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DYNAMIC CHARACTERISTICS OF A NEW MACHINE FOR FATIGUE TESTING OF RAILWAY AXLES

DYNAMICKÉ CHARAKTERISTIKY NOVÉHO STROJE PRO TESTOVÁNÍ ÚNAVY ŽELEZNIČNÍCH NÁPRAV

Abstract

There were 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 and design of massive testing machine is also important to know the basic dynamic charakteristics of whole system.

Abstrakt

Byly provedeny návrhové výpočty nového zkušebního stroje. Tento nový zkušební stroj je určen pro testování únavy železničních náprav. Železniční nápravy jsou vystaveny působení ohybu a rotaci (odstředivé efekty). Pro projekt a návrh masivního testovacího stroje je také důležité znát základní dynamické charakteristiky celého systému.

1 INTRODUCTION

The detailed study of fatigue of railway axles started more than 150 years ago by german engineer August Wohler, see also [1], [2], [7] 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 [3], [4] and [5].

The determination of a fatigue limit for material loaded by composed bending and rotation is described in [4]. The fatigue tests for railvay 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 frequency is decreased about 0.5%). It signs the situation when a crack is iniciated.

On the left part of Fig.1 there is miniaturized model of testing machine which is made in 1:4 scale.

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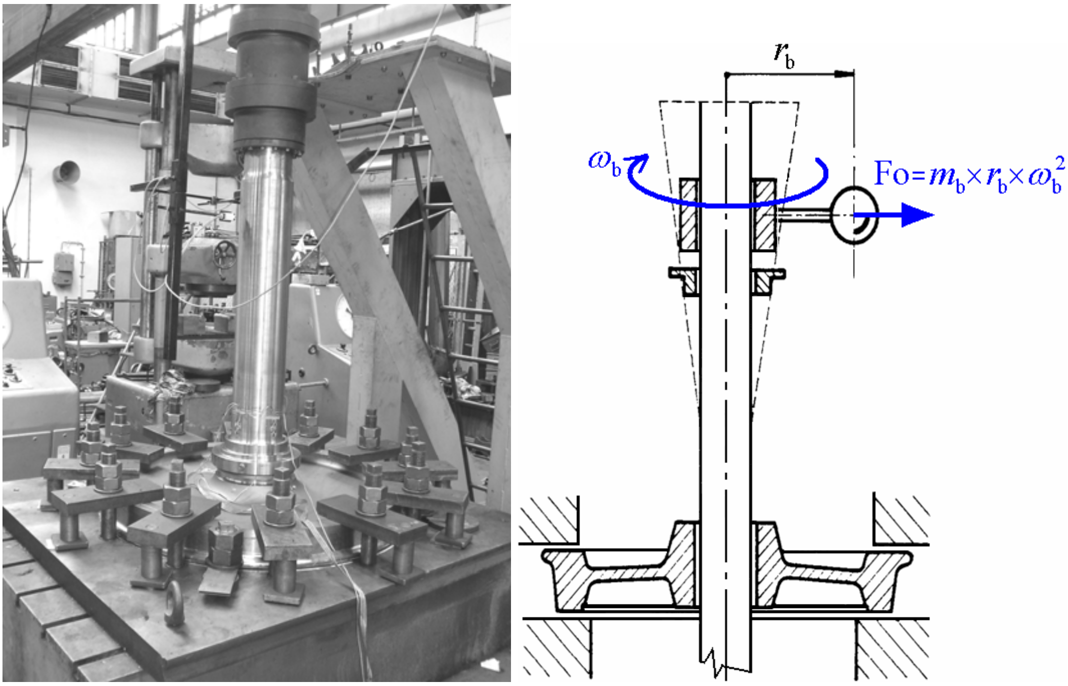

Fig. 1 Miniaturized Model and Principle of Testing Machine Based on Loading by Centifugal Forces Fo.

2 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 calculated and solved via FEM (MSC.MARC/MENTAT software).

The FE model and its simplifications are shown on Fig.2. For FE mesh was used brick elements with 20 nodes. The massive base (botton part) of the testing machine in made of concrete and the railvay axle (upper part) is made of steel.

