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COLD REGION MINE SHAFT ANTI-FREEZING TECHNOLOGY BASED ON LOW-TEMPERATURE WASTE HEAT UTILIZATION FROM UNDERGROUND MINE DRAINAGE

To solve the ground icing problem at the mine wellhead in winter and overcome the high energy consumption of traditional anti-freezing solutions, this paper developed a mine intake air anti-freezing system using underground drainage waste heat. Based on the ANSYS simulation platform, a 3D cold-hot air mixing model of the mine entrance section in the constant-temperature rock-soil layer was established to simulate the airflow temperature field distribution at the wellhead under extreme low temperature and multi-condition heat exchange, and the influence of the rock-soil layer on the heat transfer process was quantitatively analysed. Aiming at near-zero power consumption heat exchange, a finned-tube heat exchanger was developed: key thermal parameters were calculated based on energy conservation and heat transfer theory to optimise fin tube specifications, structure and layout; resistance checks, wall temperature calculation and material selection were completed to improve heat exchange efficiency. The results show that the system can stably heat mine intake air to above 2°C through non-contact heat exchange with underground drainage as the natural heat source, which meets mine anti-freezing standards. This system can replace traditional high-energy anti-freezing solutions, realising the dual goals of mine safety and energy conservation.

Keywords: Mine return air; Wellhead anti-freezing; Heat pipe heat exchanger; ANSYS; Numerical simulation

1. Introduction

The mine tunnel is the core passage for ventilation, transportation and personnel passage in a mine. Its operational stability directly affects the production safety and operational efficiency of the mine [1]. The freezing of the mine tunnel ground in winter is a core seasonal safety risk

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in high-altitude mining areas, which can easily lead to transportation failures, personnel injuries and emergency evacuation disruptions, etc. [2]. The “Coal Mine Industrial Mining Design Specifications” of China clearly stipulates that for all types of intake shafts under low-temperature thresholds, if there are water seepage and drainage facilities, air heating equipment at the shaft opening must be installed to implement the control of ice formation at the shaft opening. Yang et al. [3] developed a gravity split heat pipe system adapted to mine conditions, raising -16°C fresh air by $12\text{--}16.5^{\circ}\text{C}$ for unpowered low-temperature waste heat recovery. The Colorado Coal Mine project [4] established a full heat transfer model and realised over 500 m long-distance waste heat transport with 78% thermal efficiency, heating fresh air to 5°C at -18°C . Xin et al. [5] clarified the effects of heat pipe dimensions, fins and tube layout on heat transfer, supporting structural optimisation. To overcome the high energy consumption of traditional electric heating, this paper presents a novel energy-saving antifreezing method using underground wastewater waste heat for intake air heating and energy recycling. Numerical simulation is conducted to analyse the temperature field at the mine entrance and compare the performance of antifreezing schemes under various conditions. Globally, industrial waste heat recovery and sustainable energy utilisation have drawn extensive scholarly attention. Jouhara et al. [6] classified waste heat recovery technologies across various temperature zones, providing a macro-theoretical framework for low-grade waste heat recycling. For core heat-exchanging components, Chaudhry et al. [7] reviewed the latest advances in heat pipe heat exchangers (HPHEs), clarifying their unsteady heat transfer mechanisms and passive heat transfer characteristics. Srimuang et al. [8] further quantified the impacts of working fluids, work structures, and fin geometries on HPHE efficiency, offering component-level tools for optimising mine heat exchangers. Specifically for low-enthalpy mine water utilisation, Athresh et al. [9] proposed an open-loop single-shaft ground source heat pump (GSHP) system for flooded mines, verifying its technical feasibility for winter heating. Al-Habaibeh et al. [10] performed a full-season performance analysis on this system; their empirical data demonstrated that due to the stable temperature of underground mine water, the system maintains a high Coefficient of Performance (COP) even under extreme low-temperature impacts. These international studies on waste heat recovery, HPHE mechanisms, and mine-water heat pumps validate the feasibility of low-grade mine waste heat utilisation, providing a solid academic benchmark for the design and numerical simulation of the novel non-contact mine tunnel anti-freezing system presented in this paper.

2. Overall design of the wellbore anti-freezing system

This study focuses on mines with a depth of over 800 meters, where the water temperature inside the mine is above 20°C . The wastewater discharged from the mine is directed to a water tank at a height of 0 meters from the wellhead. The water stored in this 0-meter-high water tank has a static pressure and flows into the heat source side of the heat exchanger. The internal medium of the heat exchanger exchanges heat to the output side, heating the cold air ranging from 0 to -20°C that flows through the heat exchanger. This cold air then enters the mine shaft through the negative pressure air of the mine. As shown in Fig. 1.

The schematic diagram of the underground drainage low-temperature heat exchange anti-freezing well technology is shown in Fig. 2.

The scheme of the underground water low-temperature heat exchange mine anti-freezing control system is shown in Fig. 3.

