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Strength and life estimation for assigning the useful life of wheelpairs of high-speed flat wagons

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Abstract. Aim. *The most vital unit of railway rolling stock is a wheelpair, as a broken wheel or axle may have catastrophic consequences. Therefore, before the production of a high-speed flat wagon designed for operation at speeds of up to 140 km/h, which is unique for the 1520 mm gauge space, could commence, it was required to research the applicability of the standard wheelpair for high-speed movement. Ensuring the safe operation of a wheelpair involves compliance with the requirements that are to be confirmed by means of assessment of strength and durability parameters [1]. Product conformity assessment may be based on the requirements of standards, whose voluntary fulfilment ensures compliance with [1], or other documents.* **Methods.** *The paper describes the computational and experimental methods used for confirming the strength and estimating the life (durability) of wheelpair elements in the probabilistic setting. As experimental data, the authors used the results of full-scale bench testing of wheelpairs for fatigue using the method of rotational bending as it best approximates the loading conditions in operation. The results confirmed the endurance limits of the axle and wheel as parts of an assembled wheelpair. Using design analysis, the authors examined the stress-strain state of the wheelpair caused by installation and operational loads in various running modes.* **Results.** *The conducted studies confirmed the wheelpair's compliance with the requirements of [1–3] in terms of safety factors of fatigue strength and endurance, which eliminates the possibility of hazardous situations in the course of high-speed flat wagon operation. The time to fatigue crack nucleation in wheelpair components was evaluated using the fatigue resistance figures of the parts and equivalent amplitudes of dynamic stress caused by operational loads. It appears that this assessment allows establishing – with the assumed probability of destruction – the assigned useful life of a wheelpair axle at 32 years, which corresponds to the assigned useful life of the flat wagon according to the combined criterion. Corresponding standards and regulations required for developing the container-carrying flat wagon are being updated and a new State Standard is being developed.* **Conclusion.** *The conducted conformity assessment established that the flat wagon wheelpair meets the safety requirements of [1] and ensures the absence of unacceptable risks associated with harm to life and health of people, animals and plants, the environment and property of individuals and companies in the course of flat wagon operation.*

Keywords: *wheelpair, stress-strain state, fatigue resistance, endurance limit, S-N curve, life, assigned useful life.*

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Introduction

Today, a six-axle container-carrying 80-foot flat wagon with three-axle freight bogies with the operational speed of up to 140 km/h and axle load of up to 20.5 tons is under development jointly with Federal Freight for the purpose of gross weight freight container transportation. The flat wagon as part of a fixed train set is unique within the 1520-mm gauge space in terms of its design features and permissible speeds of operation. The platform is suitable for operation both as part of trains consisting of similar wagons, and as single flat wagons as part of a freight train throughout the 1520-mm gauge railway network (Russian Federation, CIS countries, Finland, Poland, Ukraine, Latvia, Lithuania, Estonia, Georgia).

In accordance with Annex 2 of [1], the flat wagon is subject to mandatory confirmation of conformity (in the form of certification). In accordance with Article 6, Item 5 of [1], product conformity assessment [1] may be based on the requirements of standards, whose voluntarily fulfilment ensures compliance with [1], or other documents.

The most vital component of the flat wagon is a wheelpair that – as a unit and its components (axle and wheel) – is subject to mandatory certification [1, 2]. The above regulatory documents specify safety requirements that are to be confirmed by assessing the strength and durability (life) of the axles and wheels of wheelpairs. Thus, Article 4, Item 57 of [1] contains the following requirement for rolling stock: "wheels, axles ... of wheelpairs of railway rolling stock ... of freight wagons shall have a safety margin of static strength and a required fatigue resistance margin that guarantee resistance to the formation and development of defects (cracks) during the useful life per the design documentation."

[3] is the supporting standard for the purposes of strength assessment of axles and wheels of wheelpairs (of freight and passenger cars) and non-motored wheelpairs of electric multiple units (including high-speed).

