www.czasopisma.pan.pl PAN POLSKA AKADEMIA NAUK www.journals.pan.pl

Arch. Min. Sci. **64** (2019), 4, 785-796

Electronic version (in color) of this paper is available: http://mining.archives.pl

DOI 10.24425/ams.2019.131066

QI YUDONG*[#], CHENG WEIMIN**, XIN SONG**

THEORETICAL ANALYSIS OF TEMPERATURE RISING FOR CHILLED WATER IN THE LONG DISTANCE TRANSPORT PIPELINES IN COAL MINE

ANALIZA TEORETYCZNA WZROSTU TEMPERATUR OCHŁODZONEJ WODY W TRANSPORCIE RUROCIĄGAMI NA DUŻE ODLEGŁOŚCI W KOPALNIACH WĘGLA

In order to provide sufficient cooling capacity for working and heading faces of the coal mine, chilled water is often transported a long distance along pipelines in deep mine, which inevitably results in its temperature rising owing to heat transfer through pipe wall and the friction heat for flow resistance. Through theoretical models for temperature increasing of the chilled water were built. It is pointed out that the temperature rising of the chilled water should be considered as a result of the synergy effects of the heat transfer and the friction heat, but theoretical analysis shows that within engineering permitting error range, the temperature increasing can be regarded as the sum caused by heat transfer and fraction heat respectively, and the calculation is simplified. The calculation analysis of the above two methods was made by taking two type of pipe whose diameters are *De*273 × 7 mm and *De*377 × 10 mm, with 15 km length in coal mine as an example, which shows that the error between the two methods is not over 0.04°C within the allowable error range. Aims at the commonly used chilled water diameter pipe, it is proposed that if the specific frictional head loss is limited between 100 Pa/m and 400 Pa/m, the proportion of the frictional temperature rising is about 24%~81% of the total, and it will increase with high flow velocity and the thin of the pipe. As a result, the friction temperature rising must not be ignored and should be paid enough attention in calculation of the chilled water temperature rising along pipe.

Keywords: heat transfer; friction heat; model; theoretical analysis

W celu zapewnienia odpowiedniego chłodzenia dla urządzeń górniczych wykorzystywanych do prac wydobywczych i urządzeń do drążenia tuneli w kopalni konieczny jest transport ochłodzonej wody rurociągami, nierzadko na znaczne odległości w obrębie kopalni podziemnej. Transport rurociągami nieuchronnie prowadzi do wzrostu temperatury wody wskutek wymiany ciepła poprzez ścianki rurociągu i wskutek tarcia związanego z oporem przepływu. Opracowane zostały modele teoretyczne wzrostu temperatury wody ochładzanej; na ich podstawie wskazano, że wzrost temperatury wody rozpatrywać należy jako oddziaływanie efektu synergii pomiędzy wymianą ciepła i ciepłem tarcia. Analiza teoretyczna wykazuje jednak, że

* SHANDONG PROVINCIAL KEY LABORATORY OF CIVIL ENGINEERING DISASTER PREVENTION AND MITIGATION, SHANDONG UNIVERSITY OF SCIENCE AND TECHNOLOGY, QINGDAO 266590, SHANDONG, CHINA;

** KEY LABORATORY OF MINE DISASTER PREVENTION AND CONTROL, SHANDONG UNIVERSITY OF SCIENCE AND TECHNOLOGY, QINGDAO 266590, SHANDONG, CHINA)

[#] Corresponding author: yudong_qi@163.com

przy poziomie błędu dopuszczalnego w praktyce inżynierskiej, wzrost temperatury wody traktować można jako zwykłą sumę wpływu wymiany ciepła i ciepła tarcia, tym samym znacznie upraszczając procedurę obliczeniową. Weryfikację wyników obliczeń otrzymanych w oparciu o dwie wymienione metody przeprowadzono poprzez zbadanie przepływu wody w kopalni węgla, rurociągiem o długości 15 km złożonym z dwóch rodzajów rur, o wymiarach *De* 273 × 7 mm i *De* 377 × 10 mm. Wykazano, że różnica wyników uzyskanych dla obydwu metod nie przekraczała 0.04°C, przy założonym dopuszczalnym poziomie błędu. W odniesieniu do typowych rodzajów rur używanych do przesyłania wody ochłodzonej proponuje się ograniczenie dopuszczalnego spadku ciśnienia w przewodzie w granicach 100 Pa/m-400 Pa/m. Udział wzrostu temperatury wskutek oddziaływania sił tarcia waha się w granicach 24%-81% i rośnie wraz ze wzrostem prędkości przepływu i ze zmniejszaniem grubości ścianek rury. Jest więc rzeczą oczywistą, że wzrost temperatury wody wskutek oddziaływania sił tarcia nie może być pomijany i winien zostać odpowiednio uwzględniony przy obliczeniach wzrostu temperatury wody w rurociągu.

