

Homologacija elementov prototipnega vozila

Test Certificate of Elements for Vehicle Prototype

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Za vključitev vozila v slovenski vozni park je treba pridobiti homologacijski dokument vozila. V primeru izdelave unikatnega prototipa vozila je dolžnost izdelovalca, da zagotovi ustrezna potrdila o trdnostni in/ali funkcijski ustreznosti vgrajenih elementov v vozilo.

V prispevku je opisan trdnostni preračun nosilnih elementov podvozja vozila Šinkovec MM. Za pridobitev kolesnih obremenitev je bil zgrajen večmasni dinamični model vozila, na podlagi simuliranja vožnje v obratovalnih razmerah pa so bile te obremenitve tudi izračunane. Obremenitve so bile nato vnešene na elemente prednje in zadnje obese, čemur je sledil računalniško podprti trdnostni preračun zahtevanih elementov. Preverjala se je možnost trenutne in utrujenostne porušitve opazovanih elementov.

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(Ključne besede: vozila, podvozja vozil, izračuni trdnostni, homologacija)

For a vehicle to be registered for use in Slovenia it is necessary to obtain a motor vehicle approval document. In the case of manufacturing a unique prototype vehicle it is the task of the manufacturer to provide all the necessary certificates relating to the strength and/or functional suitability of the built-in elements.

This paper presents strength calculations on the chassis elements of the Šinkovec MM vehicle. In order to obtain the data on possible loads acting on the wheels, a multi-mass dynamic model of the vehicle has been developed and on the basis of the simulation of roads under different driving conditions these loads were then calculated. They were then applied to elements of the front and rear suspension, and a computer supported strength analysis of the elements was made. The possibility of both instant and fatigue failure of the discussed elements was verified.

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(Keywords: vehicle, chassis, strength calculations, homologation)

0 UVOD

Za trdnostni preračun nosilnih elementov konstrukcij vedno potrebujemo obremenitve, ki jih morajo ti elementi prenesti. Za trdnostni preračun nosilnih elementov podvozja športnega vozila MM so bile te obremenitve pridobljene s pomočjo simuliranja na osebnem računalniku. Na podlagi teh izračunov je sledil trdnostni preračun vseh obes, pri čemer se je preverjala nevarnost trenutne in utrujenostne porušitve.

Kolesne obremenitve so bile pridobljene z analizo dinamike dveh modelov vozila. Pri prvem modelu je bilo vzmetenje vozila zanemarljivo, izračunane pa so bile mogoče obodne sile na kolesa pri zaviranju oz. pospeševanju in bočne sile pri vožnji v ovinek v ustaljenih razmerah. V drugem modelu – vzmetenje vozila – je bila analizirana dinamika vozila pri vožnji čez oviro. Iz tako

0 INTRODUCTION

For strength calculations of structural supporting elements it is necessary to know the loads which the supporting elements must sustain. For the strength calculations of the suspension elements of the Šinkovec MM vehicle, the loads were determined by computer simulation. On the basis of these simulations the strength calculations of all the wishbone suspensions were performed to assess the risk of instant and fatigue failure.

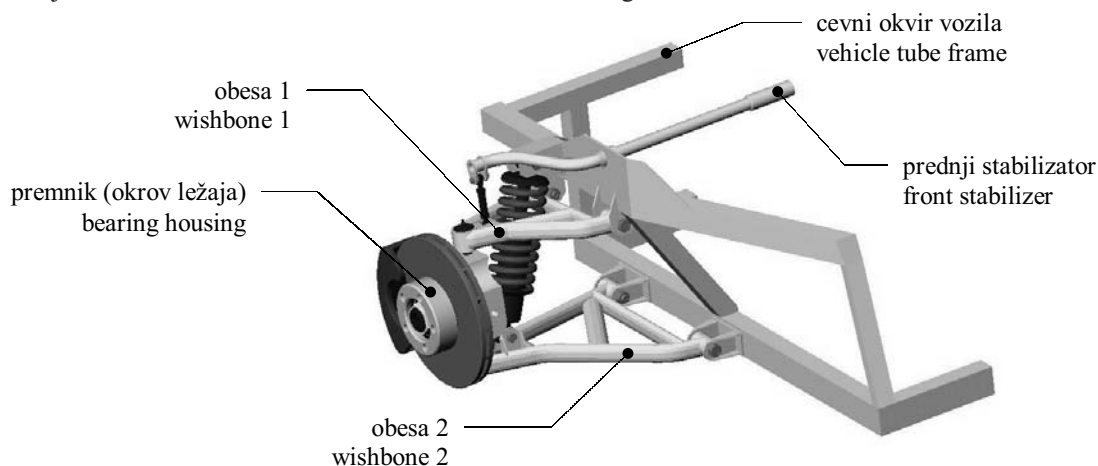
By analysing two dynamic vehicle models, the wheel loads were determined. In the first dynamic model of the vehicle no suspension damper or spring elements were considered and the model was required only for the calculation of the possible tangential forces acting on the wheels during vehicle braking or acceleration and for the possible lateral forces for a vehicle driving on a bend. Second model was the multi-mass dynamic model of a vehicle with all the suspension elements, and this model was

dobljenih obremenitev so bile izbrane največje obremenitve, ki so bile uporabljene za obremenitve modelov obes oz. za izračun poteka napetosti po postopku končnih elementov (MKE).

