Analysis of granular materials vibrorheology of a railway sanding system

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Abstract. In this research work, after allowing the experiment to run, we proved the possibility to dose sand in the sand feeding system with the effect of vibratory displacement. We set the disturbing frequency and amplitude, at which the maximum system performance is achieved. The sand consumption curves were obtained depending on the oscillation frequency, pulse shape, and tilt angle. A model proposed to describe the effect of vibrodisplacement in a sand feeding system. It was established that disturbances cause sand oscillations at multitude frequencies, coinciding with which resonance phenomena occur. In practice, it means the increase in consumption.

1. Introduction
Currently, the railway industry is changing at rapid pace with numerous innovative technologies and introduction of new rolling stock. The next generation of rail vehicles, both domestic and foreign, has been successfully tested and put into operation. The rolling stock has improved dynamic and traction-performance characteristics due to the advanced mechanical part, electrical equipment, control systems, the introduction of asynchronous traction drive, up-to-date microprocessor diagnostics and control systems. The above-mentioned factors lead to an increase in traction properties, reliability of locomotives, safety of train movement, which makes it possible to extend the train mass and the length of the locomotive run.

However, despite the significant improvements in the rolling stock, rail transport still experiences an urgent problem of increasing and stabilizing the value of the coefficient of wheel-rail adhesion [1]. Unstable adhesion in the process of wheel rolling often leads to its breakdown and the occurrence of such negative phenomena as spinning when pushing and braking skid. To date, sanding in a compressed air jet under the wheels of locomotives still remains the most common and effective way to stabilize and increase the adhesion value. For these purposes, the rolling stock is equipped with a pneumatic sanding system.

In various locomotives the sand feeding system is based on the general principle involving taking sand from the nozzle and transporting it to the wheel-rail contact area (figure 1). Depending on the design features and purpose of the traction units, as well as the regions of operation, the systems differ in the length of the supply pipeline, the capacity of the sand bunkers, the amount of sand consumed, the feeding method (continuous and momentum). Nevertheless, they have common features, namely "winter-summer" seasonal setting of the sandbox nozzle without the adjustment possibility considering the actual weather and climatic conditions during operation. An attempt to automate the sand feeding is
the introduction of a pulsed sand feeding, installed parallel to the manual one in the driver’s cab. However, there is an excess sanding both in the manual control (continuously), and in the case of automatic detection of spinning (impulsively, at a predetermined interval), which is explained by the fixed setting of the sandbox nozzles.

Numerous researches indicate that the amount of sand in the wheel-rail contact area has an optimal value, which varies depending on the actual operating conditions [2]. Excessive amount of sand reduces the coefficient of adhesion which is assumed a significant disadvantage, as sand contributes to additional abrasive wear of rubbing surfaces, increases resistance to movement, clogs the ballast section of the track structure, increases the rolling stock downtime during filling at sand plants, which leads to notable costs.

Nowadays, the rolling stock is not equipped with a system for monitoring the actual amount of sand fed when moving on the running line. The most common problems in the sand feeding are the clogging of air passages with sand particles and the misalignment of sandbox nozzles. In the first case, the defect leads to a system failure, and in the second, to uncontrolled over-consumption of sand, which may cause "unexpected" emptying of the sand bunker and sanding termination. The reason for the clogging air passages is the poor quality of the sand used, its physical and chemical properties, particle-size distribution, humidity, etc. The poor quality adjustment of the sand feeding in maintenance depots contributes to the misalignment of sandbox nozzles.

The adjustment bolt is locked, when aligning sandbox nozzles. It should be noted that it is possible to tighten the locknut with insufficient force, which under the action of dynamic loads leads to a loosening of the connection with the subsequent spontaneous unscrewing of the bolt. Some series of locomotives uses adjusting bolts with metric or inch threads, which have a coarse pitch. The most common are the bolts with M10 and 7/16 "UNC threads, whose thread pitch is equal to 1.5 and 1.8 mm respectively. The given fact also increases the tendency to self-loosening.