Słowa kluczowe: wymiana ciepła, ciepło tarcia, model, analiza teoretyczna

1. Introduction

With the improvement of mechanization and the increasing depth of coal mine, the number of hot mines is increasing and the thermal disaster is becoming more severer. In order to control the thermal disaster, pipelines are laid in the roadway for chilled water and cooling capacity transportation in most refrigeration systems of the mine, and the transportation distance is about a few kilometers and even more than ten kilometers (Yang Deyuan & Yang Tianhong, 2009; Chu Z., Ji J., Zhang X. et al., 2016). The temperature rising and cooling loss of the chilled water along the pipelines are both high, so it should be fully considered (Qi Yudong, 2010; Wojciechowski, 2013).

Because temperature difference between the pipeline inside and the tunnel air, the temperature rising and cooling loss of the chilled water along the pipelines is always considered inevitable and only can be controlled in an acceptable range. Enhancing the insulation thickness and choosing fine insulation materials are common measure. For another reason, the chilled water transported by the complicated and enormous pipeline system and even stays over ten hours, so the hydraulic disorder is very common and even more serious. In order to supply sufficient chilled water to the demand of different coal faces and heading faces, optimizing the pipelines layout, diameter and control valve for better hydraulic resistance balance are all important (Chan & Chow, 2007; Dalla Rosa Li & Svendsen, 2011; Jie et al., 2012; Söderman, 2007; Li & Svendsen, 2012). For the long piping network of the chilled water system, first only know where and how much the cooling losses, then can take proper measures to improve the efficiency of transmission and distribution system. The cooling losses in transmission and distribution system are affected by pipe diameters and the insulation material used, as well as the temperature of the chilled water in the supply and return pipes. Pipe diameters have an impact on pressure loss in the cooling system and consequently on the electrical energy consumption of pumps. The electrical energy consumption of pump is also influenced by the flow rate of the chilled water and it is proportional to the third power of flow rate (Tol & Svendsen, 2012). For the central air-conditioning system, the supply and return water temperature difference in the circulation network greatly affect the energy consumption of the cooling system (Lin et al., 2001; Cygankiewicz, & Knechtel, 2014). For a given cooling load, the energy consumed by the circulating pumps in the network is determined by the temperature difference between the supply and return water. Decreasing the temperature difference increases the flow rate of the chilled water and the energy consumption of the pumps. The energy consumption of the pumps is almost turned into heat because of the fraction among fluid and between fluid and inner of the pipelines while the water flowing along the pipe, and causes the water temperature increasing (Bar et al., 2010; Rishel Durkin & Kincaid, 2006). The temperature increasing produces a negative effect to mine cooling, because the fractional heating rising the chilled water temperature and equivalent to offset the cooling capacity of the chilled water. As a result, special attention needs to be paid to the temperature rising of the chilled water along the pipelines (Pirouti et al., 2013; Zhu Yingxin et al., 2008).

The reason/energy of fractional temperature rising for chilled water is the pump or the electric motor, electrical energy is converted into mechanical energy, then converted into thermal energy and causes the chilled water temperature rising (Petitjean, 1994). The temperature rising caused by fraction can be calculated by formula (1) (Qi Yudong, 2010; Szlązak, Obracaj, Swolkień et al., 2016):

$$\Delta t_1' = \frac{N \cdot \eta}{\rho c V} \tag{1}$$

Here, $\Delta t_1'$ stands for temperature rising for pump work, °C; N is the shaft power of the pump, W; η is the pump efficiency, %; ρ stands for the density of the water, $\rho = 1000$ kg/m^3; c is the specific heat of water, $c = 4.186$ kJ/(kg.°C); V stands for the pump flow rate, m^3/s.

But formula (1) only gives the total average temperature rising for friction. As shown in Fig. 1, we can calculate $\Delta t_1'$ by formula (1), and it means the frictional temperature rising of the chilled water along the whole pipelines, that is from the outlet of the pump, along different branches and return to the entrance of the pump. But we can't know the fractional temperature rising between arbitrary two points along the flow, such as from II to III, III to IV, and so on.

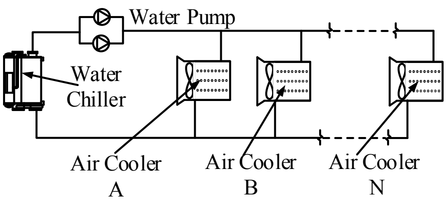


Fig. 1. Schematic of Mine Cooling

As we all know the temperature increasing and cooling loss along the pipelines of the chilled water is the total effect of heat transfer for temperature difference and fraction heat (Chandel Misal & Beka, 2012; Rishel et al., 2006; Henze & Floss, 2011). But some calculation problems still exist in different references about the temperature rising and heat transfer of the pipelines currently; for example,

(1) In most cases only the heat transfer through the pipe wall and the insulation materials are considered, while the temperature rising for friction is always ignored. The frictional temperature rising could be ignored or not? What proportion of the frictional temperature is in the total temperature rising? But all these we can't give a definite answer.

