stabilizer bars, calculation, construction

Adam-Markus WITTEK*, Hans-Christian RICHTER
ThyssenKrupp Bilstein Suspension GmbH
Oeger St. 85, 58095 Hagen, Germany
Bogusław ŁAZARZ
Silesian University of Technology, Faculty of Transport
Krasińskiego St. 8, 40-019 Katowice, Poland
**Corresponding author*. E-mail: adam.wittek@thyssenkrupp.com

STABILIZER BARS: Part 1. CALCULATIONS AND CONSTRUCTION

Summary. The article outlines the calculation methods for stabilizer bars. Modern technological and structural solutions in contemporary cars are reflected also in the construction and manufacturing of stabilizer bars. A proper construction and the selection of parameters influence the strength properties, the weight, durability and reliability as well as the selection of an appropriate production method.

STABILIZATORY SAMOCHODOWE: Część 1. OBLICZENIA I KONSTRUKCJA

Streszczenie. W artykule przedstawiono zarys metod obliczeniowych stabilizatorów samochodowych. Nowoczesne rozwiązania technologiczno-konstrukcyjne we współczesnych samochodach znajdują również odzwierciedlenie w konstrukcji i produkcji stabilizatorów. Prawidłowa konstrukcja i dobór parametrów mają wpływ na cechy wytrzymałościowe, ciężar, trwałość oraz niezawodność jak i wybór właściwej metody produkcyjnej.

1. INTRODUCTION

The function of stabilizer bars in motor vehicles is to reduce the body roll during cornering. The body roll is influenced by the occurring wheel load shift and the change of camber angle. Decisive is the steering performance which may be purposefully adjusted towards understeer or oversteer when designing the stabilization. So the stabilizer bars increases the travelling comfort and to a considerable extent the driving safety [10].

Stabilizer bars are non-bearing spring elements in vehicles. In contrast to all bearing springs, which are loaded by the static forces also in resting condition, the stabilizer bars are normally loaded during the driving phases only [4]. As resilient components of the chassis, stabilizer bars are connectors between axle and body as well as between the wheels of an axle. The position of stabilizer bars is selected in such a way that the anti-roll suspension stiffens – the rotation of the body about the vehicle's longitudinal axis is made difficult – without simultaneously hindering the vertical suspension, i.e. motion of the body towards the vertical axis. For this purpose, the stabilizer bar is arranged in the axle in such a way that the back comes to rest approximately at the level of the wheel centers across the driving direction. The bearings of the stabilizer bar support themselves against the body; stabilizer bars do not contribute to static support of the weight of the body against the axle and remain unloaded during synchronistic downward or upward deflection of the spring.

When the body leans due to centrifugal forces acting in the transverse direction of the vehicle, the so-called reciprocal suspension comes about. This means that at the bend of the road the bend-external wheel deflects downwards and the bend-internal wheel deflects upwards. As a result of this, the stabilizer bar arms are deflected in the opposite direction and the back is twisted [9, 10].

The body roll during cornering could be reduced also by the selection of a harder vertical suspension, but it would have a negative impact on the driving comfort. The stabilizer bars contribute thus considerably to the improvement of comfort of the motor vehicles.

2. CALCULATION AND CONSTRUCTION

Stabilizer bars for the chassis of motor vehicles are usually U-shaped bars of spring steel with circular or circular ring cross-section, so they form a bow with the back and the arms. As few as possible kinks preferably in a single plane should be planned when constructing stabilizer bars.

