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Methods and results of evaluating the dual-power electric train crew elements service life

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Abstract. The article assesses the long-term strength of the vibration damper assembly of a double power transmission of electric trains manufactured by «Hyundai-Rotem Company». The technique is described and the analysis of the service life of the elements of the chassis of the electric train is made. The presence and nature of micro cracks in the loaded elements of the vibration damper assembly was evaluated after a full cycle of vibration bench tests. The influence of vertical vibrations of the electric train body on the level of dynamic load and the service life of the bearing elements of the vibration damper assembly is considered. The service life of the vibration damper at the end of the service life has been estimated by comparing working hours and “statistical playback”. The calculation of the resource by the method of "statistical reproduction" confirms the possibility of operating the vibration damping module (second modernization) for 50 years from the date of construction. The influence of the vibration damper position on its dynamic loading is checked. The main stages and sequence of tests for estimation the resource of the longitudinal vibration damper module of a double-fed electric train are also considered, including methodological, calculated and experimental confirmation of the reliability and durability of the unit, which was studied as part of a comprehensive work.

1. Introduction

This article is devoted to a review of the work, carried out by specialists of the Dnipro National University of Railway Transport named Academician V. Lazaryan (DNURT) in assessing the long-term strength of the vibration damper assembly of the dual electric power train.

The purpose of the research was:

- assessment of the presence and nature of microcracks in the loaded elements of the vibration damper assembly after a full cycle of bench vibration testing;
- evaluation and consideration of the influence of the vertical oscillations of the body of the electric train on the level of dynamic loading and the life of the bearing elements of the vibration damper assembly;
- estimation of the end-of-life service life of the vibration damper by methods of comparing the operating hours and “statistical playback”;
- formation of conclusions regarding the quality of construction and the durability of the vibration damper assembly.



- verification of the conformity of the materials [12,13] of the units of increased reliability of the pivot beam of the interregional electric train (hereinafter referred to as EP) of the Hyundai-Rotem double feeder (hereinafter referred to as the vibration damper assembly) used in conducting bench vibration tests and in operation

The vibration damping unit of the bogie of the electric train (Figure 1) is intended to improve the dynamic performance of the crew, namely to prevent the development of resonance oscillations of wagging when traveling at critical speeds. The hydraulic damper 1 02R-2217-001 [1] (Figure 1, a) is connected to the frame of the bogie and the body bolster. The bracket of the damper 2 on the frame of the body is connected to it by bolted connections and is located on the tide at the side of the pivot beam. The tide is strengthened with kerchiefs 3 (Figure 1, b).

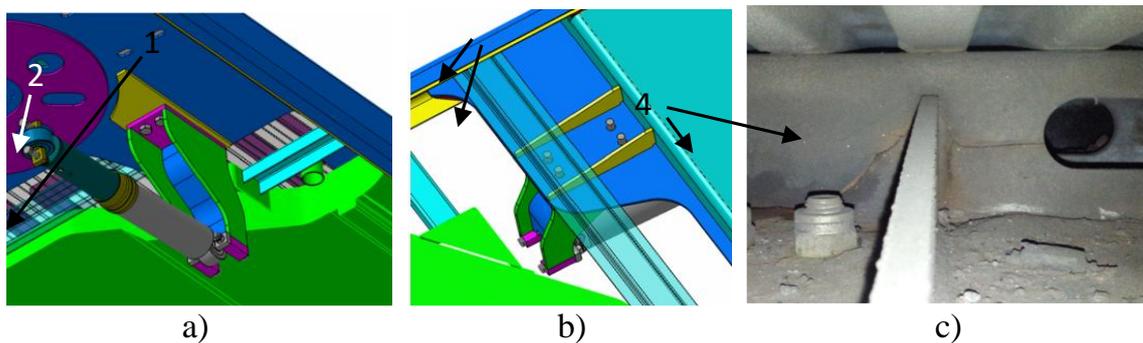


Figure 1. Unit of vibration damping device:

a) view from damper side; b) top view; c) crack of the side wall of the body bolster.

After a short (about 2 years) operation period, fatigue cracks 4 (Figure 1, c) were found [3] on the side wall of the pivot beam. The appearance of cracks, in addition to the intense dynamic loading of the node, could be due to the presence of a stress concentrator near the kerchiefs and residual welding stresses.