Strength assessment

Items 4.3.11 and 4.3.12 of [3] contain requirements for the fatigue resistance, static strength and fatigue endurance coefficients of the axle and wheel as part of a wheelpair for wheelpairs not specified in Annex A. According to Annex A of [3], mass-produced wheelpairs have a maximum design speed of 120 km/h. Therefore, a wheelpair with the design speed of 140 km/h (Fig. 1) is to prove compliance with the requirements of Items 4.3.11 and 4.3.12 of [3].

Estimating the fatigue strength assurance coefficients of wheelpair components requires an experimental confirmation of the fatigue endurance obtained as the result of fatigue tests and S-N curve construction.

The results of benchmark fatigue tests of axles and wheels by the method of circular bending as the most similar to the loading conditions of operation [4] previously conducted by JSC VNIKTI experts were used as experimental data.

Following the wheelpair manufacturing process, three axle test objects and three wheel test objects (modified axles with temporary hubs and wheels with modified axles) were installed in the testbench (Fig. 2). Resistive strain sensors were mounted in line in the longitudinal and tangential directions as sockets on the most heavily loaded side of the wheel. Then the measurement circuits were mounted and the stress amplitudes (deformations) were measured using a measuring and computing system. The test object was loaded by rotating an unbalanced mass installed at the end of its axle . The bending moment created stress amplitudes (deformations) within the test object. The required values of the stress amplitudes were established by changing the rate of rotation of the unbalanced mass.

The fatigue test results confirmed an actual fatigue endurance with a 50% probability of failure of the most heavily loaded part of the axle behind the wheel seat equal to 180 MPa (against the required 160 MPa [2]) and the actual fatigue endurance of the all-rolled wheel equal to 225 MPa (against the required 150 MPa [2]) on a standardized test basis of 50 and 20 mil load cycles [3].

The estimation of the strain-stress state of the wheelpair showed that the maximum stress amplitudes that occur in the most heavily loaded part of the wheel in operation (Fig. 3) amount to 97 MPa. Based on wheel test results and computed coefficients that take into account the dependence of the fatigue strength on the total average cycle stress in operation and on bench tests, the fatigue resistance margin is defined that amounts to 2.54 against the minimal allowed 1.5.

The axle calculation showed that the most heavily loaded part of the axle is behind the wheel seat where the

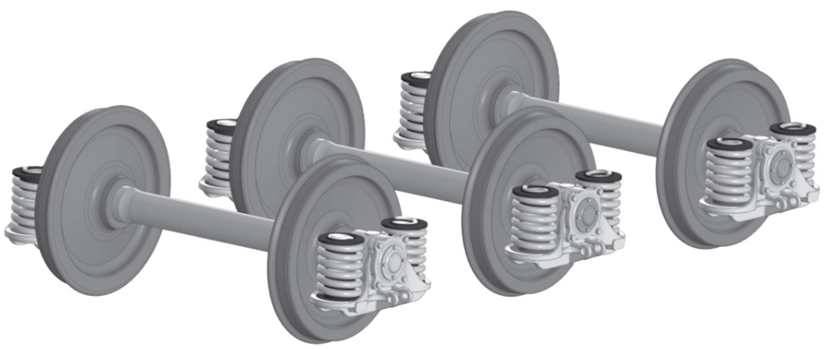


Fig. 1. Wheelpairs of a three-axle bogie of a high-speed flat wagon

a) b)

Fig. 2. Testbench with installed test objects, axle (a) and wheel (b)

maximum amplitude of the mechanical stresses in operation is 144 MPa, taking into account the fatigue endurance identified as the result of the tests, the safety coefficient is 1.25 with the minimum allowable value of 1.2.

Thus, the above calculations and experimental evaluation has confirmed the wheelpair's compliance with the requirements of [1,3] in terms of the fatigue resistance figures and endurance limits.

Life evaluation

According to Article 5, Item 3 of [1]: "The safety of railway rolling stock and its components shall be ensured by: ... c) establishing the assigned useful life of products, as well as conducting maintenance and repairs with the required frequency; ... f) establishing the criteria for identifying limit states of the product."