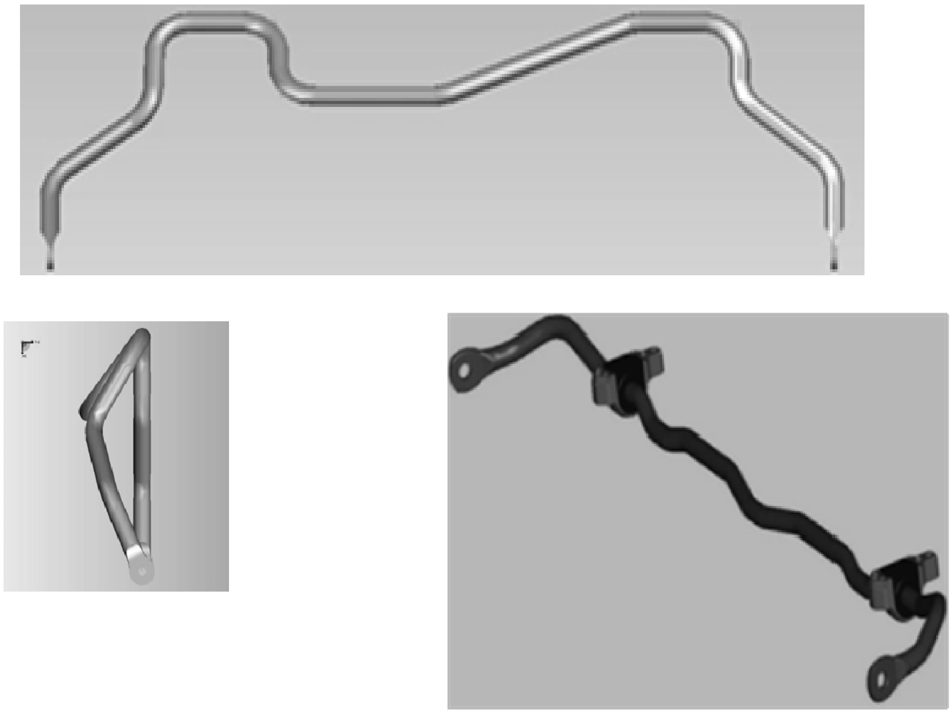

Fig. 1. Form of stabilizers
Rys. 1. Kształty stabilizatorów

This makes the manufacturing of the parts easier. However, as a rule, the stabilizer bars do not lie in a single plane, but – for the purpose of avoidance of other chassis parts – are arranged spatially bent, angled and cranked in a partly bizarre way (fig. 1, 2). Their U-shape is principally maintained [1, 3, 4]. Fig. 2 contains examples of different shapes of stabilizer bars.

As large bending radii as possible should be selected. The inner bending radius must have at least the size of the bar diameter. The arm ends are shaped differently for the purpose of force transmission and steering. Generally, when constructing stabilizer bars, effort must be made to minimize the weight, e.g. by shortening the arms while maintaining constant stabilizer bar action (fig. 4) [1, 3, 4].

Most commonly, stabilizer bars are mounted in rubber or plastic (fig. 3), and each shape of bearing requires an appropriate shape of the bar end. The plastic bearing, the rubber stiffness and the rubber strain have an influence on all stabilizer bars or the wheel rate, respectively.

