Modelling the dynamics of an undercar generator with a v-belt drive of an isothermal railway vehicle

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Abstract. In this research work, mathematical modelling has been performed and the oscillations of the isothermal railway vehicle on the KVZ-I2 bogies with the v-belt drive of the undercar generator have been investigated. As a result, the natural frequencies of oscillations of the railway vehicle have been determined, the crew oscillations caused by random track irregularities have been investigated, the influence of the irregularities of the bogie frame on the generator oscillations has been evaluated.

1. Introduction
One of the solutions for increasing the power of the autonomous power supply system for railway vehicles is a v-belt drive of the undercar generator from the middle part of the wheelset axle [1, 2]. In the paper [2] the options of v-belt drive designs from the middle part of the wheelset axle and their estimated power are described. The issues of modelling the railway vehicle dynamics with the v-belt drive of the undercar switched reluctance generator are described in the articles [1, 3, 4].

Most studies on the dynamics were carried out using the mathematical model of the rail vehicle oscillations in a vertical longitudinal plane [1, 2], which made it possible to identify the main behaviour patterns of the object studied. However, in certain cases, more sophisticated models are required, including those describing spatial oscillations.

The purpose of this study is mathematical modelling and the research of the oscillations of the isothermal car on the KVZ-I2 bogies with a v-belt driven undercar generator. The following tasks were set: to determine the natural frequencies of the rail vehicle oscillations, to investigate the crew’s oscillations in case of random track irregularities, to evaluate the influence of irregularities of the bogie frame on the generator oscillations.

2. Methods
Figure 1 shows the design of the undercar generator drive. The feature of this drive is the generator suspension system, in which the generator is mounted on the head stock of the bogie frame by means of a joint. In the middle of the wheelset axle, the driving pulley of the belt drive is mounted. The following pulley is located on the generator shaft.
The software options used to study the drive dynamics were the UMLoco and Simpack programs. In "UMLoco" the spatial model for the rail vehicle with the generator drive was created. The model developed in Simpack was used to analyze the stability of the generator oscillations. To determine the mass and inertia characteristics of the bodies, a three-dimensional model of the bogie was created in the Kompas3D program.

We present a brief description of the elements of the rail vehicle model. The elastic properties of the primary suspension spring were calculated in UMLoco using a three-dimensional model.

The secondary suspension was made as a separate subsystem, taking into account the nonlinear properties of the spring. The “Spring Fancher” element was used as a spring model, with \( \beta \to 0 \); the friction coefficients for compression and tension were assumed to be identical. As a result, the spring force characteristic had the form

\[
(\xi) = c(f_{st} + \xi)(1 + \varphi \, sgn \, \dot{\xi}),
\]

where \( c \) is the spring stiffness; \( f_{st} \) – static deflection; \( \varphi \) – relative friction coefficient; \( \xi \) – additional deflection counted from static deflection. The operation of the center plate arrangement and side bearings was described, taking into account the contact interaction in these elements.

The generator with the belt drive and tensioning device was created as a separate subsystem. The tensioner and the belt set were described by an elastic element with a linear force characteristic.

The appearance of the bogie model in the Universal Mechanism is shown in figure 2.
The model of the generator suspension system is shown in figure 3. The letter O denotes the joint, the letter A is the axis of the following pulley, the point of the tensioner force application is indicated by the letter B.

![Drive Model in the Simpack program.](image)

Technical characteristics of drive design options are presented in table 1.