According to Article 4, Item 7, "the design of the railway rolling stock and its components chosen by the designer (developer) shall be safe over the course of the assigned useful life, the assigned storing life, as well as withstand the effects and loads that they may be subjected to in the course of operation."

Therefore, the safety of the axles and wheels of wheelpairs shall be ensured over the course of the assigned useful life that is to be specified on the basis of their limit state and life taking into account a certain safety margin.

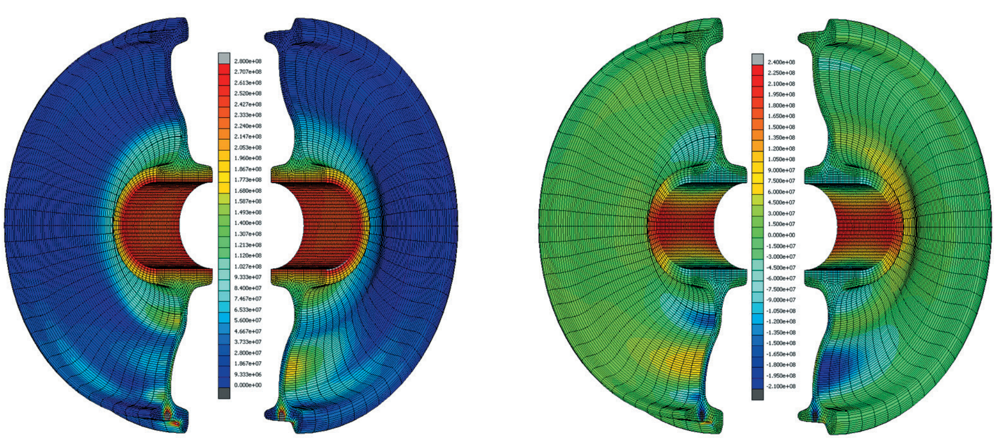
Since rolling stock is used throughout the railway network (therefore, the operating conditions – temperature, current track condition, terrain, etc. – can differ significantly, which affects the load intensity), the life of the axle and wheel in terms of fatigue damage accumulation is to be specified subject to the totality of differences in the operation of a specific type of rolling stock. Axle or wheel failure may have detrimental consequences, therefore crack nucleation is taken as the limit state criterion (the process of its formation is not controlled).

Estimating the life of the axle and wheel of a wheelpair requires identifying the following:

– strength characteristics of the parts in probabilistic terms;

– operational loading.

Based on the obtained data, the life is defined with the required probabilistic parameters. This life assessment



a) b)

Fig. 3. Distribution of equivalent (a) and radial stresses (b) with the wheel tread worn to the limit from movement in curves

of the axle and wheel of a wheelpair was performed using the analytical and experimental method.

The analytical solution [5] implies calculations that involve the part's fatigue strength characteristics (based on the results of fatigue bench tests) and the amplitude (equivalent) of the dynamic stress caused by the operational loads. For this purpose, out of the S-N curve (the second sloped branch) we find the number of the loading cycles of the part before the exhaustion of the load-carrying capacity (failure) N_f:

$$\sigma_{-1p}^{m_2} \cdot N_0 = \sigma_{\text{ea}}^{m_2} \cdot N_p = \text{const} \quad (1)$$

where σ_{-1p} is the fatigue endurance of the part based on N_0 test cycles;

σ_{ea} is the equivalent amplitude of stress;

m_2 is the slope of the curve.

The calculation used an S-N curve with two sloped branches: for numbers below 10^7 cycles, the slope of the left-hand branch of the curve $m_1 = 7$, for numbers above 10^7 cycles, i.e., for the right-hand branch, $m_2 = 2m_1 - 1 = 13$.

