Improving the pneumatic actuator of the locomotive sand feeding system by increasing the outlet flow velocity

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Abstract During the study, the main characteristics and parameters of the gas jet injector of the pneumatic actuator for the locomotive sand feeding system were calculated based on the specified input and output values of gas flows. The process of ejecting an additional volume of ambient air in order to increase the outlet flow velocity of the air-sand mixture from the outlet pipe of the locomotive sand feeding system was simulated by means of the FlowSimulation package tools of the SolidWorks CFD software for solid modeling. The reduction in the compressed air consumption relative to the standard locomotive sand feeding systems has been achieved.

1. Introduction
To improve adhesive properties of the traction rolling stock in the traction mode and reduce stopping distance during braking in adverse operating conditions such as presence of moisture or oil contamination on the rail surface, and also when the locomotive is moving on an incline or on an scent, delivering quartz sand particles in the compressed air flow to the wheel-rail contact area by means of pneumatic sand feeding systems for locomotives is applied.

During operation of the existing sand feeding systems on mainline electric locomotives produced in series with a maximum sand discharge of 1500 g/min, the velocity of the air-sand mixture flow is slightly more than 10 m/s [1], and with the known methods of supplying additional air to the pipeline, the velocity does not exceed 20 m/s with the doubled consumption of the compressed air amount.

From the technical and scientific literature [2, 3], as well as the authors' own research, it was established [4] that in the absence of a cross wind, the delivery of sand to wheel-rail contact area can be carried out at fairly low speeds of the air-sand mixture up to 10 m/s.

It is also established that with a cross wind of 20 m/s, in order to deliver more than 90% of the sand volume discharged from the outlet pipe, it is necessary to ensure the air-sand mixture flow velocity of at least 40 m/s [4]. In this regime, the sand particles are almost not blown away during the delivery process from the outlet pipe to the rail surface, this fact allows us to state that the air-sand mixture becomes resistant to the cross wind equal to 8 points on the Beaufort scale.

In order to improve the reliability of the locomotive sand feeding system, various technical solutions are used, such as: creating various designs of sand distributors, supplying an additional volume of compressed air to the sand distributor body as an attempt to increase the air-sand mixture flow velocity, however, this leads to excessive consumption of air and sand reserves of the locomotive, but a significant increase in the velocity of sand particle movement is not achieved [5].
Based on the above, to improve the operational reliability of the pneumatic actuator of the locomotive sand feeding system, the upgrade is proposed based on supplying an additional volume of air using a gas jet unit – an injector. As indicated above, it is recommended to achieve the air-sand mixture flow velocity of above 40 m/s from the outlet pipe.

2. Methods
To solve the given problem of increasing the outlet flow velocity of the air-sand mixture from the outlet pipe of the locomotive sand feeding system up to 40 m/s and more, it is proposed to apply the ejection effect that is supplying the additional volume of ambient air by means of underpressure generated by the high-pressure flow in the gas jet unit body.

We will define the parameters of an equal-phase (air-to-air) gas jet injector without a diffuser with a conical nozzle and determine its main geometric parameters according to the diagram shown in figure 1.

The initial data for calculating the injector are taken from the main problem of increasing the outlet flow velocity of the air-sand mixture up to minimum 45 m/s while reducing the compressed air consumption relative to the standard sand feeding system, with a maximum discharge of quartz sand being 1500 g/min, that corresponds to the mass flow rate \( G_{m} = 0.01 \text{ kg/s} \).

In the sand feeding systems of most locomotives of various types and purposes produced in series, the working pressure of compressed air is in the range from 0.75 to 0.9 MPa, thus the minimum value is assumed to be 0.5 MPa, as for the diesel locomotives of the series TGK-2 and TGK-2M, operating with the sand distributor of the P32-149sb type.

![Diagram](image)

**Figure 1.** Design scheme for modeling the air-sand mixture flow: 1 – working nozzle; 2 – receiving chamber; 3 – mixing chamber; 1-1 – outlet cross-section of the working nozzle; 2-2 – inlet cross-section of the mixing chamber; 3-3 – outlet cross-section of the mixing chamber; S-S – random cross-section of the mixing chamber.

