Computation of a torsion spring stabilizer bar rigidity and fatigue resistance

I Geonea[1], N Dumitru[1], A Rosca[1], N Craciunoiu[1], L Racila[1] and C Copilusi[1]

[1]Faculty of Mechanics, University of Craiova, Craiova, Romania.

igeonea@yahoo.com

Abstract. The stabilizing bar is the suspension element that reduces the sliding of the car by stopping any kind of uneven vertical movement between the wheels to which it is connected. The stabilizer bar is a piece designed to increase stability during curves, in the case of sharp turns or maneuvers, and to enhance vehicle maneuverability. In general, sports cars are equipped with such stabilizing bars both on the front and rear. In this paper we present research on the experimental determination of the characteristic and rigidity of the torsion bar springs used in suspensions of motor vehicles. The determination of the torsion bar spring characteristic is done by two methods: analytical and experimental. The results obtained by the two methods are similar. In the second part of the paper there is discussed a test bench used for the experimental determination of fatigue resistance of the torsion bar spring. Also, angular deformations of torsion bar with resistive strain gauge transducers are monitored during fatigue applications. The results of the fatigue application are also studied by numerical simulation in the ANSYS FEA program.

1. Introduction

Stabilizers serve to reduce rolling oscillations and increase vehicle stability in turn. The most widespread stabilizers are U-shaped, with the heads attached, directly or through intermediate rods, to the car body or suspension arms, and the middle part hinged to the body.

The main purpose is to obtain a proper design of the stabilizer bar and also to obtain proper torsion stiffness for this component [1]. Because the stabilizing bar is subjected to alternating twisting stresses, the material and its construction must be safe to operate in order to prevent fatigue breakage. For this purpose, a series of achievements are presented in the literature showing the study of fatigue resistance of the stabilizing bar, proposing optimal constructive variants [2, 3].

Most fatigue studies use the finite element method, especially the ANSYS program [4]. The influence of factors of the production process as well as of the material on the fatigue resistance is presented in [5]. Different methods of finite element modeling are presented in [6]. In [7] the construction of a stand for fatigue testing of stabilizing bars is presented. In paper [8] it is considered a crank and connecting rod mechanism which performs rotation fatigue loads upon a stabilizer bar.

The main purpose of the present research study is the experimental computation of the characteristic and rigidity of torsion bar spring. Also another purpose is to compare the experimental results with theoretical results, and determination of fatigue resistance of torsion bar spring, for torsion solicitation, with an alternant symmetric cycle.

2. Theoretical considerations

The curve that represents the dependence between the load that is acting upon the spring and the elastic deformation of the spring produced by the load, measured on the load direction, is called spring characteristic.

For metallic springs, that respects the Hooke law and they are without interior friction, the characteristic is linear (with constant rigidity), as presented in Figure 1. For these springs, based on the Figure 1, is defined the rigidity as in equation (1):

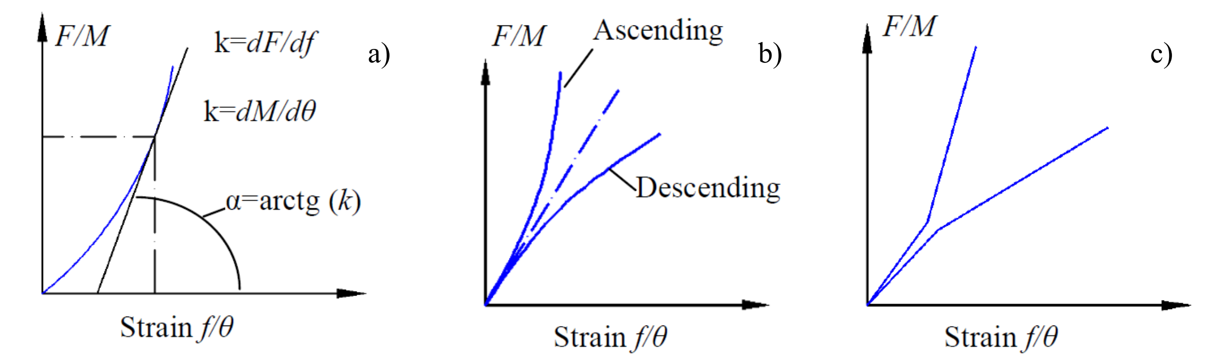

Figure 1. Torsion bar spring characteristic: a) -nonlinear characteristics; b)-progressive and regressive; c) - in steps.

$$c = \frac{F_i}{f} = tg\alpha = \frac{F}{f};\ c' = \frac{T_i}{\varphi_i} = tg\alpha' = \frac{T}{\varphi} \quad (1)$$

where: c is the spring rigidity; f is the bending displacement, in (mm); φ is the rotation, in (radians).

