DESIGN PROCEDURE FOR LOADING CAPACITY CALCULATIONS FOR CLASSIC AUTOMOBILE DIFFERENTIALS

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SHRNUTÍ

Článek se zabývá návrhem dimenzování ozubení pro různé typy automobilních diferenciálů a děličů momentů. V navrženém postupu jsou použity nejnovější výpočtové postupy podle norem ISO a DIN. Do výpočtu jsou aplikována hypotetická spektra provozního zatížení, která jsou odvozena od vlastních měření.

KLÍČOVÁ SLOVA: OZUBENÍ, DIFFERENCIÁL, DĚLIČ MOMENTU, ÚNOSNOST, SPEKTRUM ZATÍŽENÍ, ŽIVOTNOST

ABSTRACT

This article deals with the procedure for loading capacity calculation for various types of automobile differentials and torque dividers. The latest calculation methods in accordance with ISO and DIN standards are applied in this procedure. Hypothetical spectra of working load derived from own measurements are applied.

KEYWORDS: GEARS, DIFFERENTIAL, TORQUE DIVIDER, LOADING CAPACITY, LOADING SPECTRUM, SERVICE LIFE

The proposed calculation procedures relate to a differential with spur and bevel gears and various division ratios of torque. The division ratio is defined by relationships $J_D = T_1/T_2 \geq 1$; $T_1 \geq T_2$; $T_U = T_1 + T_2$, where indexes 1, 2 are marked with an output torque, and their sum T_U is usually torque on the planet carrier. Gears of mechanical differentials are significantly different from other gear drives, because they have a specific mesh and method of loading.

The relative circumferential speeds are very low; for example, the gears of an axis differential have a relative circumferential velocity close to zero when driving straight ahead. Frequencies of load cycles at all load levels are very low and therefore the design approach differs from classic kinematic gears. The vehicle drive train scheme can be very diverse with a variety of differentials location, as shown in Table 1.

TABLE 1: Examples of basic car drive variants

TABULKA 1: Příklady základních variant pohonu automobilu

type of engine	scheme	caption
4×2		steered, not driven
4×4		steered, driven
6×6		not steered, not driven
8×8		not steered, driven
e.g. 10×4 10×8 12×4 12×8	These combinations are solved by adding more not driven rear axle to applications 8×4, 8×8 or driven rest connect without inter-axle differential.	Axial differential inter-axle divider inter-axle divider

FIGURE 1: Drive train scheme for 6×6 vehicles

OBRAZĚK 1: Schéma pohon u vozidla 6×6

The scheme for fully 6×6 vehicles is shown in Figure 1, where each differential type is marked.

D_D ...torque divider $J_D = T_Z/T_P \geq 1$
 T_Z ...drive of rear two axles (tandem) [Nm]
 T_P ...drive of front axle [Nm]
 D_M ...inter-axle differential (divider) $J_D = T_2/T_3 = 1$
 D_N ...axle differential $J_D = T_R/T_L = 1$
 $T_{R/L}$...driving torques for right (R) and left (L) axle shaft [Nm]

The following are examples of some mechanical differentials and their power loads. An example of a frequently used design

for an axial differential with bevel gears and four planets is shown in Figure 2.

The scheme and design of a differential with spur gears and division ratio $J_D = 1$ as shown in Figure 3 is used less frequently. The design of a differential lock that provides connection between carrier with sun gear is shown in Figure 3. The differential locks are used primarily in off-road vehicles, where there is a risk of reduced mobility when the wheels slip. When the differential lock is used on a road with high adhesion, parasitic torques arise and increases the gear load.