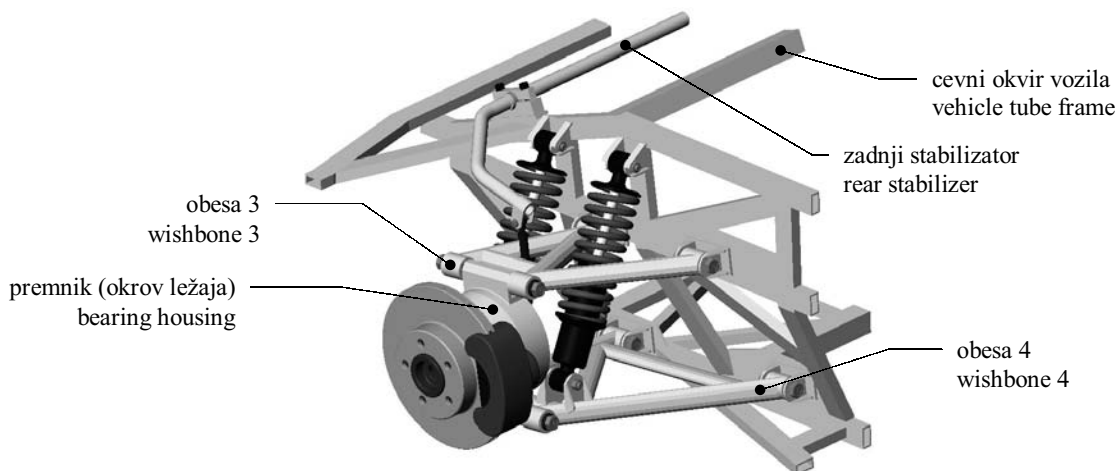
Pri preverjanju na utrujenostne porušitve so bile upoštevane povprečne obremenitve med življenjsko dobo, ki so bile izračunane na podlagi izkustvenih faktorjev ([1] do [3]).

1 MOGOČE KOLESNÉ OBREMENTITVE

Imena nekaterih elementov obešenja so zaradi jasnosti in lažjega razumevanja podana na naslednjih dveh slikah:



Sl. 1. Tridimenzionalni kinematični model prvega desnega obešenja
Fig. 1. Three-dimensional kinematic model of front right suspension



Sl. 2. Tridimenzionalni kinematični model zadnjega desnega obešenja
Fig. 2. Three-dimensional kinematic model of rear right suspension

1.1 Največje kolesne obremenitve

Največje kolesne obremenitve so bile uporabljene za trdnostni preračun posameznih obes pri preverjanju na trenutno porušitev.

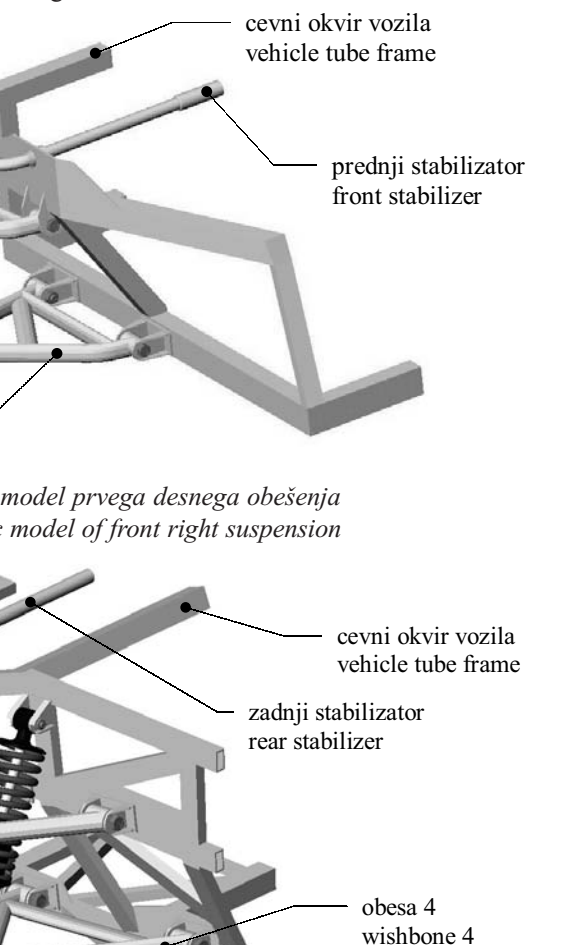
Za iskanje najbolj neugodnih režimov vožnje je bil uporabljen togi model vozila (sl. 3). Analizirana je bila kinematika vožnje pri močnem

the basis for the analysis of the vehicle's dynamic response when hitting a bump on the road. Maximum loads were then chosen for loading the wishbone suspensions and a computer-aided stress calculation was performed by the final elements method (FEM) analysis.

In the case of verifying fatigue failure the average loads during the fatigue life were used which were calculated on the basis of empirical factors ([1] to [3]).

1 POSSIBLE WHEEL LOADS

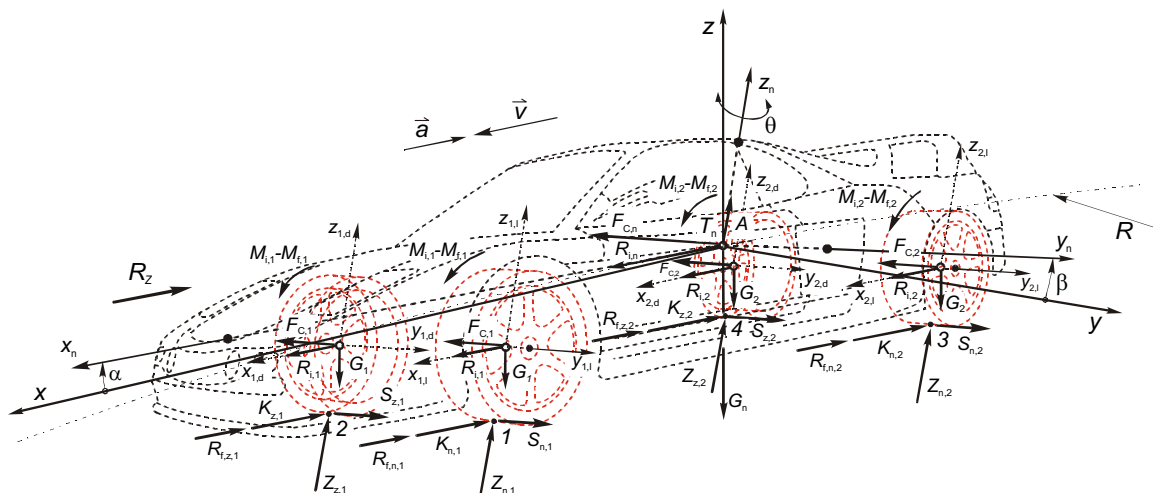
In the following two figures for reasons of clarity the names of some of the suspension elements are given:



1.1 Maximum wheel loads

Maximum wheel loads were used for strength calculations of all wishbones where the possibility of instant failure was checked.

