UDC: 623.438.3(047)=20
COSATI: 19-03, 20-11

Characteristics of torsion bar suspension elasticity in MBTs and the assessment of realized solutions

Rade Stevanović, BSc (Eng)[1)]

On the basis of available data and known equations, basic parameters of suspension systems in a number of main battle tanks with torsion bar suspension are determined. On the basis of the conclusions obtained from the presented graphs and the table, the characteristics of torsion bar suspension elasticity in some main types of tanks are assessed.

Key words: **main battle tank, suspension, torsion bar, elasticity, work load capacity.**

Principal symbols

a	– swing arm length
c	– torsion bar spring rate
C_{rs}	– reduced torsion bar spring rate in the static wheel position, suspension characteristic derivate in the static wheel position
d	– torsion bar diameter
f_d	– positive vertical spring travel, static-to-bump movement
f_s, f_m	– rebound and total vertical spring travel
F_s	– vertical wheel load on the road wheel in the static wheel position
g	– gravitational acceleration
G	– shear modulus for torsion bar steel
l	– torsion bar working part length
m_0	– tank sprung weight
N	– road wheel number
θ_s, θ_m	– torsion bar twist angle in the static wheel position and the bump stop
ν_z	– frequency of natural bounce oscillations of the vehicle
ν_φ	– frequency of natural pitch oscillations of the vehicle
τ_{dm}	– nominal shear stress of the torsion bar
φ_s	– swing arm angle in the static wheel position
ω_z	– angular frequency of bounce oscillations of the vehicle
E_p,E_{pu}	– work load capacity and the total available work load capacity
ε_p	– specific work load capacity
η	– work load capacity of the volume unit of the torsion bar working part
η_u	– total available work load capacity of the volume unit of the torsion bar working part
m_r,V_r	– weight and volume of the torsion bar working part.

Introduction

TORSION bar suspension systems are applied in all serial main battle tanks produced in Russia (former Soviet Union), the US and Germany, in one type of French and one Japanese main battle tank, and in some British tanks for export as well. All other producers from other countries, whose main battle tanks had the torsion bar suspension, mainly used the licence or the experience of the previously mentioned countries.

The characteristics of torsion bar suspension systems in main battle tanks in serial production until the 60s for the use in NATO forces (M-47 and M-48) and the Warsaw Treaty countries (T-54 and T-55), were of nearly the same (low) level, but still higher than the British tank Centurion characteristics, with BOGEY suspension, which was also in serial production at that time. Torsion bars were made of standard spring steel at that time, the technology was simple, and the presetting level small. Similar estimations can be given for torsion bars of main battle tanks from the 60s: AMX-30, T-62 and M-60A1. In the LEOPARD 1 torsion bar production technology was raised to a higher level. Significant improvement of suspension characteristics started in the 70s, by producing torsion bars out of nickel-chro-mium alloy steels of high purity, by using electro slag refining, and by introducing a variety of strengthening operations, including presetting, surface finish rolling and shoot peening. In this way, the nominal shear stress of torsion bars in the T-72, LEOPARD 2 and M-1 ABRAMS tanks increased for about 50%. Finally, the increasing of the torsion bar nominal shear stress for further 20%, was realized by introducing the technology of presetting during tempering, applied in some improved versions of T-72 T-90S tanks.

Apart from positive vertical spring travel and suspension work load capacity, the level of the realized nominal shear stresses in torsion bars is one of the main parameters of suspension quality. For a general analysis there is a need to take into account the influence of the frequency of natural bounce oscillations in a vehicle, i.e. the torsion bar spring rate, as one of main factors influencing the ride quality. This analysis considers some other parameters in order to better determine suspension characteristics.

