconsistent with the national environmental protection policies and regulations. The pollutant emissions from such boilers are not satisfying the environmental standards and they are under the national and local governments' clear requirements to be demolished and shutdown within a certain time limit. When such boilers shall be eliminated and dismantled, the coal mining companies must seek for practical and feasible environmentally friendly heating alternatives to provide a strong guarantee for the company's sustainable and healthy development and heat users.

Thermal demand projects have been continuously developed in recent years [2-4]. In the process of coal mining, the return air temperature in most mines is maintained above 10°C all year round, and that in the heat-damaged mines is maintained above 25°C all year round. Therefore, this part of the
mine exhaust air can be reasonably used as the heat source for the heat exchange. In this work, the water source heat pump system using such exhaust heat was designed and tested to meet the heat supply demand of a mine in northwestern Shanxi.

2. System design

2.1. Project overview
The project belongs to a mine of a group company. The mine has a designed coal production capacity of 1.20 million tons/year. The mine is located remotely and in winter, the weather is severely cold. So the air intake shaft is frost-proof and two 2 tons/h boilers and one 1 ton/h hot blast stove are utilized for heating. Due to the high energy consumption and the severe pollution problems associated with the existent boiler heating system, the original boiler heating system in the mining area has been replaced by the exhaust air sprinkler heat extraction-heat pump heating system. This heating system mainly serves to avoid the freezing of the air inlet shaft of the panel 307. The water source heat pump technology is used to recover the waste heat from the mine exhaust [5-6]. The main research and development target is to demonstrate the performance of this new heating system composed of a spray heat extraction device, a water source heat pump unit and a high-efficiency terminal heat dissipation equipment.

2.2. System calculation
According to "Code for Design of Heating, Ventilation and Air Conditioning of Industrial Buildings", the outdoor air calculation parameters of this site are as follows: the outdoor heating temperature in winter is -16.3°C; the outdoor ventilation temperature in winter is -10.6°C, the extreme lowest temperature is -27.2°C; the average extreme lowest temperature all year round is -24.3°C and the average wind speed in winter is 2.8m/s[7]. In this mine, according to the professional mine process design, the exhaust air volume flow rate is 115 m³/s, the intake air volume flow rate is 68 m³/s and the remaining air is supplied from the air intake shaft in another industrial site which is not within the design scope of this heating system.

According to the "Coal Safety Technical Regulations", the average extreme lowest temperature all year around of -24.3°C is used to calculate the heat demand for the air intake. The temperature of the air intake shaft is assumed to be 2°C, and the calculated heat demand for the air intake is 2540kW. Therefore, the heat supply capacity of the heat pump unit needs to exceed this heat demand of 2540kW.

For the designing, it is assumed that the energy efficiency of the heat pump unit is 4.0, and the heat pump unit is calculated to take the heat of 1905kW. So when designing the system heat extractor, it must be ensured that the heat extraction exceeds 1905kW.

As stated above, the mine exhaust air volume flow rate is 115m³/s, and the design value of the air enthalpy difference before and after the spray heat extractor is 16.5kJ/kg. The designing and calculation of other accessories will not be reported here.

2.3. System configuration
The water source heat pump unit in the heating system consists of five HE600 units with the nominal heating capacity of 584.8kW, the rated return water temperature of 50/40°C, the nominal input power of 121.47kW and the refrigerant (R22) charge of 120kg. There are 2 sets of the spray heating devices, one for the use and another for the backup, with the rated intake air volume flow rate of 115m³/s and the rated heat absorption capacity of 1905kW. Five wellhead heaters with the air supply capacity of 13.9m³/s are installed, where one heater has the rated heat supply capacity of 756kW with the motor power of 15kW. There are four heating circulating water pumps, two pressurized water pumps, one submersible sewage pump, one automatic filter, one all-round water processor and one set of the water softening device. The heating system is equipped with ten control cabinets and one transformer. The
schematic diagram of the system configuration is shown in Figure 1 and the picture of the machine room is shown in Figure 2.

![Figure 1. The schematic diagram of the heating system.](image1)

![Figure 2. Picture of the machine room.](image2)

3. **Test data analysis**

The following test data analyses were conducted to evaluate the performance of the air heating system.