For investigating the harshest driving conditions the rigid vehicle model was used (Fig. 3). The kinematics of driving under severe breaking and



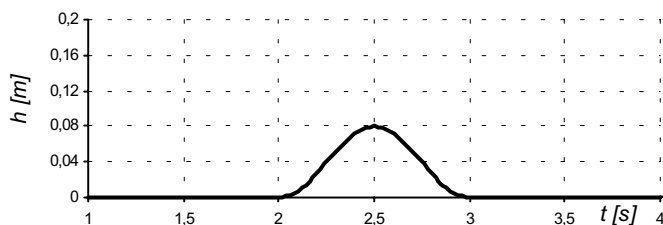
Sl. 3. Togi dinamični model vozila
Fig. 3. Rigid dynamic vehicle model

zaviranju oz. pospeševanje vozila po ravni cesti ali v ovinek.

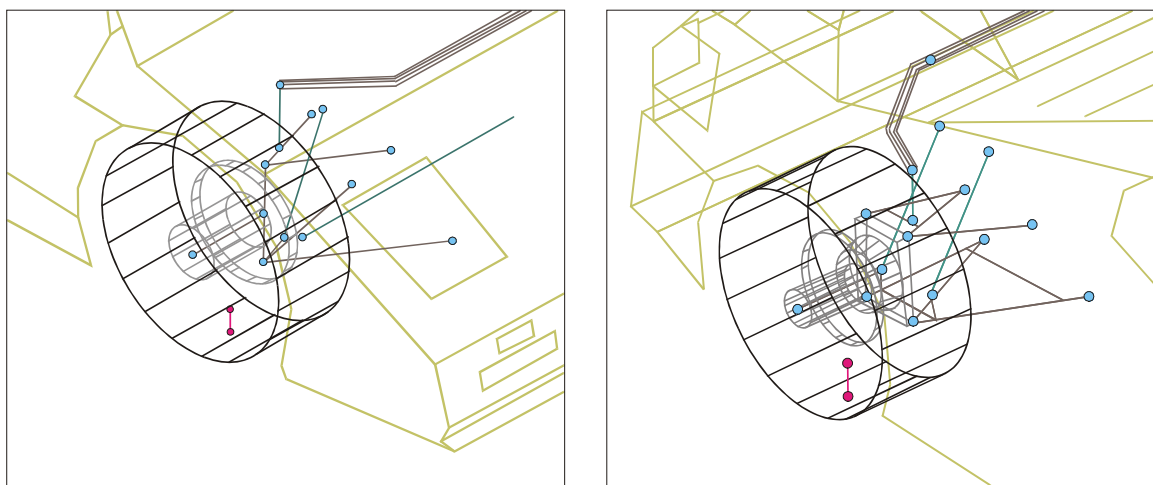
Največje obremenitve so bile pridobljene z analizo dinamike večmasnega dinamičnega modela vozila (sl. 5) pri najbolj neugodnih voznih pogojih (hitrost v , pospešek a , trenutni polmer ovinka R , klanec α , nagib cestišča β , koeficienti sojemanja (trenja) v točki nakotaljenja pnevmatike po cestišču μ_H in $\mu_{S,maks}$, itn.) in hkratnem vznujanju s časovno funkcijo, ki je popisovala grbino na cesti (sl. 4).

hard acceleration on a straight road and on a bend were analysed.

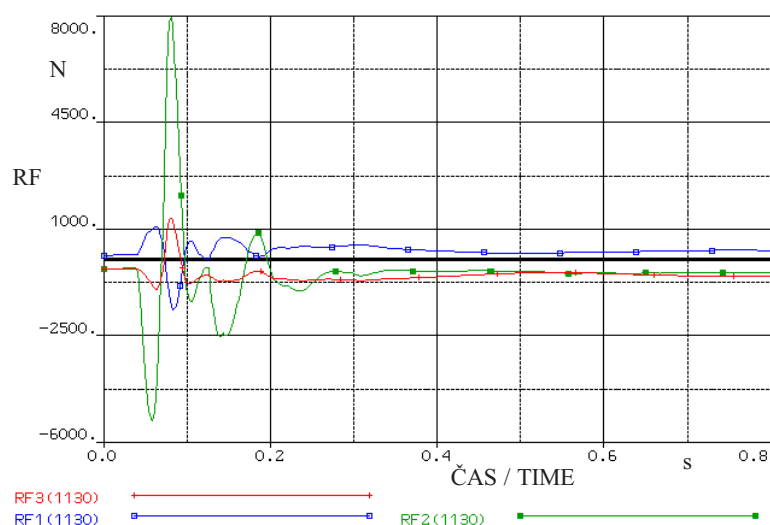
Maximum loads were acquired by analysis of the multi-mass dynamic model of vehicle (Fig. 5) under the harshest driving conditions and a simultaneous forced time function that represented a bump on the road (Fig. 4). The harshest driving conditions were represented by velocity v , acceleration a , present radii of the road bend R , acclivity α , banking of the road β , coefficient of friction at the rolling point of tires μ_H and μ_S .



