NÁVRHOVÉ VÝPOČTY STROJE PRO TESTOVÁNÍ ÚNAVY ŽELEZNIČNÍCH NÁPRAV

PROPOSAL CALCULATION FOR A MACHINE FOR FATIGUE TESTING OF RAILWAY AXLES

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Abstrakt

Byly provedeny návrhové výpočty nového zkušebního stroje. Tento nový zkušební stroj je určen pro testování únavy materiálu v železničních nápravách. Na železniční nápravy působí ohyb a rotace (účinky odstředivé síly). Pro celý projekt velkého zkušebního stroje je také důležité znát základní dynamické charakteristiky celého systému.

Klíčová slova: únava materiálu, zkušební stroj, železniční nápravy, dynamické charakteristiky

Abstract

There was done some proposal calculations for a new testing machine. This new testing machine is determined for a fatigue testing of railway axles. The railway axles are subjected to bending and rotation (centrifugal effects). For the project of massive testing machine is also important to know the basic dynamic charakteristics of whole system.

Keywords: fatigue, testing machine, railvay axles, dynamic characteristics

INTRODUCTION

The detailed study of fatigue of railway axles started more than 150 years ago by german engineer August Wohler, see also [1] etc. Today′s european standards define material quality of railway axles including requirements for chemical composition, material behaviour, stress-strain calculations in individual points of axle -cross-sections, fatigue testing and its evaluation, see standards [2], [3] and [4].

The determination of a fatigue limit for material loaded by composed bending and rotation is described in [3]. The fatigue test for railway axles which are made in actual size are very important for verifications of all calculations.

At the present days, in the Czech Republic only SVÚM in Prague provides fatigue tests using Sinco-TEC rezonator. Hence a new kind of rezonator is designed, see Fig.1. The shaft exciter is described via centrifugal force Fo /N/.

The whole fatigue test of railway axles is controled via computer and strain gauges. The fatigue test is finished when the stiffness of system is decreased (the testing frequence is decreased about 0.5%). It signs the situation when a crack is iniciated.

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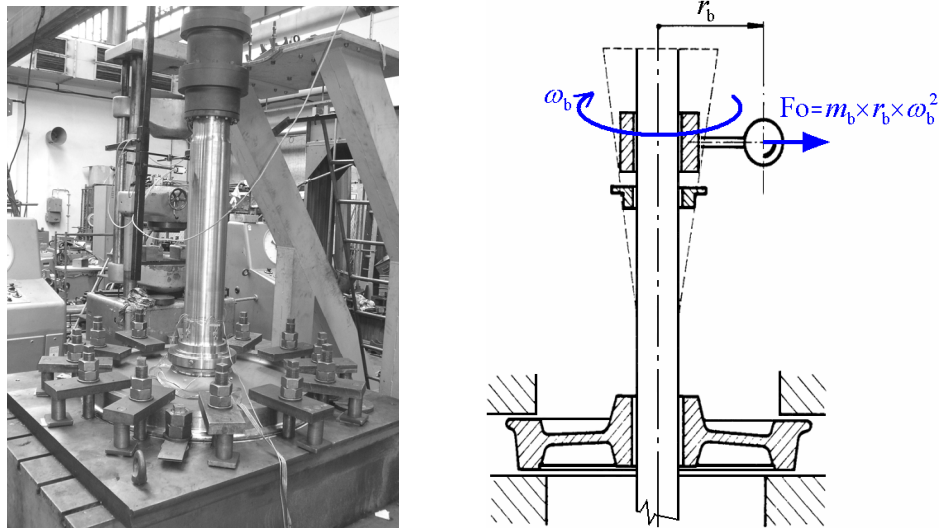[2] MSc. Rostislav FAJKOŠ, Bonatrans a.s., Ostrava, rfajkos@bonatrans.cz

