Study of Heat and Mass Transfer in Ventilation Shafts of Deep Mines in the Case of Airflow Reverse

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Abstract. This article presents the results of a simulation of conjugate heat transfer between cold air and warm shaft lining during airflow reverse at a hard rock mine. A comparison of the numerical results with collected experimental data was carried out. A significant discrepancy between the two was revealed, indicating that not all heat sources and additional air heating mechanisms were taken into account in the simulation. Possible reasons for the discrepancy are discussed and the most probable reasons are identified for further study. A quantitative assessment of the possible factors considered is carried out. Further tasks for research are determined.

1. Introduction

In the event of an accident at an underground mine, in which smoke or gassing of the mine workings occurs, in some cases reversal of the airflow is used. Technically, it is carried out by changing the direction of rotation of the fan impeller or by changing the position of special folding doors in the ventilation duct. As a result, the direction of air movement changes in the entire shaft or part of it. In addition to this, regulatory requirements in Russia and several CIS countries [1] require the reverse operation of the fans to be tested every six months. In northern regions, such as Norilsk in Russia, the air temperature on the surface in winter can drop to values of $-45 \, ^\circ C$ and below [2]. The ventilation (upcast) shafts, through which air is supplied from the surface to the underground mine workings during airflow reverse, often have no air heaters at the top. As a result, the cold air enters the shaft during the winter reverse, and the shaft walls (which are often wet) begin to freeze. This can cause breakdowns in the engineering systems located in the ventilation shafts.

It should be noted that the air supply (downcast) shafts are usually equipped with air handling (conditioning) systems for air heating to a temperature of $+2 \, ^\circ C$ or higher. In normal ventilation mode, air is supplied through downcast shafts to underground mine workings, and the problem of freezing the shaft walls and the equipment in them does not occur. If ventilation shafts are equipped with air handling systems, airflow reverse would not be a problem for them; however, due to cost savings, mining enterprises usually do not install air heaters on the ventilation shafts. The possible explanation is that the reverse time is short (1–4 hours) and the shaft does not have time to freeze.

This article discusses the problem of airflow reverse in the winter season for a hard rock mine in the Norilsk region. This study is concerned with the air-conditioning system, which was installed at the mine studied in 2019–2020. According to the project for the underground air-conditioner construction, the water–air heat exchangers, which are used to release heat from the system, should be installed in the main ventilation airways near the ventilation shaft. When the cold air is supplied in reverse ventilation mode to the heat exchangers, they can fail and require repair with replacement of the elements. For this reason, it became necessary to regulate and improve the airflow reverse procedure, by determining the parameters at which airflow reverse is permissible (outside air temperature, reverse duration, etc.). The development of measures for the regulation of airflow reverse requires both experimental studies of the air movement during the reverse and simulation of heat and mass transfer in the mine workings, in order to predict temperature fields in the shaft in possible hazardous and emergency situations.
Obtaining reliable model data requires adjusting the parameters of the mathematical model of heat and mass transfer processes for a specific object, by conducting an experimental study on that object. In this article, we will discuss the experience of using a mathematical model of conjugate heat transfer in the space of a ventilation shaft, the shaft’s concrete lining, and the surrounding rock mass under conditions of airflow reverse and cold air entering the shaft.

2. Mathematical model

A coupled model of heat and mass transfer in the ventilation shaft was developed [3–7], which facilitates calculation of the air temperature at any depth of the shaft. This model takes into account the following physical processes:

- heating of air in the main fan;
- convective heat exchange between air and the shaft lining;
- heating of air due to its adiabatic compression during downward movement in the shaft [8, 9];
- decrease of the shaft lining and rock mass temperatures over time.

In reverse ventilation mode, outside air flows primarily through the fan (see Figure 1). As a result, the air is heated. The temperature increase can be determined from the fan power according to the formula:

\[ \Delta T_f = \frac{N_f}{\rho a_0 c_v Q^*} \]

where \( Q^* \) is the fan discharge in reverse ventilation mode, m³/s; \( \rho a_0 \) is the air density at the surface (or at the fan inlet), kg/m³; \( c_v \) is the specific heat capacity of air, kJ/kg; and \( N_f \) is the power transmitted to the fan impeller, kW.

