On the calculation of air flow rates to ventilate closed-type stations in subway with the double-track tunnel

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Abstract. Metro is not only the most promising kind of public transport but also an important part of infrastructure in a modern city. As a place where large groups of people gather, subway is to ensure the required air exchange at the passenger platforms of the stations. The air flow rate for airing the stations is also determined based on the required temperature, humidity and MAC of gases. The present study estimates the required air flow rate at the passenger platform of the closed-type subway station with the double-track tunnel given the standard air temperature, humidity and gas concentration, as well as based on the condition of the specified air flow feed and air changes per hour. The article proposes the scheme of air recirculation from the double-track tunnel to the station.

1. Introduction
Microclimate in subway underground structures is provided by tunnel and station ventilation systems. Choice of equipment and operation mode parameters depends on air flow rates calculated in terms of heat-humidity balance in main line tunnels and at Metro stations. Modern subway construction trends imply erection of closed-type stations and double-track tunnels. Case studies are construction of double-track tunnel in Frunze radius in Saint Petersburg and Kozhukhov line in Moscow. Of actual significance is the development of scientific-methodological fundamentals of design of closed-type subway with double-track tunnels under severe continental climate conditions, as the experience in design of similar subways is not available so far.

The basic specific feature of closed-type stations alternatively to other types is aerodynamic isolation of passenger platforms from track sections and line tunnels with no respective heat and humidity exchange between these subway sections. This explains existence of separate aeration modes for closed-type stations and double-track tunnels [1].

The airflow performance of a ventilation system is selected based on heat and humidity balance in passenger platforms and track sections. Microclimate parameters at stations and tunnels are normalized by the Russian Federation acting acts [2, 3].

To calculate a ventilation air flow rate at passenger platforms it is necessary to calculate amount of released heat, and humidity, and repugnant substances and to derive an equation of heat, humidity balance and hazardous substance balance in order to determine a required assimilating capacity of the fresh air flow.

2. Calculation methods
The principle components of the station heat balance are passengers and personnel, escalators, heat exchange between a station and a track tunnel, light systems, heat exchange between peripheral soil...
mass and a station, subway stationary equipment. In the case of the lack of a permanent water flow the main sources of moisture and CO\textsubscript{2} at a station are passengers and personnel. The basic components of heat balance in a tunnel and a track section are trains, passengers, lighting systems, cable-channels, heat exchange between a station and a track section, stationary equipment, combines trailer-secondary substations (if waste air is removed from a station), heat exchange between subway infrastructure and a peripheral soil mass. The principal source of moisture and CO\textsubscript{2} in a tunnel is passengers.

Heat generation (average per an hour) \( q_p, \text{kW} \), for trains is determined from formula [4]:

\[
q_p = \frac{2 \cdot (T_1 - T_2 \pm \sum A_G) \cdot N_{td}}{24 \cdot 3600 \cdot 1000},
\]

where \( T_1 \) is kinetic energy of a train at the initial slow-downing stage, J; \( T_2 \) is kinetic energy at the final braking stage, \( T_2 = 0 \text{ J} \); \( \Sigma A_G \) is job of gravitation forces in the given section, it is assumed that \( \Sigma A_G = 0 \); 2 is a magnitude of twoness of traffic; \( N_{td} \) is a number of train couples per a day.

Heat flow \( q_{pc}, \text{kW} \), from a track section to passenger platforms by heat transmission through a partition can be calculated from formula [5]:

\[
q_{pc} = \frac{F}{1000} \cdot \frac{(t_u - t_a)}{\alpha_{t-p} + \sum \delta_m \cdot \lambda_n + \alpha_{s-p}},
\]

where \( F \) is area of a partition wall, m\textsuperscript{2}; \( t_u \) is temperature in a track section, °C; \( t_a \) is air temperature at a station, °C; \( \alpha_{t-p} \) is air heat transfer factor in a track section to a partition wall, W/(m\textsuperscript{2}·°C); \( \alpha_{s-p} \) – is air heat transfer factor in a station to a partition wall, W/(m\textsuperscript{2}·°C); \( \delta_m \) is thickness of the \( i \)-th wall layer, m; \( \lambda_n \) is heat conductivity factor of the \( i \)-th the layer of the wall, W/m·°C; \( n \) is number of wall layers, u.