FIGURE 2: Scheme and design of axis differential with bevel gears
OBRÁZEK2: Příklad konstrukčního řešení kuželového osového diferenciálu

FIGURE 3: Scheme and design of a differential with spur gears
OBRÁZEK 3: Schéma a příklad konstrukčního řešení diferenciálu s čelními koly

FIGURE 4: Planetary differential (divider)
OBRÁZEK 4: Planetový diferenciál (dělič)

A torque divider (inter-axle differential) is used to drive the front and rear axles for $J_D \gg 1$. The division ratio is dependent on the engine and the total power load, or on the load ratio of the rear and front axles. The mechanical torque divider is implemented using the planet as shown in Figure 4.

The division ratio is $J_D \doteq 1.7 \sim 5$. Most transmissions have $J_D \doteq 2$ for application on a 4×4 with rear double mounting or 6×6 with simple mounting. The ratio of axle load when applied to a 6×6 with rear double mounting for off-road vehicles (dump trucks) can be considerably greater $J_D \doteq 3 \sim 5$. When deciding on the choice of divider the fact that in practically all driving modes the rear axle is overload due to acceleration, group, etc. should be taken into account. An applied example of inter-axle differential $J_D \doteq 2$ is shown in Figure 5.

FIGURE 5: Inter-axle differential with division ratio $J_D \doteq 2$
OBRÁZEK 5: Mezinápravový diferenciál s poměrem dělení $J_D \doteq 2$

The load of the differential for dimensioning of parts is very difficult to define. Maximum torque from the motor and adhesive load, which has an influence on the differential box (cage), is usually determined according to Table 1. Let's suppose that the differential is loaded from the engine torque acting on the differential cage as per the following equation:

$$T_{DM} = T_M \times i_{DM} \times m_D \times \eta_{DM}$$

where

- T_M maximum engine torque [Nm],
- i_{DM} maximum ratio between the engine and differential cage [-],
- m_D proportion of the torque corresponding to the controlled differential [-]
- $\eta_{DM} = 1$, for comparative calculation the efficiency is usually neglected.

For example, from Figure 1 it will follow that:

- drivetrain 4×2 $m_D = 1$ (for front and rear);
- drivetrain 4×4 $m_D = 0.5$ for divider $J_D = 1$
- drivetrain 6×6 for divider with performance $J_D \neq 1$, $m_{Dp} = 1/(1+J_D)$, $m_{Dz} = J_D/(1+J_D)$. This means that if $J_D = 2$ then $m_{Dp} = 0.33$ for front axle and $m_{Dz} = 0.67$ for rear axle.

If the vehicle isn't equipped with an inter-axle differential (divider), then hypothetically we can assume that the division ratio J_D is given by the axle load ratio $J'_D = G_z/G_p$. If the front axle drive can be turned off, then according to the cited relationships $m_p = 1/(1+J'_D)$ and $m_z = 1$ for drivetrain 4×4 or $m_z = 0.5$ for drivetrain 6×6 . Similarly, for the drivetrain 8×8 . This reasoning is valid providing that the inter-axle in a double-axle always has ratio $J_D = 1$.

Control of a differential torque transmission is performed by calculating adhesion torque according to the equation:

$$T_{DA} = G_N \times g \times \varphi \times R_d \times \frac{1}{i_N} = 9,81 \times G_N \times \frac{R_d}{i_N}, [\text{Nm}]$$

where

- G_N axle or double-axle load driven by differential [kg],
- g gravitational acceleration [$\text{m} \times \text{s}^{-2}$],
- $\varphi = 1$ agreed coefficient of adhesion [-],
- R_d dynamic radius of tire [m],
- i_N ratio between solved differential and tire [-].