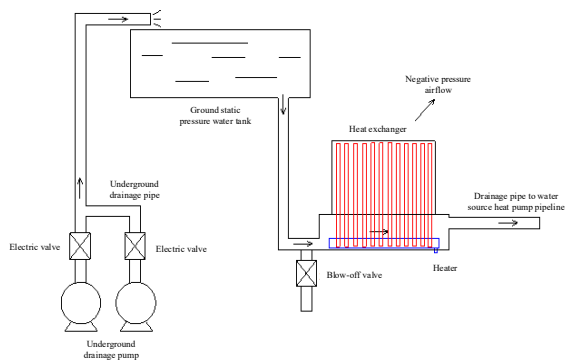


Fig. 1. Underground Drainage Low-Temperature Heat Exchange Anti-Freezing Well Technology

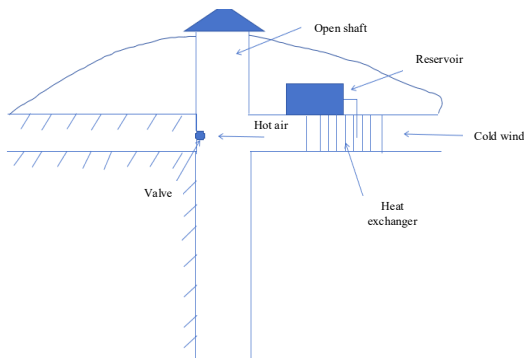


Fig. 2. Schematic Diagram of Underground Drainage Low-Temperature Heat Exchange Anti-Freezing Well Technology

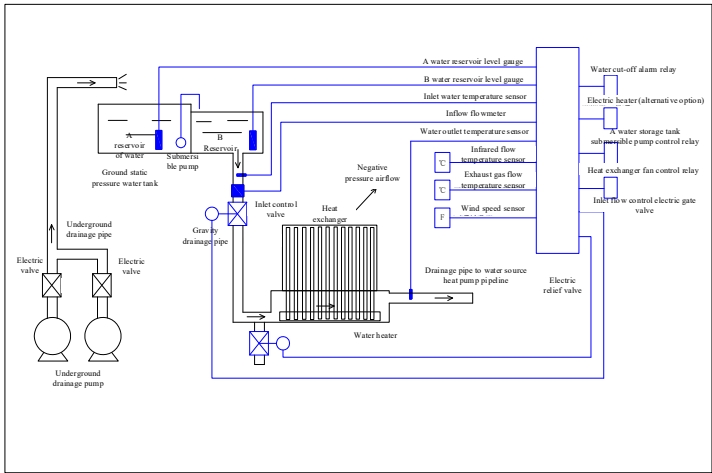


Fig. 3. Principle of the underground water low-temperature heat exchange mine anti-freezing control system

3. Numerical simulation of the mixing of hot and cold air at the entrance section of the mine tunnel

Based on the lowest winter temperature in this region and the temperature requirements for preventing freezing of the tunnels, the simulation software was used to conduct a simulation of the mixed hot and cold air at the entrance section of the tunnel, in order to determine the optimal parameters such as the heat exchange temperature and the position of the incoming air.

3.1. Establishment of simulation model

3.1.1. Simulation Model

For the mine tunnels located in the constant-temperature rock strata, the internal air temperature is no longer affected by the surface environment but is continuously heated by the surrounding hot rock strata. This study takes a constant-temperature depth of 20 to 30 meters underground as the benchmark. The area corresponding to 20 to 30 meters below the top of the mine tunnel is set as the lower boundary of the rock and soil region; that is, the height of the rock and soil region is set at 20 to 30 meters. At the same time, the width of the rock and soil region around the mine tunnel is set at 10 meters, which is used as the analysis range.

3.1.2. Simplified Assumptions

On the premise of ensuring the reliability of the simulation results, the following reasonable simplifying assumptions are made for the model.

The following model simplifications and boundary condition adjustments are adopted in this study:

- (1) Ignore local heat exchange interference inside the mine tunnel, as its intensity is far lower than the core heat exchange process and has negligible impact on the overall temperature field;
- (2) Neglect the influence of outdoor air humidity, considering the extremely low humidity in northern winter has little effect on the temperature change of mixed air;
- (3) Exclude the compression heat generation effect of the inclined shaft, which is limited in inclined tunnels and has minimal impact on heat exchange efficiency;
- (4) Simplify the boundary conditions of the hot air channel, set the wall to an ideal smooth state and adopt a unified wind speed boundary, to reduce non-core variable interference and focus on the temperature change law of cold-hot air mixing. The above simplifications do not affect the reliability of the core conclusions of the study.

3.1.3. Operating parameters for preventing freezing in mines

At the entrance of the mine tunnel, there is generally no need to install a rolling shutter door. Air is directly supplied through the inlet (TABLE 1).

(1) Numerical solution settings

After completing the grid import and verifying that there were no unreasonable negative volume elements within the computational domain, the Y-axis gravitational field, energy equation,

TABLE 1

Operating Parameters of Mine Shaft

Environmental temperature °C	Cold air volume m ³ /s
–20, –25, –30	30, 40, 50

and standard k- ϵ turbulence model were activated. Regarding the medium definition, the air physical property parameters were taken from the software material library, while the rock and soil materials were customised based on the constitutive relationships given in the following table. In terms of the solution strategy, the SIMPLE algorithm was selected to handle the pressure-velocity coupling (TABLE 2).