We assume conditionally for the purpose of simplifying the conclusions that before entering the mixing chamber in the section between the plane 1-1 coinciding with the outlet section of the working nozzle and the inlet cross-section 2-2 of the cylindrical mixing chamber, the working and ejected flows do not mix (the initial section of the submerged ejection jet). And the pressure of the working flow \( p_{w} \) in the outlet cross-section of the nozzle is equal to the pressure of the medium the outlet flow goes into, in our case the ambient pressure. We also assume that the cross-section of the high-pressure injecting flow in the cross-section 2-2 is equal to the cross-section diameter of the working nozzle outlet. In the outlet section of the mixing chamber 3-3, the flow has a uniform velocity profile.

Since the interacting media are air and air, we will take the adiabatic index of diatomic gases (air) \( k = 1.4 \) and the specific gas constant of air \( R_{g} = 287 \text{ N-m/kg-K} \).

We will set the values of gas flows for further calculations of the main parameters of the injector and its modeling with the FlowSimulation package of the SolidWorks CFD software. The static pressure of the injecting gas in front of the nozzle (before entering the nozzle) is \( p_{0} = 405300 \text{ Pa} \), the density is \( \rho_{0} = 4.8 \text{ kg/m}^{3} \) respectively. The ejected flow pressure is assumed to be equal to the ambient...
pressure \( p_0 = 101325 \) Pa. The potential pressure at the outlet of the mixing chamber is taken with a small excess relative to the ambient pressure, where the air-sand mixture flows out, \( p_1 = 111457.5 \) Pa. Then the inequality \( p_s > 1 \) \( p_{man} \) is guaranteed to hold for the outlet flow at the outlet cross section of the pipe. When the total gas pressure \( p_s \) is in the range of \( 1.1 p_{man} < p_s < p_{man}/\Pi_r \), the gas flows out at subsonic speeds and the gas cannot be considered incompressible in calculations, although the pressure at the outlet cross-section is equal to the ambient pressure [6]. We assume a dimensionless empirical correction coefficient for the velocity in the outlet section of the mixing chamber to be equal \( \varphi_s = 0.8 \), taking into account various friction losses and uneven flow for injectors without diffusers with a small diameter nozzle [7]. For all flows, the temperature is \( r = t_0 = t_3 = 293.2 \) K.

The diameter of the outlet pipe of the standard sand feeding system for most locomotives produced in series is 32 mm, this value is the diameter \( d_3 \) of the outlet cross-section of the injector mixing chamber. Based on the above, we assume the outlet flow velocity of the air-sand flow from the outlet pipe to be equal to 45 m/s.

Using gas-dynamic functions, we obtain the parameter of the critical velocity for the specified parameters from the formula [8]

\[
a_{cr} = \frac{2 \cdot k}{k+1} \frac{p_i}{\rho_i}
\]

where \( p_i \) and \( \rho_i \) are the flow pressure and flow density, respectively.

The relative pressure for the critical flow of the diatomic gas is \( \Pi_r = 0.528 \) Pa. Since the relative pressure at the given expansion degree of the gas jet unit is \( p_i/p_r = \Pi_r = 0.25 < \Pi_r = 0.528 \) Pa, the operation of the gas jet unit is carried out with a supercritical degree of expansion and then the velocity of the ejecting flow in the cross-section 2-2 will be \( \omega_i > \omega_{cr} \), i.e. higher than the critical velocity due to the fact that only the part of the complete expansion work is converted into kinetic energy in the nozzle. The gas expands to the critical pressure \( p_{cr} \), and the critical velocity of the nozzle outlet flow \( \omega_{cr} \) is set. The minimum pressure at which the critical flow regime can occur is \( p_{r.min} = p_i/\Pi_r = 191801 \) Pa.