In accordance with the above, the first unit upgrade was proposed (Figure 2).

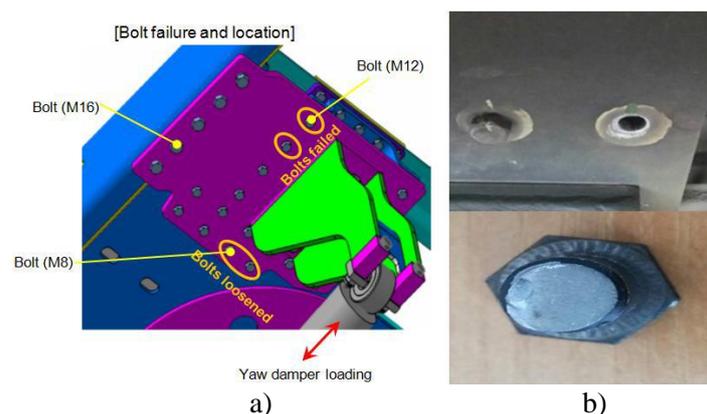


Figure 2. Unit vibration damper wobble, upgrade 1:
a) view from the damper side; b) failure of bolted connections.

This modernization (according to the results of the strength tests carried out by the DNURT [2]) has improved the dynamic loading of the unit. However, there was a problem associated with loosening the bolted joints (Figure 1.2, b) and the subsequent cutting of the bolts.

At the moment, the second upgrade of the high reliability unit has been proposed (Figure 3).

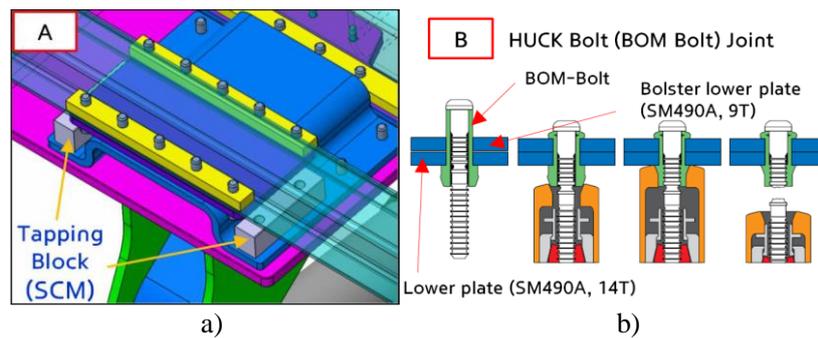


Figure 3. Unit of vibration damping oscillator, modernization 2:
a) top view; b) bolted connections.

Its meaning lies in the use of self-locking bolted joints in conjunction with the replacement of nuts with rigid centering blocks (tapping block) and bolts with a crimping ring (huck bolt).

Qualitative indicators of this modernization are the object of research of this work.

Conclusions: The vibration damping unit of the bogie turned out to be a weak place for the crew part of the EP design, which required its two-stage modernization.

2. Methods and composition of research objects strength and resource studies

The resource estimates are obtained by calculation and compared with the unit's calculated resource as indicated below.

The ratio of the resource of the investigated design to the resource of the analogue (when performing vibration tests) d is found as the inverse relation of the developments:

$$d = \frac{D_d}{D_a} \quad (1)$$

In the indicated expression: D_d is the specific (per unit of distance) operating time of the design in use; D_a – specific (per unit of time) operating time of the module – analog in the testing process. Then the resource of the structure under study will be determined as follows:

$$R_d = \frac{d \cdot T_a}{L_d \cdot R_a}, \quad (2)$$

where L_d is the average annual mileage of the structure under investigation during operation; T_a – time tests of analog; R_d – resource, expressed in terms of the lifetime of the structure under study; R_a – the resource gained during the test.

To carry out “statistical playback” a model of endurance is used, generally corresponding to the model adopted in the Norms [4]. In this case, as a cyclic load, a harmonic perturbation is adopted with the most unfavorable statistical parameters of the Gaussian distribution (m is the average value of the perturbation, σ is its standard deviation) obtained during strength tests [2].