TABLE 2

Physical Properties Parameters of Materials

Materials	Thermal conductivity W/m·K	Specific heat capacity J/kg·K	Density kg/m ³
Soil and Rock	2.5	921.1	2500

(2) Boundary condition setting

To ensure simulation reliability and fit the actual mine tunnel operating conditions, four boundary conditions are set for the numerical model:

- 1) The mine tunnel cold air inlet is set as a pressure inlet with a static pressure of –500 Pa, and the temperature matches the local winter ambient temperature;
- 2) The end outlet of the heat exchange section is set as a pressure outlet with a static pressure of –550 Pa, calculated via the mine suction ventilation energy equation;
- 3) The section walls of the rock-soil domain are set as insulated walls to eliminate non-core heat exchange interference;
- 4) The domain bottom boundary is set as a constant wall temperature of 277.7 K ($\approx 4.55^\circ\text{C}$), corresponding to the rock-soil constant temperature layer. All settings do not affect the core simulation conclusions.

3.2. Numerical simulation of the cold air mixing temperature field at the entrance section of the mine

3.2.1. Numerical simulation of cold air mixing at the entrance section of the mine tunnel

The main analysis focuses on the influence of the constant-temperature rock and soil layer on the cold air, as well as the direct impact of the cold air from the external environment on the mine tunnel. In this case, three scenarios are simulated (Fig. 4).

From the cloud chart in the figure, it can be seen that after the cold air enters from the right air inlet after passing through the mine passage, the temperature of the cold air hardly changes, indicating that the rock layer has almost no effect on the cold air. Also, because the cold air flows downward, the temperature change above the mine is not very significant. Therefore, in summary, the influence of the constant-temperature zone rock layer on the incoming cold air can be ignored, and the air outlet is in the environment with the lowest temperature.

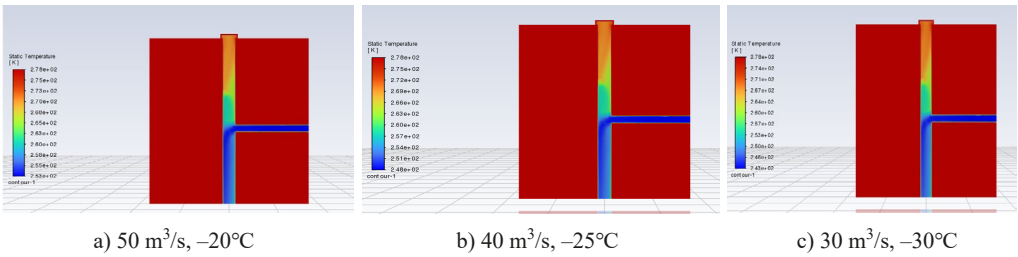


Fig. 4. Cross-sectional temperature map of the mine passage entrance showing the variations in wind temperature and wind speed on the right side

3.2.2. Analysis of the warm air temperature field at the entrance section of the mine passage

Now, assume that after passing through the heat exchanger, the airflow enters the inlet as warm air. Suppose the airflow rate after passing through the heat exchanger is 30 m³/s. Then the parameters of the warm air are as follows.

TABLE 3

Position Where Warm Wind Blows Out

The distance between the heat exchanger and the wellhead m	The temperature of the exhaust outlet °C
0, 5, 15, 20	-2, 0, 2, 5

Since the temperature of the rocks has almost no effect on the temperature inside the mine tunnel, the external rock layer can be disregarded.

The simulation results show the situation of warm air entering at 0 m (Fig. 5).

The simulation shows the situation of warm air entering at a distance of 5 meters (Fig. 6).

The simulation results show the situation of warm air entering at a distance of 15 meters (Fig. 7).

The simulation results show the situation of warm air entering at a distance of 20 meters (Fig. 8).

By analysing the situation of hot air entering the mine shaft entrance, the findings determined that at -2 and 0°C, the temperature difference between the shaft and the top was still significant. If there is moisture in the rock layer, ice formation could still occur. Comprehensive assessment indicates that heating the incoming air to 2°C is the best solution for preventing freezing in the mine. This temperature strictly meets the minimum requirements of safety regulations and can effectively prevent the shaft from freezing. Although higher stability requirements are imposed in complex conditions, precise airflow organisation and operation management can fully ensure the freezing prevention effect. Raising the target temperature to 5°C is an excessive guarantee that will lead to unnecessary energy consumption and operating costs, and does not conform to the technical principle of energy conservation and consumption reduction. Therefore, 2°C is the optimal choice for achieving both safety and economic goals.