Sl. 4. Predstavitev grbine na cestišču
Fig. 4. Representation of a bump on the road



Sl. 5. Detajl prednjega in zadnjega obešenja večmasnega dinamičnega modela vozila
Fig. 5. Detail of front and rear suspension of multi-mass dynamic vehicle model



Sl. 6. Primer diagrama dinamičnih sil v zgornjem krogelnem zgibu prednjega desnega obešenja pri analizi dinamičnega odziva vozila pri naletu na oviro

Fig. 6. Example of dynamic-force diagram in front upper spherical joint while hitting a bump on the road

Na podlagi izračunanih dinamičnih odzivov vozila pri naletu na grbino, so za trdnostni preračun posameznih obes pri preverjanju na trenutno porušitev, bile preverjane naslednje obremenitve:

On the basis of the calculated dynamic vehicle response while hitting a bump on the road the strength calculations for the possibility of instant failure for the following loads were performed:

Preglednica 1. Največje obremenitve obese 1

Table 1. Maximum loads on wishbone 1

i	pogoji vožnje / driving conditions	grbina / bump	X_{P1} N	Y_{P1} N	Z_{P1} N
10a	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-10,76\text{m/s}^2$, $v=100\text{km/h}$, $\mu_H=1,067$, $\mu_S=0,167$	┌	7394	2898	-600
10b			-1299	8727	1890
22	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-6,3\text{m/s}^2$, $v=153\text{km/h}$, $\mu_H=0,587$, $\mu_S=0,592$	└	4697	-4505	-865

Pri izračunu statičnega ravnotežja mirujočega vozila so normalne sile Z na prednji kolesi 3480 N in na zadnji kolesi 4370 N. Zaradi boljše predstave o velikosti obremenitev (preglednica 1), so bile obremenitve normirane z vrednostjo normalne obremenitve Z enega kolesa (preglednica 2).

Calculations of the equilibrium forces for the standing vehicle shown that the normal force on the front wheels was 3480 N and on the rear wheels 4370 N. For the sake of a better presentation of the force amplitude (Table 1) the forces were normalized with the normal force Z on one wheel (Table 2).

Preglednica 2. Največje normirane obremenitve obese 1

Table 2. Maximum normalized loads on wishbone 1

i	pogoji vožnje / driving conditions	grbina / bump	X_{P1} N	Y_{P1} N	Z_{P1} N
10a	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-10,76\text{m/s}^2$, $v=100\text{km/h}$, $\mu_H=1,067$, $\mu_S=0,167$	┌	2,12	0,83	-0,17
10b			-0,37	2,51	0,54
22	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-6,3\text{m/s}^2$, $v=153\text{km/h}$, $\mu_H=0,587$, $\mu_S=0,592$	└	1,35	-1,26	-0,25

Preglednica 3. Največje obremenitve obese 2

Table 3. Maximum loads on wishbone 2

i	pogoji vožnje / driving conditions	grbina / bump	X_{P2} N	Y_{P2} N	Z_{P2} N
10	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-10,76\text{m/s}^2$, $v=100\text{km/h}$, $\mu_H=1,067$, $\mu_S=0,167$	┌	-22250	7858	-8327
28	$R=60\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-1,78\text{m/s}^2$, $v=90\text{km/h}$, $\mu_H=0,156$, $\mu_S=0,693$	┌	-113	-4350	-7532
22	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-6,3\text{m/s}^2$, $v=153\text{km/h}$, $\mu_H=0,587$, $\mu_S=0,592$	└	-11790	16970	-7055

Preglednica 4. Največje normirane obremenitve obese 2

Table 4. Maximum normalized loads on wishbone 2

i	pogoji vožnje / driving conditions	grbina / bump	X_{p2} N	Y_{p2} N	Z_{p2} N
10	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-10,76\text{m/s}^2$, $v=100\text{km/h}$, $\mu_H=1,067$, $\mu_S=0,167$	[	-6,93	2,26	-2,39
28	$R=60\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-1,78\text{m/s}^2$, $v=90\text{km/h}$, $\mu_H=0,156$, $\mu_S=0,693$	[	-0,03	-1,25	-2,16
22	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=12,0^\circ$, $a=-6,3\text{m/s}^2$, $v=153\text{km/h}$, $\mu_H=0,587$, $\mu_S=0,592$	[	-3,39	4,88	-2,03

Preglednica 5. Največje obremenitve obese 3 in 4

Table 5. Maximum loads on wishbones 3 and 4

i	pogoji vožnje / driving conditions	grbina / bump	Z N	K N	S N
19	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=0,0^\circ$, $a=-6,05\text{m/s}^2$, $v=123\text{km/h}$, $\mu_H=0,579$, $\mu_S=0,595$	[	-5633	8338	-8569
42	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=0,0^\circ$, $a=3,57\text{m/s}^2$, $v=122\text{km/h}$, $\mu_H=0,603$, $\mu_S=0,585$	[	-7501	-10520	-10210
43	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=0,0^\circ$, $a=6,431\text{m/s}^2$, $v=93\text{km/h}$, $\mu_H=0,961$, $\mu_S=0,36$	[	-6985	-15020	-5625

Preglednica 6. Največje normirane obremenitve obese 3 in 4

Table 6. Maximum normalized loads on wishbones 3 and 4

i	pogoji vožnje / driving conditions	grbina / bump	Z N	K N	S N
19	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=0,0^\circ$, $a=-6,05\text{m/s}^2$, $v=123\text{km/h}$, $\mu_H=0,579$, $\mu_S=0,595$	[	-1,29	1,91	-1,96
42	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=0,0^\circ$, $a=3,571\text{m/s}^2$, $v=122\text{km/h}$, $\mu_H=0,603$, $\mu_S=0,585$	[	-1,72	-2,41	-2,34
43	$R=200\text{m}$, $\alpha=0,0^\circ$, $\beta=0,0^\circ$, $a=6,431\text{m/s}^2$, $v=93\text{km/h}$, $\mu_H=0,961$, $\mu_S=0,36$	[	-1,6	-3,44	-1,29

1.2 Spremenljive kolesne obremenitve

Za trdnostni preračun posameznih obes pri preverjanju na utrujenostno porušitev smo uporabili spremenljive obremenitve, ki se pojavijo na vozilu pri vožnji naravnost po srednje dobri poti. Velikost teh obremenitev je bila izračunana na podlagi izkustvenih dejavnikov po literaturi ([1] in [2]).

1.2 Variable wheel loads

For strength calculations in the case of fatigue failure the variable loads that emerge during the driving on medium-good roads were used. The magnitudes of these loads were calculated on the basis of the empirical factors in references ([1] and [2]).

2 PRIKAZ REZULTATOV TRDNOSTNEGA PRERAČUNA

Trdnostni preračun posameznih obes je bil izveden z MKE, prikazana napetostna polja po posameznih obesah pa so Misesove primerjalne napetosti.