Two versions of testing machines with different dimensions (described by parametrers: Rhrid, Lhrid, Tbeto, Ruchy, Rbeto, Rpruz /mm/) and with 12 or 16 non-linear springs were solved via FEM, see Fig.3. The springs are described by non-linear stiffnesses in radial and axial directions and also by linear damping in radial and axial directions. Damping properties of concrete and steel (elastic materials) were described by Rayleigh material damping.

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 (steady-state values, see Fig.4) is possible to calculate the radial displacement $u_{RAD} = \sqrt{u^2 + v^2}$ /m/ and axial displacement w /m/ in the end of shaft (i. e. in the Node 583). These displacements depend on frequency n_b /Hz/, see Fig.4 and 5. Finally can be calculated maximal bending stresses σ_0 /MPa/ in the shaft using basic formulas for clamped beam.

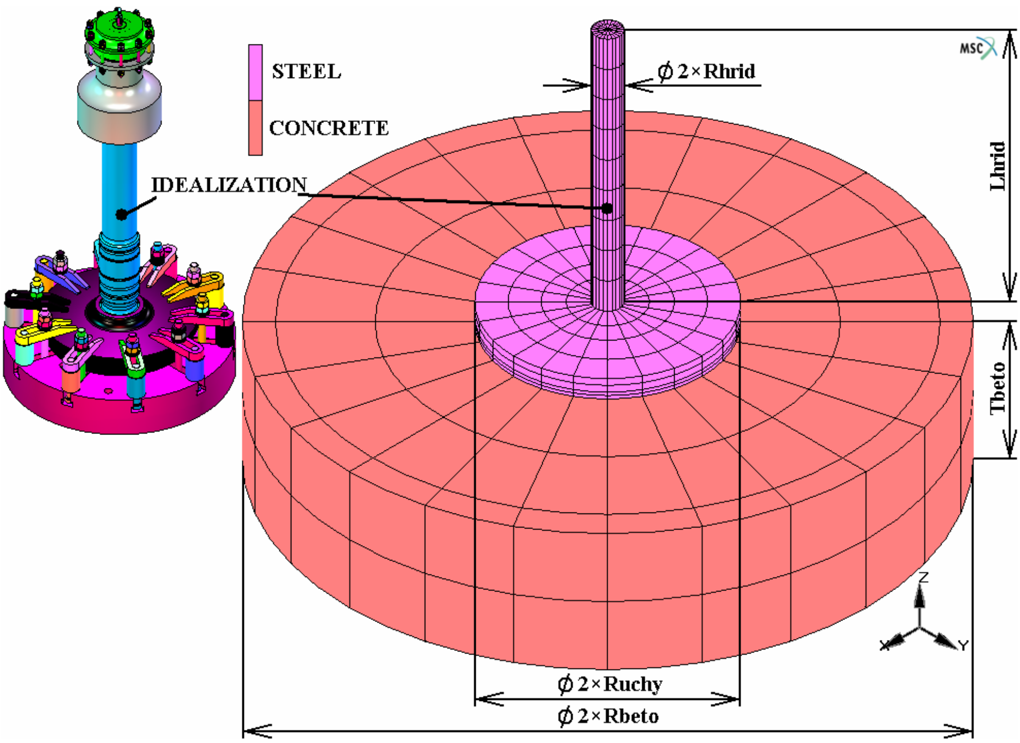

Fig. 2 The FE Model and Accepted Simplifications.