(2) Entrance temperature of the chilled water is known generally, but the temperature of the outlet or the changing rule along the pipelines needs to be predicted in the long distance transport for practical engineering project, and there is no formula or equation to expressing them.

(3) The cooling loss of the heat transfer along the pipelines is under the assumed outlet temperature or a constant temperature of the chilled water (Rawlins, 2007; Petitjean, 1994), but in fact the temperature of the chilled water increases along the direction of the flow. So it is not suitable to calculate the cooling loss under an assumed outlet temperature. Therefore, such problems about the temperature change rule of the chilled water along the pipe will be analyzed theoretically in this paper.

The temperature difference between the start and the terminal or the temperature change rule along the pipelines is mainly affected by the temperature difference between the inside and outside of the pipe and the friction of the flow. Based on such consideration, two methods for the calculation of the terminal temperature of the chilled water are given. The first method is that the temperature rising is caused by flow friction and heat transfer respectively. And after the calculation of the temperature rising of the friction Δt_1, and the rising of the heat transfer Δt_2 respectively, the total temperature rising $\Delta t'$ (or temperature difference of the start and the terminal) is their sum, that is $\Delta t' = \Delta t_1 + \Delta t_2$. The second method is that the temperature rising is caused by the synergistic effect of the heat transfer and the friction, so the influence of the heat transfer and friction on the temperature rising Δt should be considered simultaneously, which is not done as simply as their sum. In this paper, the above two methods will be compared theoretically, and the influence of heat transfer and frictional heat on the total temperature rising will be also analyzed.

2. Theoretical analysis of the temperature rising under the respective effect of the heat transfer and frictional heat (First Method)

Both heat transfer through the pipe wall and frictional resistance are all affect the temperature rising of the chilled water along the pipe. So it is not appropriate or even mistake for only considering heat transfer or friction. But we can know and compare their relative influence for chilled water temperature increasing.

2.1. Temperature rising caused by frictional heat

As is shown in Fig. 2, the total length of the chilled water pipe is L with its inside radius r_1, entrance temperature t_0, but its outlet temperature t_1 is unknown. The basic assumptions of the model are: (1) The local resistance is neglected and it is considered that the temperature rising of the friction is only caused by fraction-head loss; (2) The heat transfer between the inside and

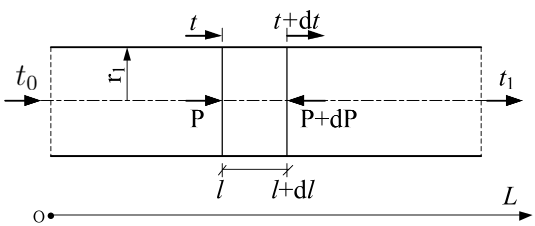


Fig. 2. Model for temperature rising of fractional heat

outside of the pipeline is neglected for temperature difference; (3)The inside of the pipe is adiabatic and the energy loss of friction is transformed into heating and absorbed by the chilled water totally. (4) Flow is constant; (5) Physical parameters of the water are constant.

As is shown in Fig. 2, here the elemental volume with the length dl is selected at random, with entrance temperature of the chilled water t °C, exit temperature (t + dt) °C, and the frictional resistance dp (dp = R_mdl) Pascal. Then the following energy balance may be made for the element length dl:

The energy difference of the income and left of the elemental volume during dτ=. The work done by the frictional resistance during dτ. So Eq. (2) is obtained:

$$\begin{cases} \pi r_1^2 \rho c v \mathrm{d}t = \pi r_1^2 v R_m \mathrm{d}l \\ t_{l=0} = t_0 \end{cases} \tag{2}$$

Here, r_1 stands for the inside radius of the pipe; υ is the velocity of the chilled water; ρ stands for the density of the water, ρ = 1000 kg/m^3; c is the specific heat of water, c = 4.186 kJ/(kg.°C); t stands for temperature, °C; R_m stands for specific frictional resistance, Pa/m; l stands for the length of the pipe, m.