Under the torsion-only load, based on the calculation scheme in Figure 2, we can write the relationships for stresses and angular deformation as in equation (2).

$$\tau_{t\max} = \frac{T}{W_p} = \frac{16Fa}{\pi \cdot d^3} \le \tau_{at};\ \varphi = \frac{Tl}{GI_p} = \frac{Fal}{G\frac{\pi d^2}{32}} = \frac{32Fal}{G\pi d^4} = \frac{2\tau_t l}{Gd} \quad (2)$$

where: T is the applied torsion, in (N·mm); F- applied force, in (N); a- cantilever length of the spring, in (mm); d is the bar diameter; l is the spring length (mm); G is the material transversal elasticity module, in (MPa); φ –is the angular deformation of the spring, in (mm); τ_t – is the tangential stress, in (MPa).

To design, from the load is established the spring diameter d, and from the deformation the length l. The clamping of these levers is done by adjusting the shape of the corresponding ends of the springs: eccentric, flattened, hexagonal, square, triangular, grooves. To increase the fatigue strength, the shape from Figure 2, a) is recommended.

3. Experimental test rig

In Figure 2, a) is the drawing of a stabilizer at the rear axle of a car. The front stabilizer consists of a U-shaped bar which is transversely mounted to the longitudinal axis of the car in two rubber bushings fixed by the frame. The bar ends are connected by means of the rods to the support plates of the springs. If the deformations of the two springs are equal, the torsion bar will rotate freely in the rubber sleeves without getting into action. At turns, when the body is tilted laterally and one of the springs will compress more, the bar will work as a torsion bar and opposes a transverse tilt resistance of the car. With the help of stabilizers, the lateral tilting of the body can be reduced by 20 ... 30% without changing the suspension characteristics. Stabilizers are fitted to most cars as well as to some buses. The rigidity of the stabilizer can be determined with the equation (3):

$$k_{st} = G_c \cdot h\left(\frac{Y}{\alpha G_a} \pm 1\right) - k'_g \qquad (3)$$

in which: G_c - is the weight of the car body; h - the distance from the center of gravity to the roll axis; Y - lateral force; α - the roll angle in radians; k'_g - the overall angularity of the front and rear suspension without a stabilizer. It is recommended that the stabilizer mounts only if the roll angle is greater than 7(°). If the stabilizer is mounted only on the front axle the car will be under the turn. If the car is fitted with both front and rear axle stabilizers, the ratio between the front and rear axle rigidities of the front axle shall be between 1.2 and 1.6. In order to calculate the characteristic and rigidity, the torsion bar spring, with shape and constructive dimensions indicated in Figure 2, is mounted on the fatigue test bench. The kinematic scheme of the fatigue test bench is presented in Figure 3. Notations used in Figure 3, that is the test rig kinematic scheme, have the following significations: 1, 2 – actuation gearboxes; 3 – rocker; 4- connecting rods; 5 – links with connecting devices; 6 – vehicle roll rods; 7 –front stabilizer bar; 8 – flexible blocks for stabilizer bar; 9- rear vehicle stabilizer bar.