1) Military Technical Institute (VTI), Katanićeva 15, 11000 Beograd

Equation deriving

On the basis of known basic equations (see Fig.1):

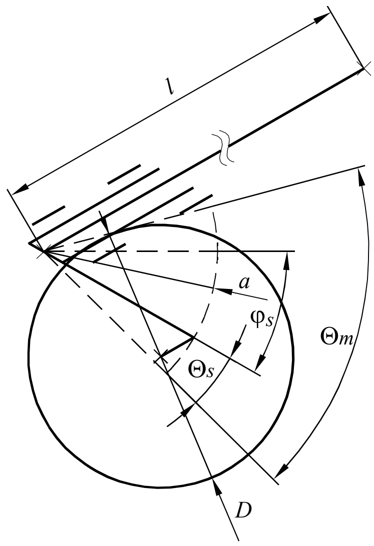

Figure 1

$$c = \frac{\pi G d^4}{32 l} \quad (1)$$

$$\theta_s = \frac{F_s a \cos\varphi_s}{c} \quad (2)$$

$$\theta_m = \frac{2 l \tau_{dm}}{G d} \quad (3)$$

$$F_s = \frac{m_0 g}{N} \quad (4)$$

$$C_{rs} = \frac{c\left(1-\theta_s \tan\varphi_s\right)}{\left(a\cos\varphi_s\right)^2} \quad (5)$$

$$\omega_z = 2\pi\nu_z = \sqrt{\frac{C_{rs}}{m_0/N}} \quad (6)$$

a formula for the torsion bar twist angle in the static wheel position was derived from the following:

$$\theta_s = \left(\frac{a}{g}\omega_z^2\cos\varphi_s + \tan\varphi_s\right)^{-1} = \left(4,026 a\nu_z^2\cos\varphi_s + \tan\varphi_s\right)^{-1} \quad (7)$$

and the torsion bar twist angle in the bump stop as well

$$\theta_m = \tau_{dm}\sqrt[4]{\left(\frac{l}{G}\right)^3\frac{\pi}{2c}} \quad (8)$$

From (2) and (4), the torsion bar spring rate c can be obtained from the following relation

$$c = \frac{m_0 g a \cos\varphi_s}{N\theta_s} \quad (9)$$

and the torsion bar twist angle at the bump stop, is a function of one parameter θ_s, (ν_z)

$$\theta_m = \tau_{dm}\sqrt[4]{\left(\frac{l}{G}\right)^3\frac{\pi}{2}\frac{N\theta_s}{m_0 g a\cos\varphi_s}} \quad (10)$$

The angles θ_s and θ_m, φ_s, length l, shear modulus G, sprung weight m_0 and road wheel number N are used in determining the positive vertical spring travel, the total available work load capacity and the torsion bar suspension work load capacity, by equations

$$f_d = a\left[\sin\left(\theta_m-\theta_s-\varphi_s\right)+\sin\varphi_s\right] \quad (11)$$

$$E_{pu} = N\tau_{dm}^2\sqrt{\left(\frac{l}{2G}\right)^3\pi c} \quad (12)$$

$$E_p = E_{pu} - \frac{Nc\theta_s^2}{2} - m_0 g f_d \quad (13)$$

By substituting the torsion bar spring rate from eq.(9), equations for the total available work load capacity of torsion bars and the suspension work load capacity are obtained, as functions of one variable θ_s, (ν_z)

$$E_{pu} = N\tau_{dm}^2\sqrt{\left(\frac{l}{2G}\right)^3\pi\frac{m_0 g a\cos\varphi_s}{N\theta_s}} \quad (14)$$

$$E_p = E_{pu} - \frac{m_0 g a\theta_s\cos\varphi_s}{2} - m_0 g f_d \quad (15)$$

The specific work load capacity, obtained by dividing the work load capacity with the sprung weight is as follows:

$$\varepsilon_p = \tau_{dm}^2\sqrt{\left(\frac{l}{2G}\right)^3\pi\frac{Nga\cos\varphi_s}{m_0\theta_s}} - \frac{ga\theta_s\cos\varphi_s}{2} - gf_d \quad (16)$$

The total work load capacity of the volume unit of the torsion bar working part, is obtained from [7]

$$\eta_u = \frac{E_{pu}}{V_r} = \frac{\tau_{dm}^2}{4G} \quad (17)$$

It can be concluded from eq.(17) that the total available work load capacity of the torsion bar volume unit is the function of square nominal shear stress and depends only on materials and torsion bar production process.