Convective heat exchange between the air and the shaft lining is calculated using the following one-dimensional heat balance equation for air in the vertical shaft section of length \( dx \)

\[ \rho_a c_v \frac{dT}{dx} = \frac{P_a}{\rho_a} \frac{d\rho_a}{dx} + 4 \alpha \frac{m}{V} (T_m - T), \]

the constitutive relation

\[ \rho_a = \rho a_0 \left( \frac{P_a}{P_{a0}} \right)^{\frac{1}{\gamma}}, \]

and the boundary condition

\[ T|_{x=0} = T_a + \Delta T_f, \]

where \( \rho_a \) is the air density in the part of the shaft studied, kg/m³; \( c_v \) is the specific heat capacity of air, kJ/kg; \( x \) is the coordinate along the axis of the shaft, m; \( P_a \) is the barometric pressure of air, Pa; \( D \) is the shaft diameter, m; \( \alpha \) is the heat transfer coefficient, W/(m²·°C); \( V \) is the air velocity, m/s; \( T_m \) is the temperature of the shaft wall, °C; \( T \) is the air temperature in the space of the shaft, °C; \( \gamma \) is the adiabatic exponent for air (which is equal to 1.4); and \( T_a \) is the air temperature at the fan inlet, °C.
The coordinate system $Ox$ is chosen in such a way that at the surface of the shaft we have $x = 0$, and at depth $L$, which corresponds to the underground level of the mine, $x = L$. The shaft has a circular cross-section – this property is used to derive equation (2).

The heat transfer equation for the shaft lining and surrounding rock mass with corresponding boundary and initial conditions can be written as:

$$\frac{\partial T_m}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r} \left( a_m r \frac{\partial T_m}{\partial r} \right),$$  \hspace{1cm} (5)

$$\lambda_m \frac{\partial T_m}{\partial r}_{r=R_s} = \alpha (T - T_m),$$  \hspace{1cm} (6)

$$T_m |_{r=R_{out}} = T_0(z),$$  \hspace{1cm} (7)

$$T_m |_{t=0} = T_0(z),$$  \hspace{1cm} (8)

where $a_m$ is the thermal diffusivity of the concrete lining/rock mass, $m^2/s$; $r$ is the radial coordinate, $m$; $\lambda_m$ is the thermal conductivity of the rock mass, W/(m·°C); $R_s$ is the radius of the shaft, $m$; and $R_{out}$ is the radius of thermal influence, $m$. The radius of the thermal influence represents the distance from the shaft at which, in the period of simulation studied, the temperature does not change significantly as a result of the thermal effect of cold air. This parameter was determined in the course of preliminary calculations.

To simplify the calculation, the following assumptions were made in the model:

- The concrete shaft lining and rock mass are homogeneous and isotropic bodies; variations in the thermophysical properties of different rock layers are not taken into account.
- The phase transitions of moisture in the permafrost rock mass in the depth interval 0–100 m are neglected.
- The shaft surface is dry; moisture exchange is not taken into account.
- The cross-section of the shaft is constant along the entire length.
- Air velocity is constant along the entire length of the shaft.
- The heat capacity of air does not depend on its thermophysical properties.
- The initial distribution of the shaft wall temperature is assumed to be equal to the air temperature in normal ventilation mode before the start of the reversal.
- All power supplied to the main fan impeller is spent on heating the air.

