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Static and Dynamic Experimental Stress – Strain Analysis of Axles Loaded by Rotation Bending

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Abstract. Safety assessment of railway axles is based on fatigue strength design. During the design overall lifetime, railway axles undergo a very high number of loading cycles under rotating bending. Due to safety and reliability reasons, fatigue strength design has to be very carefully evaluated. Both theoretical modeling and extensive experimental fatigue testing including full scale tests have to be performed, whereas such tests have to be performed correctly with a high precision. In the paper, problems of dynamic forces during fatigue tests of railway axles are pointed out. Though static calibration can be carried out with a very high precision, dynamic forces can cause a significant redistribution of static stresses along the axle. An example is given and discussed.

Introduction

Safe operation of railway vehicles has been one of the most important issues of railway transport since first serious crashes with tragic consequences [1]. The importance of safe axle operation problems even rapidly increases together with recent increasing speed of passenger trains, weight saving design requirements and necessity to consider damage tolerance principles in axle designs [2].

Railway axles are usually press fitted or shrink fitted in wheels and so two main characteristics of resistance against fatigue damage have to be evaluated and proved: (i) sufficient fatigue resistance of the smooth part of the axle on its free surface and (ii) fatigue resistance under press fit (or shrink fit). Railway axle design and certification standards lay down minimum values of fatigue strength for the material used, evaluated not only on small rotating bending specimens, but also on full scale axles. Such tests are quite expensive, including the specimens – actual axles. It is therefore necessary to carry out the full scale tests with a high precision.

Accredited full scale axle fatigue tests are essential and unconditionally requested steps of certification process of axles to be used for manufacture of railway vehicles. These tests must be performed with a high precision of fatigue loading, usually better than 2 %. Moreover, the loading at a critical position of axle surface has to be uniform along the whole circumference. Such requirements can only be satisfied, if the tests are performed using very sophisticated modern facilities like new generation of Sincotec machines with very special, patented round basis. In addition, careful and exact static measurement and machine calibration has to be carried out before the test starts.

Previous generation of rotating bending full scale fatigue tests facilities used quite massive eccentric parts to introduce the necessary loading. As a result of the high weight of the eccentric masses, the test frequency was quite low, usually bellow 15 – 17 Hz. The disadvantage of such tests, being performed with target number of cycles 10^7, was long testing time, almost 200 hours. Recent test facilities have been using much less massive eccentrics and consequently, test frequencies are around 30 Hz or even more. However, another disadvantageous phenomenon became an issue: additional centrifugal forces.

The paper contains results and discussion of several aspects of a typical fatigue test programme of axles. Actual distribution of stresses near axle press fit in comparison with nominal stresses on one hand and effects of centrifugal forces one the other hand are particularly analysed and pointed out.

Experiments

A full scale model of a 230 mm diameter axle of approximately 2 m length, made from C45 steel, shrink fitted in a hub, was fixed by the hub to the base of the Sincotec rotating bending machine. Beyond standard calibration procedure recommended by the machine manual, quite comprehensive experimental stress-strain analysis programme, both static and dynamic, was carried out.

A high number of strain gauges (SGs) were bonded in the axial direction on the axle surface. The first area was close to the hub edge, at distances 5 mm, 25 mm, 41 mm and 55 mm (Fig.1). Further measurement was performed at distances 250 mm, 500 mm and 750 mm from the hub edge, always at several circumferential positions (Fig.2). Static loading was applied near the top end of the axle at the distance of 1715 mm from the hub edge, in the lateral direction. Static loading was exactly measured by load cell. During static loading, values in both tension and compression sides were monitored.

Dynamic rotating bending load during fatigue test was controlled by means of SGs located at the distance 250 mm from the hub, as recommended. However, in addition, dynamic values of all SGs were independently monitored by a HBM Spider 8 device. Note that the load frequency was considerably high, approximately 30 Hz.