Hence, according to Eq. (2), the temperature change rule of the chilled water (t) with the length of the pipe (l) is obtained, as is shown in Eq. (3):

$$t = \frac{R_m l}{\rho c} + t_0 \tag{3}$$

If the total length of the pipe is L, the temperature rising of friction (Δt_1) is obtained, as is shown in Eq. (4):

$$\Delta t_1 = t - t_0 = \frac{R_m L}{\rho c} \tag{4}$$

2.2. Temperature rising caused by heat transfer between pipe inside and outside

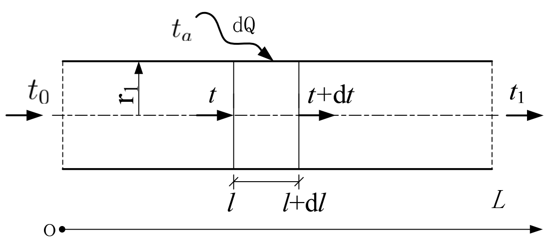


Fig. 3. Model for temperature rising of heat transfer

Because of the temperature difference between the pipe inside and outside, heat transfer or cooling loss through the pipe wall is inevitable. As is shown in Fig. 3, the basic assumptions of the model are: (1) The constant temperature of the tunnel air is t_a °C; (2) The influence of friction heat on the chilled water is neglected; (3) Flow is constant; (4) Both the physical parameters of the pipe and insulation materials are constant. So according to the energy conservation, we can gain: Internal energy difference of the income and left of the elemental volume during per unit of time = Heat gain through the pipe wall because of temperature difference inside and outside during per unit of time. Eq. (5) can be obtained.

$$\begin{cases} \pi r_1^2 \rho c v \mathrm{d}t = 2k\pi r_3 (t_a - t)\mathrm{d}l \\ t_{l=0} = t_0 \end{cases} \tag{5}$$

Here, t_a is the temperature of the tunnel air; t_0 stands for the entrance temperature of the chilled water; k is the heat transfer coefficient of the pipe, and the pipeline structure is shown in Fig. 4. The heat transfer coefficient k is given as follows Eq. (6).

$$k = \frac{1}{\frac{r_3}{h_{in} r_1} + \frac{r_3}{\lambda_1} \ln \frac{r_2}{r_1} + \frac{r_3}{\lambda_2} \ln \frac{r_3}{r_2} + \frac{1}{h_0}} \tag{6}$$

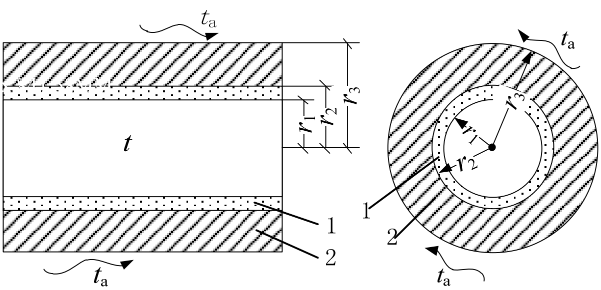


Fig. 4. Sketch illustrating for the structure of the pipe

Here, r_2 is outside radius of the pipe; r_3 stands for the outside radius of insulation material; λ_1 is the thermal conductivity of the pipeline; λ_2 is the thermal conductivity of the insulation material; h_{in} stands for the pipe inside convective heat transfer coefficient, and h_0 stands for the pipe outside convective heat transfer coefficient.

The temperature (t) changing rule with the transport distance (l) of the chilled water is given from Eq. (5)

$$t = t_a - (t_a - t_0) \cdot exp\,(-\frac{2kr_3 l}{r_1^2 \rho c v}\,) \tag{7}$$

If the length of the pipe is L, the temperature rising between the start and the terminal of the chilled water for heat transfer is as follows,

$$\begin{aligned} \Delta t_2 &= t - t_0 \\ &= t_a - (t_a - t_0) \cdot exp(-\frac{2kr_3 L}{r_1^2 \rho c v}) - t_0 \\ &= (t_a - t_0)\,[1 - exp(-\frac{2kr_3 L}{r_1^2 \rho c v})] \end{aligned} \tag{8}$$

2.3. Temperature rising under the respective effect of heat transfer and frictional heat (First Method)

As is stated above, if the temperature rising of the chilled water is regarded as the sum of frictional temperature rising and heat transfer temperature rising respectively, from Eq. (4) and (8), the terminal temperature rising of the chilled water ($\Delta t'$) is:

$$\begin{aligned}\Delta t' &= \Delta t_1 + \Delta t_2 \\ &= \frac{R_m L}{\rho c} + (t_a - t_0)\left[1 - exp\left(-\frac{2kr_3 L}{r_1^2 \rho c v}\right)\right]\end{aligned} \tag{9}$$

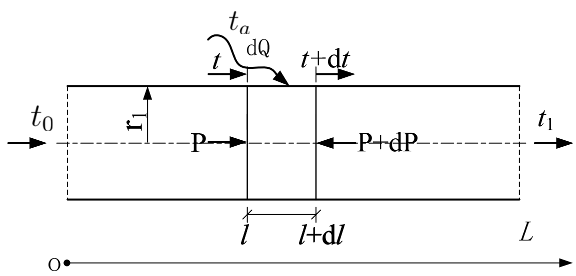


Fig. 5. Model of temperature rising under synergistic effect of heat transfer and friction