Fig.1 Model and Principle of Testing Machine Based on Loading by Centifugal Forces Fo
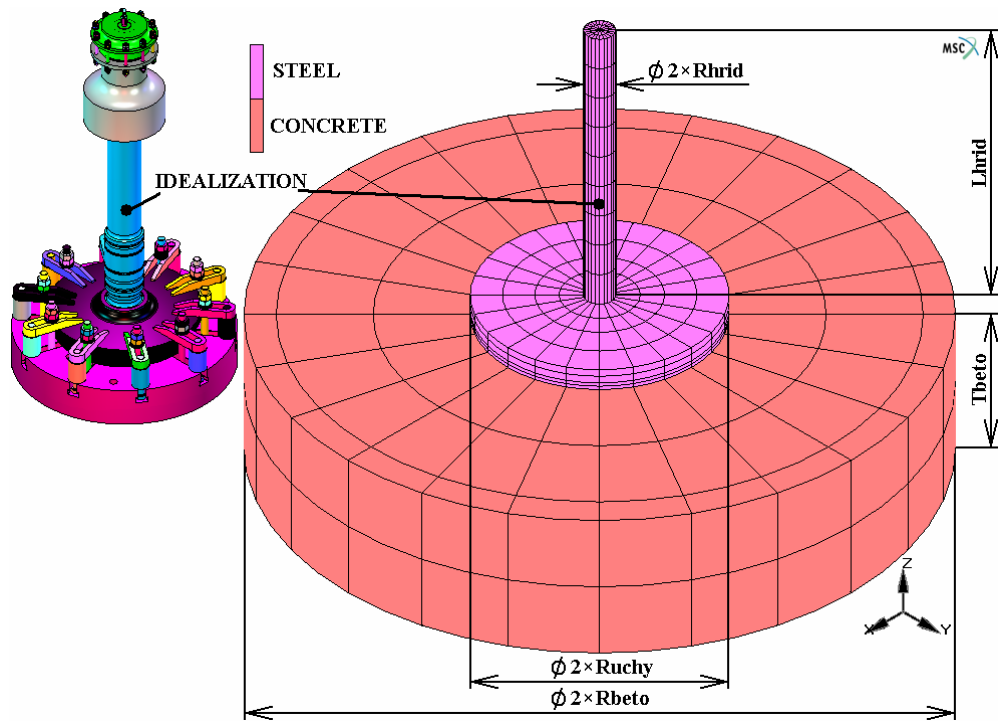

Fig.2 The FE Model and Accepted Simplifications

DYNAMIC BEHAVIOUR OF MACHINE FOR FATIGUE TESTING OF RAILWAY AXLES

For the right proposition of a new machine for fatigue testing of railway axles (rezonator) is very important to know the basic dynamic characteristics of whole system. These dynamic characteristics are solved via FEM (ANSYS and MSC.MARC/MENTAT software).

The FE model and its simplifications is shown on Fig.2.

The base (botton part) of the testing machine in made of concrete and the upper part is made of steel. Two versions of testing machines with different dimensions and with 12 or 16 springs were solved, see Fig.3. The springs are described by non-linear stiffnesses in radial and axial directions. Damping propperties of concrete and steel (elastic materials) was described by Rayleigh material damping.
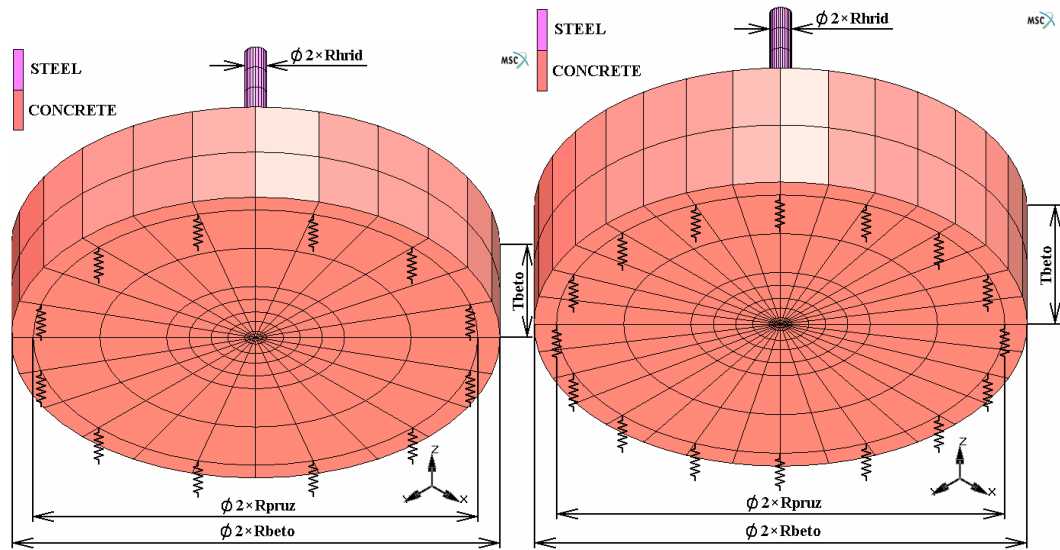