The numerical modeling of heat transition between a track section and a station through open gateway of a track section enabled to establish that warm air mainly passed trough an upper one third of gateway opening. A warm air flow through open gateways of a track section \( q_{og}, \text{kW} \) was calculated from formula [5]:

\[
q_{og} = \vartheta \cdot \frac{h \cdot b \cdot \rho_a \cdot c_a \cdot (t_u - t_a) \cdot n_c \cdot n_d \cdot N_{td} \cdot 2 \cdot \tau_s}{24 \cdot 1000 \cdot 3600},
\]

where \( \vartheta \) is air velocity in the upper one third of gate opening, m/s, determined experimentally; \( h \) and \( b \) are dimensions of gate opening, m; \( \rho_a \) is air density, kg/m\textsuperscript{3}; \( c_a \) is isobar heat capacity of air, J/(kg·°C); \( t_u \) is temperature in a track section, °C; \( t_a \) is temperature at a station, °C; \( n_c \) is number of cars in a train, u.; \( n_d \) number of door openings per a car, u.; \( \tau_s \) is stopping time at a station, J is assumed 30 s; 2 is a magnitude of twoness of train traffic.

Heat generation of escalators, light systems and power substations are calculated under procedures, described in [6]. Heat generation of stationary equipment \( q_{eq}, \text{kW} \), is determined from formula [7, 8]:

\[
q_{eq} = \frac{P_m \cdot (1 - \eta) \cdot K}{\eta},
\]

where \( P_m \) is rated motor capacity of a facility, kW; \( \eta \) is engine efficiency; \( K \) is activity ratio.

Heat flow from station platform and rooms to a soil mass per a year is of reverse character. Herewith heat loss of a station in a soil mass in the first year of exploitation differs from heat loss at a routine operation mode. Heat loss of passenger platform and a track section of a station in a soil mass in the initial operation period \( q_{pp,ini}^{\text{pp}}, \text{W} \) is calculated under procedure described in [6].

Specific heat loss at a routine operation mode in passenger platform and a track section \( q_{pp,ro}^{\text{pp}}, \text{W/m}^2 \) are calculated from:

\[
q_{pp,ro}^{\text{pp}}(\tau) = B(h) + A(h) \cdot \cos(0.0172 \cdot \tau + \omega(h)),
\]
where $A(h)$ amplitude of specific heat flow fluctuations, W/m$^2$; $B(h)$ is average annual specific heat flow, W/m$^2$; $\omega(h)$ is initial phase of fluctuation of specific heat flow, rad; 0.0172 is angular factor, 1/day. Coefficients $A(h)$, $B(h)$, $\omega(h)$ are determined in terms of station depth $h$, m.

Heat loss in a peripheral soil mass of a running meter of double-track tunnel is calculated under the procedure [9].

Heat generation $q_{pas}$, W, moisture production $W_{pas}$, kg/h, and CO$_2$ emission $B_{pas}$, kg/h, on the part of passengers and personnel are calculated by formulas:

- passenger platform of the station:
  
  $$ q_{pas}^s = q_{pas} \cdot n_{pas} ; \quad W_{pas}^s = w_{pas} \cdot n_{pas} ; \quad B_{pas}^s = b_{pas} \cdot n_{pas}, $$

  (6)

where $q_{pas}$, $w_{pas}$, $b_{pas}$ are total heat generation by a passenger calculated by [6] and $n_{pas}$ – number of passengers at a platform;

- two-track tunnel and a track section:
  
  $$ q_{pas}^n = q_{pas} \cdot n_{pas} \cdot n_{t} ; \quad W_{pas}^n = w_{pas} \cdot n_{pas} \cdot n_{t} ; \quad B_{pas}^n = b_{pas} \cdot n_{pas} \cdot n_{t}, $$

  (7)

where $q_{pas}$, $w_{pas}$, $b_{pas}$ are total heat generation by a passenger calculated by [6] and $n_{pas}$ is number of passengers in a train; $n_{t}$ is number of trains per a track section, calculated by formula:

  $$ n_{t} = \frac{2 \cdot N_{t/d}}{24 \cdot 3600} \left( \frac{l - l_{sd} - l_{acc}}{\vartheta} + \tau_{s} + \tau_{sd} + \tau_{acc} \right) \leq 2, $$

  (8)

where $N_{t/d}$ is number of train pairs per a day; $l$ is track span length, m; $l_{sd}$, $l_{acc}$ are length of slowdown and acceleration sections, respectively, m; $\vartheta$ is operating speed of train motion, m/s; $\tau_{s}$, $\tau_{sd}$, $\tau_{acc}$ rated time of stopping, slowdown, and acceleration of a train, s.

Rated heat amount $q_{vent}$, kW, humidity $W_{vent}$, kg/h, and CO$_2$ emission $B_{vent}$, kg/h, intended for removal by station ventilation systems and calculated by formulas:

  $$ \sum_{i=1}^{k} q_{i} = q_{vent} ; \quad \sum_{i=1}^{k} W_{i} = W_{vent} ; \quad \sum_{i=1}^{k} B_{i} = B_{vent}, $$

(9)

where $i$ is number of a toxic source; $k$ is quantity of toxic sources, $q_{i}$, $W_{i}$, $B_{i}$ amount of toxic emission of the $i$-th source;

The air flow rate of overall-exchange ventilation and air conditioning systems is computed under the procedure described in [10] to provide sanitary-hygienic conditions, fire-and-explosion safety and to eliminate condensate generation. The air flow rate should be evaluated separately for warm, cold, and transitional periods of the year in terms of assimilation of heat and moisture generation and mass of harmful substance emission, approving the maximum values of the resultant air flow rate data.