3. Baseline data for research. Visual inspection of the presence of visible cracks

Bench vibratory tests of the damper damping unit EP [7] are divided into two stages: the first stage – cyclic loading tests from 0 to 9 000 000 cycles; the second stage is cyclic loading tests from 9 000 000 to 10 000 000 cycles.

Before the second stage of the test, a visual PT control of the presence of visible cracks was performed. The order of this and subsequent (final) control did not contradict the requirements of national regulatory requirements [6]. The control was carried out with the participation of the representative of the HRC Testing Center, Mr. Do Yoon Kim (certificate DY-451/G, valid until Nov.6 2020). The control points are shown in Figure 4.

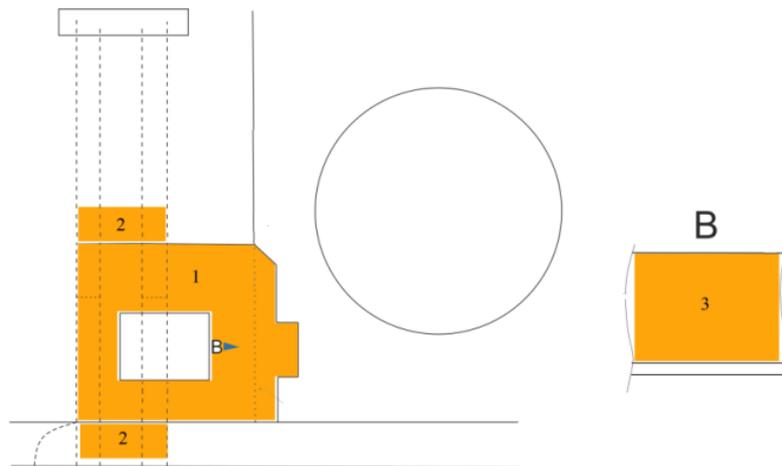


Figure 4. The places of penetrant control.

In Figure 4: zone 1 – all areas of bolted connections of the support plate of the wobbling damper support bracket; 2 – transverse beam in places near the support plate; 3 – side wall of the pivot beam in the vicinity of the support plate of the vibration damper support.

After the procedures for the preparation of control zones were completed, an intermediate visual control of the presence of cracks in the load-bearing elements of the structure was made (Figures 5, 6).



Figure 5. Marking of places and results of intermediate penetrant control zone 1.

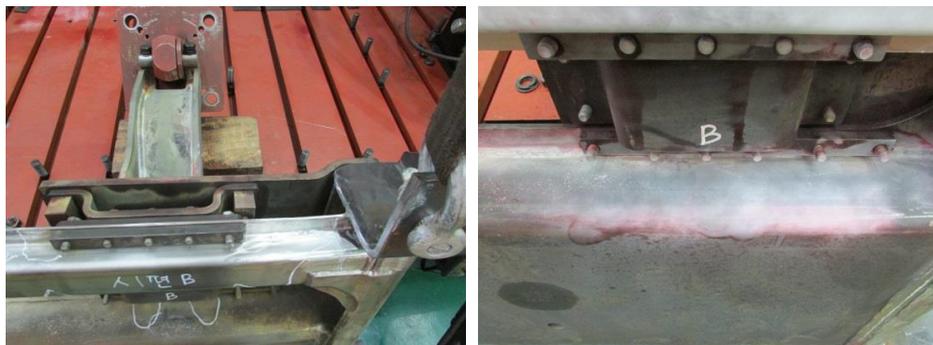


Figure 6. Marking of places and results of intermediate penetrant control of zones 2 and 3.

4. Evaluation of the operating times of the load-bearing elements of the vibration damping assembly of the bogie during strength and bench testing

During the dynamic strength tests [2] dynamic processes were registered according to the scheme of location of the sensors, shown in Figure 7. This sensor placement scheme was also used when

performing wall-vibration (resource) tests at the “Hyundai-Rotem Company” Test Center with the participation of representatives of the DNURT.

