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Application of Elastic Fastenings of Equipment to Increase Vibration Frequency of the Wagon Body

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ABSTRACT

Improving dynamic performance is a priority when designing new rolling stock. The frequency of natural bending vibrations of the body is one of the most important standardised indicators, a preliminary assessment of which allows obtaining optimal body designs.

The objective of the work is to assess the prospects for the use of elastic fastenings of equipment to increase the natural vibration frequency of wagon bodies of suburban electric trains. Calculations were based on the finite element method and block Lanczos method.

It is shown that it is advisable to use the rigid area tool and linearly elastic finite elements to calculate the frequencies in the simulation. The main ranges of fastening stiffness are highlighted, where the effect of using elastic supports is different. It is proposed to determine the stiffness of fastenings according to a given vibration frequency of the equipment. When the equipment is rigidly attached, the relative mass of the equipment does not affect the body bending vibration frequency. With elastic fastening, a greater effect can be achieved with a larger relative weight of the equipment. The effect of using resilient mounts increases with heavier equipment located closer to the centre of the body. It is shown that the effect of shear admittance of fastenings on the body vibration frequency is within 1 % and may not be considered in the simulation. In the considered example of a wagon body of a suburban electric train, the use of elastic supports allows an increase in the frequency of oscillations of the body by 3–10 %.

Keywords: *railway, body, electric train, fastening stiffness, finite element method, modal analysis, natural frequency, wagon dynamics.*

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INTRODUCTION

The development of modern railway transport involves an increase in design speeds while ensuring traffic safety. An increase in speed of movement inevitably causes difficulties in ensuring the dynamic qualities of the running gear. One of the ways to solve this problem is to reduce the weight of wagons. Optimisation of the design of the running gear to reduce weight is relevant for any speed of movement, since it leads to a decrease in wear in the wheel-rail system, a reduction in energy consumption and an increase in passenger capacity [1].

One of the elements of the wagon with the highest specific weight is the body. However, reduction in the mass of the bodies leads to a decrease in overall rigidity that negatively affects frequency of natural vibrations, the level of vibration acceleration, and, accordingly, the indicators of smoothness. An increased level of dynamic loading of the body can lead to fatigue damage [2]. Therefore, development of various damping systems is relevant. For the entire vehicle, the correct choice of parameters of two stages of spring suspension is first of all important [3; 4]. Choosing the most efficient body design in terms of weight and rigidity is also important to ensure dynamic performance. One of the normalized parameters of dynamic properties is the first frequency of natural bending vibrations of the body (FNBV). According to Russian domestic regulatory documents, its value for a fully equipped car body of multiple unit rolling stock should not be less than 8 Hz[1]. This parameter allows an express assessment of the dynamic qualities of the body without building a complete dynamic model. The higher is FNBV value, the lower the average levels of vertical acceleration will be and, other things being equal, the lower will be the smoothness of the ride. An increase in the value of the first FNBV can be achieved with the help of new structural materials, optimisation of the body structure and the location of the internal ribs of extruded panels, and variation in the location of window and door openings [5–8]. In case of using a rigid fastening of undercarriage equipment, an increase in the value of FNBV can be achieved with the help of a rational arrangement of boxes [9]. Meanwhile, the elastic attachments of equipment can also affect the dynamic qualities of the vehicle [10].

Studies of high-speed rolling stock show that the use of elastic supports, with the correct choice of rigid and damping properties, makes it possible to reduce the amplitudes of vibration accelerations in the vertical and transverse directions and improve the smoothness of the ride [11]. Suspension parameters must be carefully selected to avoid resonance during joint vibrations of equipment and body [12].

Vibration damping is possible due to the use of active and passive vibration dampers, vibration isolating supports. Due to the high cost of dampers, vibration isolating supports are most often used. However, in this case, it is not possible to achieve high values of the damping coefficient due to significant heating of the isolating material, which increases the likelihood of its aging and destruction [13].

Dynamic calculations can be performed analytically, using the differential equation of motion and a beam model of the body [14], or using three-dimensional finite element models [11]. The latter method is the most accurate, as it allows to adequately consider the actual rigidity of the body [15]. Until now, studies of influence of flexibility of equipment fastenings have been carried out for high-speed rolling stock without considering the influence of standing passengers [10; 12; thirteen].

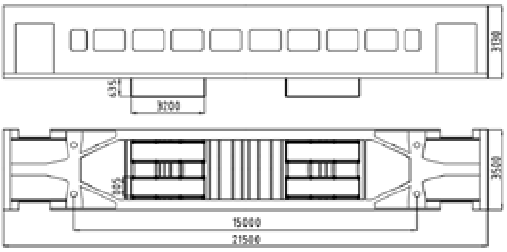
The *objective* of this work is to assess the prospects of using flexible supports for fastening equipment of wagons of suburban electric trains marked by intense vibrations of the frame structure.