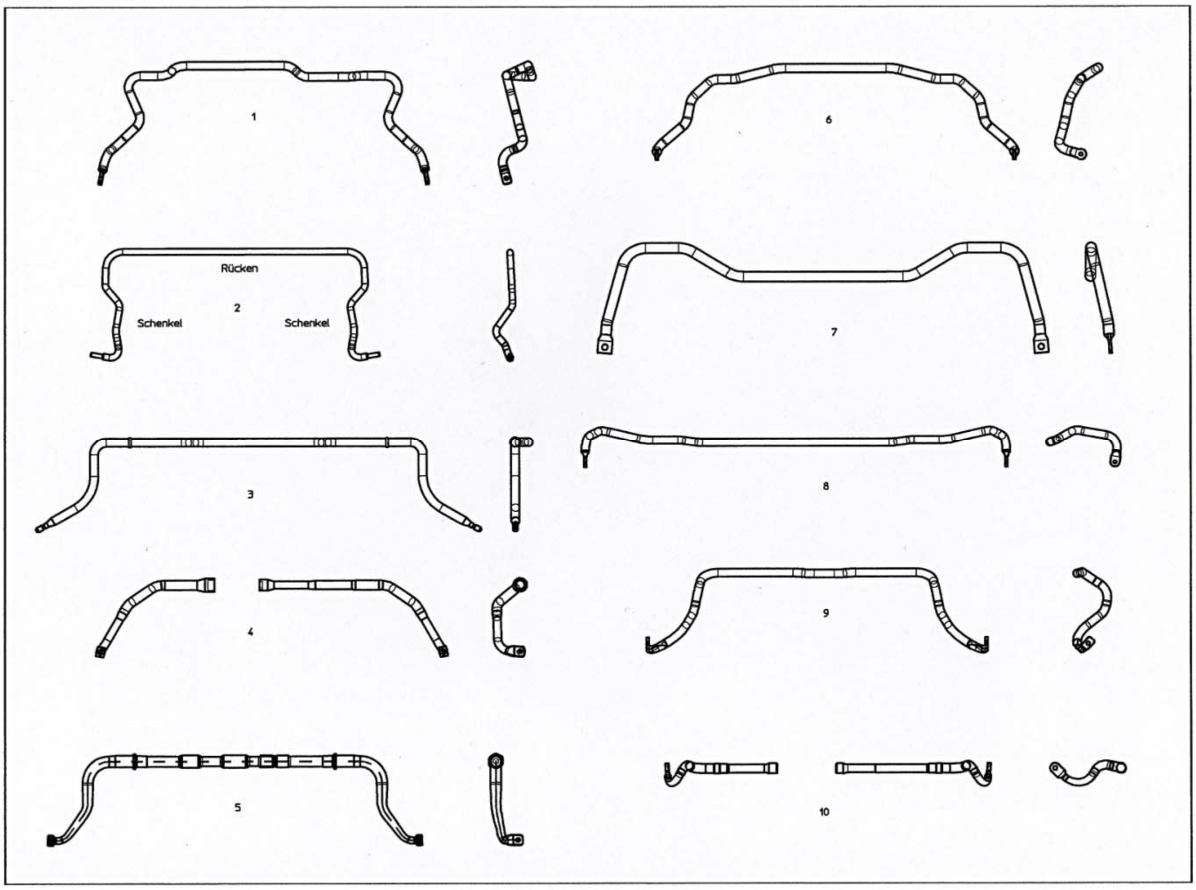

Fig. 2. Shapes of stabilizer bars - examples
Rys. 2. Kształty stabilizatorów samochodowych – przykłady

Fig. 3. Plastic bearings for stabilizer bars
Rys. 3. Przykłady łożysk stabilizatorów samochodowych

Stabilizer bars are manufactured mostly of round stock with rolled, drawn, peeled or ground surface. Bars that additionally take over the axle location functions are manufactured principally of ground or peeled primary material. Stabilizer bars are loaded only in turns. In contrast to the load-bearing spring elements, there are no requirements in respect of the relaxation performance. Therefore, also considering the notch sensitivity, the heat treatment strength is selected lower than in case of load-bearing spring elements [1, 3, 4].

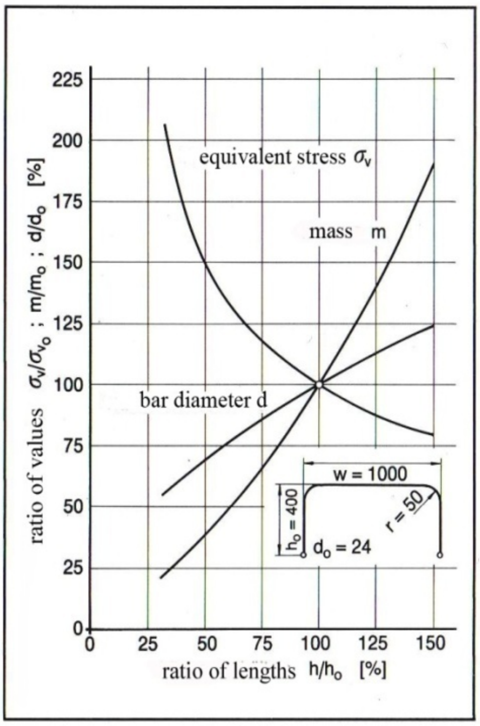

Fig. 4. Optimization of the weight of stabilizer bars by shortening the arm length with unchanged stabilization effect
Rys. 4. Redukcja ciężaru stabilizatorów poprzez skrócenie ramion przy niezmienionej sztywności

The curb weight of a car has been increasing steadily in the course of years due to increased requirements with regard to safety and convenience equipment. In order to counteract the tendency, components with substantial weight saving potential have been identified. The consideration of the load of stabilizer bars has shown that the max. load is on the outer edges of the diameter. The load decreases inwards to the neutral stage to a mean stress of $\sigma_{vm} = 0$. Theoretically, the solid stabilizer bar could be hollowed out without affecting the function. Therefore, the tubular stabilizer bars come into question with increasing frequency. In case of a tubular stabilizer bar, the weight may be reduced in comparison with the solid stabilizer bar of equal shape, with equal stabilization effect and adequate maximum stress [10].