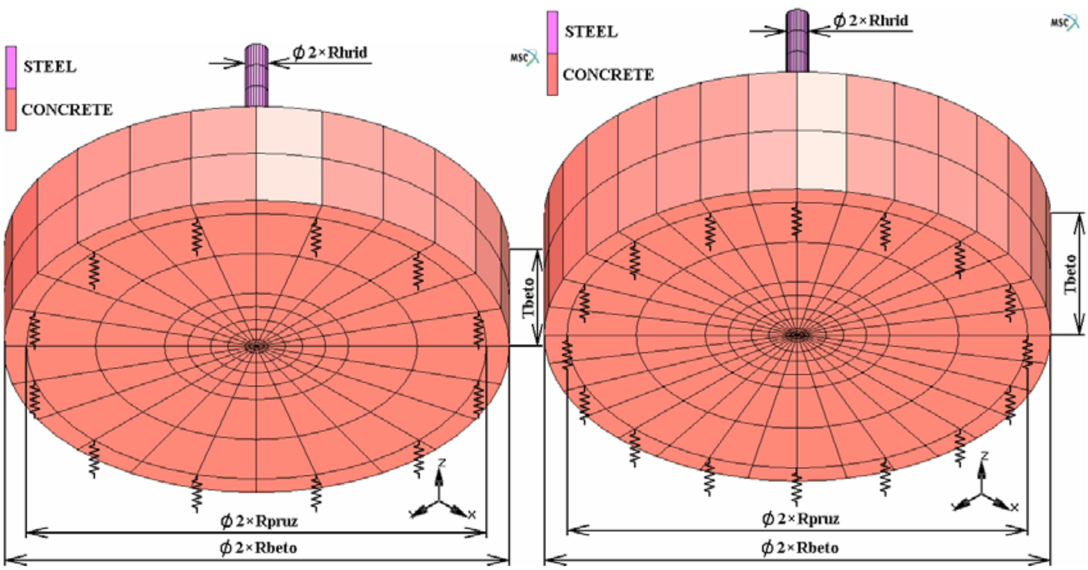

Fig. 3 The FE Models with 12 and 16 Non-Linear Springs.

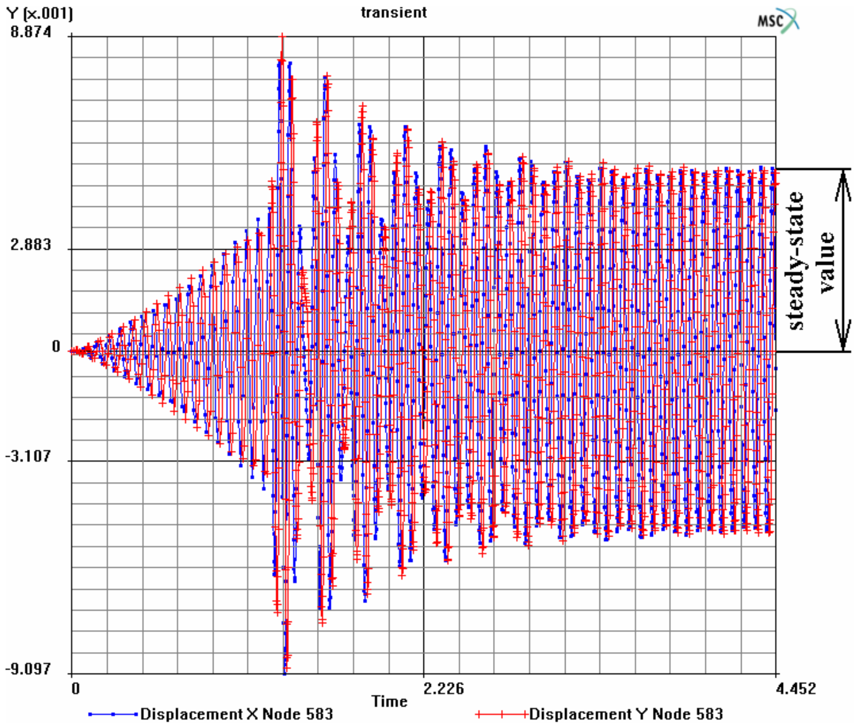

Fig. 4 The Dependencies of Displacements u, v /m/ on Time t /s/ (Transient Analysis, frequency n_b =25Hz). Results acquired by FEM.