3. Temperature rising under the synergistic effect of heat transfer and friction (Second Method)

Strictly speaking, the temperature rising of the chilled water is caused by the synergistic action of the inside fraction heat and heat transfer between the inside and outside of the pipeline, but it is not the sum of temperature rising caused by separate effect. The basic assumptions of the model are: (1) Both the heat transfer and frictional heat are all absorbed by the chilled water totally; (2) The constant temperature of the tunnel air is t_a °C; (3) Flow is constant; (4) The physical parameters of the chilled water and insulation material are constant. As is shown in Fig. 5, the following energy balance may be made: The chilled water internal energy difference of the income and left of the elemental volume during per unit of time = The synergistic heat gain caused by internal flow friction and heat transfer between the inside and outside of the pipe during per unit of time. Eq. (10) is obtained.

$$\begin{cases} \pi r_1^2 \rho c v \mathrm{d}t = \pi r_1^2 v R_m \mathrm{d}l + 2k\pi r_3 (t_a - t)\mathrm{d}l \\ t_{l=0} = t_0 \end{cases} \tag{10}$$

Based on Eq. (10), the function of the chilled water temperature (t) with the length of the pipe (l) can be expressed as follows:

$$t = t_a - \left(t_a - t_0 + \frac{R_m r_1^2 v}{2kr_3}\right)\exp\left(-\frac{2kr_3 l}{r_1^2 v \rho c}\right) + \frac{R_m r_1^2 v}{2kr_3} \tag{11}$$

If the length of the pipe is L, the synergistic temperature rising (Δt) of the frictional heat and heat transfer for temperature difference can be expressed as Eq. (12):

$$\Delta t = t - t_0 = \left(t_a - t_0 + \frac{R_m r_1^2 v}{2kr_3}\right)\left[1 - \exp\left(-\frac{2kr_3 L}{r_1^2 v \rho c}\right)\right] \tag{12}$$

As is stated above, the temperature rising of the chilled water is caused by the synergistic effect of the friction and temperature difference. If the temperature rising of the synergistic effect of the friction and temperature difference is Δt, and the sum of temperature rising caused by friction and heat transfer respectively is $\Delta t'$, how much is the difference between Δt and $\Delta t'$? And can use $\Delta t'$ substitute Δt? The two methods and their difference according to the practical pipeline of the mine will be analyzed in the following.

4. Theoretical comparative analysis of the temperature rising of the chilled water

The parameters of the pipelines used in the mine are usually $De273 \times 7$ mm, $De325 \times 8$ mm, and $De377 \times 10$ mm (Yang Deyuan & Yang Tianhong, 2009). As a demo, only two type of pipe $De273 \times 7$ mm and $De377 \times 10$ mm are used in this paper and their lengths are both 15 km. Because the pipelines are seldom cleaned, the absolute roughness (K) is supposed to be 5×10^{-4} m. Assume the entrance temperature of the chilled water (t_0) is 3°C, kinematic viscosity (v) is 1.548×10^{-6} m^2/s, the air temperature of the roadway (t_a) is 28°C; the insulation material of the pipe is polyurethane and taking the high humidity level of the tunnel air into account, the thermal conductivity (λ) is assumed 0.05 W/(m·°C) and the thickness is 50 mm. From Eq. (6), the heat transfer coefficients of the two type pipe are 0.84 W/(m^2·°C) and 0.88W/(m^2·°C) respectively. Both Robert Petitjean (1994) and Lu Yaoqing (2008) suggested that the specific frictional head loss would better be limited within 100 Pa/m to 300 Pa/m, and the maximum should not be over 400 Pa/m. If the specific frictional head loss is limited to 100 Pa/m~400 Pa/m, the velocity range of the two types of pipes are (1.5~3.0) m/s and (1.8~3.6) m/s respectively (Henze & Floss, 2011; Petitjean, 1994).

Assuming $m = (2kr_3L)/(r_1^2 v\rho c)$, according to the above parameters, the value range of m for the above two types of pipes is (0.022~0.045) and (0.013~0.025) respectively. So $m \ll 1$ for the ordinary type of pipelines. And assuming $f(L) = \exp(-m) = \exp\left(-(2kr_3L)/(r_1^2 v\rho c)\right)$, then we can express $f(L)$ by the first and the second order of Maclaurin-series expansion (Prosperetti, 2011), and they can be expressed as expressions (13) and (14).

$$f(L) = exp(-\tfrac{2kr_3L}{r_1^2 v\rho c}) = 1 - \tfrac{2kr_3}{r_1^2 v\rho c}L + 0(L) \tag{13}$$

$$f(L) = exp(-\tfrac{2kr_3L}{r_1^2 v\rho c}) = 1 - \tfrac{2kr_3}{r_1^2 v\rho c}L + \tfrac{2k^2r_3^2}{r_1^4 v^2\rho^2 c^2}L^2 + 0(L^2) \tag{14}$$