RESULTS

Body Design and Calculation Procedure

The body of a suburban train wagon taken as an example for further study (Pic. 1), made of structural steel, consists of a frame, side and end walls and a roof. For the body material, Young's modulus is set to $2{,}1 \cdot 10^5$ MPa and Poisson's ratio is 0,3. A structure with two planes of symmetry is considered, which, under symmetric vertical loading, allows using a quarter of the body in calculations. The body frame has a classic design, consisting of a closed profile body frame, cross bearers, span bolsters and reduced centre sills. The span bolsters are connected to the body frame with side frame diagonals. The vestibules are located in the end parts of the body, there are exits to the lower platforms. In the undercarriage space there are four equipment boxes of the same weight. The

[1] GOST [Russian state standard] 33796-2016. Multiple unit rolling stock. Requirements for strength and dynamic properties, P. 3.



Pic. 1. Structural diagram of an electric train wagon body (the author's diagram).

structure is loaded with its own gross weight, including the weight of passengers sitting and standing on the free floor area at the rate of 7 persons/m²[2]. The location of all internal elements of the wagon is considered on the basis of the design documentation. Loads are applied by varying the density of the respective body parts.

The body is located on elastic supports with a vertical stiffness of 0,66 MN/m, which corresponds to the total stiffness of two suspension stages for a gross weight of 55000 kg of a wagon body. Since only vertical vibrations are considered, stiffness of supports in other directions is not taken into account in all cases, with the exception for a separate assessment of their impact on FNBV. For four boxes of undercarriage equipment, an assessment was made of the influence of rigidity of equipment fastenings on FNBV. Each box weighs from 275 to 1100 kg. The rigidity of the box was modelled in two ways: using the full frame model of the box and using the rigid area. The rigid area assumes the mutual limitation of translational degrees of freedom for the main node corresponding to the centre of mass of the box, and the secondary nodes, which are located at eight points of the box attachment – the lower nodes of elastic elements that simulate flexible attachments. The upper nodes of these elastic elements are located on the brackets and bearers of the body at the attachment points of the equipment.

Frequencies of natural vibrations are determined using ANSYS software package by the finite element method using the block Lanczos method. To simulate the body and equipment, four-node thin shell elements, one-node mass elements, and two-node linear elastic elements are used. The average shell finite element size in the model is 50 mm.

Analysis of Simulation Results

The results of calculating FNBV with stiffness of fastenings from 0 to 1000 MN/m are shown in Pic. 2. Simulation of equipment boxes with a rigid area leads to an error in determining the upper FNBV up to 1 % (left curve) and the lower FNBV up to 3,5 % (right curve) for a given body design. In further calculations, the rigidity of the equipment is modelled using a rigid area.

Natural vibration frequencies of equipment boxes as a rigid whole body with low rigidity of fastenings do not cause bending vibrations of the body. Their oscillations occur at frequencies of 3–6 Hz, which exceed the frequency of body bouncing oscillations, but do not exceed FNBV. With an increase in rigidity of fastenings, the natural vibrations of the boxes begin to cause bending vibrations of the body, the amplitudes of which increase. At a certain stiffness value of fastenings, the amplitude of body oscillations on the lower FNBV (at which the boxes and the body experience in-phase oscillations) become comparable to the amplitudes on the upper FNBV (oscillations of the boxes and the body in antiphase). With a further increase in rigidity of fastenings, bending vibrations of the body are determined by the lower FNBV (Pic. 2). In the limiting state, when the rigidity of fastenings is high (the absence of elastic fastenings), the masses of the body and equipment also vibrate in one and the same direction.

Thus, it is possible to distinguish three conditional ranges of rigidity of equipment fastenings, which affect the dynamic qualities of the body. The first range of low stiffness (Pic. 2) is characterized by low vibration frequencies of the equipment and does not meet safety requirements of railway transport due to possible resonance with vibrations of the body on spring suspension. The second range of medium stiffness is transitional and its left part contains mostly the values of the fastening stiffness that are the most optimal from the point of view of improving the dynamic qualities of the body. However, when the stiffness values correspond to its right side, it is possible that low FNBV appear, which devalues the use of elastic fastenings. Finally, in the third range of high stiffness, the body experiences in-phase vibrations with the equipment boxes at frequencies lower than the frequency when the equipment is rigidly attached.