3. Results of simulation and experimental study

The model described was implemented numerically in the computer algebra package Wolfram Mathematica. The numerical solution was performed by the method of finite differences, using the Runge–Kutta scheme of the fourth order. The model parameters were based on the collected data from the object under study – the VS-9 shaft of a mine in the Norilsk region – $L = 850$ m, $D = 8$ m, $T_{a} = -50$ °C, $V = 8$ m/s. The shaft lining is made of monolithic concrete M300, and the rock mass is composed of fractured basalts. The heat transfer coefficient $\alpha$ was taken on the basis of the classical concepts of heat transfer in the shafts according to the Shcherban formula [10, 11]:

$$\alpha = 3.4 \cdot \frac{V^{0.8}}{D^{0.2}}$$  \hspace{1cm} (9)

The simulation results are shown in Figure 2. The lines show the temperature distribution in the shaft for different durations of airflow reverse. As can be seen from the figure, the calculated air temperature at the start of the reversal at a depth of 850 m (the depth of joining with the underground level) was $-17.8$ °C, and after half an hour it was $-23.6$ °C and continued to fall. At this temperature, heat exchangers that use water will quickly fail. Another difficulty in this case will be the evacuation of mine workers to the surface along the ventilation shaft. At such low air temperatures and high velocity, frostbite is likely.
Thus, numerical simulation in the first stage of the study yielded disappointing conclusions about the possibility of reversing the airflow in the cold season at the mine studied. In the second stage of the study, a full-scale experiment was carried out during a test reversal at the mine in November. It was planned that experimental studies of the reversal in situ would allow us to validate the model. The air temperature at the surface at the time of the measurements was –23 °C. The temperature measurements in the shaft were carried out at the joining of the shaft with the underground level at a depth of 850 m. The measurement results are summarized in Table 1. This table also presents the results of calculations of the model, when the temperature of air at the surface was set to –23 °C.

| Duration of airflow reverse | Measured value | Calculated value |
|----------------------------|----------------|------------------|
| 24 min                     | 12.7 °C        | –4.0 °C          |
| 37 min                     | 12.1 °C        | –4.5 °C          |
| 60 min                     | 8.0 °C         | –5.1 °C          |
| 91 min                     | 5.8 °C         | –5.7 °C          |

From the comparison, it was found that the experimental and calculated temperatures differ greatly from each other: the difference ranges from 16.7 °C to 11.5 °C with a decreasing difference over time. The heat exchange between the air and the shaft walls actually occurs much faster in the experiment than in the model. At the same time, the air has a smaller thermal effect on the surrounding concrete lining and rock mass, since over time the air temperature at the shaft bottom drops in the experiment more slowly than in the model. This suggests that either the model uses parameters that do not correspond to reality, or the model does not take into account some important physical processes that take place in practice.

The main parameter responsible for the heat exchange between the air and the shaft lining and, accordingly, air heating, is the heat transfer coefficient, which in the model is determined by the Shcherban formula (9). First of all, we decided to analyze this formula. From the primary source [10], it follows that the coefficient in this formula can vary within wide limits, depending on the type of surface and the average roughness height. For this reason, we tried to adjust the model by choosing this parameter in such a way that it would provide us with calculated values that are consistent with the results of field measurements. It was found that the heat transfer coefficient had to be increased by more than 100 times, which, despite the inaccuracy of the formula, is unrealistic.

The heat transfer coefficient (9) strongly depends on the air velocity in the flow core. In conditions of high temperature gradients of the air flow, the velocity field can be significantly distorted due to the influence of thermal convection [11]. The effect of thermal convection is determined by such...
dimensionless complexes as the Reynolds number (Re) and the Rayleigh number (Ra). In the problem under consideration, we tried to quantify these complexes:

\[
\text{Re} = \frac{VD}{\nu} \approx 7.3 \times 10^8, \quad \text{Ra} = \frac{\beta g \Delta T D^3}{\nu \chi} = 4.8 \times 10^{14} \quad (10)
\]

Where \( \nu \) is the kinematic air viscosity, \( m^2/s \); \( \beta \) is the coefficient of thermal expansion, \( ^{\circ}C^{-1} \); \( g \) is the gravitational acceleration; \( \Delta T \) is the temperature difference, \( ^{\circ}C \); \( \chi \) is the air thermal diffusivity, \( m^2/s \).