It is proved that compared to other threads metric ones have a higher coefficient of friction [3]. With equal outer diameters, metric threads with fine pitch differ from threads with coarse pitch, smaller profile height and, as a result, smaller thread lifting angle and efficiency of the threaded pair. Reducing the efficiency of threads with fine pitch is a consequence of the increased work of friction forces; therefore, compared with threads with a coarse pitch, threads with fine pitch are more reliable in terms of self-loosening. This conclusion is also confirmed by the fact that with the same friction angle of a metric thread equal to 30, the lifting angle should be less to fulfill self-braking condition. The lower its value, the more resistant to self-loosening is the threaded connection. Thus, reducing the thread pitch on the adjustment bolt helps to improve the sandbox nozzle adjustability under operating conditions.
The studies conducted in VNIIZhT (Russian Railway Research Institute) also confirm the unsatisfactory adjustment capability of the sandbox nozzles [4]. As can be seen from Figure 2, ON3-64 sandbox nozzles, the most common ones installed in electric locomotives, can supply a sufficiently small amount of sand to the wheel-rail contact area. The greatest amount of sand on electric rolling stock, up to 1500 g/min, is consumed in winter. Such a consumption rate can be ensured when the air passage is opened. It is locked with an adjusting bolt in the range from ¼ to 3/8 of the turn to leave the air gap equal to 0.19-0.28 mm. It is worth mentioning that it is rather difficult to leave such an insignificant air gap, since the malfunctioning technologies prevent the nozzle bodies of sandboxes from providing the necessary alignment of the adjusting bolt with the hole closed by it. As the graphs demonstrate, a small increase in the cross section of the air passage is sufficient for the nozzle performance to grow dramatically: when the adjusting bolt is released by ½-3/4 turns, the sand mass flow rate exceeds the maximum allowable value.

Figure 2. Dependence of the sand mass flow rate by ON3-64 nozzle and the velocity of the air outflow from the end nozzle (D<sub>out</sub> = 20 mm) on the position of the adjusting bolt: I – the nozzle performance; II – velocity of the air outflow; 1 – C<sub>n</sub> = 3 kG / cm²; 2 – C<sub>n</sub> = 5 kG / cm²; 3 – C<sub>n</sub> = 6 kG / cm²

The goal of the paper is to improve the sanding system of the traction rolling stock using the vibrorheological properties of sand. The research objectives focus on designing a physical model of the locomotive sanding system, conducting an experiment on the sand vibrodisplacement by varying the exposure amplitude and frequency, suggesting a model for describing the processes, identifying the frequencies at which the system has the highest level of performance.

2. Methods
The laboratory assembly is a complex of devices, including:
- Vibrobench VEB Robotron-Meßelektronik "Otto Schön" Dresden, type 11077 (ese 221);
- Frequency generator;
- Integrated accelerometer with matching unit based on AVR microcontroller;
- The software to register the acceleration sensor indications;
- Notebook;
- Electronic weighting unit;
- Simulation model of the sandbox nozzle cavity with Du = 40 mm membrane;
- Digital level;
- Digital chronometer;
High quality railway silica sand: 95% quartz content, 0.1-2 mm particle size.

The laboratory assembly is shown in the figure 3

**Figure 3.** The laboratory assembly: 1 Simulation model of the sandbox nozzle cavity; 2 Frequency generator; 3 Integrated accelerometer; 4 Latex membrane; 5 Plunger; 6 Rod.

The experimental methodology.

The digital level is used to set a vibrobench and fix a simulated model of a sandbox nozzle cavity, tilted at an angle of 60 degrees relative to the horizon. Under its own weight, silica sand is poured into the upper neck to the maximum level. Next stage involves turning on the generator with a pre-set shape, frequency and amplitude of the output signal and starting the timer. If the force action of the piston is sufficient, then the layer of silica sand particles is converted to a fluid-like state and flows out from the nozzle cavity model into the container. The generator is switched off after a specified time. Its values are fixed. The mass of sand inside the container is then weighed. The data obtained from the accelerometer is automatically stored in a file with the date and time of the experiment. The previously described algorithm is repeated as many times as necessary. The experiment is summarized by analysis and interpretation of findings and drawing conclusions.

**3. Results and Discussion**

The following data represent the results from an experiment defining the sand mass flow rate. The figure 4 shows the mass flow rate when exposed to rectangular pulses.

**Figure 4.** Sand mass flow rate when exposed to rectangular pulses, depending on the oscillation frequency of the generator.
The undermentioned figure 5 shows the sine wave oscillation.

![Graph showing sand mass flow rate affected by sinusoidal oscillation and generator frequency.](image1)

**Figure 5.** Sand mass flow rate affected by the sinusoidal oscillation and the generator frequency.

As can be seen from the figures 4-5, a rectangular pulse shape demonstrates smoother curve and the absence of extreme in 17-18 Hz frequency range. The process of extreme emergence can be traced with power spectral density accelerations.

Consider generator oscillations at 26 and 27 Hz frequencies. 26 Hz frequency corresponds to a maximum point in the figure, and 27 Hz frequency is close to a local minimum. The figure 6 shows calculated power spectral density of the process. We assume that the remaining frequencies characterize the natural oscillations of the system; then the convergence of the disturbing frequency with one of the natural frequencies will obviously cause a resonance, which is observed in this case. The accelerations power spectral density at a disturbing frequency equal to 27 Hz is shown in the figure 7. The vibrobench disturbance corresponds to 26.7 Hz frequency, which apparently contributes to the oscillations appearance at a frequency of 24 Hz; however, since these frequencies are relatively far from each other, no resonance occurs.