The concern of the stabilizer bar calculation is to consider the numerous influences and various requirements based on the strain and stress relations in such a way that the designed stabilizer bar satisfies the requirements of

the strength test within the scope of which the compliance with the permitted stress, safety, load capacity or endurance limits are tested

and the requirements of

the function test within the scope of which the compliance with the requested stabilizer bar rate, the forces and stabilizer bar travels within the specified tolerances, the vibration performance and other requirements are tested [6].

2.1. Strength test

A force applied on the bar ends of a U-shaped bent solid stabilizer bar causes bending stress as well as torsional stress at the bar. While torsional stresses prevail at the back of the bar, the bending stresses are particularly great in the area of the arms [1, 3, 4, 6].

The permitted equivalent stress σ_V may be calculated according to the equation

$$\sigma_V = \sqrt{\sigma^2 + 3\tau^2} \tag{1}$$

where: σ - bending stress, τ torsional stress.

From the permitted torsional and bending stress values. In most cases, during stress analysis it will be found that the maximum equivalent stress and consequently the vulnerable cross-section is at the transition radius from the back to the arm. The position (stress maximum) is determined by the shape of the arm and the relationship bending radius/arm length [1, 3, 4].

The torsional stresses may be calculated from

$$\tau = \frac{M_t}{W_p}, \tag{2}$$

were: M_t – torque moment, W_p – modulus of twist.

For bar backs with round profile

$$\tau = \frac{16Ft}{\pi d^3}, \tag{3}$$

where: t – lever arm (fig. 5), F - force, d – diameter of stabilizer bar.

And the bending stresses from

$$\sigma = \frac{M_b}{W}, \tag{4}$$

where: M_b –bending moment, W – modulus of section.

For bar backs with round profile

$$\sigma = \frac{32Fb}{\pi d^3}, \tag{5}$$

where: b – lever arm (fig. 5).

The equivalent stress σ_v may be calculated according to the equation

$$\sigma_V = \frac{16F}{\pi d^3}\sqrt{4b^2 + 3t^2}. \tag{6}$$

The value of strains at the transition radius depends on the distances h_1+r and h_2 fig. 6 [3] shows different arm shapes of a stabilizer bar with circular profile.

Depending on the angle w^o defined in (fig. 7), the torsional and bending stresses as well as the equivalent stress resulting from the summing-up may be calculated:

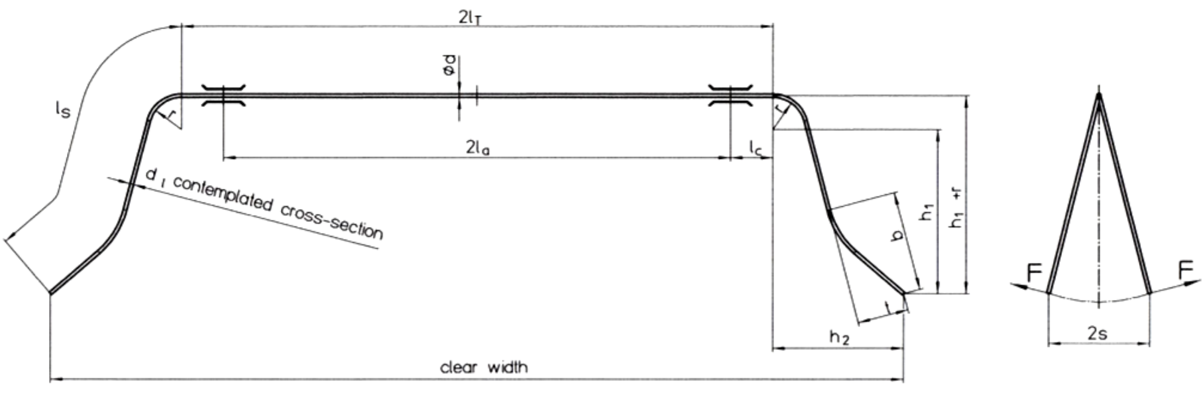

Fig. 5. Exemplary embodiment and transmission of forces
Rys. 5. Przykładowy model zastępczy do obliczeń wytrzymałościowych stabilizatora