Also, FNBV values were obtained for various weight fractions of equipment. The values of stiffness of elastic fastenings were taken at a given natural vibration frequency of the equipment as of a body with one degree of freedom:

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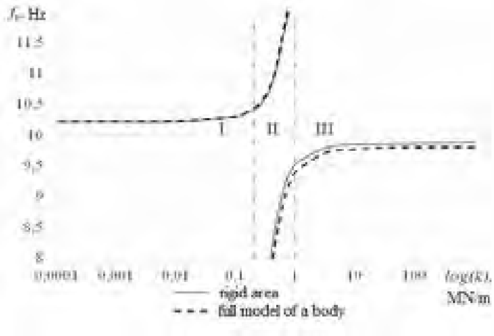
[2] GOST 33796-2016. Multiple unit rolling stock. Requirements for strength and dynamic properties, P. 3.



Pic. 2. FNBV with different rigidity of fastening of undercarriage equipment with a total weight of 4400 kg (developed by the author).

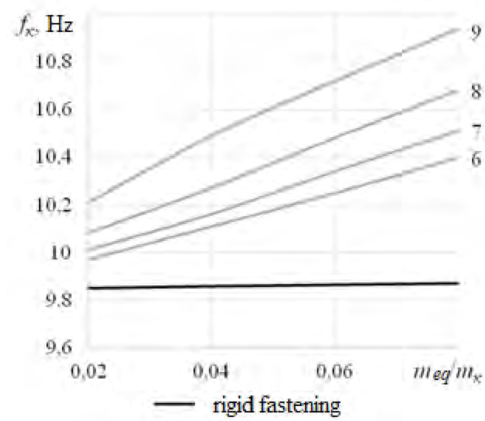
$$k = \frac{4\pi^2 f^2 m_{eq}}{n}, \quad (1)$$

where f is frequency of natural vibrations of the equipment;

m_{eq} – mass of the equipment;

n – number of box fasteners.

The range of equipment vibration frequencies 6–9 Hz is considered. For all considered fastening stiffnesses, the upper FNBV is proportional to the value of the specific weight of the equipment (Pic. 3). Consequently, the greater is the weight of the equipment located on the body, the greater increase in the upper FNBV can be achieved.



Pic. 3. Dependence of FNBV on specific weight of the equipment (on the right – values of f, Hz) (developed by the author).

When setting a certain frequency of the box for selection of the stiffness of fastenings, it should be borne in mind that the actual natural frequencies of the boxes can be, as the calculation shows, 2–5 % less than the indicated frequency. This is due to the additional flexibility of the fastening areas in the body bearers.

To effectively reduce body vibrations with a relative weight of the equipment of 0,02–0,08, the frequency of the equipment should be approximately 12–42 % lower than FNBV on spring suspension, taking into account the flexibility of the equipment fastenings.

With a given mass of equipment, the greatest influence of the equipment on FNBV can be achieved by increasing the rigidity of its fastenings. However, the stiffness values in this case must belong to the left side of the transition range in order to avoid the appearance of resonant oscillations at the lower FNBV. For example, Pic. 4 shows the forms of bending vibrations of the body for a specific gravity of the equipment of 0,08 and vibration frequencies of the equipment of 6 and 9 Hz. If at a frequency of 6 Hz the amplitudes of body oscillations at the lower frequency are an order of magnitude less than the amplitudes at the upper one (Pic. 4a), then at a frequency of 9 Hz the amplitudes of the oscillations differ by a factor of two (Pic. 4b). Thus, the use of the equipment fastening rigidity higher than 0,44 MN/m (which corresponds to the equipment vibration frequency of 9 Hz) in this case can lead to the undesirable appearance of resonant oscillations at the lower FNBV of 8,38 Hz (Pic. 4b). In fact, this will mean a negative effect of the use of flexible fastenings, since FNBV with a rigid fastening of equipment in this case is 9,85 Hz.

When the equipment is rigidly attached, FNBV on flexible supports does not depend on the specific weight of the equipment. At the same time, it is known that there is a proportional increase in FNBV on rigid supports with an increase in the specific weight of rigidly fixed equipment [9].

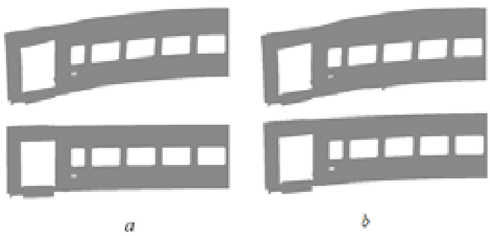
The influence of arrangement of boxes of undercarriage equipment was estimated by calculations for models with four, two and one boxes of the same total mass of 4,4 tons. Each box had eight fastenings (Pic. 5).