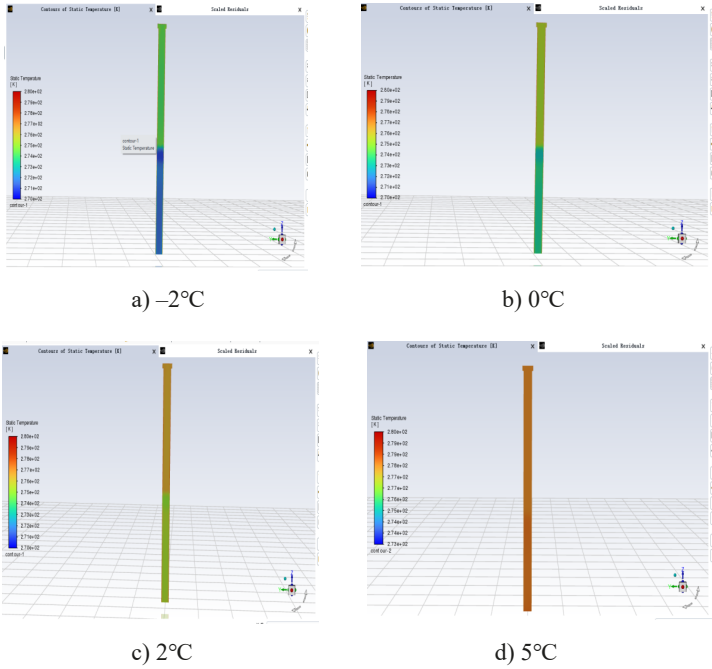


Fig. 5. Temperature contour map of the cross-section at 0m from the wellhead

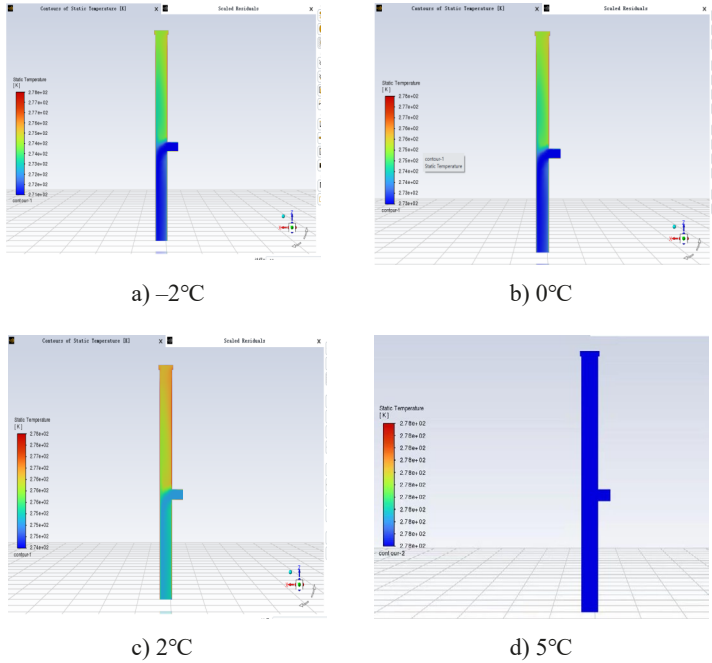


Fig. 6. Temperature contour map of the cross-section 5 meters away from the wellhead

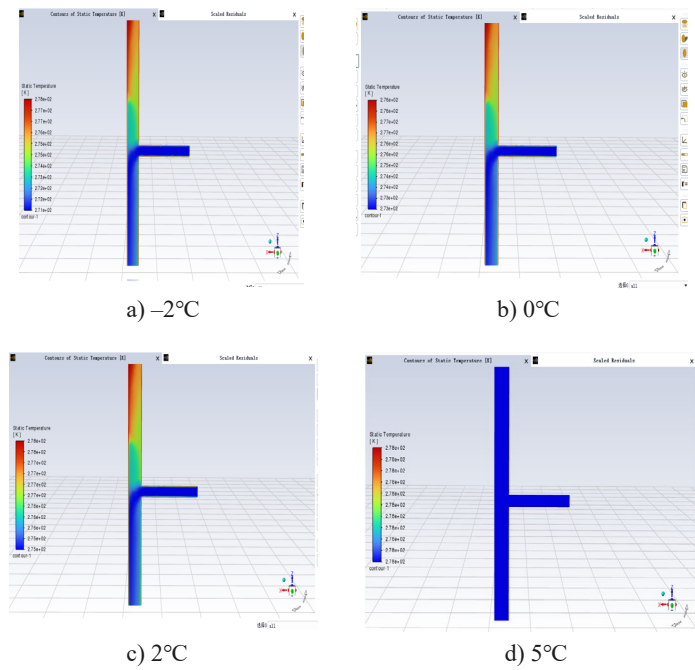


Fig. 7. Temperature cross-sectional map at a distance of 15 meters from the wellhead

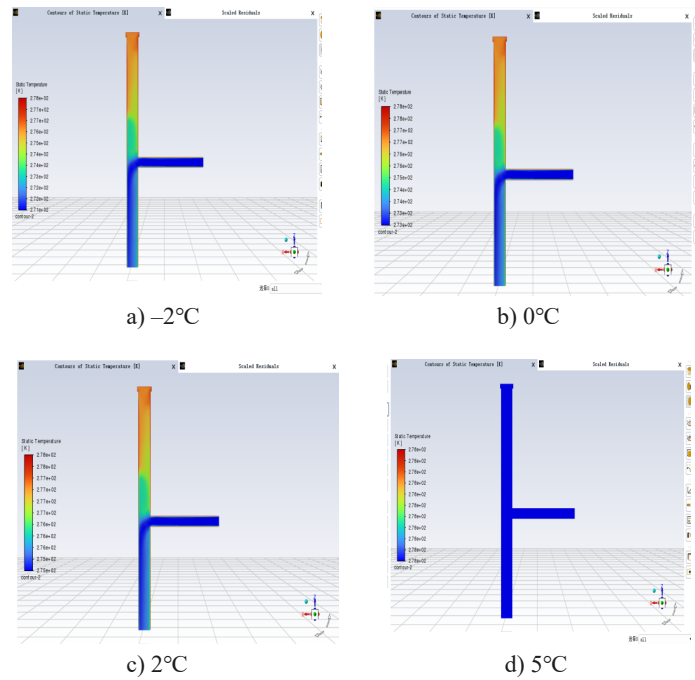


Fig. 8. Temperature cross-sectional map 20 meters away from the wellhead

4. Design and Parameter Optimisation of Low-Temperature Waste Heat Utilisation Heat Exchangers

The core of this article's low-temperature waste heat utilisation lies in the design and parameter optimisation of the efficient heat exchanger. In terms of design, the system adopts finned tube heat exchangers. This chapter focuses on the specification of the heat exchanger and determining its optimal parameters.