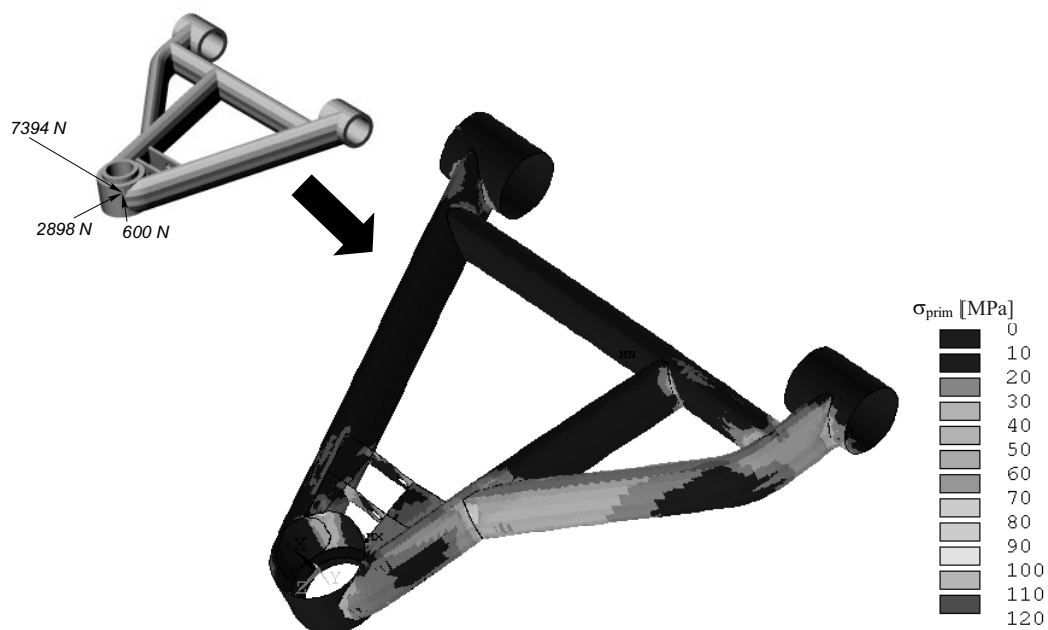
2 RESULTS OF STRENGTH CALCULATION

The computer-aided strength calculation was performed by FEM analysis. Stress fields on the wishbones represent the Mises equivalent stress.

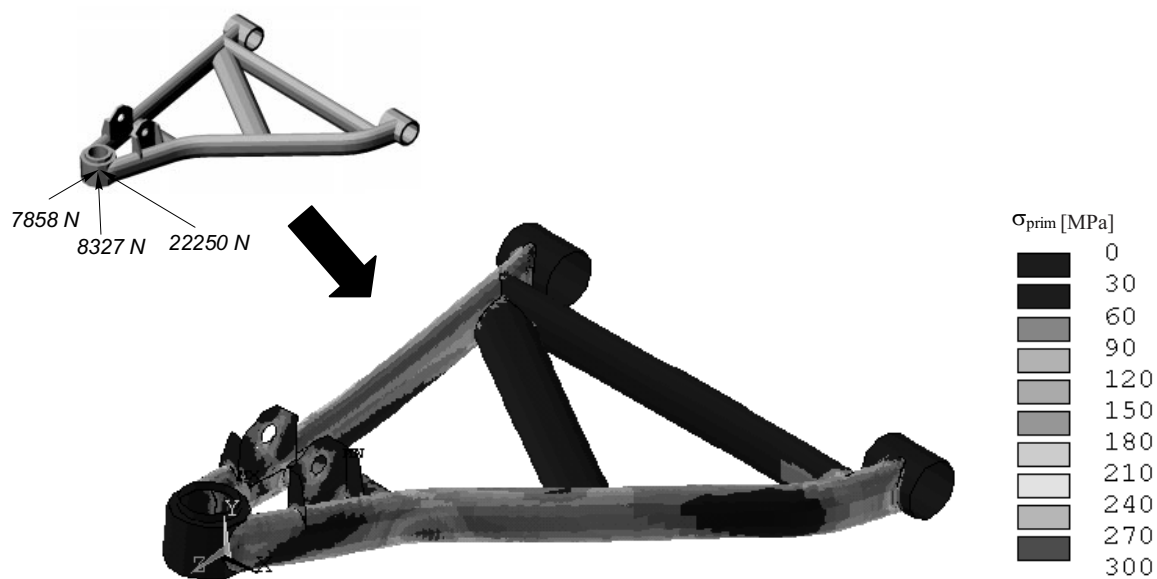
Preglednica 7. Spremenljive obremenitve posameznih obes

Table 7. Variable loads on individual wishbones

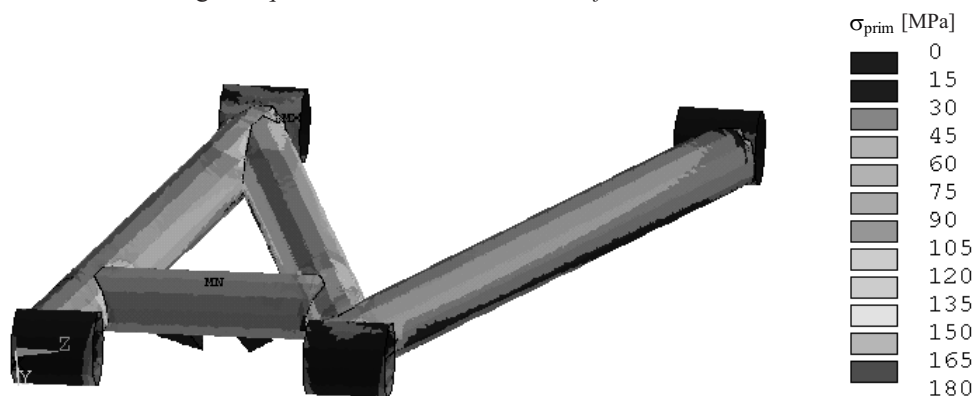
i	pogoji vožnje / driving conditions	obesa / wishbone	X N	Y N	Z N
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P1	-206	2339	20
			-124	485	0
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P2	808	-3410	-6426
			451	-401	-2756
i	pogoji vožnje / driving conditions	obesa / wishbone	Z N	K N	S N
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P3	7195	1469	605
			4212	1469	-605
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P4	7195	1469	605
			4212	1469	-605