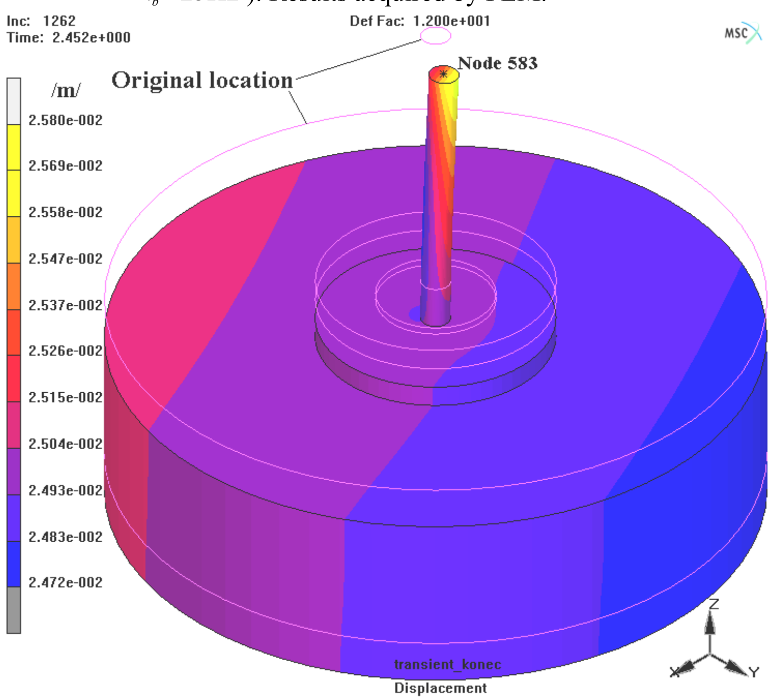

Fig. 5 The Total Displacement on Time $t = 2.452\ s$ (Transient Analysis, 16 springs, frequency $n_b = 25\ Hz$).

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

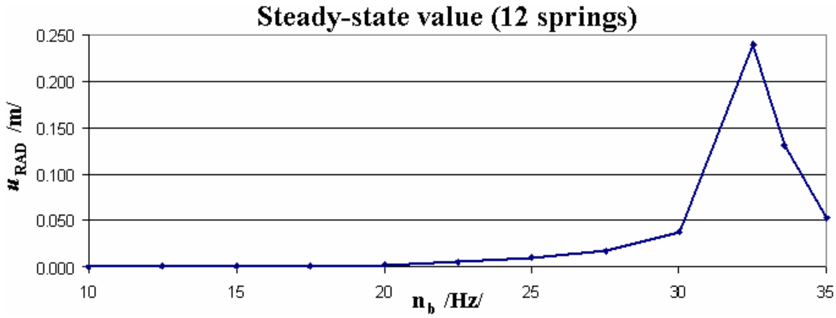

Fig. 6 The Dependence u_{RAD} on Frequency $\mathrm{n_b}$ (Transient Analyses, 12 Springs).

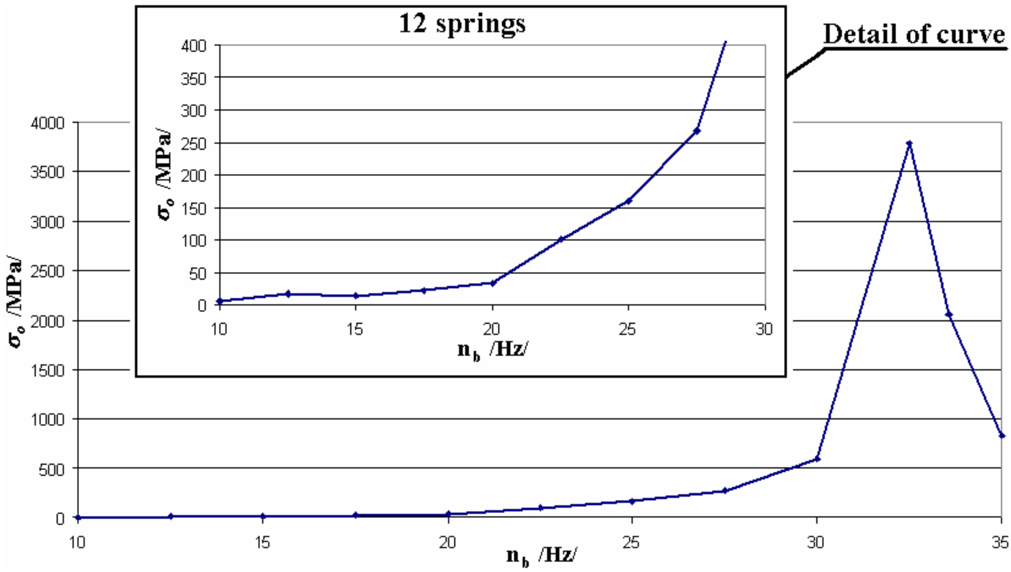

Fig. 7 The Dependence of σ_0 on Frequency $\mathrm{n_b}$ (Transient Analyses, 12 Springs).

The dependencies $u_{RAD} = f(n_b)$ and $\sigma_0 = f(n_b)$ for 16 springs are shown on Fig.8 and 9.