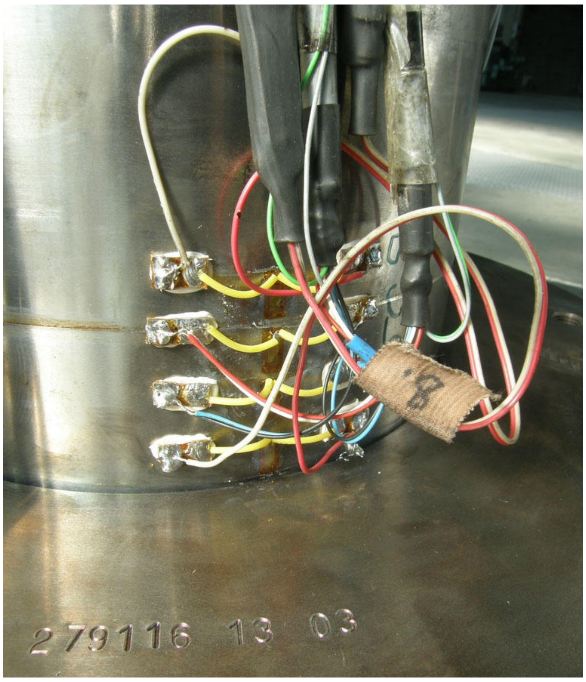

Fig. 1. Chain of strain gagues near hub edge.

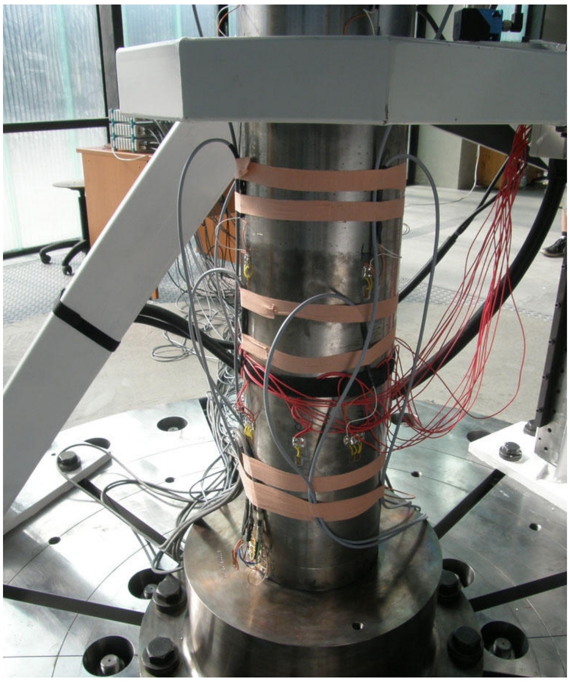

Fig. 2. Axle with all strain gauges at different disrance from hub.