If we estimate the value of the heat transfer coefficient \( \alpha_{nat} \) for natural convection using Churchill correlation [12] and compare it with the heat transfer coefficient \( \alpha_{for} \) from (9) for purely forced convection, then we get that

\[
\frac{\alpha_{nat}}{\alpha_{for}} = \frac{0.151 \lambda}{D} \frac{Ra^{1/3}}{Pr} \left( 1 + \left( \frac{0.436}{Pr} \right)^{9/16} \right)^{16/27} \cdot \frac{1}{3.4} \frac{1}{D^{0.2}} \approx 2.13, \quad (11)
\]

where \( Pr \) is the Prandtl number for turbulent airflow (~ 0.7). Estimated calculations show that the role of thermal convection in this case is large, and the corresponding heat transfer coefficient is more than two times higher than when considering only forced convection. Nevertheless, such an increase in the intensity of heat transfer still does not explain the experimentally observed temperature drops in the shaft. For this, the heat transfer coefficient should increase by 1–2 orders of magnitude.

Another option for adjusting the model consisted of the simultaneous change of two parameters—the initial temperature of the rock mass and the heat transfer coefficient. In this case, due to an increase in the initial temperature of the rock mass, it was possible to achieve consistent values with a heat transfer coefficient increase of only 2–3 times. As a result, it was concluded that the model does not take into account some important physical mechanisms [13], which lead to air heating in the shaft.

In this stage of the study, a new research task was formulated—to modify the mathematical model (1)–(9) so that it is capable of predicting air temperatures in the shaft that correspond to the experimental ones. For this, in the first stage it is necessary:

- to determine the possible reasons for the discrepancy between the calculated and experimental data based on actual data about the shaft;
- to estimate the contribution of potential heat sources to the resulting air temperature increase;
- to take into account the potential heat sources in the mathematical model and to recalculate the temperature field.

4. Possible reasons for additional air heating

It should be noted that the model presented (1)–(9) behaves well when calculating heat distribution in mines for long periods of time (more than one year) in normal ventilation mode. In this case, the shaft lining and several meters of the surrounding rock mass have time to reach air temperature. Seasonal temperature fluctuations certainly have some small impact; however, for deep mines this impact is often ignored [7]. Since the rock mass has a temperature close to the air temperature in the shaft, calculation of the heat transfer in the shaft assumes that the following physical factors can be ignored: the presence of metal structures (cast-iron tubing) with non-zero heat capacity in the shaft [14], the presence of moisture in the shaft, and ventilation drift [15–17]. In addition, earlier the model was tested mainly under normal ventilation mode, when the temperatures in the shaft have strictly positive values.

However, when reversed cold air enters the wet shaft, an additional source of heat due to moisture crystallization may be significant [15]. These are not all of the possible factors leading to additional heat input into the shaft atmosphere, but we started our analysis with them.

The presence of moisture in the shaft can intensify the heat transfer between the air and the shaft walls and lead to a higher value of \( \alpha \). The presence of thick-walled metal structures of non-zero heat capacity in the shaft can lead to the fact that the cooling of these metal structures, as well as the concrete lining and rock mass, will occur more slowly over time. Mathematically this leads to the need to reduce the thermal diffusivity \( \alpha_{nat} \) in (5). In the following subsections, we will make a simplified quantitative assessment of these additional temperature factors.
4.1. Influence of cast-iron tubing

According to preliminary estimates, the majority of the metal structures in the VS-9 shaft are made up of cast-iron tubing with weight 3,575 tons. The weight of the rest of the structures is insignificant and is not taken into account. The presence of tubing can have an effect not only on the rate of cooling of concrete lining and rocks, which are mentioned above, but also on the coefficient $\alpha$. Due to the rougher inner surface of the tubing, the actual experimental heat transfer area and heat transfer coefficient can be greater than those in the model [16], which in combination with a higher temperature at the surface will give a more intense heat inflow to the air. To assess the influence of cast-iron tubing on air heating, a quantitative analysis was carried out. For this, the following initial data were taken (see Table 2).