$$w^{o} = \arccos(-3r / h_1 \sqrt{\left(\frac{h_2}{h_1}\right)^2 + 1}) - arctg\frac{h_2}{h_1} \tag{7}$$

$$\sigma_{V\max} = \frac{F}{W}\sqrt{h_1\left[\left(\frac{h_2}{h_1}\right)^2 + 1\right] + 3r^2} \tag{8}$$

The equations describe the place and size of the maximum equivalent stress in the transition area between the back and the arms [3] is the angle at which the transfer of equivalent stress is zero.

$$\frac{d\sigma_V}{dw^{\circ}} = 0 \tag{9}$$

The equivalent stress reaches its maximum at these points.

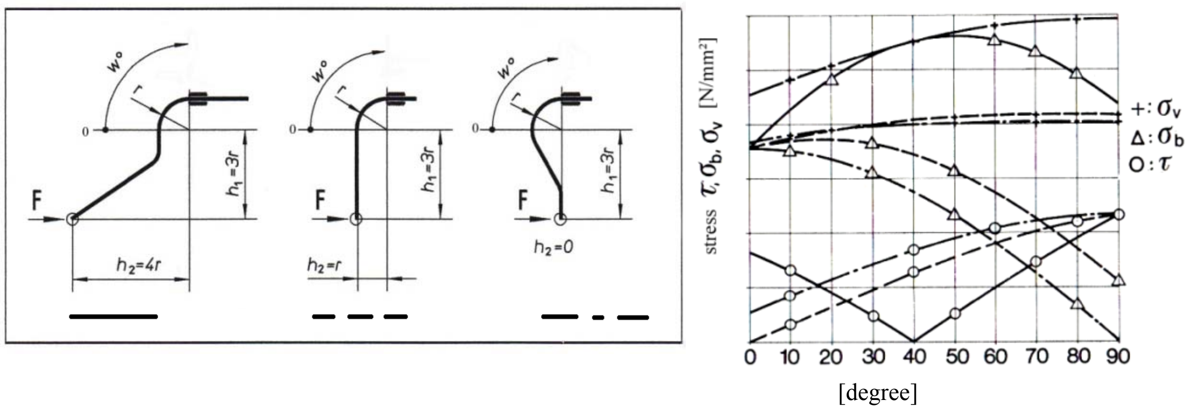

Fig. 6. Load τ, σ_b and σ_v at the transition radius from the shaft to the arm of a stabilizer bar
Rys. 6. Naprężenia τ, σ_b und σ_v w przejściu promienia z części prostej do ramienia pręta stabilizatora

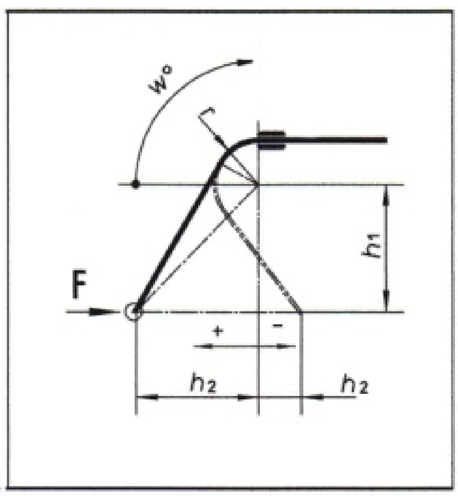

Fig. 7. Change in shape of the arm in the transition area
Rys. 7. Zmiany geometrii ramion stabilizatora w strefie promienia

Due to different requirements on stabilizer bars and numerous influences on the strength of materials, the characteristic value of the planned material is not always used in full value when dimensioning.