Experimental Results and Discussion

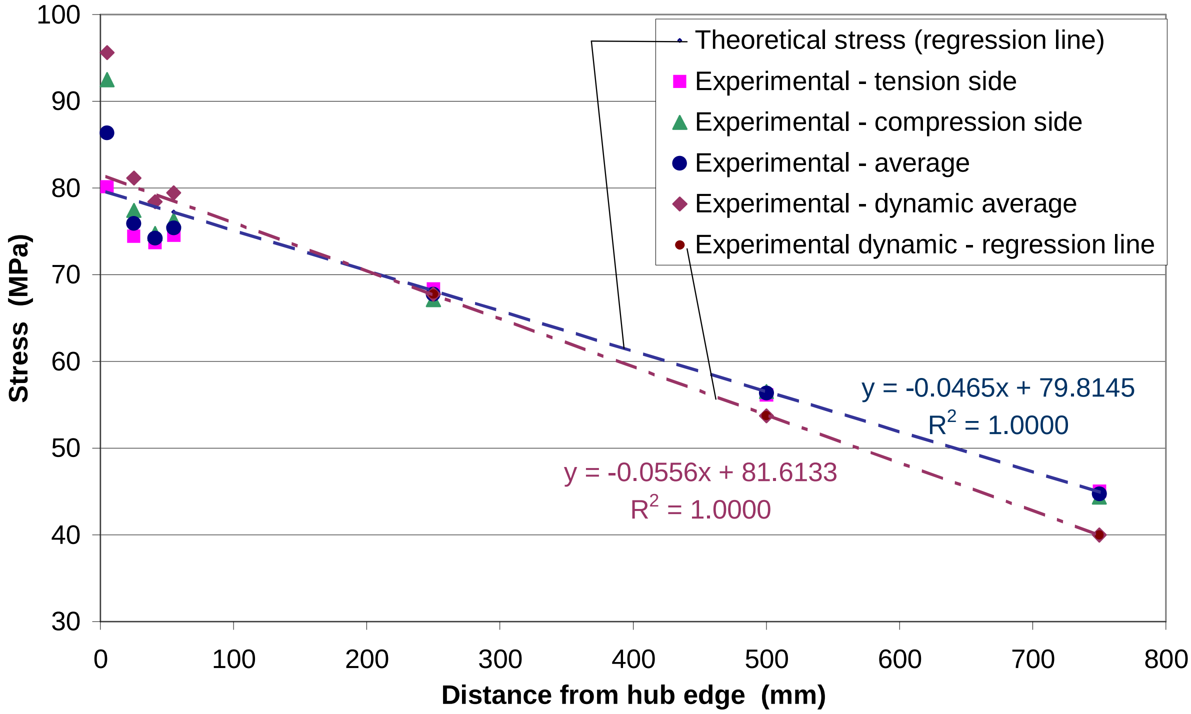

Fig. 3. Experimentally evaluated stress values on the axle surface during static and dynamic bending in comparison with theoretical values.

Results of the experimental stress analysis are in Fig. 3. For the conversion of measured strain values to stresses, E-modulus 206 GPa, evaluated separately by standard static tensile tests of small specimens, was considered.

Disregarding the SG chain near the hub, where some strain redistribution was expected, which will not be discussed in this contribution, there is an excellent agreement between theoretical and experimental stresses at the distances 250 mm, 500 mm and 750 mm. Note that the calibrating static force at the point 1715 mm from the hub was 55.72 kN. On the other hand, dynamic stresses measured during rotating bending of the axle with frequency ca. 30 Hz, when dynamic bending force was introduced by a rotating eccentric mass at the axle top end, differ from static stresses significantly with the exception of the distance 250 mm from the hub, there were attached Sincotec machine controlling SGs. The slope of the regression lines is significantly different. The phenomenon is explained by an occurrence of additional dynamic centrifugal forces.

Deformation of the axle during static calibration corresponded more or less to a beam fixed at its bottom end and loaded near the top end by lateral force acted in a single point. Calculation of lateral displacement at the point of load is not difficult, using simple relevant formulae (e.g. [3,4]). For the above mentioned load and distance, this displacement is 3.4 mm. Since the machine eccentric mass is not big to reach high test frequency, ca. 30 Hz, such quite small distance plays an important role, as centrifugal force is given by product m*a, where m is mass and a centrifugal acceleration ω^2*r, r being rotation radius, close to 3.4 mm in the specific case. m is mass of the specific axle segment being considered, contributing to a kind of axle self loading. If as a simplified example, axle segment of the length 0.5 m is considered (top quarter of the axle), its self-loading centrifugal force would correspond to lateral load 19 kN, which is 34 % of the lateral force during static calibration. Note that in previous type of Sincotec machine with much robust eccentric mass, test frequency only was ca. 15 Hz and the corresponding centrifugal force only 4.8 kN, just 8.6 % of the static calibration force. That is why centrifugal forces could be almost neglected with the previous type of the machine.

An exact calculation of the axle loading by centrifugal forces can be made by integration of centrifugal forces function along the whole axle. The "self-loading" function is continuous and smooth, being highest near the axle top and zero at the hub edge. This axle "self-loading" effect results in the fact that the axle is eventually more linear in comparison with static calibration force, whereas bending is concentrated just to the area near the hub. The effect is not negligible and has to be considered during testing and subsequent use of the results.

Conclusions

The most important results of the study can be summarized as follows:

- Static experimental stress analysis corresponded almost exactly to theoretical values at distances remote from the hub edge. There was a considerable stress redistribution near the hub edge, as expected.
- Dynamic stresses corresponded to static stresses just at the distance of 250 mm from the hub edge, i.e. at the position of controlling SGs. However, the slope of the dynamic stress dependence on the distance from the hub significantly differed from the slope of the static loading, by approximately 20 %. The main source of the difference was a contribution of centrifugal forces of the axle mass itself to excitation forces introduced by the machine eccentric mass at quite high load frequency – 30 Hz.
- Dynamic stresses measured along the whole axle circumference were almost ideally self consistent, within precision better than 2 %.

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