The resulting curve allows determining the median value of durability that corresponds to the 50% probability of failure. For the purpose of calculating the failure with the probability of (P)%, according to normal distribution tables, the corresponding quantiles are determined and the equation is solved:

$$U_p = \frac{1-\tilde{n}}{\sqrt{\tilde{n}^2\nu_{\sigma_{-1\partial}}^2 + \nu_{\sigma_a}^2}}, \quad (2)$$

out of where the relative safety factor $\tilde{n}$ is determined and, based on the latter, the overload factor n_p is defined:

$$n_p = \tilde{n} \cdot n, \quad (3)$$

where $n = \dfrac{\sigma_{a_{\max}}}{\sigma_{-1\partial}}$ is the actual loading factor;

$\nu_{\sigma_{-1\partial}}$, ν_{σ_a} are the variation coefficient of normally distributed values of the fatigue endurance σ_{-1p} and maximum stress $\sigma_{a_{\max}}$ in operation, respectively.

The variation coefficients characterize the scattering of the corresponding values: the lower are the coefficients, the more stable are the results in terms of design and manufacturing processes and operating condition.

Thus, the fatigue endurance is recalculated for the (P)% probability of failure according to the formula:

$$\sigma_{-1\partial} = \overline{\sigma}_{-1\partial}(1 - U_p \cdot \nu_{\sigma_{-1\partial}}). \quad (4)$$

Under the accepted variation coefficients of normally distributed values of the axle fatigue endurance $\nu_{\sigma_{-1\partial}} = 0.1$, maximum operating stress $\nu_{\sigma_a} = 0.2$, probability of failure $P = 0.1\%$, quantile $U_p = -3.09$, according to formula (1) we deduce $N_p = 8.2 \times 10^9$ loading cycles.

According to dependence $n_N = \varphi\,(n_\sigma)$ between the safety factors in terms of stress and durability, we deduce $N_{\text{sum}} = 2.9 \times 10^9$ loading cycles to the exhaustion of the load-bearing capacity. Upon recalculation into running kilometres, the life of an axle will be $8.7 \cdot 10^6$. Based on the established average annual run of a high-speed flat wagon of 250 ths km, the estimated operating life with the adopted probability of failure of 0.1% will be 34.7 years.

The wheels as part of a wheelpair are replaceable components, and, in operation, wheels are rejected when they reach the limit state in terms of geometry, which occurs between 300 and 800 ths km. The estimation of a wheel's life has shown that life in terms of the fatigue of the disc part is significantly higher than the run in service, meaning that the wheel will not fail.

The calculation assumed that the stress-related characteristics of the material do not change over the course of the useful life, are not subject to corrosion and other hostile environments. Given the limited scope of experimental data, as well as the probability of its growing scattering due to the impossibility of taking into account all the affecting factors for the purpose of ensuring the dependability and safety of axle and wheel operation, it appears possible to establish the assigned useful life of an axle at 32 years, which corresponds to the assigned useful life of a flat wagon according to the combined criterion.

Thus, the above calculation determined the durability of elements of a wheelpair as a vital part of rolling stock.

Conclusions

The above calculations and experimental strength evaluation have confirmed the wheelpair's compliance with the requirements of [1, 3] in terms of the fatigue resistance figures and endurance limits. The paper presents the results of calculation and experimental evaluation of the life of the axle and wheel of a wheelpair of a high-speed flat wagon currently under development. The evaluation is based on an S-N curve in the probabilistic setting. The assigned useful life of the axle was defined at 32 years.

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The authors' contribution

Volokhov G.M. analysed regulatory documents as regards rolling stock safety confirmation.

Oganian E.S. analysed literature on the estimation of the life of railway rolling stock components.

Gajimetov G.I. defined the problem, analysed and selected the methodological approaches to testing, analysed the results and prepared the conclusions.

Knyazev D.A. carried out benchmark fatigue tests of axles and wheels of wheelpairs of a high-speed flat wagon.

Chunin V.V. assessed the stress-strain behaviour and carried out strength calculations.

Timakov M.V. carried out life assessment and defined the assigned useful life of a wheelpair axle.

Conflict of interests

The authors declare the absence of a conflict of interests.