To determine whether there is an area in which the compressor operation is impossible under the condition \( q_3 > p_i/p_r \) \( q_{r,s} \), where \( q_{r,s} \) is the superficial mass velocity in any cross-section of the mixing chamber of the gas jet unit and \( q_3 \) is the superficial mass velocity in the outlet cross-section of the outlet pipe, which are determined by the relative pressures and the superficial velocities in the corresponding cross-sections according to gas-dynamic functional dependencies [7].

\[
\Pi_r = \Pi_r x \Pi_r cr = 0.132
\]

Hence \( q_{r,s} = 0.6 \) kg/s/m², and \( q_3 = 0.27 \) kg/s/m², then \( q_3 < p_i/p_r q_{r,s} \), thus, the area in which the injector operation is impossible is absent. Since in this regime the achievable ejection coefficient is \( u \leq u_{pr} \leq 0 \), the occurrence of the second limiting regime is physically impossible in the conditions under consideration due to \( u_{pr} < 0 \), and therefore it should not be taken into account in the calculation. Accordingly, achieving the ejection coefficient of the first limiting regime is also impossible, because the second limiting regime always occurs before the first one \( u_{pr} < u_{pr1} \). When the mixture flow velocity at the outlet of the mixing chamber is equal to the speed of sound, the implementation of the third limiting regime is impossible, because under the conditions of the problem to be solved it is necessary to achieve the air flow velocity of more than 45 m/s.

The achievable ejection coefficient is determined by the expression for the injector without a diffuser with a cylindrical mixing chamber [7]

\[
u = \frac{f_{3}}{f_{r,cr}} \frac{p_i}{p_r} w_3 - 1 = 8.5
\]

where \( f_3 \) is the area of the outlet cross-section of the outlet pipe, \( f_3 = 0.000804 \) m²; \( f_{r,cr} \) is the critical cross-section area of the working nozzle for the given mass flow rate \( G_{r,cr}=0.01 \) kg/s; \( p_r \) and \( t_r \) were given above; the diameter of the critical cross-section of the working nozzle is determined equal to \( d_{r,cr} = 2.9 \) mm; \( w_3 \) is the velocity coefficient for the outlet cross-section of the mixing chamber which
is taken for ease of calculating the injector without a diffusor [7] and equal to the ratio of the superficial mass rate \(q_s\) to a relative pressure \(\Pi_r\), \(w_j=0.285\) m/s.

For the isentropic flow, the cross-section area for the ejected flow in the cross-section 2-2 is equal \(f_{n,2}=f_3\cdot f_{cr,2}=0.000796\) m².

Another important parameter for the geometric dimensions of the gas jet injector is the length of the mixing chamber, which is determined by the ratio \(L_k=6\ldots10\cdot d_3\) [7] and ranges from \(L_{k,\text{min}}=192\) mm to \(L_{k,\text{max}}=320\) mm.

The dependence of the mass flow rate of the ejecting gas on the pressure in front of the working nozzle is determined as

\[
G_{m_r}(p_r) = \frac{f_r \cdot \omega_{cr} \cdot p_r}{R_{gaz} \cdot t_r} \quad \text{if} \quad \frac{p_n}{p_r} \leq \Pi_{cr} \\
G_{m_r}(p_r) = \frac{f_r \cdot \omega_{cr} \cdot p_r}{R_{gaz} \cdot t_r} \cdot \left[1 - \left(\frac{p_n}{p_r}\right)^{k-1}\right] \quad \text{if} \quad \frac{p_n}{p_r} > \Pi_{cr}
\]

(1)

(2)

The dependence of the velocity of the outlet flow from the critical cross-section of the working nozzle on the pressure in front of it is determined by the expressions

\[
\omega_r(p_r) = \omega_{cr} \quad \text{if} \quad \frac{p_n}{p_r} \leq \Pi_{cr} \\
\omega_r(p_r) = \omega_{cr} \cdot \left[1 - \left(\frac{p_n}{p_r}\right)^{k-1}\right] \quad \text{if} \quad \frac{p_n}{p_r} > \Pi_{cr}
\]

(3)

(4)

The dependence of the mass flow rate of the ejecting gas on the cross-section diameter of the working nozzle and the pressure in front of it in the critical flow regime is determined by the formula

\[
G_{i_k}(d_r) = \frac{\pi \cdot d_r^2}{4} \cdot \frac{\omega_r \cdot p_{i_r}}{R_{gaz} \cdot t_r}
\]

(5)

where \(i\) is the value of the compressed air pressure in front of the nozzle in the range from 0.2 MPa to 0.9 MPa, indicated by a number of integers 2...9, respectively.