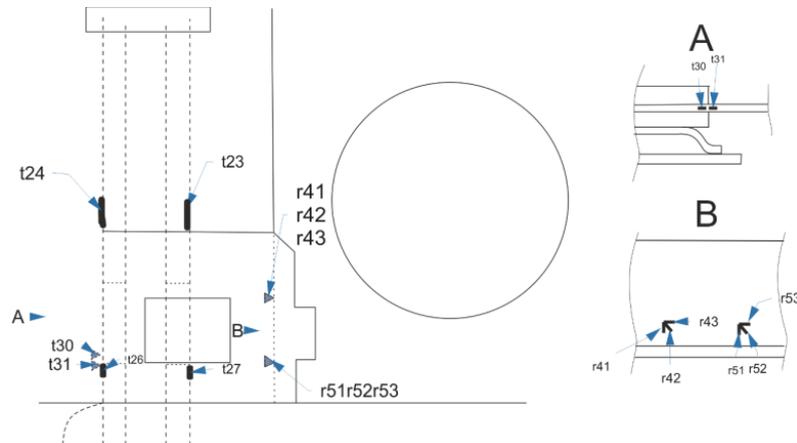


Figure 7. Scheme of the distribution of strain gauges during dynamic strength and bench vibration (resource) tests.

The most loaded element is the transverse beam in the area of connection with the support plate of the vibration damper bracket on the body frame. This makes it possible to use its stress state (sensor t30) as an indicator of the operating time for running strength and bench vibration testing.

A typical example of the dynamic loading of this point, determined during dynamic strength tests, is shown in Figure 8.

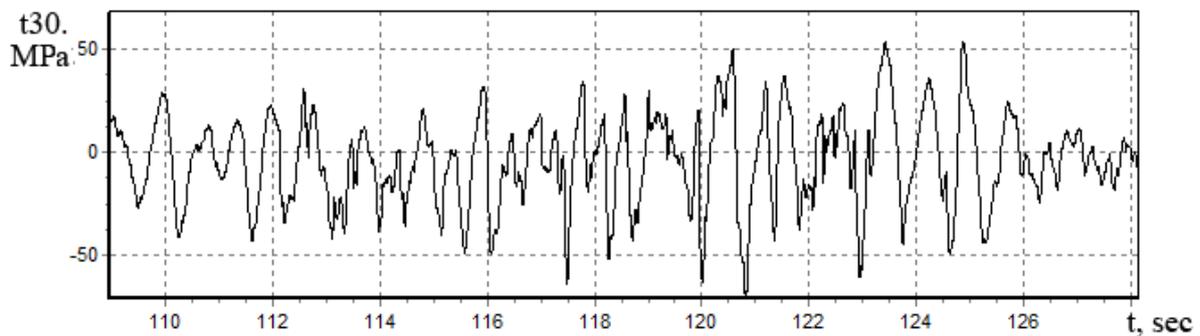


Figure 8. A typical example of the process of dynamic loading of the transverse beam of the body frame (experiment 6, entry 29).

To evaluate the level of dynamic loading, three records were selected which are characterized by the highest level of process variance: experiment 4, record 26 and 29 and experiment 6, record 29. These processes were schematized according to the rain method according to [9]. The results of determining the operating time D_d – the specific (per unit distance) operating time of the structure under investigation during the operation are given in Table 1.

Table 1. Work during realization of different experiments.

Experiment / Record	6 / 29	4 / 29	4 / 26
Work on 1 km, MPa9	1.16 ¹⁴	1.22 ¹³	6.3 ¹³

The coefficient of the power-law curve of endurance m was assumed according to [10] as $m = 16 / K_\sigma$, where $K_\sigma = 1.72$ was taken from [7] and amounted to $m = 9.3$.

5. Research results. Analysis of the results of intermediate and final visual inspection of the presence of visible cracks

After the work on an intermediate visual control of the presence of visible cracks, no contrasting penetrate residues were found in the regions of the individual bolts of the base plate, which were additionally examined (Figure 9) using a 25-500x portable optical computer microscope U500X.