In order to simplify the calculation, with the above formulae (13) and (14) substituted in Eq. (8) and (9), they can be transformed as follows,

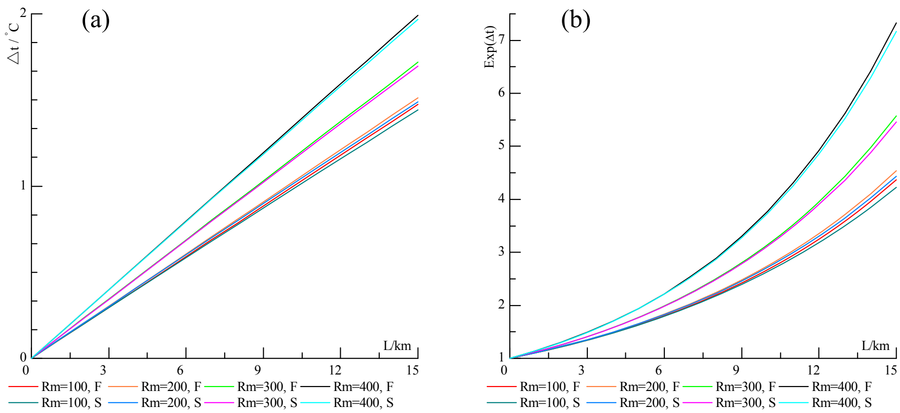
$$\Delta t_2 = \frac{2(t_a - t_0)r_3 kL}{r_1^2 \rho c v} \tag{15}$$

$$\Delta t' = \frac{R_m L}{\rho c} + \frac{2kr_3L(t_a - t_0)}{r_1^2 \rho c v} \tag{16}$$

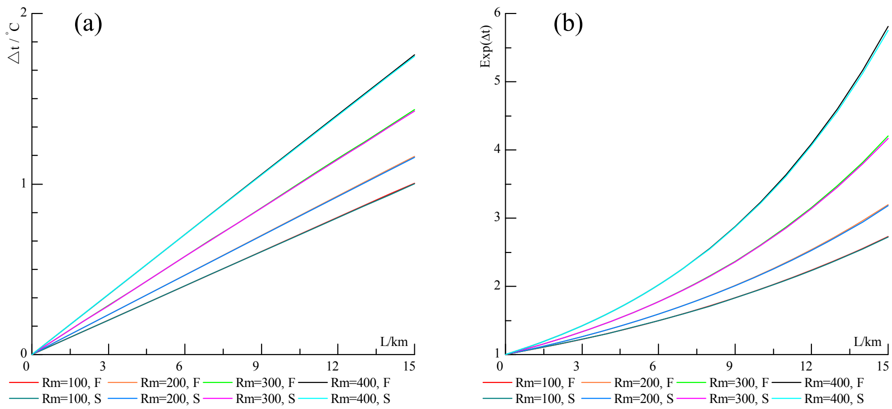
According fomulae (16) (First method) and (12) (Second method), we can gain the temperature increasing along the length of the pipeline. Fig. 6 and Fig. 7, are the temperature increasing along the length of pipeline $De273 \times 7$ and $De377 \times 10$. Because the two curves are too close and in order to distinguish easily, so the ordinate is the exponential functions with natural constant in Fig. 6(b) and Fig. 7(b). From the figures we can see the temperature increasing lines of the two model are very close and nearly coincidence.

The formula (16) is a simplified equation by the first method and the formula (12) is deduced by strict logic mathematical reasoning by the second mehod; in fact, $\Delta t'$ is not equal to Δt and their difference is shown as follows:

$$\begin{aligned}\Delta t'' &= \Delta t' - \Delta t \\ &= \frac{R_m L}{\rho c} + \frac{2kr_3 L(t_a - t_0)}{r_1^2 \rho c v} - \left(t_a - t_0 + \frac{R_m r_1^2 v}{2kr_3}\right)\left[1 - \exp\left(-\frac{2kr_3 L}{r_1^2 v \rho c}\right)\right]\end{aligned} \tag{17}$$



(F — First method, S — Second method)
Fig. 6. Temperature increasing along the length of the pipeline ($De273 \times 7$)



(F — First method, S — Second method)
Fig. 7. Temperature increasing along the length of the pipeline ($De377 \times 10$)

With Eq. (13) substituted in Eq. (17) $\Delta t'' \approx 0$, that is $\Delta t \approx \Delta t'$, so Δt can be replaced by $\Delta t'$. In order to study the error between Δt and $\Delta t'$, substitute Eq. (14) in Eq. (17), and then the error is shown as follows:

$$\Delta t" = \frac{2(t_a - t_0)k^2 r_3^2 L^2}{r_1^4 v^2 \rho^2 c^2} + \frac{kR_m r_3}{r_1^2 \rho^2 c^2 v} L^2 \tag{18}$$

Substitute the above value of heat transfer coefficient (k), the inner and outer radius (r_1 and r_3), pipe length (L = 15 km) etc. in (18), and then we can gain the errors of temperature rising (Δt) under different specific frictional resistance between 100~400 Pa/m, which are shown in Table 1.