Permitted stresses which result as a quotient from the strength value regarded as ultimate stress $(R_m, R_e, R_{p0.2}, R_{p0.01}, \sigma_{bE}, \tau_{tF})$ and the required safety $S = S_{erf}$ are taken into account [6, 7, 8].

$$\sigma_{zul} = \sigma_{ertr} / S \tag{10}$$

$$\tau_{zul} = \tau_{ertr} / S \tag{11}$$

In some cases, characteristic values as shear modulus G and shear-spring limit (torsional elasticity limit) τ_{tE} are required for the design of torsionally strained stabilizer bars, but they are usually missing. Using approximation notations,

$$G = \frac{E}{2(1+\upsilon)} \tag{12}$$

where: υ - Poisson ratio, E – modulus of elasticity.

$$\tau_{tE} = \sigma_{bE} / 1{,}73 = 0{,}578\sigma_{bE} \tag{13}$$

a calculation is possible ($\upsilon = 3/10$ set). When the abovementioned material data is missing, the tensile strength R_m is used as a basis for the calculation of permitted stress τ_{zul} [6, 7, 8].

2.2. Function test

2.2.1. Rough determination of the total spring travel 2s

The rough calculation assumes that a part of the stabilizer bar is stressed only by twisting and the other only by bending. Participation of the back in the total spring travel:

$$2s_{T1} = \frac{2(h_1 + r)^2 l_T}{GI_p} F \tag{14}$$

where: I_p – polar moment of inertia.
Participation of the arm in the total spring travel:

$$2s_{B2} = \frac{2l_s^{\,3}}{3EI} F \tag{15}$$

where: I – moment of inertia about axis.
The total spring travel results from:

$$2s = 2(s_{T1} + s_{B2}) \tag{16}$$

$$2s = 2\left[\frac{(h_1 + r)^2 l_T}{GI_p} + \frac{l_s^{\,3}}{3EI}\right] F \tag{17}$$

The force F is calculable from the equation (17). Generally, the rough calculation yields greater forces than the measurements. This can be also traced back to the fact that the straining of rubber bearing is left out of consideration. Therefore, it is recommended to reduce the calculated force by about 10%. In case of simple geometries of stabilizer bar arms, the errors are of the order of 5%, however, in case of arms which, due to their shaping, allows assuming high torsion rates, the error may increase considerably [3, 4].

2.2.2. Exact determination of the total spring travels 2s

In case of an exact calculation, strains occurring in each profile are taken into consideration. Here, too, simplifying assumptions are made:

- The flexibility of the arms is small in relation to their length,
- In unloaded condition, the stabilizer bar lies in a single plane,
- The bearings are rigid,

- The stabilizer bar is symmetrical,
- As a result, only one side of the stabilizer bar is calculated.

Participation of the back in the total spring travel resulting from torsion:

$$s_{T1} = \frac{(h_1 + r)^2 l_T}{GI_p} F \tag{18}$$

Resulting from bending:

$$s_{B1} = \frac{F(l_c + h_2)^2}{3EI} \left\{ l_a + l_c \left[1 + \frac{h_2(l_0 + 2h_2)}{(l_c + h_2)^2} \right] \right\} \tag{19}$$

Participation of the arm in the total spring travel resulting from torsion:

$$ds = \frac{Ft^2}{GI_p} dl \Rightarrow s_{T1} = \frac{F}{GI_p} \int_0^{l_s} t^2 dl \tag{20}$$

Participation of the arm in the total spring travel resulting from bending:

$$ds = \frac{Fb^2}{GI} dl \Rightarrow s_{B1} = \frac{F}{GI} \int_0^{l_s} b^2 dl \tag{21}$$

The total spring travel 2s for the entire bar gives then (16)(17)(18)(19):

$$2s = 2(s_{B1} + s_{T1} + s_{B2} + s_{T2}) \tag{22}$$

3. CONCLUSIONS

The engineering data for stabilizer bars are specified by car manufacturers. These physical characteristics must not be altered by the stabilizer bar manufacturers. The described calculation methods serve thus the purpose of determining the following:

1. Whether the most important physical characteristics such as stabilizer bar rate, geometrical data (such as bending radii and planes) have been chosen correctly.
2. Whether the stress concentration, in particular in radii areas, remains comparable to the other stabilizer bar constructions within permitted limits.
3. Whether the chosen, possible method of production guarantees that the stresses in critical areas remain under the permitted limit.
4. Whether the geometrical requirements of the car manufacturer for bars are feasible in the series production.