### Table 2 – Parameters of cast-iron tubing and air

| Specific heat capacity of cast-iron tubing ($c_t$), J/(kg°C) | Mass of tubing ($m_t$), kg | Average temperature of tubing ($T_{i \text{ave}}$), °C | Mass airflow in reversal case ($G_a$), kg/s | Specific heat capacity of air ($c_a$), J/(kg°C) | Average difference between measured and simulated temperature ($\Delta T_a^\text{ave}$), °C |
|------------------------------------------------------------|---------------------------|---------------------------------------------|---------------------------------|---------------------------------|---------------------------------|
| 540                                                        | 3 575 000                 | 16.2                                        | 540                             | 1005                            | 14.5                            |

The idea was to determine the time during which the tubing would cool to 0 °C and provide such quantity of heat to the air that would explain the mismatch between the measured and calculated temperatures. This time span of influence was determined by this simple formula:

$$t = \frac{c_t \cdot m_t \cdot T_{i \text{ave}}}{c_a \cdot G_a \cdot \Delta T_a^\text{ave} \cdot 60} \approx 66 \text{ min}$$  \hspace{1cm} (12)

As a result, it turned out that the thermal energy of the tubing when it is cooled to a temperature of 0 °C is enough to heat the air by 14.5 °C for 66 minutes. This time span is comparable to the airflow reverse time.

4.2. Influence of shaft water

It was determined experimentally that water in the shaft appeared for two reasons:

- due to water inflows from aquifers through cracks in the lining and gaps in the tubing;
- due to condensation of moisture from the humid air, when it rises up the shaft [17].

As a result, at the start of the airflow reverse, the shaft walls are wet, and the ventilation drift is flooded by water (the average height of the water layer in the drift is nearly 10 cm). When the airflow is reversed, dripping moisture can be carried away from the ventilation drift into the shaft. The heat exchange between the cold air and warm droplet moisture and the subsequent crystallization of moisture droplets lead to a significant heat flow from water to air.

To estimate the time span of the possible effect of water on air heating, the following initial data were taken (see Table 3).

### Table 3 – Parameters of water and air

| Specific heat capacity of water ($c_w$), J/(kg°C) | Mass of water ($m_w$), kg | Initial average temperature of water ($T_{w \text{ave}}$), °C | Specific heat of water crystallization ($\lambda_w$), J/kg | Mass airflow in reversal case ($G_a$), kg/s | Specific heat capacity of air ($c_a$), J/(kg°C) | Average difference between measured and simulated temperature ($\Delta T_a^\text{ave}$), °C |
|-------------------------------------------------|---------------------------|---------------------------------------------|---------------------------------|---------------------------------|---------------------------------|---------------------------------|
| 4180                                            | 59 400                    | 16.2                                        | 330 000                         | 540                             | 1005                            | 14.5                            |

The time span of influence was determined by this formula:

$$t_w = \frac{(c_w \cdot T_{w \text{ave}}^\text{top} + \lambda_w) \cdot m_w}{c_a \cdot G_a \cdot \Delta T_a^\text{ave} \cdot 60} \approx 50 \text{ min}$$  \hspace{1cm} (13)
As a result, it turned out that the thermal energy of the water, when it is cooled to a temperature of 0 ℃ and completely crystallized, is enough to heat the air by 14.5 ℃ for 50 minutes. This time span is also comparable to the airflow reverse time.

5. Conclusion
As a result of the study, the most probable causes of additional air heating in the shaft were identified and their quantitative assessment was made. Based on the results obtained, the tasks of the next stage of the study were determined:

1. Collection of additional experimental data when carrying out airflow reverse in various mines:
   - air temperature and relative humidity (at the surface and at the underground levels);
   - temperature of shaft walls;
   - presence of moisture in the shaft and the ventilation drift;
   - presence of water inflows in the shaft from the cracks in the lining;
   - geometry of cast-iron tubing.
2. Comparison of the experimentally measured and numerically predicted results.
3. Search for and description of additional heat exchange mechanisms in various special cases; determination of the actual values for the heat transfer coefficient in mine shafts.
4. Improvement of the existing model of coupled heat and mass transfer in the shaft.

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