The determination of the work load capacity of the volume unit of the torsion bar working part was derived by expression (17)

$$\eta = \frac{E_P}{V_r} = \frac{E_p}{E_{pu}}\eta_u \quad (18)$$

The torsion bar diameter, calculated by the formula obtained by eqs.(1), (2) and (4) is as follows:

$$d = \sqrt[4]{\frac{32l}{\pi G}\frac{m_0}{N}\frac{ga\cos\varphi_s}{\theta_s}} \quad (19)$$

Determination of suspension parameters in main battle tanks

The basic suspension parameters are calculated for nine well-known main battle tanks with the torsion bar suspension and the results are given in Table 1.

Table 1.

Number	1	2	3	4	5	6	7	8	9
Tank type	T-55	M-60	LEO PARD 1	AMX-30	T-72	T-80U	LEO PARD 2	M-1 ABRAMS	T-90S
In set from:	1955	1962	1965	1966	1973	1976	1979	1980	1993
M_b [kg]	*36700*	*46700*	*41000*	*36000*	*41000*	*46000*	*55150*	*54500*	*46500*
M_0 [kg]	*32950*	*42000*	*37500*	*32500*	*37430*	*42000*	*50500*	*49500*	*42000*
N	*10*	*12*	*14*	*10*	*12*	*12*	*14*	*14*	*12*
a [mm]	*250*	*406.4*	*400*	*340*	*250*	*350*	*500*	*508*	*250*
l [mm]	*1930*	*1920*	*2010*	*1900*	*2110*	*2000*	*2126*	*1920*	*2110*
d [mm]	*52*	*58.7*	*53*	*52.5*	*47*	*47*	*63*	65.1	*47*
φ_s [°]	15.6	18.8	21.1	15.6	28.2	22	19.9	21.9	31.1
θ_s [°]	15.37	15.9	18.68	19.47	21.81	34.16	16.96	13.08	23.78
θ_m[°]	52	47.3	57.4	48.4	88.8	81.34	56.7	57.1	104.7
f_d [mm]	**157**	**220**	**265**	**170**	**275**	**280**	**340**	**381**	**320**
f_{st} [mm]	61	100	112	104	73	160	139	*102*	75
f_m [mm]	218	320	377	274	348	440	470	483	395
ν_z [Hz]	**1.89**	**1.45**	**1.34**	**1.42**	**1.54**	**0.99**	**1.26**	**1.45**	**1.45**
c [Nm/°]	506	830	525	534	309	326	990	1249	309
E_p [kNm]	58.4	82.1	*91.5*	37.6	138.9	70.7	184	287	205
E_{pu} [kNm]	120	194.5	211.2	119	255	256	389	498	355
ε_p [m²/s²]	**1.77**	**1.95**	**2.44**	**1.16**	**3.71**	**1.68**	**3.61**	**5.8**	**4.87**
η [Nm/cm³]	1.42	1.32	1.47	0.84	3.16	1.5	2.04	3.21	4.66
η_u [Nm/cm³]	2.92	3.123	3.4	2.65	5.81	5.42	4.19	5.57	8.07
τ_{dm} [N/mm²]	954	987	1012	910	1347	1301	1143	1318	1588
m_r[kg]	325	570	548	323	345	327	738	702	345
Data source		[2]	[4]	[3]			[4,5]	[6]	

The calculation was carried out with the values of the following parameters:

a – swing arm length
l – torsion bar working part length
m_0 – tank sprung mass
N – road wheel number.

The torsion bar diameter d was taken as a known parameter in most of the tanks. The shears modulus of $7.8 \cdot 10^{10}$ N/m^2 was accepted. Data for work load capacity or rebound travel were used for some tanks. All known parameters are shown in italics.

The calculation procedure is as follows:

- Angle φ_s was determined on the basis of clearance, road wheel diameter and track height.
- Angle θ_s was determined from eq.(7).
- Angle θ_m was determined from eq.(10).
- Positive vertical spring travel fd was determined from eq.(11). Since the effect of load in the track in the static wheel position is eliminated by eq.(9), higher values of the positive vertical spring travel than declared, ones were obtained.
- The rebound travel was determined from the following relation:

$$f_{st} = a\left[\sin\left(\theta_s + \varphi_s\right) - \sin\varphi_s\right] \tag{20}$$

- The total vertical spring travel f_m is the sum of the positive vertical spring travel f_d and the rebound travel f_{st}.
- The frequency of free bounce oscillations of the vehicle ν_z, was calculated from eq.(19). In the case of the LEOPARD 1 was used eq.(15), and in the case of the M-1 ABRAMS eq.(20).
- Total work load capacity E_{pu} was determined from eq.(14).
- Work load capacity E_p was determined from eq.(15).
- Torsion bar spring rate c was calculated according to eq.(1).
- Mass and volume of the torsion bar working part m_r and V_r, were determined using dimensions d and l; the density of steel is 7850 kg/m^3.