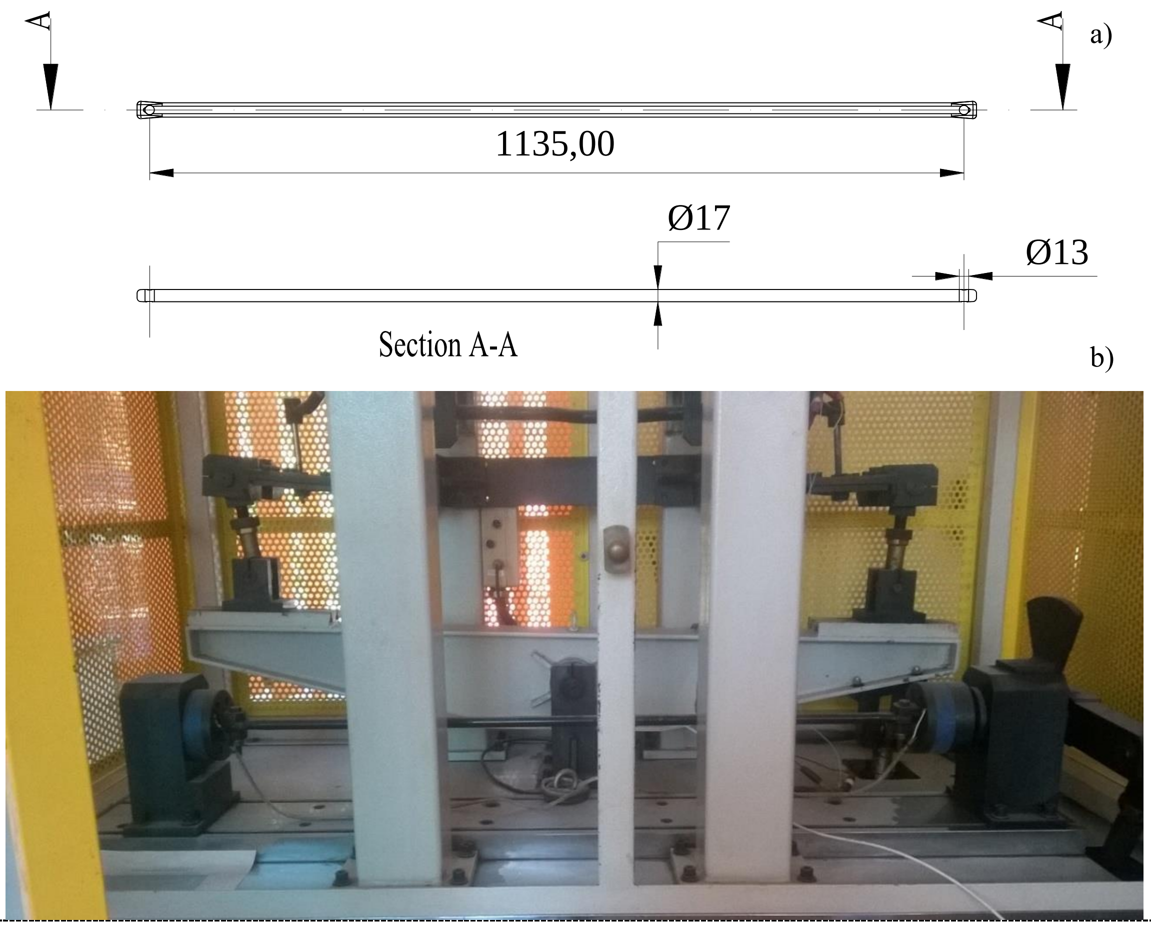

Figure 2. Geometric dimensions of the torsion bar spring a), and mounting of test rig b).

The control panel of the fatigue test rig of the torsion bar springs is presented in Figure 4, a). The control panel allows the following actions and features: manual mode start or automatic mode start of the gear motor, necessary for the torsion spring fatigue mechanism; emergency stop button; indicator for number of cycles of solicitations (number of cycles performed and the number of maximum specified cycles); the electric motor rpm indicator; deformations indicator; general key for start – stop.

The torsion bar spring, with the shape and dimensions indicated in Figure 2, is mounted on the torsion fatigue test rig. The torsion solicitation is achieved with crank rod rocker mechanism. The rod length can be adjusted with a screw-nut mechanism. In this way is modified the angular amplitude of torsion solicitation. The measurement equipment mounted on the test rig, consist in strain gauges, which measures the deformations produced by the torsion. In Figure 4, b) is observed the strain gauge transducer that measures the angular deformations of the torsion spring stabilizer.