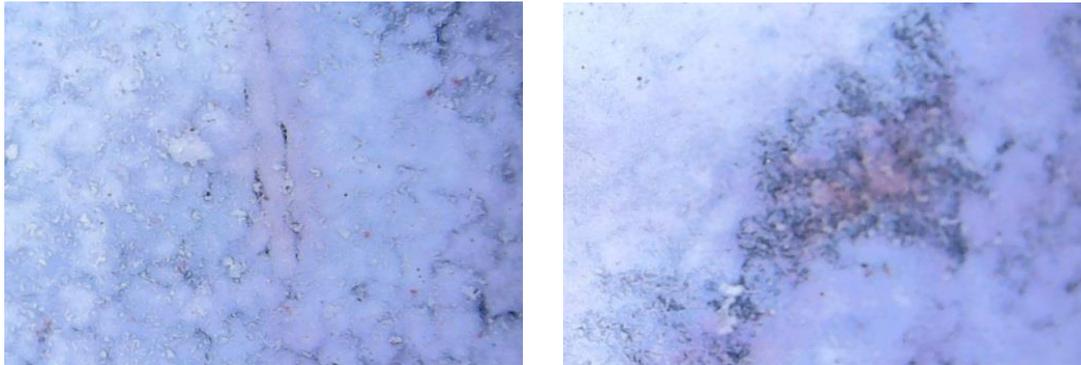


Figure 9. Additional investigation of possible locations of cracks with the use of an optical computer microscope U500X.

A study of all the selected control zones showed no visible cracks. This allowed us to proceed to the second stage of the cyclic loading of the structure without any observations. After the completion of the second test phase, the final PT control of the presence of visual cracks was repeated. After performing all the necessary operations, no cracks were found (examples – Figure 10).

So, after two stages of cyclic loading, visible cracks in the loaded structural elements are not revealed. The vibration damping unit of the wagon of the second modernization truck sustained a cyclic loading with a maximum operational amplitude of the load (9 kN) in the volume $N_0 = 10\,000\,000$ cycles. This volume of tests is taken into account for the total operating time when assessing its final service life.

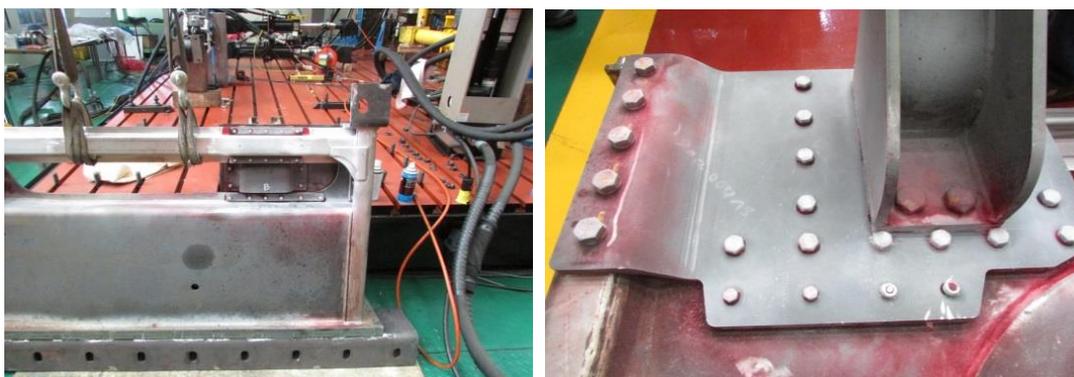


Figure 10. The results of the final penetrant control.

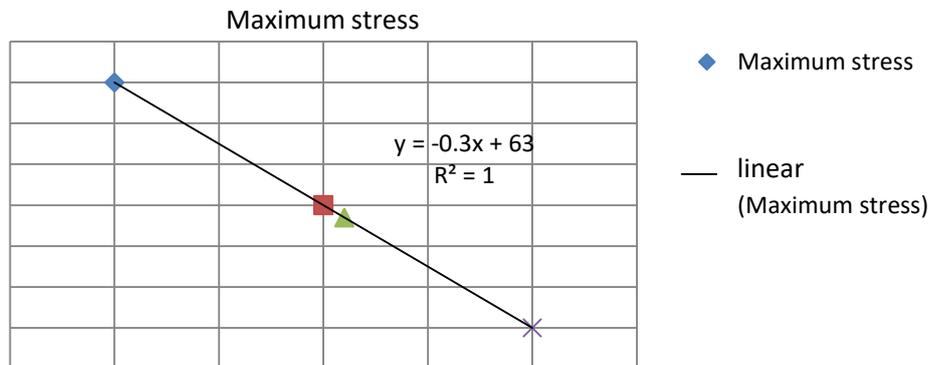
6. Evaluation of the influence of vertical dynamics on the loading of the vibration dampener assembly of oscillations of a bogie of an electric train bogie

Below, carried out special calculations of the stress-strain state of the vibration damper assembly [8], namely for cases of horizontal load application, as well as with a deviation of 10 in the positive and negative directions.