3.1. *Spray heat transfer-water spray coefficient* [8-9]

\[ \eta = \frac{W}{G} \]

where \( \eta \) is the water spray coefficient, kg (water)/kg (air), \( W \) is the spray water flow rate, kg/h and \( G \) is the processed exhaust air mass flow rate, kg/h. According to the measured and recorded data, the spray water flow rate was 382.5 m³/h. According to the measured wind speed data, the average air velocity behind the baffle was 2.58 m/s. The reserve area was 45 m² and the processed exhaust air volume flow rate was 116.1 m³/s. Then the spray coefficient (water-air ratio) can be calculated as follows:

\[ \eta = \frac{W}{G} = \frac{382.5}{116.1 \times 3.6 \times 1.1} = 0.83 \]

3.2. *Spray heat exchange-heat exchange efficiency coefficient*

\[ \eta_1 = 1 - \frac{t_{s2} - t_{w2}}{t_{s1} - t_{w1}} = 1 - \frac{5.8 - 4.2}{10.2 - 1.6} = 81.4\% \]

where \( t_{s1}, t_{s2} \) are air wet bulb temperature before and after treatment, °C.

\( t_{w1} \) and \( t_{w2} \) are Initial and final temperatures of water spraying, °C.

3.3. *Spray heat transfer-thermal contact coefficient*

\[ \eta_2 = 1 - \frac{t_{2} - t_{s2}}{t_{1} - t_{s1}} = 1 - \frac{6.1 - 5.8}{11.0 - 10.2} = 62.5\% \]

where \( t_1 \) and \( t_2 \) are the dry-bulb air temperatures before and after the spray heat extractor, respectively, °C

3.4. *Spray heat exchange-air mass flow rate*

\[ \nu = \frac{G}{3600S} = \frac{116.1 \times 1.1}{45} = 2.84 \]
where \( \nu \) is the air mass flow rate of the spray heat exchange, kg/(m\(^2\)·s) and \( S \) is the spray heat exchange area, m\(^2\).

### 3.5. Spray heat exchange-air heat release

The heat released from the exhaust air is calculated as:

\[
Q_1 = G \times (h_2 - h_1) = 116.1 \times 1.1 \times (32.5 - 22.4) = 1289.87\, kW
\]

where \( h_1 \) and \( h_2 \) are the air enthalpy before and after the spray heat extractor, respectively, (kJ/kg)

### 3.6. Spray heat exchange-water heat absorbance

The heat absorbed by water is calculated as:

\[
Q_2 = (W \times \frac{1000}{3600}) \times c \times (t_{w2} - t_{w1}) = 382.5 \times 1.163 \times (4.2 - 1.6) = 1156.60\, kW
\]

where \( c \) is the specific heat of water, kJ/(kg·°C)

### 3.7. Spray heat exchange-effective utilization of heat

\[
\eta_3 = \frac{Q_2}{Q_1} = \frac{1156.60}{1289.87} = 89.7\%
\]

### 3.8. Spray heat transfer-resistance calculation [10]

1. Water resistance:

\[
H_W = 1180b\mu P, \, Pa
\]

where \( b \) is the coefficient determined by the direction of the water spray and the air movement and \( P \) is the water pressure in front of the nozzle, MPa

2. Nozzle tube row resistance:

\[
H_p = 0.1Z\nu^2 \frac{P}{\rho}, \, Pa
\]

where \( Z \) is the tube row number, \( \nu \) is the cross-sectional wind speed in the spraying chamber, m/s and \( \rho \) is the density of air, kg/m\(^3\)

3. Water baffle resistance:

\[
H_d = \sum \zeta (\nu_d)^2 \frac{P}{\rho}, \, Pa
\]

where \( \sum \zeta \) is the resistance coefficient of the baffle

4. Total resistance:

\[
H = 1180b\mu P + 0.1Z\nu^2 \frac{P}{\rho} + \sum \zeta (\nu_d)^2 \frac{P}{\rho} = 1180 \times 0.075 \times 0.83 \times 0.1 + 1.2 \times 2.58^2 \times 1.2/2 + 10 \times 1.2 \times 2.58^2 \times 1.2/2 = 60.06\, Pa
\]

According to the pressure recording instrument, the pressure difference between the inlet and exit of the water spray heat exchange section was 68-10=58Pa. This is the atmospheric pressure difference value because the pressure values at the inlet and exit were used to calculate this pressure difference. This data shows that the designed resistance is within the required range. Figure 3 is the physical picture of the waste heat recovery device, and Figure 4 is its field resistance test.