4.1. Basic parameters of the heat transfer medium

After conducting preliminary design, the author has determined the main technical parameters of the preheater as follows:

The operating parameters of this finned tube heat exchanger are as follows: air flow rate is $79.3 \text{ Nm}^3/\text{h}$, inlet temperature is -23°C , outlet temperature is 2°C ; water flow rate is $125 \text{ Nm}^3/\text{h}$, inlet temperature is 19°C , outlet temperature is 2°C ; the equipment power consumption is 2658 kW , and the total heat transfer area is 5757 m^2 . The structural parameters of the equipment are: overall dimensions are $2500 \times 6259 \times 4300 \text{ mm}$, total weight is 74066 Kg ; the finned tubes are of $\phi 60 \times 5 \text{ mm}$ specification, with a length of 2.5 m , a total of 3048 tubes, arranged in an array of 64 tubes per row and 49 rows, with a longitudinal spacing of 83 mm and a transverse spacing of 96 mm .

The specific parameters of the fins are as follows: thickness 1.6 mm , height 10 mm , pitch 8 mm , and the number of fins on a single finned tube is 312 . Based on the final design parameters of the finned tube air preheater (such as the number of fins, arrangement and spacing), the following finned tube layout diagram of the heat exchanger was generated, as follows (Figs. 9 and 10).

After calculation and design, the main technical parameters and structure of the finned tube air preheater have been determined:

- (1) Thermal parameters: The design heat load is 2568.55 kW , total heat transfer area is 5711.65 m^2 .

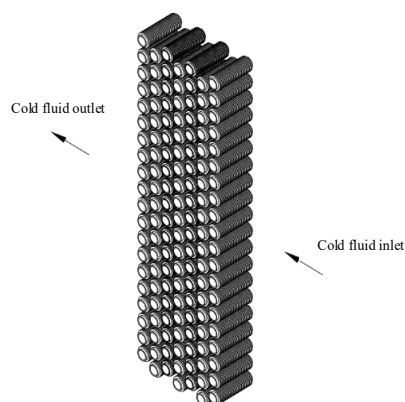


Fig. 9. Cross-sectional view of the heat exchanger

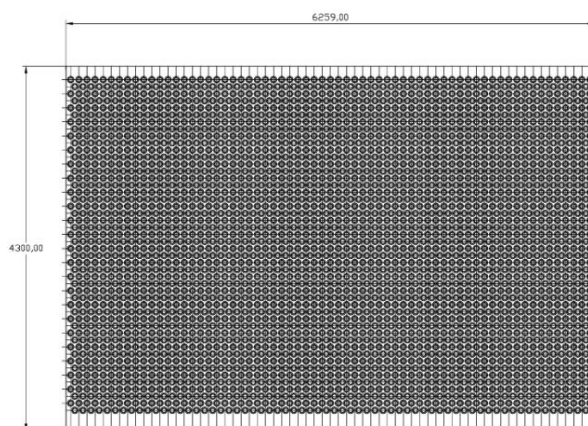


Fig. 10. Layout Diagram of Heat Exchanger Fin Tubes

- (2) Tube bundle arrangement: The total number of finned tubes is 3048, arranged in a $64 \text{ (rows)} \times 49 \text{ (columns)}$ pattern, with tube spacing (horizontal/vertical) of 96/83 mm.
- (3) Structural dimensions: The length $\times$ width $\times$ height of the preheater is $2500 \times 6259 \times 4300 \text{ mm}$.

4.2. Establishment of performance simulation model for finned tube heat exchanger

4.2.1. Geometric model establishment and simplification assumptions

1) Establishment of geometric model

Given the large scale of the research object ($64 \text{ rows} \times 49 \text{ fin tubes}$), the cost of complete modelling calculation is excessively high. Therefore, in this study, a simplification strategy based on symmetry was adopted: a representative unit composed of half a fin tube and half of the fin pitch flow channel was selected for simulation (Fig. 11). Additionally, to avoid the influence of backflow, the model inlet and outlet were extended by 1.2 times and 3.6 times the outer diameter of the fin tube, respectively.

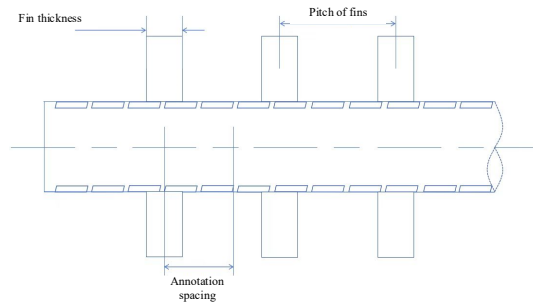


Fig. 11. Schematic diagram of the calculation area for finned tubes

The simplified geometric model is shown in Fig. 12. In the simulation, low-temperature air enters from the left inlet, flows over the finned tubes, absorbs heat and rises in temperature, reproducing the convective heat transfer process.