The values of specific work load capacity, frequency of natural bounce vehicle oscillations, positive vertical spring travel, total work load capacity of the volume unit and work load capacity of the volume unit are graphically presented in Figs.2 to 6.

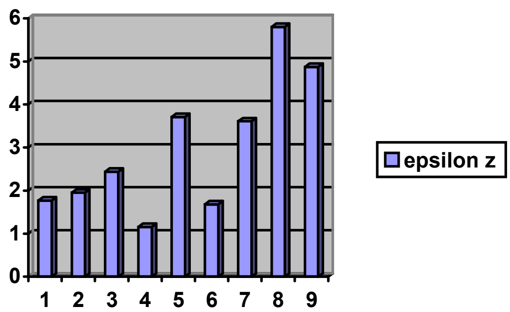

Figure 2. 1. T-55, 2. M-60A1, 3. LEOPARD 1, 4. AMX-30, 5. T-72, 6. T-80U, 7. LEOPARD 2, 8. M-1 ABRAMS, 9.T-90S

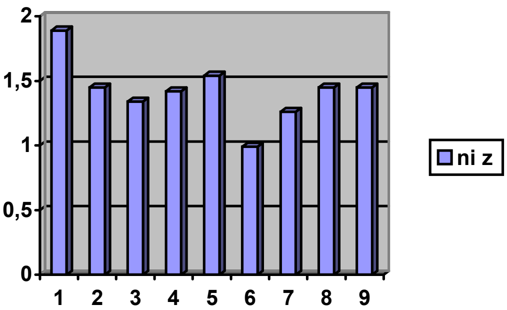

Figure 3. 1. T-55, 2. M-60A1, 3. LEOPARD 1, 4. AMX-30, 5. T-72, 6. T-80U, 7. LEOPARD 2, 8. M-1 ABRAMS, 9.T-90S

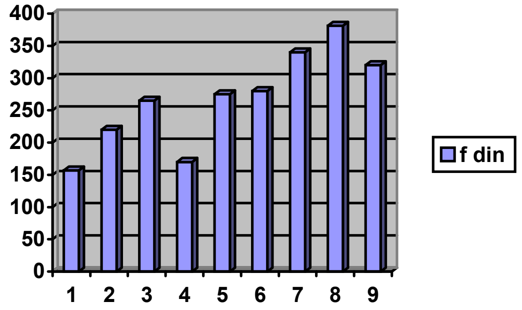

Figure 4. 1. T-55, 2. M-60A1, 3. LEOPARD 1, 4. AMX-30, 5. T-72, 6. T-80U, 7. LEOPARD 2, 8. M-1 ABRAMS, 9.T-90S

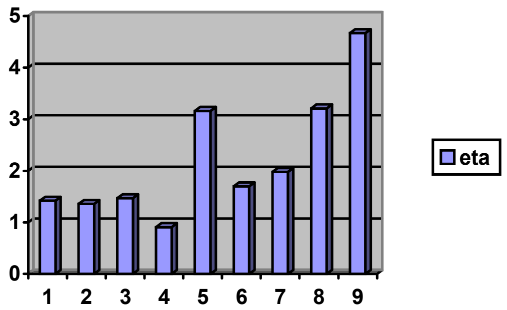

Figure 5. 1. T-55, 2. M-60A1, 3. LEOPARD 1, 4. AMX-30, 5. T-72, 6. T-80U, 7. LEOPARD 2, 8. M-1 ABRAMS, 9.T-90S

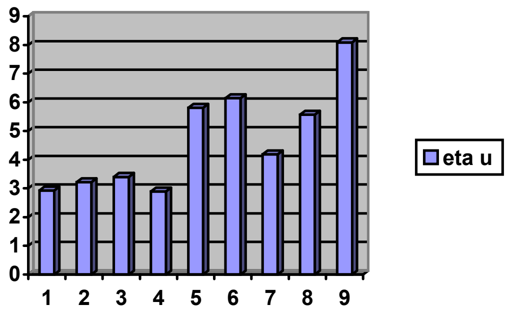

Figure 6. 1. T-55, 2. M-60A1, 3. LEOPARD 1, 4. AMX-30, 5. T-72, 6. T-80U, 7. LEOPARD 2, 8. M-1 ABRAMS, 9.T-90S

The graph in Fig.2, presents the specific work load capacity ε_p. This characteristic can be considered as a main one of a suspension system.