Sl. 7. Primerjalne napetosti na obesi 1 za obremenitveni primer $i=10a$
Fig. 7. Equivalent stress on wishbone 1 for load case $i=10a$



Sl. 8. Primerjalne napetosti na obesi 2 za obremenitveni primer $i=10a$
Fig. 8. Equivalent stress on wishbone 2 for load case $i=10a$



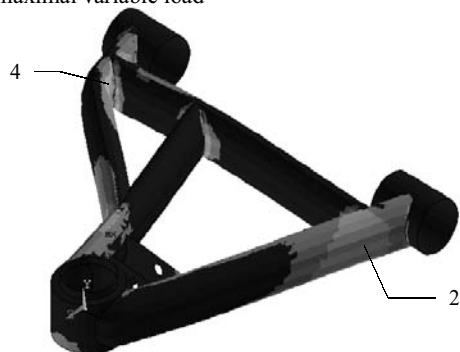
Sl. 9. Primerjalne napetosti na obesi 3 za obremenitveni primer $i=43$
Fig. 9. Equivalent stress on wishbone 3 for load case $i=43$



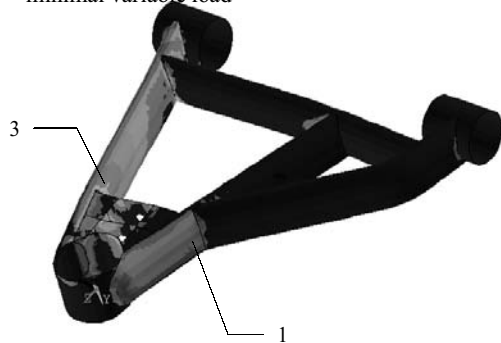
Sl. 11. Primerjalne napetosti na obesi 4 za obremenitveni primer $i=43$

Fig. 11. Equivalent stress on wishbone 4 for load case $i=43$

največja spremenljiva obremenitev
maximal variable load



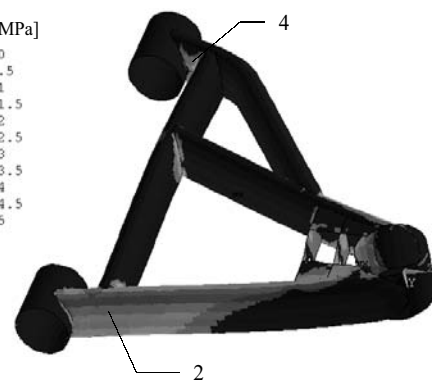
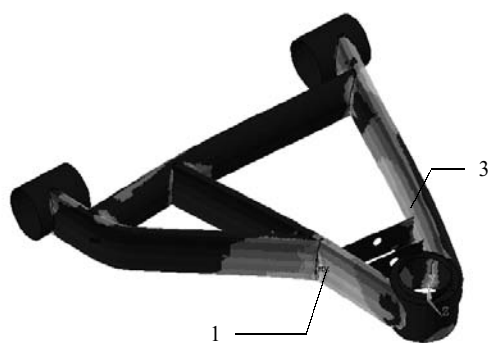
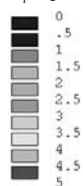
najmanjša spremenljiva obremenitev
minimal variable load



σ_{prim} [MPa]



σ_{prim} [MPa]



Sl. 12. Potek primerjalnih napetosti na premi 1 pri spremenljivih obremenitvah

Fig. 12. Equivalent stress on wishbone 1 for variable loads

2.1 Preverjanje na trenutno porušitev

V celoti dobimo največje napetosti pri obremenitvenem primeru $i=10a$, zato je prikazano napetostno polje le za ta primer. Obremenitveni primer $i=10a$ označuje vožnjo pri močnem zaviranju

2.1 Instant failure verification

The result of the stress calculation for the load case $i=10a$ is shown because the highest overall stress appeared in this load case. The case represented driving under severe braking while hitting a bump

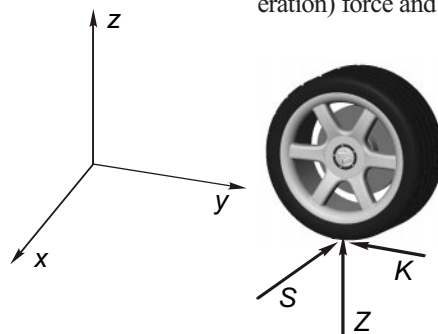
Preglednica 8. Normirane spremenljive obremenitve posameznih obes

Table 8. Normalized variable loads on individual wishbones

i	pogoji vožnje / driving conditions	obesa / wishbone	X N	Y N	Z N
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P1	-0,06	0,67	0,01
			-0,04	0,14	0
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P2	0,23	-0,98	-1,85
			0,13	-0,12	-0,79
i	pogoji vožnje / driving conditions	obesa / wishbone	Z [N]	K [N]	S [N]
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P3	1,65	0,34	0,14
			0,96	0,34	-0,14
/	spremenljive obremenitve, $R=10000\text{m}$, $\alpha=\beta=0^\circ$, $a=0\text{m/s}^2$, $v=10\text{km/h}$ [2]	P4	1,65	0,34	0,14
			0,96	0,34	-0,14

v ovinek in sočasen nalet na grbino z zunanjim prednjim kolesom (preglednica 1 in preglednica 2). Zunanje kolo je kolo na zunanji strani ovinka.

Obremenitveni primer $i=43$ označuje vožnjo pri največjem pospeševanju na ravni cesti in sočasen nalet na grbino z zadnjim kolesom. Vpetje zadnjega obešenja za trdnostni preračun obeh obes je bilo izvedeno tako, da je vnos obremenitev potekal prek točke nakotaljenja zadnje pnevmatike po cestišču. Te sile so naslednje: Z – normalna sila, K – obodna (sila zaviranja / pogonska sila) in S – bočna sila (sl. 10).