The results of the calculations are given in Table 2 and in graphical form – in Figure 11.

Table 2. Maximum stresses in the body structure of the vibration damper assembly.

Cases of load	Description	Max. Stresses, MPa
Case 1	Load of damper 9 kH with angle -10° for normal condition	66
Case 2	Load of damper 9 kH with angle $+10^\circ$ for normal condition	60
Case 3	Load of damper 9 kH with angle 0° for normal condition	63

**Figure 11.** Dependence of the maximum voltage in the construction on the angle of inclination of the damper.

Taking into account the installation of a damper with a positive inclination angle of 1 [1] and a linear dependence of the maximum stresses in the unit at small angles of the damper's inclination, we determine the coefficient of influence of the vertical dynamics on the stress level in the structure.

The value of $\Delta_{\sigma_{\max}}$ (according to the data in [5]) is 3.3 MPa. The coefficient K_{vi} of the influence of vertical dynamics is taken as:

$$K_{vi} = \frac{1 - \Delta_{\sigma_{\max}}}{\sigma_{\text{mid}}} = 0.947, \quad (3)$$

where $\sigma_{\text{mid}} = 62.7$ MPa is the average voltage (in the absence of vertical displacements of the frame of the car body with respect to the frame of the telegraph).

As can be seen from Figure 10, the vertical movement of the carriage frames relative to the body frame can both increase and decrease the maximum stresses in the structure. This depends on the ratio of the frequencies and phases of oscillation of wobble and vertical vibrations. Given the complexity of this analysis, we will assume that the frame of the bogie is constantly in an unfavorable position from the standpoint of stresses in the construction.

The effect of vertical dynamics on the increase in stresses in the unit of the vibration damping oscillator is taken into account by a constant coefficient $K_{vi} = 0.947$, which assumes a permanent finding of the frame of the bogie in an unfavorable position with respect to the body frame. This correction factor will be used in calculating the design life by artificially reducing the stress level during the bench test.

7. Evaluation of the end-life of the load-bearing elements of the vibration damping unit of the bogie

The method of direct comparison of the developments [11] is based on the direct use of the power endurance curve and on the fact that the total operating time C of the $\sigma^m N = C$ form [14] is constant under the condition of linear accumulation of damages in the structure.

When carrying out bench vibration tests, the damper assembly was loaded with the maximum operating force from the vibration damping side ($F = 9$ kN) with a frequency $f = 5$ Hz. The voltage at point t_{30} (Figure 7) during the bench vibration testing was [7] $\sigma_{t_{30}} = 65.5$ MPa. At the same time, the total operating time, obtained by the design, is:

$$D_a \cdot T_a = \frac{D_a \cdot N_0}{f} = \frac{(\sigma_{t30} K_{vi})^m N_0}{f}, \quad (4)$$

where: D_a – working out of the power form (see previous paragraph) per unit time; T_a – time of testing; N_0 – according to the conclusions of previous paragraph; f – according to previous paragraph; σ_{t30} – according to [7]; K_{vi} – according to the formula (3); m – according in accordance with the discussion of test results.

The results of estimating the site resource by comparing the developments are given in Table 3. The design of the wobble damper module (the second upgrade of enhanced reliability) has a significant $n_R = 210$ run time relative to the design life of 50 years.

The main idea of the method of "statistical playback" is the mathematical simulation of the process of accumulation of damages. At the same time, the input of the mathematical model receives a harmonic signal of the fundamental oscillation frequency f_e with the normal distribution of the amplitudes, while the statistical characteristics of the distribution coincide with the operational loading.

When using steel similar to SM 490A for a crossbeam, we will assume that its endurance characteristics have the same proportion [15] compared with SUS301L steel as their yield strength, i.e. 0.793. Conditionally, we will reduce the operating time when testing 0.7939 times. For this case, table 3 takes the form shown below.