Fig.3 The FE Models with 12 and 16 springs

For both versions of FE models were solved modal analyses via Lanczos method and transient analyses (starting of machines to the steady-state conditions).

From the results of transient analyses is possible to calculate the radial displacement u_{RAD} /m/ and axial displacement w /m/ in the end of shaft. These displacements depend on frequence n_b /Hz/, see Fig.4 and 5. Finally can be calculated maximal bending stresses σ_o /MPa/ in the shaft.

The dependencies $u_{RAD} = f(n_b)$ and $\sigma_0 = f(n_b)$ for 12 springs are shown on Fig.6 and Fig.7.

The dependencies $u_{RAD} = f(n_b)$ and $\sigma_0 = f(n_b)$ for 16 springs are shown on Fig.8 and Fig.9.
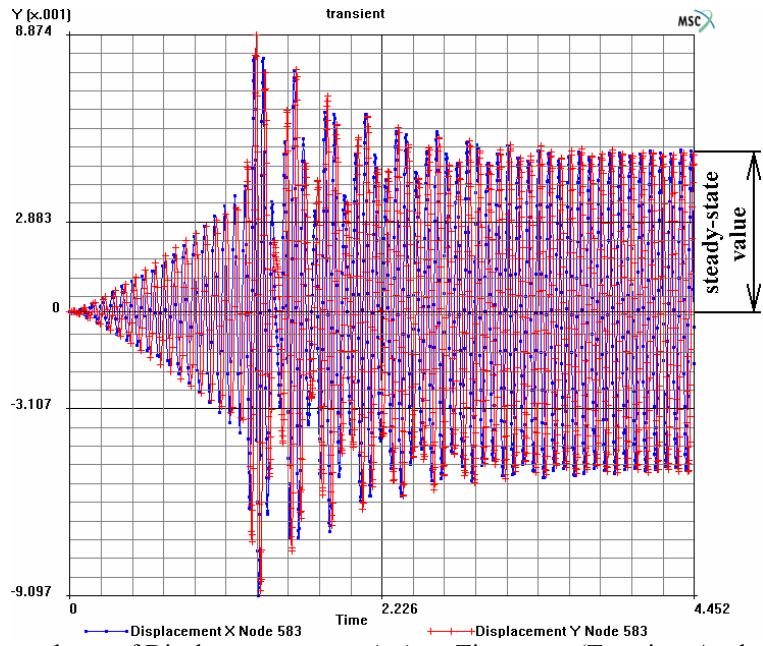
Fig.4 The Dependence of Displacements u, v /m/ on Time t /s/ (Transient Analysis, frequence $n_b = 25$ Hz)
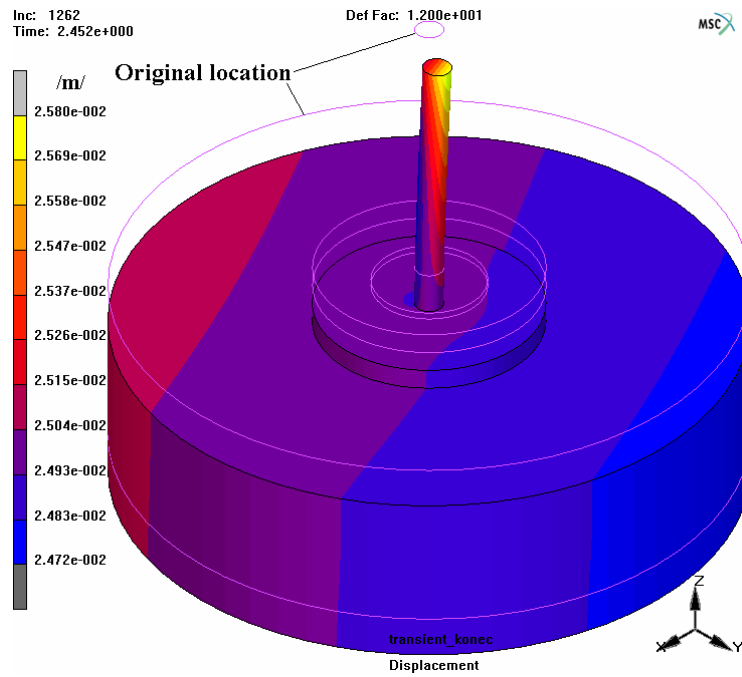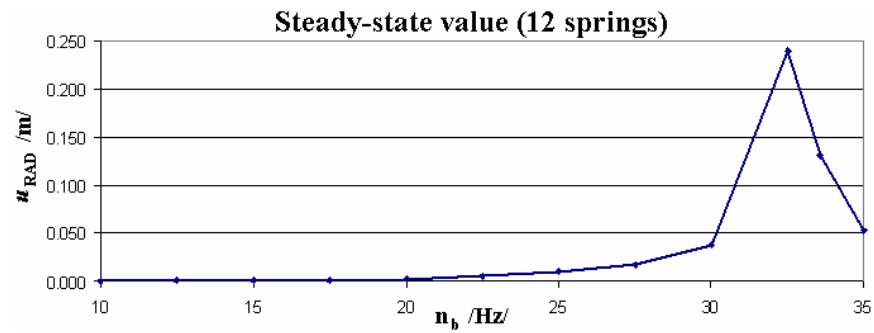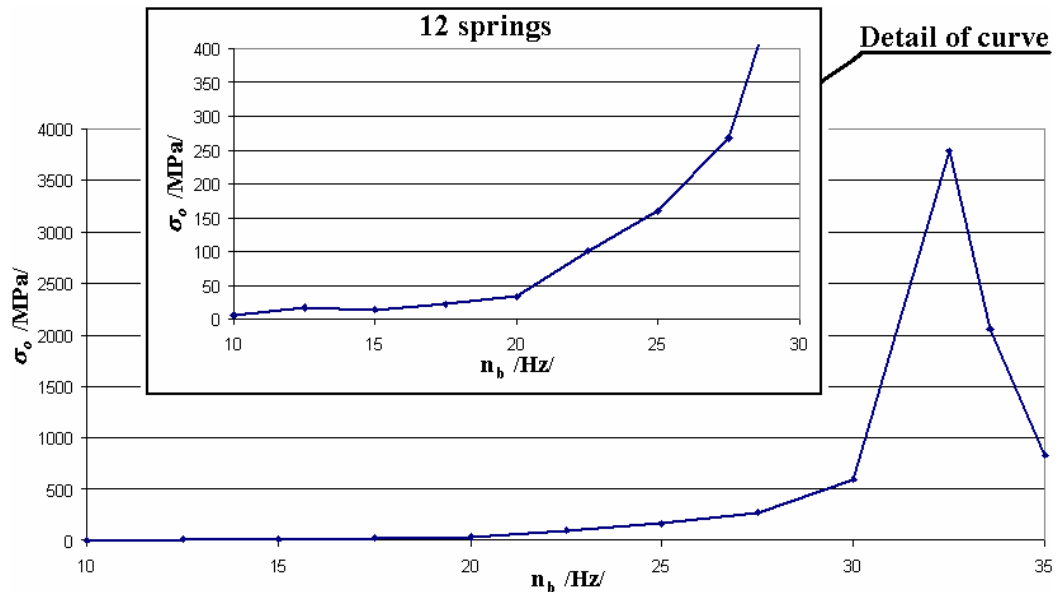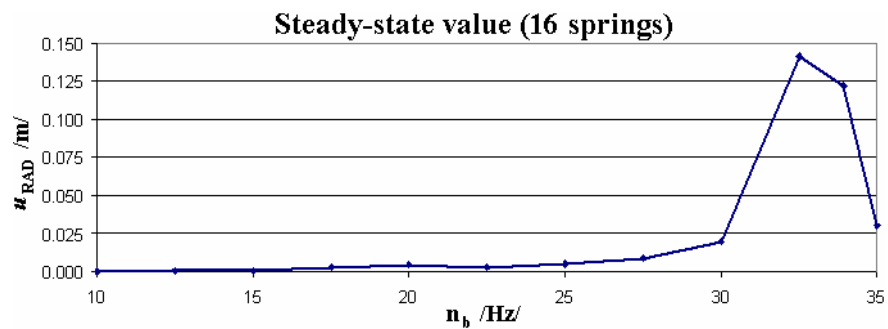