Sl. 10. Prikaz sil v točki nakotaljenja pnevmatike po cestišču

Fig. 10. Forces acting on the wheel road contact point

on the road with outer front wheel (Table 1 and Table 2). Outer wheel is the wheel on outer side of the road bend.

Load case $i=43$ represented driving with maximum acceleration on a straight road while hitting a bump on the road, with the rear wheel analysed. The entry point for loads in the strength calculation on the rear suspension elements was the wheel road contact point because of the complexity of the rear suspension geometry. Forces acting on the wheel road contact point are: Z – normal force, K – tangential (braking / acceleration) force and S – lateral force (Fig. 10).

2.2 Preverjanje na utrujenostno porušitev

Pri preverjanju na utrujenostno porušitev so bile vse obese obremenjene na dveh obremenitvenih ravneh (preglednica 7 in preglednica 8). Po izračunu primerjalnih napetosti se bile na vsaki obesi določene po 4 točke z neugodno kombinacijo napetosti (sl. 12). Za te točke so bile izračunane vrednosti srednje primerjalne napetosti, ki so bile nato vrisane v diagram dinamične trdnosti (*Smithov diagram*) za jeklo Fe 360B. Razmerje dinamičnosti napetosti:

$$r_i = \frac{\sigma_{\min,i}}{\sigma_{\max,i}} \quad (1).$$

Srednja primerjalna napetost:

$$\sigma_{sr,i} = \frac{\sigma_{\min,i} + \sigma_{\max,i}}{2} = \sigma_{\max,i} \frac{r_i + 1}{2} \quad (2).$$

2.2 Fatigue failure verification

During fatigue failure control all the wishbones were loaded on two levels (Table 7 and Table 8). On each of the wishbones the four points with the worst stress levels were determined (Fig. 12). For these points the average stress levels were calculated and used as the input data for the Smith diagram – diagram of dynamic stiffness of the steel Fe 360B.

Factor of dynamic stress:

Average equivalent stress level:

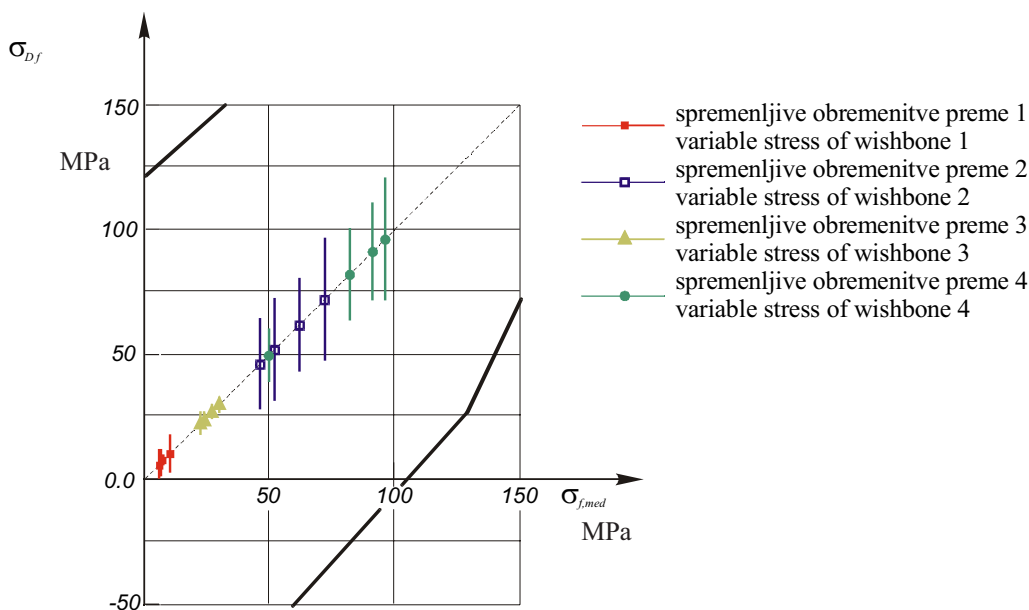
Srednje vrednosti napetosti označenih točk (sl. 12) so bile določene po enačbi (1) in (2):

$\sigma_{\max,1} = 18 \text{ MPa}$	$\sigma_{\min,1} = 3 \text{ MPa}$	$\Rightarrow$	$r_1 = 0,167$	$\Rightarrow$	$\sigma_{sr,1} = 10,5 \text{ MPa}$
$\sigma_{\max,2} = 10 \text{ MPa}$	$\sigma_{\min,2} = 3,5 \text{ MPa}$	$\Rightarrow$	$r_2 = 0,35$	$\Rightarrow$	$\sigma_{sr,2} = 6,75 \text{ MPa}$
$\sigma_{\max,3} = 12 \text{ MPa}$	$\sigma_{\min,3} = 3,5 \text{ MPa}$	$\Rightarrow$	$r_3 = 0,29$	$\Rightarrow$	$\sigma_{sr,3} = 7,75 \text{ MPa}$
$\sigma_{\max,4} = 12 \text{ MPa}$	$\sigma_{\min,4} = 0 \text{ MPa}$	$\Rightarrow$	$r_4 = 0,0$	$\Rightarrow$	$\sigma_{sr,4} = 6 \text{ MPa}$

Srednje primerjalne napetosti za detajle na preostalih obesah niso kritične, saj so vse manjše kakor pri obesi 1. Pri analizi rezultatov dinamičnega preračuna je pomemben položaj spremenljivih napetosti v diagramu dinamične trdnosti, kar prikazuje slika 13. Prikazan je diagram dinamične trdnosti za jeklo *Fe 360 B*. To je jeklo, iz katerega so bile zvarjene rezervne obese, ki so bile tudi model za trdnostni preračun.

Average stress levels for marked points (Fig. 12) were calculated using equation (1) and (2):

Average equivalent stress for the details on other wishbones were all smaller than for wishbone 1 and they were not critical. When we analysed the possibility of fatigue failure, the position of the average stress in the Smith diagram is important as we can see in Fig. 13. The Smith diagram for the steel *F 360B* is shown from reference, and this is the steel that was used for spare wishbones.