![Graph showing power spectral density of accelerations at 26 Hz.](image2)

**Figure 6.** Power spectral density of accelerations at 26 Hz.

![Graph showing power spectral density of accelerations at 27 Hz.](image3)

**Figure 7.** Power spectral density of accelerations at 27 Hz.

Thus, based on the data obtained, we can conclude that resonance contributes to the high performance of the sand feeding system as shown in the figure 6. External disturbance of the vibrobench causes additional oscillation frequencies, which depend on the disturbing frequency.
The figure 8 presents the dependence of the mass flow rate on the tilt angle.

![Figure 8. Sand mass flow rate depending on the tilt angle](image)

Perform a theoretical estimate of the results obtained using the approaches outlined in [5].

In the first approximation, the dynamic properties of the system can be described by applying computational scheme model, depicted in the figure 8. It can be shown that sand mass consumption is proportional to the air flow rate with the coefficient \( \rho S = \text{const} \).

\[
\dot{m} = \rho S \dot{v}
\]  

where \( \rho \) is sand bulk density; \( S \) – the cross-sectional area; \( \dot{v} \) – the air flow rate.

Then, the sand displacement can be approximately represented as the movement of a deterministic mass at a certain speed. We assume that \( F_0 \) force affects the mass. The given force is proportional to \( \alpha \) acceleration that acts on a certain mass \( m^* \). Therefore,

\[
F_0 = m^* \alpha \sin \omega t = m^* \dot{\xi}
\]

The scheme in the figure 9 (a) corresponds to classical mechanics, while the scheme in the figure 9 (b) correlates with vibration mechanics [5].

![Figure 9. Computational scheme](image)

The classical mechanics equation looks like that

\[
m \ddot{x} = F(\dot{x}) + F_0 + T
\]

where \( F(\dot{x}) \) is the friction force in the general depending on the displacement speed, \( T = -mg \sin \alpha \).

\[
F(\dot{x}) = \begin{cases} -f_+ & \text{if } \dot{x} > 0 \\ f_- & \text{if } \dot{x} < 0 \end{cases}
\]
\[ -f_+ < F(\dot{x}) < f_- \text{ if } \dot{x} = 0 \]  

(5)

where \( f_+ \) – specific friction force when moving in the positive direction; \( f_- \) – specific friction force when moving in the negative direction; \( m \) – mass of displaced sand.

The mechanical vibration equation [5] is written as follows:

\[ m\ddot{x} = T + V \]  

(6)

\[ m\ddot{\psi} = F(\dot{x} + \dot{\psi}) - \langle F(\dot{x} + \dot{\psi}) \rangle - m\dot{\xi} \]  

(7)

where \( V = \langle F(\dot{x} + \dot{\psi}) \rangle \) is the vibration force; \( \langle ... \rangle = \frac{1}{2\pi} \int_0^{2\pi} ... \, dt \) – mean operator.

The steady-state speed at \( m = m^* \) has the following solution

\[ \dot{x}_s = \frac{a}{\omega} \cos \frac{\pi(f_+ - T)}{f_+ + f_-} \]  

(8)

where \( f_+ \) – specific friction force when moving in the positive direction; \( f_- \) – specific friction force when moving in the negative direction.

We present the solution in the following form

\[ \dot{x}_s = a^* \cos \phi, \]  

(9)

\[ a^* = \frac{a}{\omega}, \]  

(10)

\[ \phi = \frac{\pi(f_+ - T)}{f_+ + f_-}. \]  

(11)

Thus, the speed of sand displacement is proportional to \( a^* \) value. To compare the experimental data with the theoretical estimate, we will normalize the values in accordance with the formula

\[ y = (x - x_{\min})(x_{\max} - x_{\min})^{-1}. \]

The figure 10 shows the comparison of the results.

![Figure 10. Normalized mass flow rate and accelerations](image)

As can be seen from the figure, the theoretical estimate and experimental data correlate well with each other, especially in the area of local maximums. Differences in values in 20-21 Hz frequency areas are explained by the presence of additional oscillations other than the disturbing frequency, as manifested in the figure 11.
4. Conclusions

The following conclusions can be drawn from the present study.

1. After allowing the experiment to run, we proved the possibility to dose sand in the sand feeding system with the effect of vibratory displacement.
2. We set the disturbing frequency and amplitude, at which the maximum system performance is achieved.
3. The sand consumption curves were obtained depending on the oscillation frequency, pulse shape, and tilt angle.
4. A model proposed to describe the effect of vibrodisplacement in a sand feeding system.
5. It was established that disturbances cause sand oscillations at multitude frequencies, coinciding with which resonance phenomena occur. In practice, it means the increase in consumption.
6. Increasing tilt angle has a negative impact on the efficiency of sand feeding.
7. At the bench test stage, we intend to identify the model by introducing additional masses and elastic elements, or using continuous medium models taking into account the vibrorheological properties of sand.

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