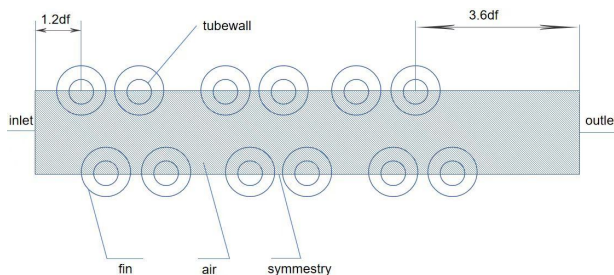


Fig. 12. Simplified Model Diagram of a Fin Tube

2) Model simplification assumptions

To establish a controllable mathematical model, the following assumptions are adopted: air satisfies the no-slip condition at the heat exchange wall; air and fluid physical properties are constant; and the temperature distribution in the flow field is uniform to simplify the model.

4.2.2. Solution Processing

1) Boundary Conditions Setting

To accurately simulate the flow and heat transfer behaviour of air within the preheater, the following core settings were established in the FLUENT solver for this study: enabling the energy equation to calculate heat transfer, using the widely applied standard k- ϵ model to characterise turbulence, and specifying the fluid medium as air. The boundary conditions for each calculation area in this numerical simulation are as follows: The fluid inlet is set as a velocity inlet boundary, with an inlet flow velocity of 38.24 m/s, an inlet fluid temperature of 250 K, a turbulence intensity of 5%, and a hydraulic diameter of 0.2845 m; the fluid outlet is set as a pressure outlet boundary, with a gauge pressure backpressure of 0 Pa, a turbulence intensity of 5%, and a hydraulic diameter of 0.2845 m; the tube wall of the finned tube is set as a constant temperature wall boundary, with the wall temperature remaining constant at 292 K; the contact area between the fluid and the solid is defined as the fluid-solid coupling interface; the symmetric area of the calculation domain adopts symmetric boundary conditions, ensuring the calculation accuracy while reducing the computational scale.

2) Setting of physical parameters

Given that the model adopts the constant physical property assumption, all physical parameters in the calculation are set as constants. Among them, the physical parameters of air are determined by the values corresponding to the average temperature at the inlet and outlet, while the physical parameters of the finned tube (solid) are taken as the standard values of the material. All physical parameters related to fluids and solids. The core thermophysical properties of the fluid and solid materials adopted in this numerical simulation are specified as follows. For the fluid medium, air, the density is 1.340 kg/m^3 , the thermal conductivity is $0.02346 \text{ W/(m}\cdot^\circ\text{C)}$, the constant-pressure specific heat capacity is $1.009 \text{ kJ/(kg}\cdot^\circ\text{C)}$, and the kinematic viscosity is $1.25 \times 10^{-5} \text{ m}^2/\text{s}$. For the solid finned tube material, the density, thermal conductivity and constant-pressure specific heat capacity are 7850 kg/m^3 , $45 \text{ W/(m}\cdot^\circ\text{C)}$ and $460 \text{ J/(kg}\cdot^\circ\text{C)}$, respectively.

4.3. Analysis of Performance Numerical Simulation Results of Finned Tube Air Preheater

4.3.1. The influence of different air inlet velocities on heat transfer performance

Numerical simulation analysis indicates that the structure of the wake zone formed on the back side of the finned tube is a key factor affecting the heat transfer and resistance performance of the heat exchanger, and the size and development degree of the wake zone are directly

determined by the magnitude of the air inlet velocity. To systematically study the influence laws of the inlet velocity on the flow and heat transfer characteristics, under the condition of keeping the structural parameters such as tube spacing, fin height and fin spacing unchanged, this study selected three typical inlet velocity conditions of $v = 10$ m/s, 20 m/s, and 30 m/s, and conducted detailed numerical simulation and comparative analysis of the velocity distribution, temperature field and pressure field on the air side of the finned tube air preheater.

1) Velocity distribution at different air inlet velocities

The velocity distribution diagrams of the finned tube heat exchanger under the inlet wind speeds of 10 m/s, 20 m/s and 30 m/s are shown in the Fig. 13.

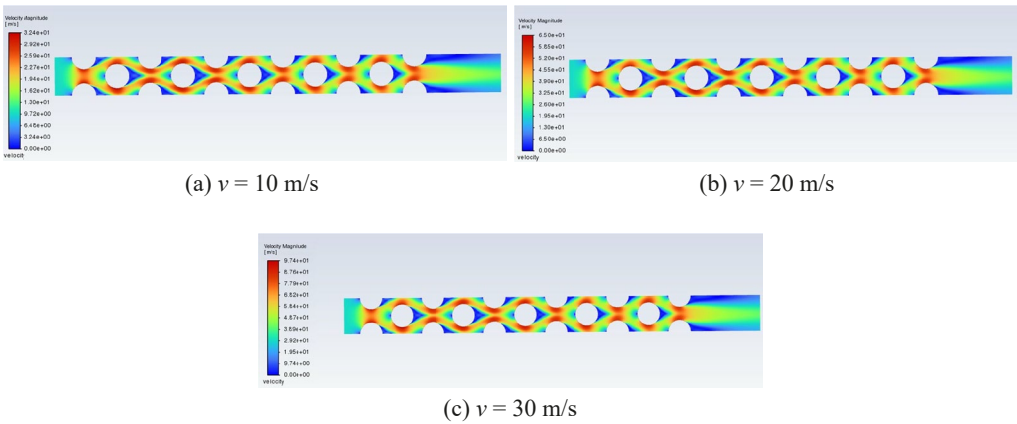


Fig. 13. Velocity distribution maps of the finned tube heat exchanger under the conditions of inlet wind speed of 10 m/s, 20 m/s and 30 m/s