Sl. 13. Prikaz dinamičnih primerjalnih napetosti v diagramu dinamične trdnosti za jeklo *Fe 360 B*
Fig. 13. Equivalent variable stress in the Smith diagram for steel *F 360B*

3 SKLEP

Izvedena trdnostna analiza na nevarnost trenutne in utrujenostne poškodbe je pokazala, da so vse obese trdnostno ustrezne za vozilo Šinkovec MM.

Poteki primerjalnih napetosti po posameznih obesah kažejo na povečanje vrednosti primerjalne napetosti okoli ostrorobnih geometrijskih stikov, kar je verjetno posledica samega modela MKE posameznih obes, saj imajo elementi okoli teh stikov določeno stopnjo popačenja oz. niso idealni. Na teh mestih so na rezervnih obesah (obese, ki so rabile za modeliranje modelov MKE) razmeroma debeli zvari, ki po eni strani verjetno omilijo omenjene konice napetosti, po drugi strani pa zvarno mesto že samo po sebi poveča verjetnost za pojav zareznihih učinkov.

Za natančnejšo sliko dogajanja bi bilo treba izvesti poglobljene teoretične raziskave (podrobneje modelirati geometrijo modela okoli zvarnih mest in točk vpetja modela na okvir vozila, preveriti vpliv

3 SUMMARY

The performed strength analysis of the possibility of instant and fatigue failure confirmed the suitability of all wishbones for the Šinkovec MM vehicle.

The distribution of equivalent stress on separate wishbones showed amplified stress levels on sharp geometric edges that were probably the consequence of wishbones' FEM models, because FEM elements on those edges were in some way deformed and not ideal. On the other hand, there were robust welds on the same edges, which were not modeled in the FEM analysis, and they could extenuate the above mentioned amplified stress levels but at the same time also amplify the probability of a notch effect.

For a thorough analysis of the problem some enhanced tasks should be carried out (detailed modelling of the geometry around the weld position and areas where the wishbones are fixed on the vehicle

elastičnosti okvira vozila na porazdelitev napetosti v posameznih obesah, preverjati materialne lastnosti po varjenju itn.) in izvesti meritve dejanskih obremenitev v kritičnih mestih na samem prototipu vozila v dejanskih obratovalnih razmerah.

tube frame, the influence of the tube frame elasticity on wishbone stress distribution and material properties after welding should be controlled, etc.). For confirmation of the stress analysis some measurement of the stress on the vehicle wishbones in real driving conditions should be performed.

4 UPORABLJENE OZNAKE 4 USED NOTATION OF SYMBOLS

Indeksi, ki so lahko poleg vsake oznake v različnih kombinacijah:	Indices that could be beside all symbols in different combinations:	
obesa 1 (2, 3, 4)	$i = 1$ (2, 3, 4)	wishbone 1 (2, 3, 4)
koordinatna os x (y, z)	$i = x$ (y, z)	coordinate axis x (y,z)
sprednja obesa	$i = j = 1$	front suspension
zadnja obesa	$j = i = 2$	rear suspension
desno kolo	d	right wheel
levo kolo	l	left wheel
notranje kolo	n	inner wheel
zunanje kolo	z	outer wheel
Oznake:	Symbols:	
višina ovire	H_0	m bump height
zavorna sila na kolesu (tudi sila pospeševanja)	K	N breaking force on the wheel (also acceleration force)
dolžina ovire	L_0	m bump length
polmer ukrivljenosti poti, po kateri se giblje težišče vozila	R	m radii of curved trajectory on which the center of mass of the vehicle is moving
bočna sila na kolo	S	N lateral force on the wheel
sile na obese v lokalnih koordinatnih sistemih posameznih obes	X (Y, Z)	N forces on wishbones in local system of coordinates
osna (normalna) sila na kolo	Z	N normal force on the wheel
vektor pospeška vozila pri zaviranju ali pospeševanju	$\vec{a}$	m/s ² acceleration vector of vehicle during breaking or acceleration
razmerje dinamičnosti obremenitve za opazovano točko i	r_i	factor of dynamic loading for observed point i
vektor hitrosti vozila	$\vec{v}$	m/s velocity vector of vehicle
kot klanca cestišča	α	°,rd acclivity angle of road
kot nagiba cestišča	β	°,rd banking angle of road
koeficient sojemanja	μ_H	coefficient of friction
specifična bočna sila - bočni koeficient sojemanja	μ_S	specific lateral force - specific lateral coefficient of friction
dinamična trdnost jekla za upogib	σ_{Df}	MPa fatigue strength for bending
srednja napetost za upogib	$\sigma_{f,med}$	MPa average bending stress
maksimalna primerjalna napetost pri spremenljivih obremenitvah v točki i	$\sigma_{max,i}$	MPa maximum Von-Misses stress at variable loads for observed point i
najmanjša primerjalna napetost pri spremenljivih obremenitvah v točki i	$\sigma_{min,i}$	MPa minimum equivalent stress at variable loads for observed point i
primerjalna napetost	σ_{prim}	MPa equivalent stress
srednja primerjalna napetost pri spremenljivih obremenitvah v točki i	σ_{sr}	MPa average equivalent stress at variable loads for observed point i

5 LITERATURA

5 REFERENCES

- [1] Mitschke, M. (1972) Dynamik der Kraftfahrzeuge. *Springer-Verlag, Berlin, Heidelberg, New York.*
- [2] Jurejevčič, T., M. Fajdiga (1988) Računska tehnika kod optimiranja podvozja avtobusa. *MVM Saobštenje, XIV-83*, str. 118-124.
- [3] Nagode, M., M. Fajdiga (1997) The influence of variable operating conditions upon the general multimodal Weibull distribution. *Reliability Engineering and System Safety 64*, str. 383-389.
- [4] Beitz, W., K.-H. Küttner (1995) Dubbel - Taschenbuch für den Maschinenbau, 18. Auflage. *Springer-Verlag, Berlin, Heidelberg, New York.*

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