TABLE 1

Errors of the temperature rising of the two methods

$\Delta t''$ [°C]	$De273 \times 7$ [mm]	$De377 \times 10$ [mm]
R_m = 100 Pa/m	0.032	0.013
R_m = 200 Pa/m	0.024	0.011
R_m = 300 Pa/m	0.023	0.011
R_m = 400 Pa/m	0.022	0.012

From Table 1 it is shown that the maximum error is not over 0.04°C by substituting Δt with $\Delta t'$ even for a long pipe (length 15 km), which is absolutely in the allowed range of engineering, so we can substitute $\Delta t'$ for Δt, and that is to say, the temperature rising along the pipe is the sum of temperature rising of frictional heat and heat transfer for temperature difference respectively. And Δt can be shown as follows:

$$\Delta t = \Delta t_1 + \Delta t_2 \tag{19}$$

So the proportion of Δt_1 and Δt_2 to the total temperature rising (Δt)can be expressed as follows:

$$m_1 = \frac{\Delta t_1}{\Delta t} \times 100\% \tag{20}$$

$$m_2 = \frac{\Delta t_2}{\Delta t} \times 100\% = (1 - \frac{\Delta t_1}{\Delta t}) \times 100\% \tag{21}$$

Under the former stated conditions, for the two types of pipes mentioned above, from Eq. (4), (15), (19)~(21), the values of Δt_1 and Δt_2 are listed in Table 2.

From Table 2 we can see that for the same pipe, both Δt_1 and m_1 increase with the increasing of specific frictional resistance, while the changes of Δt_2 and Δm_2 decrease because with specific frictional resistance increasing, frictional resistance becomes bigger and the frictional temperature rises naturally, and with the flow increasing the heat transfer through the pipe wall is almost constant, so Δt_2 decreases, and the total temperature increases because Δt_1 is higher than Δt_2. In addition, we can see from the Table 2, under the good insulation, even if the specific frictional resistance is limited between 100 Pa/m and 400 Pa/m, the frictional temperature rising proportion is over 24%, even as higher as 81%, and the proportion of friction temperature rising increases significantly with the decreasing of pipe radius. Therefore, we conclude that the friction

TABLE 2

Temperature rising and ratio for fraction and heat transfer

De **[mm]**	R_m **[Pa/m]**	Δt_1 **[°C]**	Δt_2 **[°C]**	Δt **[°C]**	m_1 **[%]**	m_2 **[%]**
$De273 \times 7$	$R_m = 100$	0.36	1.12	1.48	24.3	75.7
	$R_m = 200$	0.72	0.80	1.52	47.4	52.6
	$R_m = 300$	1.08	0.67	1.75	61.7	38.3
	$R_m = 400$	1.43	0.56	1.99	71.9	28.1
$De377 \times 10$	$R_m = 100$	0.36	0.66	1.02	35.3	64.7
	$R_m = 200$	0.72	0.47	1.19	60.5	39.5
	$R_m = 300$	1.08	0.38	1.46	74.0	26.0
	$R_m = 400$	1.43	0.33	1.76	81.3	18.8

temperature rising must not be ignored and it may be the major reason for the temperature rising, especially for the high specific frictional resistance (or high velocity) with small pipe radius.

5. Conclusions

Fraction heat along the piping network rises the temperature and produces a negative effect for chilled water transport. The fractional heat should be fully considered for long distance transport of chilled water in coal mine.

(1) The temperature rising of the chilled water is caused by the synergic effects of friction heat and heat transfer for temperature difference between the inside and outside of the pipe. The actual temperature rising of the chilled water can be considered the sum of heat transfer and friction heat respectively, and the calculation progress can be simplified.

(2) For the two types of pipelines, whose length is 15 km and whose radii are $De273 \times 7$ mm and $De377 \times 10$ mm, the calculation results show that the error is controlled within 0.04°C, which is completely within the permitted error range of the engineering and facilities.

(3) If the specific frictional head loss of the pipelines is controlled between 100 Pa/m and 400 Pa/m and the insulation is good, the temperature rising because of the friction heat is about 24%~81% of the total temperature rising of the chilled water. With the decrease of the pipe radius and increase of the specific frictional resistance (or the velocity increasing of the chilled water), the temperature rising of friction heat becomes more evident. It can be concluded that the influence of the frictional resistance on the total temperature rising should not be ignored in the transportation of the chilled water, but it should be controlled in a reasonable range to avoid a higher temperature rising for fraction.