The M-1 ABRAMS tank has the largest specific work load capacity ε_p due to a high value of the nominal shear stress and the swing arm length (a=508 mm).

The T-90S tank has a 16% lower value of the specific work load capacity, with the smallest swing arm length (a=250 mm), but with highly resistant torsion bars.

Considerably lower values of the specific work of load capacity, are found in the T-72 tank (for 36%) and the LEOPARD 2 tank (for 38%), in the first case due to half of the value of the swing arm length, and in the second case due of the lower nominal shear stress of the torsion bar.

Among the tanks of a new generation, the lowest value of the specific work load capacity is found in the T-80U tank which is lower than in the LEOPARD 1, the M-60A1 and even the T-55. This is a result of a low value of the vehicle frequency of natural bounce oscillations (see Fig. 3). The value is lower for one third than a common value and nearly double lower than the frequency of the T-55 tank.

Interesting conclusions can be drawn from the comparison between the T-80U tank and the T-72 tank (see Fig.7).

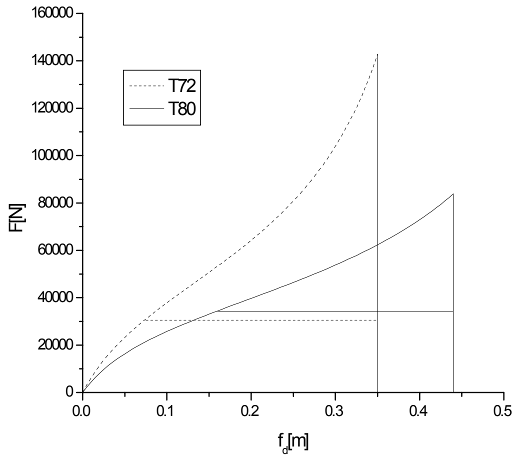

Figure 7

The T-72 and T-80U tanks have the same torsion bar diameter value, which in the case when the swing arm length ratio is 1:1.4, gives the specific work load capacity ratio of 2.2:1. The difference in the lengths of torsion bar working parts l and the m_0/N ratio have a certain influence on the specific work load capacity, but the ratio of frequencies of natural bounce oscillations has a definitive influence.

There is a more drastic difference between the frequencies of pitch oscillations of the T-80U and the T-72, because of different moments of inertia (hull length of the T-80U is 7 m and it is 6.4 m for the T-72), for the same length of the track on the ground and it can be seen from the following [1]

$$\nu_\varphi = \nu_z \frac{L_g}{i_y}\sqrt{\frac{2}{N}\sum_{i=1}^{N/2}\left(\frac{l_i}{L_g}\right)^2} \tag{21}$$

The length of the track on the ground, and the relation under the root are nearly the same values for both tank types.

Somewhat lower frequency of free bounce oscillations of the vehicle ν_z, in relation to other tanks, has the LEOPARD 2, equipped with friction dampers of comparatively low damping. Extremely large movements, are prevented by hydraulic bump stops with large damping energy, which is at the same level with the work load capacity. From this aspect, there is a need to point out to a problem of damping of extreme pitch movements in the T-80U with telescopic dampers, which, as already known, poorly eliminate heat energy.

Opposite to the T-80U, the highest level of frequencies of free bounce oscillations is found in the T-55. The fact that the specific work load capacity is higher in the T-55 can signify, that under particular ride conditions, the T-80U swing arm is more prone to hit the bump stop than its T-55 counterpart while accelerations due to oscillations, are certainly higher in the T-55 tank.