Load of axle or double-axle is usually chosen according to the static weight distribution. The actual dynamic load changes in traffic. Rear axles are loaded more precisely in the limit state (maximum) such as when riding up a hill, overloaded or during acceleration. For evaluation the load capacity can be used for the front differential axle (axles) taking into account the static load $G_N = G_p$. To calculate the adhesion torque of the rear axle (axles) a higher static load can be assumed than the static load where the ratio is about $\delta_{NZ} = (1.1 \div 1.5)$ thus $G_N = (1.1 \div 1.5) G_p$, where G_p and G_z are nominal static axle (axles) loads. This is especially true where over loading of the rear axles is not considered. The

TABLE 2: Allowable load of differential torque

TABULKA 2: Mezní zatížení moment diferenciálu

torques ratio	torques size	Recommendations for design
$T_{DM} < T_{DA}$	$T_D = T_{DM}$	front and rear axle
$T_{DM} \gg T_{DA}$ by more than 50 %	$T_D = T_{DA}$ $T_D = (1,1 \div 1,5) T_{DA}$	only front axle rear axle
$T_{DM} \geq T_{DA}$ to 50 %	$T_D = \frac{T_{DM} + T_{DA}}{2}$	rear axle

TABLE 3: Parameters for determination of load of differentials
TABULKA 3: Parametry pro stanovení zatížení diferenciálů

Type of differential	Origin of force on drive differential element d [mm]	Coefficient of uneven force distribution [-]
Bevel	$d_m = m_n \times z$ middle pitch diameter of central wheel	$\delta = 1$ for $n = 2$ or for $n = 4$ – free pin $\delta \doteq 1.2$ for $n = 4$ – rigid pin
Spur (straight teeth)	$d = m_n \times z_{12}$ pitch diameter of central wheel	$\delta = 1$ for $n = 2, 3$
Planetary	$d = d_U \doteq m_n \times (z_1 + z_s)$ center distance between satellite pins	$\delta \doteq n / (n - 1)$ for $n > 3$

resulting agreed torques acting on a planet carrier (differential cage) for the comparative strength calculations are summarized for various operating conditions in Table 2.

The forces between the teeth of all the differentials are given by the equation:

$$F_T = \frac{2000 \times T_U \times \delta}{d \times n}, [\text{N}]$$

where

$T_U = T_D$ torque moment on the driving element of a differential given by Table 2 [Nm],

n number of planetary gears or pairs of planetary gears[-],

δ degree of inequality of a force distribution (Table 3)[-],

d reference or the pitch diameter of driving element of a differential [mm].

The rotation load of differential gears, such as number of cycles, differs significantly over a variety of vehicle driving conditions.

Rotations of the differential gear are theoretically zero when driving is straight. The spinning of the differential gear occurs under the following conditions:

- driven wheel tires have different rolling radii and different load,
- driven wheels or axles move along an unequal length of the roadway (curves, transverse and longitudinal irregularities),
- uneven traction of tires on one axle (or axles for inter-axle differentials) due to uneven adhesion or different characteristics of tire slip.

The rotation frequency of differential gears depends on speed, vehicle load, traffic density, climatic conditions and other factors (adhesion, uneven off-road). Determining the load spectrum for differential gears (with respect to load cycles or rotations on load) is very difficult. For orientation and a basic idea of all relationships, load characteristics were measured on a Škoda Fabia car.

During the measurements instantaneous values of rotations on the left and right wheel axles were directly recorded. The value of the rotations difference is positive or negative. When calculating the

FIGURE 6: Evaluation of recorded differences in rotations from the left and right wheel
OBRÁZEK 6: Vyhodnocení záznam urozdílu otáček zlevého a pravého kola

TABLE 4: Rotations evaluation of the differential gears
TABULKA 4: Vyhodnocení počtu otáček ozubených kol diferenciálu