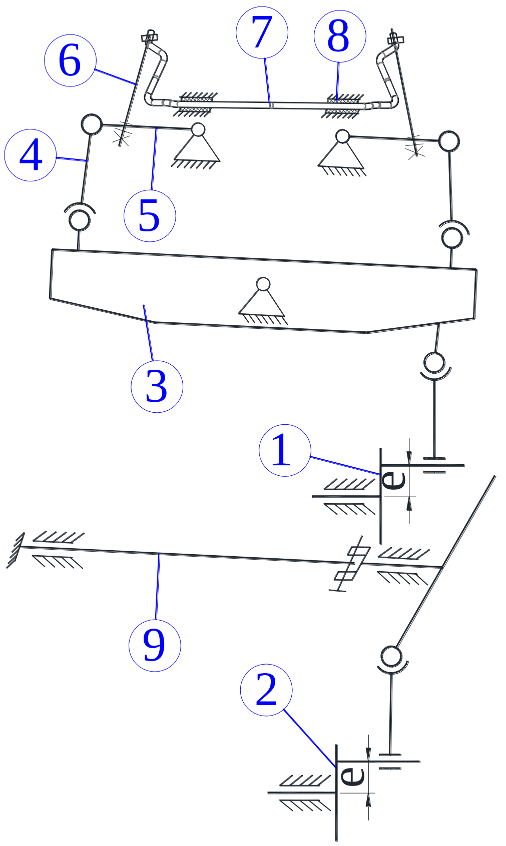

Figure 3. Test bench for torsion spring bar testing.

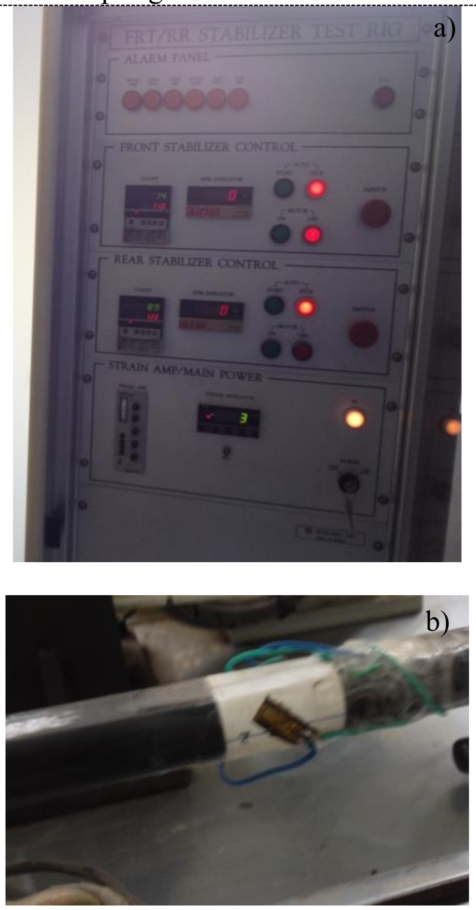

Figure 4. Control panel of test rig and. strain gauge transducer.

4. Torsion bar characteristic determination

The torsion bar spring is subjected to torsion by applying to one head of a torsion torque and the other side is fixed, as is observed in Figure 3. The torsion torque is applied by rotating of the actuation mechanism rocker.

It aims to establish the torsion bar spring rigidity, namely the dependence from the torsion torque applied to the spring head and produced angular deformation θ, in (radians) or (degrees).

$$k = \frac{dM}{d\theta} = \frac{GI_P}{l} \tag{4}$$

where: G –is the transversal elasticity modulus of the spring material, in (MPa).

Ip – is the polar moment of inertia of spring transversal section (mm^4); *d*- torsion spring diameter (mm).

Based on the geometrical dimensions of the torsion spring, *l*=1135 (mm), *d*=17 (mm), depicted in Figure 2, is computed $k_{\text{theoretic}}$ = 577,6 (N·mm/rad).

For experimental determination of the torsion spring constant, is dissembled the rod of the actuation mechanism. With a screw-nut mechanism mounted on the rocker side, is applied on the rocker 3-5 progressive loads, measured with a force transducer. Corresponding to each load is read on the pointer the angular rotation of the torsion spring bar. Taking into account the length between the point of force application and the point of rocker articulation, the experimental torsion torque is determined. The torsion torque applied to the spring, as well the angular rotation allows to draw the spring characteristic diagram.