Other tanks have vehicle frequencies of free bounce oscillations in the range of 1.34-1.54 Hz, when the optimum value, from the aspect of ride quality, is near to the minimum value, or even below it. However, designers adopted higher values of ν_z, on purpose, in order to prevent rigid bumps of the swing arm at the bump stop.

The graph in Fig.4 shows the positive vertical spring travel. In this case, according to expectations, tanks produced from 1979 on wards have a high positive vertical spring travel, (over 300 mm). Some lower positive vertical spring travels exist in tanks produced between 1970 and 1978 (250-300 mm), and also in the LEOPARD 1, but all other tanks produced before 1970 have a positive vertical spring travel lower than 250 mm. The largest positive vertical spring travel is in the M-1 ABRAMS tank.

Work load capacity of the torsion bar volume unit η (Fig.5), is a less important characteristic. It states how much of work load capacity of the torsion bar volume unit (weight) can be obtained.

The graph in Fig.6 shows only material quality and technology levels for torsion bars. The total work load capacities of the volume unit of different torsion bars η_u are related as square values of their nominal shear stresses.

Taking into account that the T-80U tank has a the slightly lower specific work load capacity, but a considerably higher positive vertical spring travel and more convenient frequency of free bounce oscillations than the T-55 and the M-60A1, there is a need to rank it higher than the mentioned tanks. The same apllies to the LEOPARD 2 in the comparison to the T-72.

Finally, the definitive ranking of main battle tanks with the torsion bar suspension from the aspect of elasticity level is: M-1 ABRAMS, T-90S, LEOPARD 2, T-72, LEOPARD 1, T-80U, M-60A1, T-55 and AMX-30.

Conclusion

The described procedure of calculation provides enough correct results which enable obtaining objective characteristics of elasticity of combat tracked vehicles with the torsion bar suspension. The presented results can be used for comparisons of different suspension systems and for considerations of further development.

References:

[1] STEVANOVIĆ,R. Potencijalna energija kao jedan od kriterijuma za izbor parametara torzionog sistema oslanjanja borbenih guseničnih vozila. Beograd, *Naučnotehnički pregled,* 2002, vol.XLII, no 2, pp.26-32.

[2] OGORKIEWICZ,R. Teledyne's Super M-60 Battle tank. *International Defense Review,* 1980, vol.13, no.8, pp.1237-1241.

[3] LEBLANC. *Mobilité des chars de combat*. Cours protesse à l'École et Techniques d'Armement, 3ème année, 1975.

[4] GREWIG,K. Fahrwerke gepanzarter Kampffahrzeuge. *Wehrtechnik,* 1985, no.2, pp.52-56.

[5] DROSEN,E. Gleiskettenfahrzeuge für dem Getechtsfeld. *Soldat und Technik*, 1985, vol.28, no.2, pp. 86-90.

[6] SCHREIER,F. A tank designed to cost – the U.S. Army's XM1. *International Defense Review*, 1977, vol.10, no.3, pp.459-468.

[7] VITAS,D., TRBOJEVIĆ,M. *Mašinski elementi 1*. Naučna knjiga, Beograd, 1966.

Received: 14.5.2003

Karakteristike elastičnosti torzionih sistema oslanjanja osnovnih tenkova i ocena izvedenih rešenja

Iz dostupnih podataka i uz pomoć poznatih formula, određeni su osnovni parametri sistema oslanjanja većeg broja osnovnih tenkova sa torzionim sistemom oslanjanja. Na osnovu zaključaka izvedenih iz predstavljenih grafika i tabele, data je ocena karakteristika elastičnosti torzionih sistema oslanjanja značajnijih tipova osnovnih tenkova.

Ključne reči: **tenk, sistem oslanjanja, torzioni štap, elastičnost, potencijalna energija.**

Estimation des caractéristiques d'élasticité des suspensions à barre de torsion chez les chars de combat

Les paramètres principaux des suspensions à barre de torsion de plusieurs types de chars de combat sont déterminés à la base de données disponibles et à l'aide de formules connues. À la base des conclusions dérivées des diagrammes et tableaux présentés les caractéristiques d'élasticité des suspensions à barre de torsion chez quelques types importants de chars de combat sont estimées.

Mots-clés: **char de combat, suspension, barre de torsion, élasticité, énergie potentielle.**