Table 3. Evaluation of site resource by direct comparison of developments.

Experiment No.– Record No.	6-29
Operating time per 1 km D_a , MPa ⁹ km ⁻¹	1.16·10 ¹⁴
Operating time, tests $D_a \cdot T_a$, MPa ⁹	4.38·10 ²³
Ratio of operating time $D_a \cdot T_a / D_d$, (residual mileage, thousand km.)	3778080
Yearly mileage L_d , residual mileage, thousand km.	360
Resource $R_1 (R_d)$, years	10494
Life length 50 years D_{d50} , MPa ⁹	2.085·10 ²¹
Run-up reserve $n_R = (D_a \cdot T_a) D_{d50}$	210
Reserve reduced to linear dependence on stresses $n_\sigma = n_R^{1/m}$	1.78
Experiment No. – No. of recording	6-29
Operating time, tests $D_a \cdot T_a$, MPa ⁹	4.38·10 ²³
Ratio of operating time $D_a \cdot T_a / D_d$, (residual mileage, thousand km.)	468522
Resource $R_1 (R_d)$, years	1300
Life length 50 years D_{d50} , MPa ⁹	2.085·10 ²¹
Run-up reserve $n_R = (D_a \cdot T_a) D_{d50}$	26
Reserve reduced to linear dependence on stresses $n_\sigma = n_R^{1/m}$	1.44

Thus, when using steel SM 490A, the design can be operated for 50 years.

The "rain" treatment [9] of the chosen most stress-laden implementation of the process of dynamic stresses at the point t_{30} gave the distribution of their amplitudes and probabilities, shown in Table 4.

The average amplitude of the dynamic stresses [4] is $M_{t30} = 12.68$ MPa, the standard deviation is $s_{t30} = 11.31$ MPa. Thus, the calculated amplitude according to the Norms [4] is $M_{t30} + 2s_{t30} = 35.3$ MPa

To determine the dominant oscillation frequency, a spectral analysis of the loading process of the damper assembly was carried out, and the spectrum of the oscillations is shown in Figure 12. The dominant frequency of the oscillations in this case was $f_e = 1.4$ Hz.

Table 4. Distribution of amplitudes of dynamic stresses.

Amplitude of the discharge of strain, MPa	Probability
5.0	0.503086
10.0	0.159122
15.0	0.108368
20.0	0.065158
25.0	0.049383
30.0	0.038409
35.0	0.024691
40.0	0.021262
45.0	0.011660
50.0	0.007545
55.0	0.006859
60.0	0.001372
65.0	0.002058
70.0	0.000686

The design response to a single overload is defined as:

$$\sigma_{-1}^+ = \sqrt[m]{\sigma_{-1}^m - \frac{\sigma_v^m}{N_0}} \uparrow \sigma_v > \sigma_{-1},, \quad (5)$$

where σ_{-1}^+ – the endurance limit of the structure after the action of a single overload; σ_{-1} – the endurance of the structure to the action of a single overload; σ_v – current amplitude of the dynamic voltage cycle; m – an indicator of the degree of the endurance curve; N_0 is the test base, on the basis of which the initial endurance limit of the structure is estimated.