Based on this experimental data, is plotted the spring experimental characteristic, respectively the dependence between the torsion torque, T in (Nm) and the spring angular rotation θ (°). The characteristic shows a linear dependence between the torsion torque and the angular rotation, as presented in Figure 5.

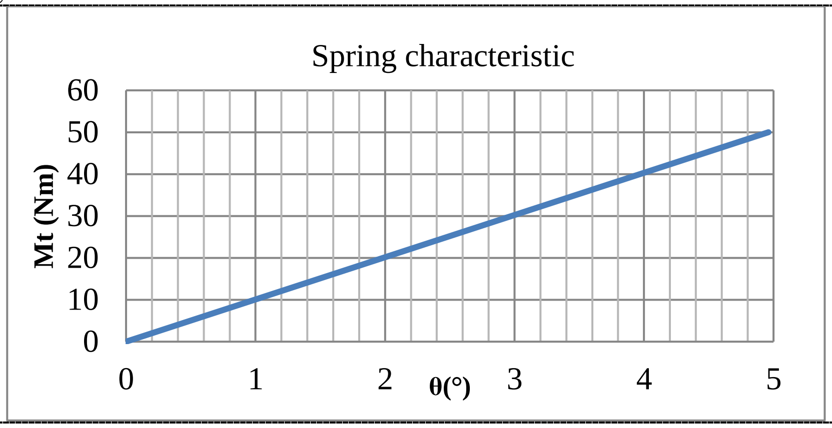

Figure 5. Experimental characteristic of torsion bar.

Based on the experimental characteristic, is computed the experimental constant, of the torsion bar spring, respectively: k_{exp}=585,872 (N·m/rad).

Is compared the spring computed constant, with that obtained experimentally, and is concluded that they have similar values.

5. Analysis of fatigue behavior in ANSYS of rear stabilizer bar

To this paragraph is studied the fatigue behavior of vehicle rear suspension stabilizer bar, with ANSYS finite element analysis software. The CAD model of rear stabilizer bar is imported in ANSYS database. The stabilizer bar is meshed in finite elements, are specified the contour conditions and loads.

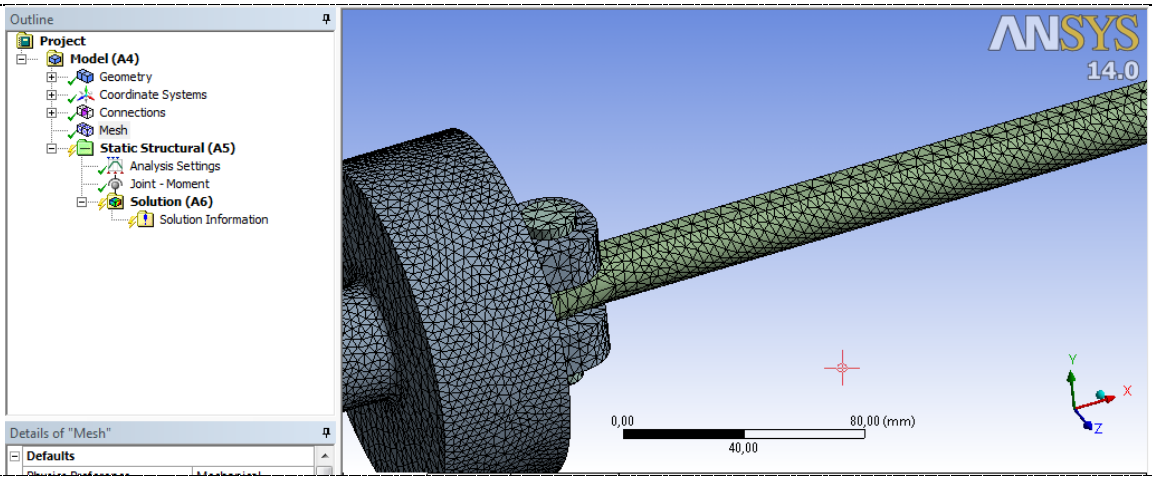

Figure 6. Stabilizer bar meshed in finite elements.