Table 1. Technical characteristics of drive design options

| Design option | Belt drive stiffness \( C_b, \text{N/ mm} \) | Tensioner spring stiffness \( Z_{sp}, \text{N/ mm} \) | Distance OA, m | Distance OB, mm | Belt tension force, N |
|---------------|---------------------------------|---------------------------------|-----------------|-----------------|---------------------|
| 1             | 880                             | 105                             | 0.23            | 0.36            | 4500                |
| 2             | 1400                            | 150                             | 0.23            | 0.4             | 6000                |
| 3             | 1500                            | 150                             | 0.3             | 0.3             | 6000                |

The studies of oscillations with random track irregularities were carried out in the time domain. On the basis of the formulas for the irregularity spectra by the International Union of Railways (UIC), irregularities were obtained for a 1000 m long track. The condition of the track was chosen to be poor. The calculations were performed for speeds of 5–160 km/h. The root mean square (RMS) was taken as a functional characteristic of the oscillations.

The stability regions of the generator oscillations were found numerically. The instability of oscillations was understood as a significant increase in the RMS values for given parameters. The irregularity was taken as the function in the form of \( z_1 = A \sin \omega t \), where \( A \) is the amplitude of the kinematic irregularities, \( \omega \) is the oscillation frequency. The irregularity was applied to the joint, by which the generator was fixed (Figure 3). The amplitude varied in the interval of 1 - 20 mm, and the range of frequency was 1 - 30 Hz.

3. Results

The power spectral density for the oscillations of the bogie frame and the generator at the speeds of 40 and 80 km/h (Figure 4) was estimated. Figure 4a shows the oscillation frequencies of the bogie frame bouncing without a generator.
Figure 4. Power spectral density of the bogie frame oscillations without a generator (a), the bogie with the generator at the speed of 40 km/h (b) and at the speed of 80 km/h (c): 1 – the speed of 80 km/h; 2 – the speed of 40 km/h; 3 – the oscillations of the frame bouncing; 4 – the angular oscillations of the generator.

The natural frequencies (Hz) and mode shapes of railway vehicle are shown in figure 5.
Figure 5. Natural frequencies (Hz) and mode shapes of the railway vehicle.

The natural frequencies of oscillations for different design options are defined further. The natural oscillation frequencies of the generator in accordance with the options are presented in Table 2.

Table 2. Eigen Frequencies

| Option number | Frequency (Hz) | Real part | Imaginary part |
|---------------|----------------|-----------|----------------|
| 1             | 6.63305        | 0         | 41.6767        |
| 2             | 8.23809        | 0         | 51.7615        |
| 3             | 8.55098        | 0         | 53.7274        |

The calculation results for the instability regions of design options 1 - 3 are presented in Fig. 6 - 8 in the form of RMS isolines with color filling.
Figure 6. Contour plot of RMS of the generator oscillations for option 1.

Figure 7. Contour plot of RMS of the generator oscillations for option 2.
4. Discussion
The power spectral density plots for the bogie acceleration without a generator and with the generator are significantly different. PSD for the bogie without a generator has one extreme. The oscillations of the bogie with the generator occur at different frequencies, which are caused by the generator and the car body oscillations. Due to the center of mass shift of the bogie frame, the angular and vertical bogie oscillations are connected with each other and the oscillations are observed to appear at the frequencies of 8-10 Hz at the speed of 80 km/h (Figure 4c).

In Figures 6–8, two instability regions can be distinguished, which expand with the increase in the irregularity amplitude, at the natural frequency of the generator and at the double natural frequency. Such behavior is characteristic of the systems with parametric excitation and in the study [3] the oscillation equation is presented, which can be reduced to the Mathieu equation.

5. Conclusions
The spatial computer model of the isothermal rail vehicle with the drive of the undercar generator has been created.

In this study, the natural frequencies of the oscillations for the rail vehicle with a belt-driven generator have been determined, which are presented in the table.

The location of the generator at the end of the bogie frame and consequently the center of mass shift result in the mutual influence of oscillations.

The developed computer model of the generator suspension system in the Simpack program allows for the calculations of design options using various irregularity models of the bogie frame.

The natural frequencies of the generator oscillations have been determined for three design options, which correspond to 6.63305 Hz, 8.23809 Hz, 8.55098 Hz, respectively.

The conditions have been found under which the parametric resonance will arise on the generator with the revolute joint suspension design.
References

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