When the equipment is located in four boxes, the upper FNBV turns out to be the smallest. When the equipment is located on the sides of the wagon in two boxes or in the centre of the body in a single box, FNBV increases by about 2–3 % (Table 1). The instability of frequency values in case of one or two boxes is explained by the different flexibility of the fastening bearers. A more distant location of the equipment from the centre of the body leads to less compensation for body oscillations in antiphase with the equipment and, accordingly, to a lower value of FNBV. Thus, it is most efficient to locate heavy equipment on elastic supports in places

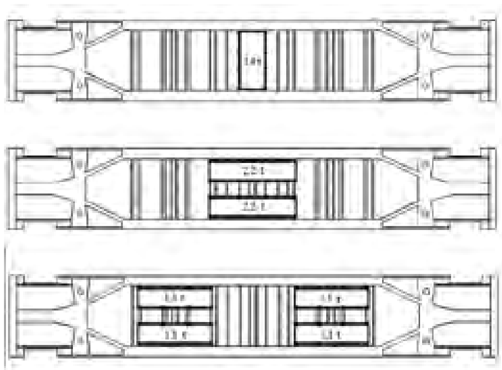
Table 1

Influence of equipment location on FNBV

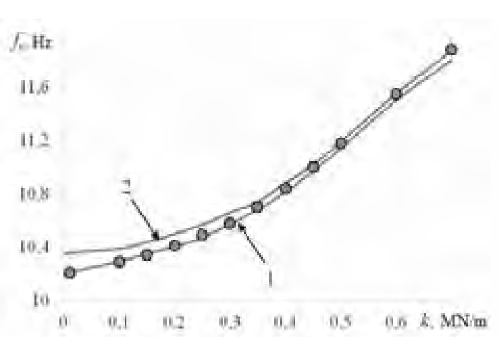
Equipment vibration frequency, Hz	FNBV with the indicated number of boxes, Hz		
	1	2	4
6	10,61	10,62	10,4
7	10,77	10,8	10,51
8	10,94	11,02	10,68
9	11,13	11,28	10,94



Pic. 4. Forms of body vibrations on the upper (top) and lower (bottom) FNBV at the equipment frequency of 6 (a) and 9 (b) Hz (author's scheme).



Pic. 5. Variants of location of the undercarriage equipment (the author's schematic diagram).



Pic. 6. Dependence of FNBV on vertical stiffness of the equipment fastenings with absolute flexibility (1), absolute rigidity (2) of fastenings in two perpendicular directions. Markers show the same rigidity of fastenings in three directions (developed by the author).

with the highest vibration amplitudes of the body, that is, near its central section. Nevertheless, it is more important to choose the optimal value of rigidity of the equipment fastenings, which allows increasing FNBV in the example under consideration by 3–10 %.

Anti-vibration mountings usually exhibit flexibility not only in the vertical, but also in the longitudinal and transverse directions [13]. To assess the effect of the shear stiffness of supports in the other two directions on FNBV, the body model was supplemented with elastic elements in the attachment areas, which ensure that the stiffness of the supports in the longitudinal and transverse directions is taken into account. Thus, each support for fixing the equipment box was modelled by three independent elastic elements.

The calculation results are shown in Pic. 6. FNBV values corresponding to curve 1 were obtained without considering shear stiffness of fastenings, to curve 2 – considering the absolute shear stiffness. The results obtained at any actual values of shear stiffness are located between these curves. For example, in the case of equal rigidity of fastenenings in three directions (markers in Pic. 6). For the values of the vertical stiffness that can be used to increase FNBV (0,1–0,7 MN/m), the differences in FNBV values at various shear stiffnesses do not exceed 1 %. Therefore, when choosing rigidity of equipment fastenings, the shear rigidity of fastenings can be neglected. This simplification is permissible in the case of a bending body shape characterised by intense vertical vibrations. However, in case of considering other modes of vibration (for example, asymmetric diamond-shaped one [16]), shear stiffnesses can have a more significant effect on frequency of natural vibrations.

CONCLUSION

The installation of equipment on flexible fastenings with the optimal selection of their rigidity leads to an increase in FNBV of a suburban electric train wagon. The preliminary selection of rigidity of fastenings can be carried out according to the values of the natural vibration frequency of the equipment.

Simulation of equipment installed on flexible supports is most efficiently performed using linear elastic elements and a rigid area tool, which allows estimating FNBV with an error of no more than 1 % compared to the method of full modelling of the equipment frame. In this case, the influence of the shear stiffness of the supports on FNBV is less than 1 % and may not be taken into account.

The heaviest equipment, when installed on flexible fastenings, is advisable to be located closer to the centre of the body for more effective compensation of bending vibrations of the body.

With a constant gross mass of the body, an increase in the relative mass of the equipment installed on flexible mountings from 0,02 to 0,08 leads to an increase in FNBV on elastic supports, and practically does not cause changes in FNBV in case of a rigid attachment of the equipment.

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