In the next step are defined the connecting bolts of the stabilizer bar with the revolute joint and fixed joint elements. To the stabilizer bar ends, where the stabilizer bar is fixed with bolts to the vehicle wheel spindle, are imposed the following conditions. One side of the stabilizer bar is fixed, as is presented previously and to the other side a torsion torque is imposed. This situation corresponds to the hypothesis when one of the wheels passes to an obstacle. The obtained results of the finite element static structural analysis, respectively the stress for present load and safety coefficient are presented in Figure 7.

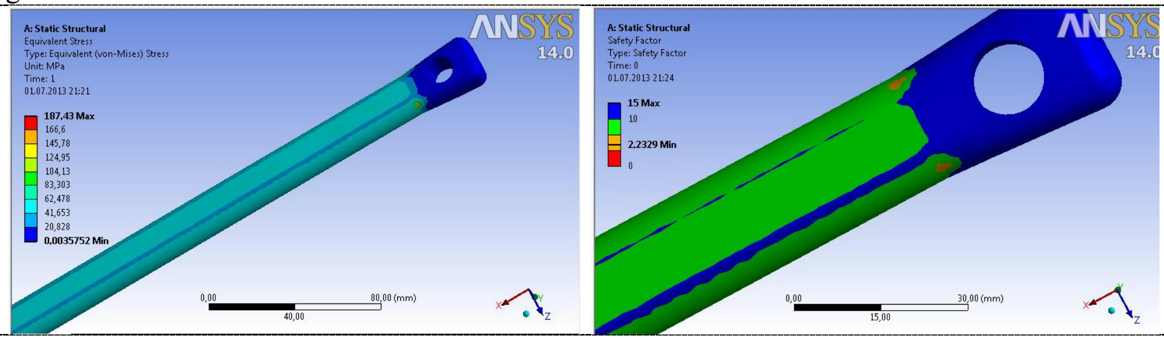

Figure 7. Equivalent von Misses stress distribution and fatigue safety coefficient.

6. Conclusions

The paper presents the theoretical and experimental determination of the characteristic and stiffness of a spring torsion spring from the suspension. The experimental analysis was performed on a fatigue test bench which stresses a rear stabilizer bar through an alternant symmetric load cycle. The characteristic diagram obtained for the torsion bar spring is linear, and the stiffness of the spring calculated theoretically and experimentally is of the same order of magnitude. Also the stabilizer bar model is subjected to a stress analysis with ANSYs software. The maximum equivalent stress obtained is 166 (MPa). The minimum safety coefficient for fatigue breakage is 2.23.

References

[1] Nikravesh P E and Gim G 1990 Ride and stability analysis of a sports car using multibody dynamic simulation Math. *Comput. Model. J* **14** pp. 953–958

[2] Marzbanrad J and Yadollahi A 2012 Fatigue life of an anti-roll bar of a passenger vehicle *World Acad. Sci. Eng. Technol. Int. J. Mech. Aerosp. Ind. Mechatron. Manuf. Eng.* **6(2)** pp. 407–413

[3] Palma ES 2002 Fatigue damage analysis in an automobile stabilizer bar *Proceedings of the Institution of Mechanical Engineers, Part D-Journal of Automobile Eng.* **216** pp. 865-871

[4] Liu S H and Li F 2012 Fatigue life analysis of the stabilizer anti-roll bar using ANSYS *Adv. Mater. Res.* **383–390** pp. 5894–5898

[5] Wittek A M, Burdzik R, Folega P, Konieczny L and Lazarz B 2015 Influence of production process and material factors on fatigue strength of tubular stabilizer bar *Arch. Metall. Mater.* **60** pp. 2485–2492

[6] Zhang L, Liu H, Xu Y, Wu S 2009 Study on Modelling Method of Anti-roll Bar Using One Dimensional Beam Element *SAE Technical Paper 2009-01-1454*

[7] Rosca A S, Craciunoiu N, Geonea I and Ploscaru C 2018 Design of a Test Rig for Vehicle Stabilizer Bar Fatigue *Study Applied Mechanics and Materials 880* (Trans Tech Publications) pp. 226-231

[8] Dumitru N, Copilusi C, Geonea I and Margine A 2016 Kinematic and Dynamic Study of a Mechanism for a Vehicle Front and Rear Stabilizer Bars *CONAT* (Brasov) pp.161-169