The calculated critical frequencies show the limits for using of testing machine.

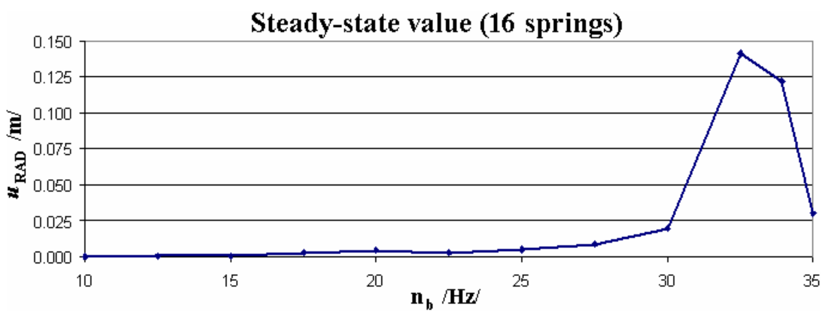

Fig. 8 The Dependence u_{RAD} on Frequency $\mathrm{n_b}$ (Transient Analyses, 16 Springs).

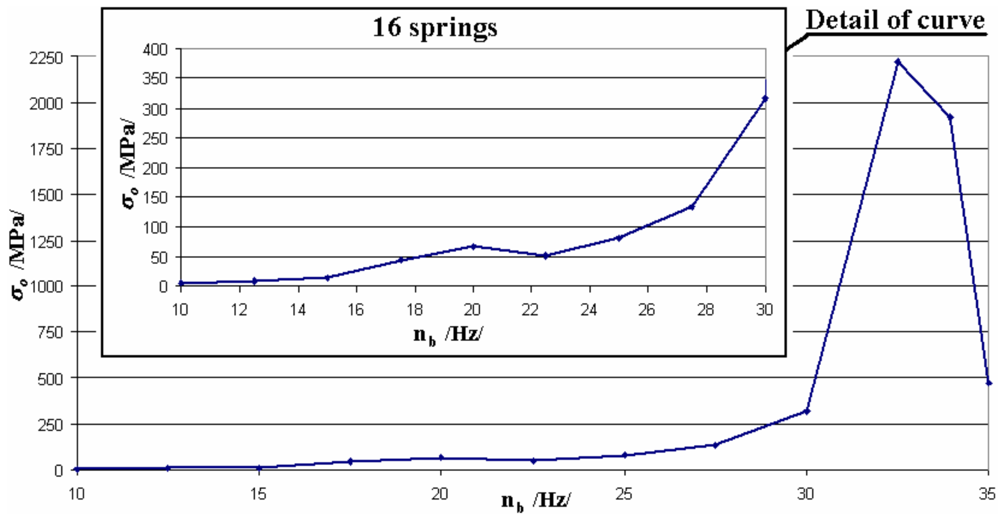

Fig. 9 The Dependence of σ_0 on Frequency n_b (Transient Analyses, 16 Springs).

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 with acceptable mistake.

3 Conclusions

The results show that the main critical (dangerous) frequencies are higher than the typical use of rezonator. Hence the proposed dimensions and springs of a new machine for fatigue testing of railway axles were used for manufacturing. Finally a new machine was produced in BONATRANS GROUP a.s. in Bohumin, CZ. For more detail about these calculations see [6].

REFERENCES

[1] DYLĄG, Z., ORŁOŚ, Z.: *Únava materiálu a její zkoušení*, Praha, SNTL, 1968.

[2] KUČERA, J.: *Úvod do mechaniky lomu (Nestabilní lom ocelových těles při statickém a dynamickém zatížení)*, Ostrava, VŠB - Technická univerzita Ostrava, pp.297, ISBN 80-248-1268-1

[3] EN 13261 Applications – Wheelsets and bogies – Axles – Product requirements

[4] EN 13103 Railway applications – Wheelsets and bogies – Non-powered axles – design Method.

[5] EN 13104 Railway applications – Wheelsets and bogies – Powered axles – design Method.

[6] 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.

[7] JURMAN, J., FRIES, J.: Konstrukce velkostrojů a jejich spolehlivost, 1. díl, Zatížení konstrukčních uzlů – teorie a praxe, Montanex a.s., ISBN 80-7225-148-1, Ostrava, Czech Republic, 2004, pp. 115.

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