The comparative analysis of different flow velocity conditions shows that when the inlet wind speed increases from 10 m/s to 30 m/s, the maximum flow velocity in the flow field (32.4, 65, 97.4 m/s) significantly increases, showing a clear positive correlation. The flow velocity distribution in the inlet section of each condition is uniform. When the gas flows through the finned tube bundle, it is deflected and circulates around due to the obstruction of the wall and the fins; the sudden contraction of the inter-tube flow channel forms a high-speed area, with the flow velocity reaching its peak. Turbulence and flow separation are clearly observed on the back side of the finned tube. As the inlet flow velocity increases, the range of the wake zone gradually shrinks, indicating that the enhancement of inertia can suppress flow separation, but even at a high flow velocity of 30 m/s, turbulence still exists, and flow separation around the finned tube is an inherent phenomenon. This conclusion can provide a reference for optimising the flow field of finned tube heat exchangers and reducing flow resistance.

2) Temperature distribution at different air inlet velocities

The temperature distribution cloud diagrams of the finned-tube heat exchanger at $v = 10$ m/s, 20 m/s, and 30 m/s are shown in the Fig. 14.

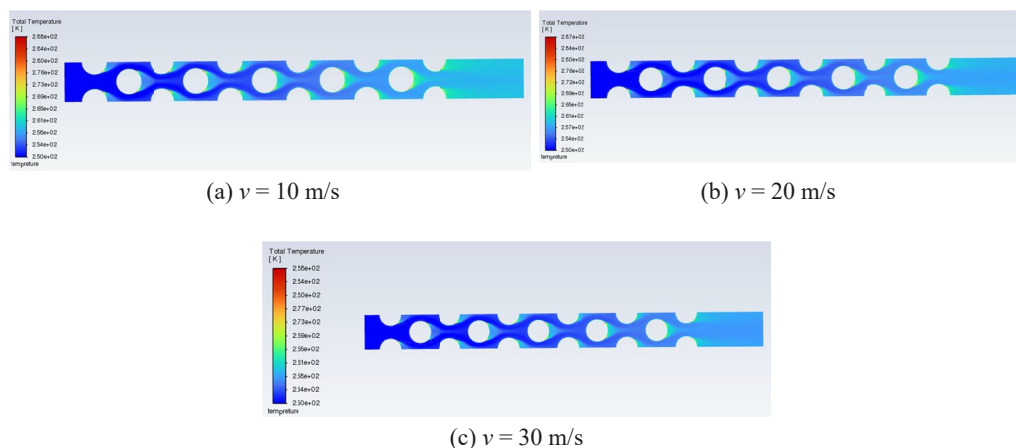


Fig. 14. Temperature distribution cloud map inside the finned tube heat exchanger at velocities of 10 m/s, 20 m/s, and 30 m/s (unit: m/s)

Under different inlet wind speeds (10, 20, and 30 m/s), the flow field temperature remained within the range of 250K to 292K. The temperature distribution in the inlet section was uniform. After the airflow entered the heat exchange zone, it underwent intense heat exchange with the finned tubes, and the temperature continuously increased along the flow direction. Increasing the wind speed would enhance the turbulence intensity of the airflow, increase the air mass flow rate per unit time, strengthen convective heat transfer, and cause the outlet temperature to rise.

The flow in the wake area on the back side of the finned tubes was sluggish. The fluid had a longer contact time with the wall, resulting in more thorough heat exchange. Therefore, the temperature was higher. This reflects the coupling characteristic between the wind speed distribution and the heat exchange efficiency: the main flow in the high-speed zone improves the overall heat exchange effect, while the local low-speed zone forms a high-temperature area due to the prolonged heat exchange time. This temperature unevenness feature can provide an important basis for optimising the finned tube layout and improving the flow field.

4.3.2. Pressure distribution under different air inlet velocities

The pressure distribution cloud diagrams of the finned tube heat exchanger inside the system under $v = 10 \text{ m/s}$, $v = 20 \text{ m/s}$, and $v = 30 \text{ m/s}$ are as shown in the following Fig. 15.

The flow field simulation results show that the system pressure decreases gradually from the inlet to the outlet. When the air passes over the finned tube bundle, it encounters resistance and forms a local high-pressure area on the windward side, while a corresponding low-pressure area appears on the leeward side. As the inlet wind speed increases, this pressure loss will significantly increase, thereby requiring the fan to bear a greater operating load.

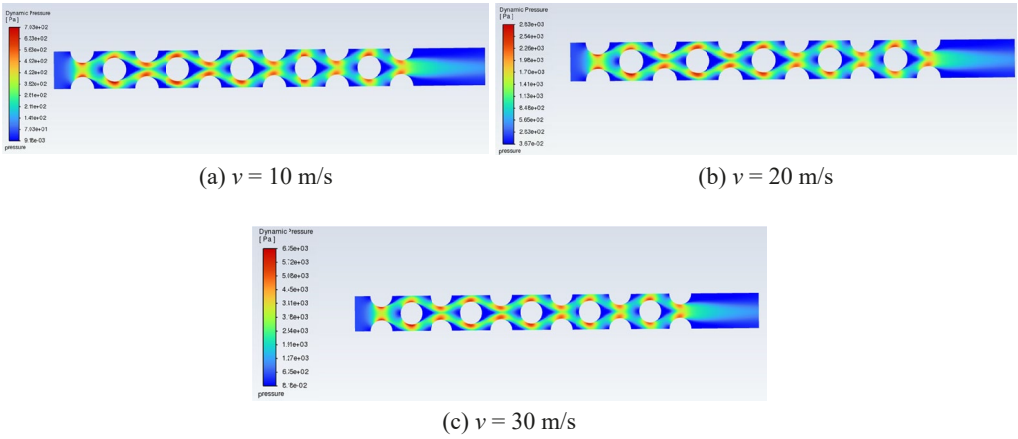


Fig. 15. Pressure distribution cloud map of the finned tube heat exchanger at $v = 10 \text{ m/s}$, $v = 20 \text{ m/s}$, and $v = 30 \text{ m/s}$ inside the device