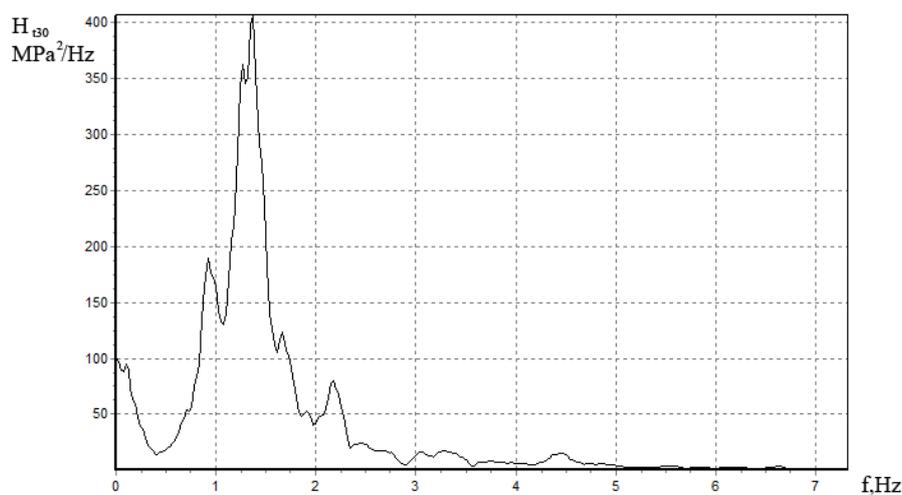


Figure 12. The spectrum of the dynamic loading process.

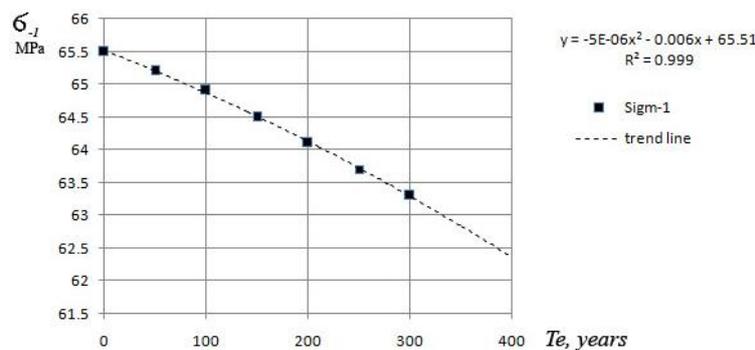
The initial data for performing "statistical playback" is given in Table 5.

Table 5. Initial data for estimating the final resource by the "statistical playback" method.

N ₀	Indicator name	Value
1	Estimated amplitude of dynamic stresses, MPa	35.3
2	Assessment of endurance limit on the basis of tests N_0 , MPa	65.5
3	Base of tests N_0	10^7
4	Endurance factor N_0	1.85
5	Estimated average annual mileage, thousand km	360
6	Average speed, km/h*	100
7	Estimated operating time per year, thousand h	3.6
8	Dominant frequency of oscillation f_e , Hz	1.4

*Note: the average speed estimation was obtained for InterCity + trains by analyzing the data of the official Internet resources <http://intercity.uz.gov.ua/> and <http://booking.uz.gov.ua/ru/>

The results of estimating the resource of the estimated "statistical playback" are shown in Figure 13 and in Table 6. It is believed that the construction begins to break down when the endurance limit is reduced to $\sigma_{-1}^+ = 0$.

**Figure 13.** Dependence of lowering the endurance limit on life.**Table 6.** Estimate the site resource by the method of "statistical playback".

N ₀	Indicator name	Value
1	Initial value of the endurance limit, MPa	65.5
2	Estimated service life for 50 years, thousand h	180
4	Maximum fixed endurance limit, MPa	110.3
5	Duration to the first exceedance of the current endurance limit, h	0.3
6	The value of the endurance limit after the estimated service life, MPa	65.2
7	Decrease in endurance limit after estimated service life, %	0.45

Thus, the calculation of the resource by the method of "statistical playback" confirms the possibility of operating the vibration damping module (second modernization) for a period of 50 years from the construction.

It is necessary to pay special attention to the fact that all the results given are obtained for normal operating conditions of EP and can not be extended to cases of emergency situations and other incidents leading to a critical disruption of the strength of the elements of its crew part.

Main conclusions

The complex of the conducted researches allows to draw the following conclusions:

1. After carrying out two stages stand cyclic loading of the bogie wagging oscillation damper (the second modernization), visible cracks in the structure loaded elements are not revealed.

2. The influence of vertical dynamics on the increase of stresses in the unit of the vibration damper is about 5 percent when the frame of the ET carriage is constantly in an unfavorable position relatively to the main frame of carriage ET.
3. Calculation of the finite value of the modernized vibration damping module resource (second modernization with increased reliability) by two methods confirms the possibility of its exploitation during 50 years from building. This conclusion does not depend of cross-beams material (SUS301L or SM 490A steel).
4. The obtained results can not be extended to cases of emergency situations and other incidents leading to a critical disruption in strength of ET crew part elements.

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