The dependence of the critical cross-section diameter of the working nozzle on the pressure in front of the nozzle and the mass flow rate of the ejecting gas is shown in figure 5 and determined by the formula

\[
d(f_r) = \sqrt[4]{\frac{4 \cdot G_j \cdot R_{gaz} \cdot t_r \cdot \omega_r \cdot p_r}{\pi \cdot \Pi_{cr}}} \cdot 1000
\]

where \(j\) is the multiplicity value for the mass flow rate of the injector relative to the standard sand feeding system \(G_{M}=0.01\) kg/s, which represents integers from 1 to 4, i.e. \(G_j=G_M/j\).

3. Results

The calculation results for the dependence of the mass flow rate of the ejecting gas \(G_r\) and the velocity of the outlet flow from the nozzle outlet cross-section \(\omega_r\) on the pressure value in front of the working nozzle \(p_r\), calculated by expressions 1-4 are presented in figure 2 and figure 3. The portion of the curve to the right of \(p_{r,\text{min}}\) corresponds to a supercritical flow from the nozzle, and the portion on the left corresponds to the subcritical one.
Figure 2. Dependence of the mass flow rate of the ejecting gas $G_r$ on the pressure in front of the working nozzle $p_r$.

Figure 3. Dependence of the velocity of the outlet flow from the critical cross-section of the working nozzle $\omega_r$ on the pressure in front of it $p_r$.

The curves shown in figure 4, based on expression 5, allow you to determine the range of possible cross-section diameters of the working nozzle in the critical flow regime at different pressure values of the compressed air supplied to the injector body. The minimum diameter of the critical section is 1.85 mm. The maximum diameter is about 4 mm, because we are interested in values within the mass flow rate of up to 0.01 kg/s.

The dependence of the critical cross-section diameter of the working nozzle $d_{cr}$ on the pressure in front of it $p_r$ and the mass flow rate of the ejecting gas $G_r$ (figure 5) shows the nature of the influence of different mass flow rate values at different pressure values on the nozzle diameter in cross-section.
1-1. The zone located to the right of \( p_{r,\text{min}} \) is interesting for consideration, since the outlet flow from the nozzle occurs in the critical regime with the supercritical expansion behind the outlet cross-section.

![Graph](image1)

**Figure 4.** Dependence of the mass flow rate of the ejecting gas \( G_r \) on the cross-section diameter of the working nozzle \( d_{r,cr} \) and the pressure in front of it \( p_r \) in the critical flow regime.

![Graph](image2)

**Figure 5.** Dependence of the critical cross-section diameter of the working nozzle \( d_{r,cr} \) on the pressure in front of it \( p_r \) and the mass flow rate of the ejecting gas \( G_r \).

4. Discussion

Based on the resulting geometric parameters of the gas jet unit obtained above, a solid model of the air injector was designed for gas dynamic process simulation by means of the CFD software (figure 6). To calculate the internal stationary problem, boundary conditions were set in accordance with the goal of the study to increase the outlet flow velocity of the air-sand mixture by means of ejecting
an additional volume of ambient air. In the inlet cross-section of the receiving chamber and on the cross-section 3-3 (figure 1), the boundary condition "Environment pressure" was set, and in front of the working nozzle "Total pressure" was set. The result of modeling the gas injector flows in the form of the trajectory profile is shown in figure 6. The parameters of the mass flow rates were: \( G_r = 0.0051 \text{ kg/s} \), \( G_n = 0.0431 \text{ kg/s} \) and \( G_3 = 0.0482 \text{ kg/s} \). The resulting ejection coefficient was \( \nu = 8.5 \) with the bulk average flow velocity \( \omega_3 = 50.986 \text{ m/s} \) in the outlet cross-section of the injector mixing chamber.

**Figure 6.** Result of calculating gas dynamic processes in a pneumatic injector: boundary conditions, the calculation area and the flow trajectory profile in the longitudinal section.

5. Conclusion

Based on the above, we can draw the following conclusions. The velocity of the air-sand mixture flow from the outlet pipe can be increased up to 45 m/s or more with the help of an air injector. To compensate for the negative impact of the cross wind blowing away quartz sand particles during delivery to the wheel-rail contact area, the increase in the outlet flow velocity may be achieved by means of ejecting an additional volume of ambient air into the body of the gas jet unit of the sand feeding system with double reduction of the mass flow rate of the compressed air from the locomotive pneumatic pipeline relative to the standard sand feeding systems at the pressure of the working nozzle less than 0.5 MPa.

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