2. Results

Cumulative average hourly heat emission/loss $q$, kW, and production of moisture $W$ and CO$_2$ emission $B$, kg/h for passenger platforms of the closed-type station (Gusinobrodskaya station, Novosibirsk Metro), track section and double-track tunnel ($l = 1990$ m) (Table 1).

The design air flow rate was calculated for: (a) track section and main track line tunnel, (b) passenger platform and rooms of the station. It is found that in public places of the closed-type station the maximum air flow rate is evaluated based on the rated air exchange condition, in a track section and main track line tunnel the maximum air flow rate is calculated considering the conditions of assimilation of heat production. For a double-track tunnel and a track section the calculated ventilation air flow rate amounted to (a) 78.93 m$^3$/s in warm season; (b) 47.24 m$^3$/s in transitional season, and (c) 20.22 m$^3$/s in cold season. The minimum fresh air flow rate for a double-track tunnel is 11.33 m$^3$/s. Calculated ventilation air flow rates for passenger platform of the closed-type station and recirculating air flow from a track section to public rooms are cited in Table 2.
Table 1. Cumulative average hourly heat emission, production of moisture and CO₂

| Period of a year | Passenger platform | Tunnel and track tunnel sections |
|------------------|--------------------|---------------------------------|
|                  | Warm period | Transitional period | Cold period | Warm period | Transitional period | Cold period |
| The first operation year | \( q, \text{ kW} \) | –396.49 | –64.49 | –59.24 | 299.06 | 963.25 | 961.94 |
|                    | \( W, \text{ kg/h} \) | 42.81 | 25.56 | 18.72 | 240.3 | 134.84 | 53.4 |
|                    | \( B, \text{ kg/h} \) | 13.86 | 13.86 | 13.86 | 60.07 | 60.07 | 60.07 |
| Routine operation year | \( q, \text{ kW} \) | –45.47 | –49.75 | –95.43 | 947.7 | 963.64 | 1091.64 |
|                    | \( W, \text{ kg/h} \) | 42.81 | 25.56 | 18.72 | 240.3 | 134.84 | 53.4 |
|                    | \( B, \text{ kg/h} \) | 13.86 | 13.86 | 13.86 | 60.07 | 60.07 | 60.07 |

Table 2. Calculated ventilation air flow rate for public rooms of the closed-type station and a share of recirculating tunnel air

| Period of a year | The first year of operation | Routine operation mode |
|------------------|----------------------------|------------------------|
|                  | Warm³ | Transitional | Cold | Warm | Transitional | Cold |
| Calculated ventilation air flow rate \( L, \text{ m}^3/\text{s} \) | 9.94 | 9.94 | 9.94 | 9.94 | 9.94 | 9.94 |
| The minimum fresh air flow rate \( L_{fa}^{\text{min}}, \text{ m}^3/\text{s} \) | 2.57 | 2.57 | 2.57 | 2.57 | 2.57 | 2.57 |
| Fresh air flow rate \( L_{fa}, \text{ m}^3/\text{s} \) | 6.17 | 2.60 | 3.98 | 6.71 | 2.57 | 4.29 |
| Share of recirculating tunnel air \( r, \% \) | 37.9 | 73.8 | 60.0 | 32.5 | 74.1 | 56.8 |

¹ – additional heat supply is required.

Recirculation share \( r, \% \), is determined from:

\[
r = \frac{L - L_{fa}^{\text{min}}}{L},
\]

where \( L \) is calculated air flow rate, \( \text{m}^3/\text{s} \); \( L_{fa}^{\text{min}} \) is the minimum required fresh air flow rate, \( \text{m}^3/\text{s} \).

To compensate heat deficit in passenger rooms of the closed-type stations (Table 1) the design of separate ventilation systems is proposed for the closed-type stations and a two-track tunnel with recirculation of tunnel air in passenger rooms of the station. This design enables to lower load on air-heating ventilation systems in winter and transitional periods (Figure 1) and to save heat energy up to 43% in the cold period of the year.
3. Conclusions
1. Average hourly heat generation, amount of moisture and CO₂ emissions designed to be assimilated by ventilation systems in passenger rooms of the closed-type station and two-track tunnel are evaluated;
2. The design air flow rate for ventilation of two-track tunnel and fresh air flow rate plus recirculating tunnel air for station ventilation are computed;
3. The new-proposed design of separate ventilation systems for the closed-type station and two-track tunnel with recirculation of tunnel air to passenger rooms of the station makes it possible to lower heat load on the ventilation air-heating systems and to save up to 43% of heat energy in the cold period of the year.

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