Fig.5 The Total Displacement on Time $t = 2.452$ s (Transient Analysis, 16 springs, frequence $n_b = 25$ Hz)

Fig.6 The Dependence u_{RAD} on Frequence n_b (Transient Analyses, 12 Springs)

Fig.7 The Dependence of σ_0 on Frequence n_b (Transient Analyses, 12 Springs)

Fig.8 The Dependence u_{RAD} on Frequence n_b (Transient Analyses, 16 Springs)
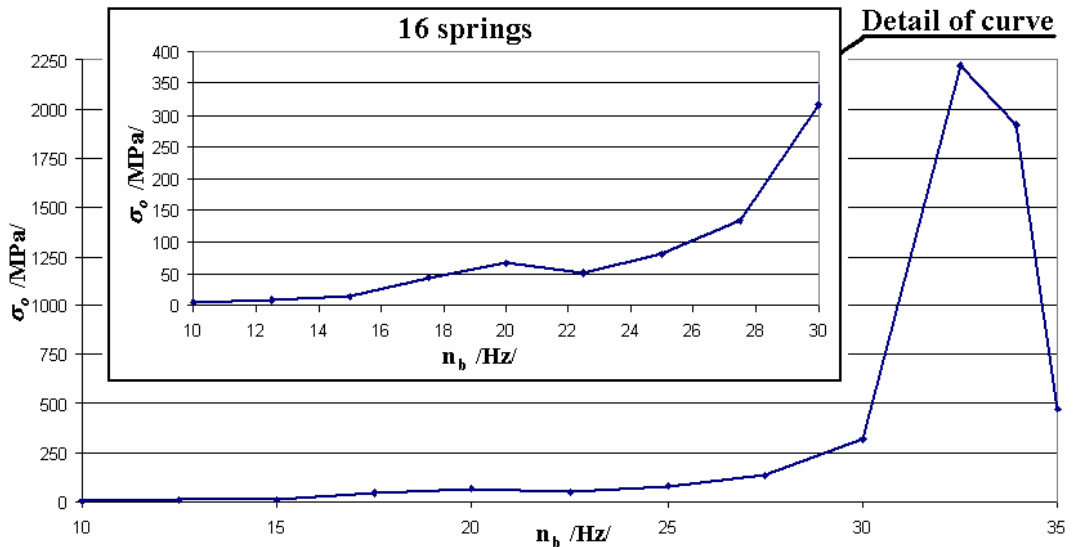
Fig.9 The Dependence of σ_0 on Frequence n_b (Transient Analyses, 16 Springs)

The higher values of bending stresses (higher than yield limit) are calculated with accepted mistakes because the plasticity of materials was not enabled. But all the basic dynamic characteristics are calculated correctly.

For more detail about these calculations see [5].

CONCLUSION

The proposed dimensions and springs of a new machine for fatigue testing of railway axles were used for manufacturing.

REFERENCES

[1] DYLĄG, Z., ORŁOŚ, Z.: *Únava materiálu a její zkoušení*, SNTL, Praha, 1968.
[2] EN 13261 Applications – Wheelsets and bogies – Axles – Product requirements
[3] EN 13103 Railway applications – Wheelsets and bogies – Non-powered axles – design Method.
[4] EN 13104 Railway applications – Wheelsets and bogies – Powered axles – design Method.
[5] FRYDRÝŠEK, K., ROJÍČEK, J.: *Výpočtová zpráva: Výpočet a optimalizace odpruženého základu určeného pro rezonanční zkoušky kol a náprav v měřítku* 1:1, katedra pružnosti a pevnosti, FS VŠB-TU Ostrava, Ostrava, 2005, pp. 25.