### 3.9. Spray heat transfer-noise test

According to the data obtained by an on-site noise meter, the noise value within 5m around the spray heat exchanger was 65dB.
4. Results and discussions

Figure 5 shows the outdoor temperature and the relative humidity data recorded by the testo174H temperature and humidity recorder on site. The blue curve in the figure is the recorded temperature data and the red curve is the recorded relative humidity data. It can be seen from this figure that the curves fluctuated greatly during the time period of 14:00-14:20 and after 15:45. The recorder started at 14:00 and its operation was affected by the ambient temperature until 14:20. After 14:20, the recorder recorded the temperature and the relative humidity normally. After 15:45, the testing personnel took the recorder back in the carrying case and started to observe the data. Therefore, the data obtained during 14:20-15:45 was taken as the test analysis data, and the remaining data was rejected as the error data. The data shown in Figure 5 was recorded every 1 minute. After analyzing the data in this figure and taking the average value, it was found that the outdoor air temperature was -7.4°C and the relative humidity was 27.6%.

The supplied air temperature from the air heating unit was measured using the testo-605i recorder shown in Figure 6 and it was combined with the APP data recording, whose recorded data chart after stabilizing the observed data is shown in Figure 7. The APP displayed the data as a real-time display interface. From the data recorded, it can be known that the supplied air temperature from the air heating unit was 27.9°C and its relative humidity was 3.7%.

Figure 8 shows the recorded data of the inlet air temperature and the relative humidity measured in the wellhead room. After the data acquisition, the test data of the wellhead room was sorted out as shown in Table 1.
Figure 6. Testo-605i recorder.  

Figure 7. APP data recorded graph.  

Figure 8. Recorded inlet air temperature and the relative humidity in the wellhead room.  

Table 1. Wellhead inlet air parameters.

| Name               | Dry-bulb air temperature | Relative humidity | Wet-bulb air temperature | Dew point temperature | Moisture content | Enthalpy Danwei (?) |
|--------------------|---------------------------|-------------------|---------------------------|-----------------------|------------------|--------------------|
| Unit               | °C                        | %                 | °C                        | °C                    | g/kg             | kJ/kg              |
| Value              | 11.5                      | 7.2               | 1.4                       | -20.5                 | 0.7              | 13.3               |
| Remark             | Rest (what does this mean?) | Rest             | Calculated                | Calculated            | Calculated       | Calculated         |
The rated air volume flow rate of the air heating unit is 13.9 m$^3$/s, the motor power is 15kW and the rated heat supply is 756kW. According to the actual test data, the outdoor air temperature was -7.4 °C, the supplied air temperature from the air heating unit was 27.8 ℃ and the temperature of the outdoor air entering the wellhead room after mixing with the hot air supplied from the air heating unit was 11.5 ℃. The actual test data showed that the actual air output of each air heating unit was 12.6 m$^3$/s. After checking and calculation, it was found that the heat supply of each air heating unit was 487.8kW and the total heat supply was 1463.4kW. The main reasons for the air heating unit not reaching the rated heat supply are that the outdoor temperature was lower than the design limit value on the test day and the temperatures of the heat source supply and the return water were different from the rated values.

The comprehensive evaluation of the performance of the air heating unit was good and the system operation achieved the design target ensuring that the wellhead room did not suffer from any adverse effects such as the icing in the outdoor low temperature environment, and the temperature of the wellhead room was maintained above the design temperature meeting the site requirements.

5. Conclusions

The test data analysis shows that the volume flow rate of the mine return air and intake air were 116.1 m$^3$/s and 66.7 m$^3$/s, respectively. The system designed has 5584.8kW water source heat pump units and 151 m$^3$/s air heating units to meet the actual heat demand on site. The spray heat exchange system achieved the following performance when the water-to-air ratio was 0.83 and the air mass flow rate was 2.84 kg/(m$^2$.s): the heat exchange efficiency coefficient was 81.4%; the thermal contact coefficient was 62.5%; the heat effective utilization rate was 89.7%; the spray heat exchange (exhaust air heat extractor) system resistance was 58Pa; the noise value of the spray heat exchanger was 65dB and COP of the water source heat pump unit was 4.6.

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