Acknowledgements

The Authors would like to acknowledge the fund provided by the National Natural Science Foundation of China (No. 51774197), a Project of Shandong Province Higher Educational Science and Technology Program (No. J15LH03), Scientific Research Foundation of Shandong University of Science and Technology for Recruited Talents (No. 2015RCJJ059), and project supported by Shandong Provincial Natural Science Foundation(No. ZR2019MEE115).

References

Bar N., Bandyopadhyay T.K., Biswas M.N., Das S.K., 2010. *Prediction of pressure drop using artificial neural network for non-Newtonian liquid flow through piping components*. Journal of Petroleum Science and Engineering **71**, 187-194.

Chu Z., Ji J., Zhang X. et al., 2016. *Development of ZL400 mine cooling unit using semi-hermetic screw compressor and its application on local air conditioning in underground long-wall face*. Archives of Mining Sciences **61**, 949-966.

Chan A.L.S., Hanby V.I., Chow T.T., 2007. *Optimization of distribution piping network in district cooling system using genetic algorithm with local search*. Energy Conversion and Management **48**, 2622-2629.

Petitjean R., 1994. *Total Hydronic Balancing: A handbook for design and troubleshooting of hydronic HVAC systems*. Sweden: Tour & Andersson AB in Ljung, Sweden.

Chandel S., Misal R.D., Beka Y.G., 2012. *Convective Heat Transfer through Thick-Walled Pipe*. Procedia Engineering **38**, 405-411.

Dalla Rosa A., Li H., Svendsen S., 2011. *Method for optimal design of pipes for low-energy district heating, with focus on heat losses*. Energy **36**, 2407-2418.

Henze G.P., Floss A.G., 2011. *Evaluation of temperature degradation in hydraulic flow networks*. Energy and Buildings **43**, 1820-1828.

Jie P., Tian Z., Yuan S., Zhu N., 2012. *Modeling the dynamic characteristics of a district heating network*. Energy **39**, 126-134.

Li H., Svendsen S., 2012. *Energy and exergy analysis of low temperature district heating network*. Energy 45, 237-246.

Lin F., Yi J., Weixing Y., Xuzhong Q., 2001. *Influence of supply and return water temperatures on the energy consumption of a district cooling system*. Applied Thermal Engineering, 511-521.

Lu Yaoqing, 2008. *Design manual for heating and air conditioning (2nd). China* Architecture & Building Press, Beijing China, 678-682.

Petitjean R., 1994. *Total Hydronic Balancing: A handbook for design and troubleshooting of hydronic HVAC system*. Tour & Andersson AB in Ljung, Sweden, 28-30.

Pirouti M., Bagdanavicius A., Ekanayake J., Wu J., Jenkins N., 2013. *Energy consumption and economic analyses of a district heating network*. Energy **57**, 149-159.

Qi Yudong, 2010. *Research of Energy Efficiency Test And Diagnosis For Thermal Disaster Mine With Ice Cooling*. Master Dissertation of Shandong University of Science and Technology, China, 3-4.

Wojciechowski J., 2013. *Application of the GMC-1000 and GMC-2000 mine cooling units for central air-conditioning in underground mines*. Arch. Min. Sci. **58**, 1, 199-216.

Prosperetti A., 2011. *Advanced Mathematics for Applications*. UK: Cambridge University Press.

Szlązak N., Obracaj D., Swolkień J. et al., 2016. *Controlling the distribution of cold water in air cooling systems of underground mines*. Archives of Mining Sciences **61**, 793-807

Rawlins C.A., 2007. *Mine cooling and insulation of chilled water transport pipes*. Journal of the South African Institute of Mining and Metallurgy **107**, 681-688.

Rishel J., Durkin T., Kincaid B., 2006. *HVAC Pump Handbook*. McGraw-Hill, 58-62.

Cygankiewicz J.J., Knechtel J., 2014. *The effect of temperature of rocks on microclimatic conditions in long gate roads and galleries in coal mines*. Archives of Mining Sciences **59**, 189-216.

Söderman J., 2007. *Optimisation of structure and operation of district cooling networks in urban regions*. Applied Thermal Engineering 27, 2665-2676.

Tol H.İ., Svendsen S., 2012. *Improving the dimensioning of piping networks and network layouts in low-energy district heating systems connected to low-energy buildings: A case study in Roskilde, Denmark*. Energy **38**, 276-290.

Yang Deyuan, Yang Tianhong, 2009. *Thermal Environment in Mine and its Control*. Metallurgical Industry Press. Beijing China, 8-10.

Zhu Yingxin, Wang Gang, Jiang Yi., 2008. *Energy consumption analysis of district cooling systems*. Heating Ventilation & Air Conditioning **38** (1), 36-40 (China).