References

1. Von Estorff H.-E.: *Technische Daten Fahrzeugfedern Teil:3 Stabilisatoren.* Stahlwerke Brüninghaus GmbH, Werk Werdohl, Hang Druck KG, Köln 1969.
2. *Technische Daten Fahrzeugfedern.* Stahlwerke Brüninghaus GmbH, Werk Werdohl, E.Anding KG, Herborn 1965.
3. Ulbricht J., Vondracek H., Kindermann S.: *Warm geformte Federn – Leitfaden für Konstruktion und Fertigung.* Hoesch Werke, Hohenlimburg Schwerte AG, W.Stumpf KG, Bochum 1973.
4. Fischer F., H.Vondracek H.: *Warm geformte Federn - Konstruktion und Fertigung.* Hoesch Werke, Hoesch Hohenlimburg AG, W.Stumpf KG, Bochum 1987.
5. Mitschke M.: *Teoria samochodu – Dynamika samochodu tom 2/ Drgania.* Wydanie 2 Wydawnictwa Komunikacji i Łączności, Warszawa 1989.
6. Meissner M.,.Schorcht H.-J.: *Metallfedern – Grundlagen, Werkstoffe, Berechnung, Gestaltung und Rechnereinsatz*. 2. Auflage, Springer Verlag, Ilmenau 2007.
7. Muhs D., Wittel H., Jannasch D., Voßiek J.: *Roloff / Matek Maschinenelemente – Normung, Berechnung, Gestaltung.* 18. Auflage, Viewegs Fachbücher der Technik, Wiesbaden 2007.

8. Jakubowicz A., Orloś Z.: Wytrzymałość materiałów. Wydanie 6, Wydawnictwa Naukowo-Techniczne, Warszawa 1984.
9. Reimpell J., Betzler J.W.: *Fahrwerktechnik – Grundlagen.* 5. Auflage, Vogel Verlag, Würzburg 2005.
10. Dziemballa H., Manke L.: *Gewichtsreduzierung durch hochbeanspruchte Rohrstabilisatoren.* ThyssenKrupp Technoforum 2004, Essen 2004.
11. Topac M., Kuralay N.S.: *Computer aided design of an anti-roll bar for a passenger bus.* http://www.turkcadcam.net/rapor/otobus-stab-cae/index.html, 23.10.2010
12. Tschätsch H., Dietrich J.: *Praxis der Umformtechnik*. 9. Auflage, Vieweg + Teubner, Wiesbaden 2008.
13. Klocke F., W.König W.: *Fertigungsverfahren 4 – Umformen.* 5. Auflage, Springer Verlag. Berlin – Heidelberg 2006.
14. Krist T.: *Handbuch für Techniker und Ingenieure.* 12. Auflage, Hoppenstedt Technik Tabellen Verlag, Darmstadt 1991.
15. Klein B.: *FEM – Grundlagen und Anwendungen der Finite-Element-Methode im Maschinen- und Fahrzeugbau*. 7. Auflage, Vieweg Studium Technik, Wiesbaden 2007.
16. Meissner M., Fischer F., Wanke K., Plitzko M.: *Die Geschichte der Metallfedern und Federtechnik in Deutschland.* 1. Auflage, Universitätsverlag Ilmenau. Ilmenau 2009.
17. Heißing B., Ersoy M.: *Fahrwerkhandbuch – Grundlagen, Fahrdynamik, Komponenten, Systeme, Mechatronik, Perspektiven.* 2. Auflage, Vieweg + Teubner, Wiesbaden 2008.
18. Prochowski L.: *Mechanika ruchu.* Wydanie 2 Wydawnictwa Komunikacji i Łączności, Warszawa 2008.

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