4.2.3. Analysis of heat transfer resistance performance at different inlet air velocities

This paper simulates under the condition of maintaining constant parameters such as tube spacing and fin spacing, comparing three speed conditions of 10, 20, and 30 m/s. Based on this, it obtains the distribution data of the maximum velocity, outlet temperature, and pressure loss of the flow field under each condition. The specific results are as follows in the TABLE 4.

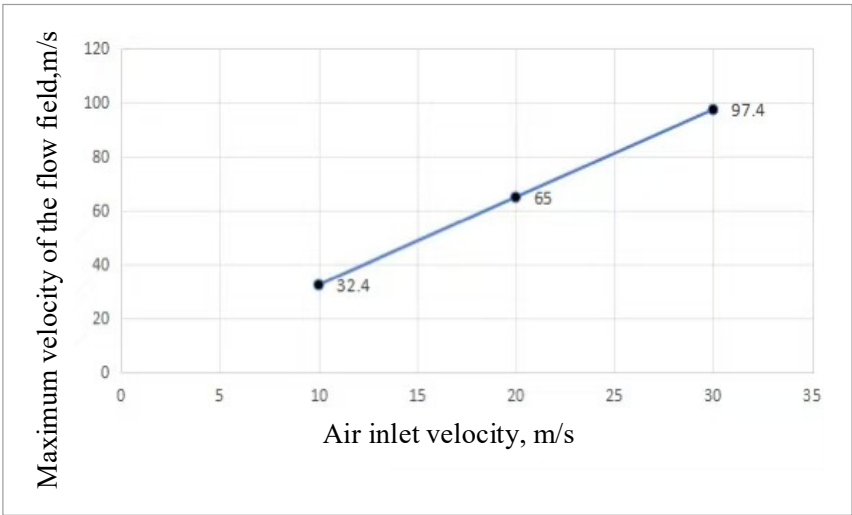


Fig. 16. Curve showing the relationship between the air inlet speed and the maximum flow field velocity

TABLE 4

The influence of different air inlet velocities on the heat transfer resistance characteristics of the heat exchanger

Air intake speed, m/s	Maximum velocity of the flow field, m/s	Outlet temperature, °C	Pressure loss, Pa
10	32.4	2	703
20	65	2	2830
30	97.4	2	6350

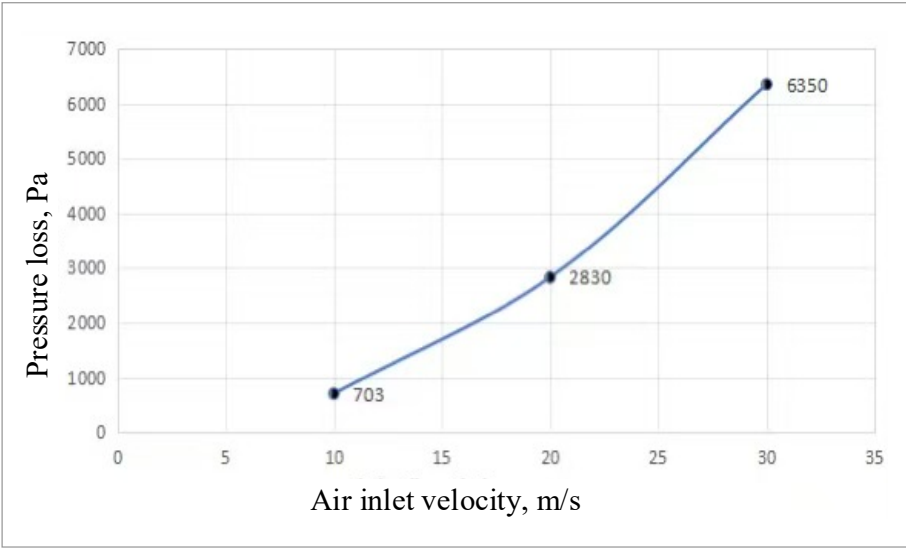


Fig. 17. Curve showing the relationship between air inlet speed and flow field pressure loss

5. Conclusion

This paper addresses the risk of shaft icing in mines during winter by utilising low-temperature waste heat from underground drainage. Through theoretical analysis and numerical simulation, a freeze-protection system design and equipment optimisation were completed. The study determined that heating incoming air to 2°C is the optimal anti-freeze temperature, and heat exchangers should be installed 5-10 meters from the shaft entrance. The optimised finned-tube heat exchanger can raise incoming air at -23°C to above 2°C, meeting anti-freeze requirements. This technology replaces traditional electric heating, enabling near-zero energy consumption operation, providing an energy-efficient and reliable solution for cold-region mines and serving as a reference for similar projects. Future work may further investigate the impact of water quality fluctuations and optimise anti-clogging design.

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