Type of roadway	Measured section	Measured values		Calculated values			
		travel time t [s]	average speed v_p [km/h]	length of the section $l = \frac{v_p \times t}{3600}$ [km]	difference of rotations $\Delta n_a = n_L - n_R $ [s ⁻¹]	rotations of central wheel on 1 km $\Delta n_i = \frac{\Delta n_a \times 3600}{v_p}$ [rot/km]	
						in section	average
motorway	motorway to Bohumín I	102.14	86.16	2.44	0.08473	3.540	3.083
	motorway to Bohumín II	463.58	81.23	10.46	0.03458	1.532	
	Klimkovice – Pustiměř	3600.0	122.35	122.35	0.14194	4.177	
town	Přívoz – areal VŠB-TU	1653.0	32.81	15.06	0.06741	7.397	9.059
	centrum – Michálkovice	1136.0	30.06	9.49	0.08522	10.206	
	Michálkovice – Radvanice	1601.0	28.97	12.88	0.08929	11.093	
	Radvanice – areal VŠB-TU	1239.6	31.34	10.79	0.06564	7.541	
roadway class I outside the city	areal VŠB-TU – Petřvald	1386.34	43.25	16.65	0.05033	4.190	5.275
	Petřvald – Kopřivnice	1004.91	52.29	14.60	0.09237	6.360	
roadway class II	Opava – Bílovec	2068.4	50.00	28.73	0.08112	5.841	5.841
Total traffic							5.815

differential loads, differences in the absolute value of the rotation difference between the left and the right half axle: $\Delta n_a = |n_L - n_R|$ are also taken into account.

An example of difference and frequency of rotations of automobile wheels during city driving is shown in Figure 6.

TABLE 5: Load spectrum parameters
TABULKA 5: Parametry spektru zatížení

type of roadway	maximal torque M_p [Nm]	relative number of spectrum cycles		degree of spectrum aggressiveness s
		for maximal load $\bar{N}_p$	total $\bar{N}_C$	
motorway	970	1×10^{-3}	1	2.2
town	1650	1.2×10^{-5}	1	1.2
roadway class I	1150	1×10^{-5}	1	1.9
roadway class II	1350	3×10^{-4}	1	1.07
total with weight 0.25	1620	1.72×10^{-6}	1	1.3

For the assessment of differential gear life it is necessary to determine the frequency rather than speed differential, depending on distance traveled. The log records the measured values of the average velocity v_p , time driving t and the average value of the relative rotation Δn_a , which was evaluated for each type of driving in Table 4. This Table lists the driving evaluated by calculating resulting frequency of sun gear rotations in test vehicles per 1 km. For lifetime calculation of differential gears a hypothetical load spectrum was created from measured values on a car, see Table 4. The load spectrum is usually expressed as torque as a function of the number of cycles (in absolute or relative values) for a defined roadway. For further use in the lifetime calculations, mathematical compensation for load spectrum was performed using the equation:

$$N_i = N_C \left(\frac{N_p}{N_C} \right)^{(\bar{M}_i)^s},$$

where

N_i cumulative number of load cycles on the i-th level,
 N_C total number of spectrum cycles,
 N_p number of cycles at the highest level,
 $\bar{M}_i$ relative medium torque on the i-th level,
 s degree of spectrum aggressiveness.

FIGURE 7: Relative cumulative frequency of torque level
OBRÁZEK 7: Poměrná kumulativní četnost výskytu hladiny momentu

For each type of road specified in Table 4 the load spectrum was evaluated (torque in relation to the cumulative number of cycles). Subsequently a substitution was performed according to the relationship with the parameters in Table 5. The measured values of the spectrum sum and their total compensation are shown in Figure 7.

The plot of the relative load spectrum from the measured values can be hypothetically accepted as a basis for estimating the load of all types of vehicles with regard to their specific destination. The maximum load corresponds to the maximum torque specified in Table 1. The number of cycles can be derived for sun gear from speed per 1 kilometer (Table 4) times the estimated expected life of the vehicle in kilometers. The estimated number of cycles of load spectrum for the basic vehicle types is shown in Table 6.

The design of automotive differential gears requires a different approach to other drive aggregates of vehicles for the following reasons:

- the frequency of maximum load for differential gears is low and it is used to reduce by tire adhesion,
- the number of loading cycles is derived from the rotation difference of driven parts (axles, half-axles) and is very low,
- formation of higher relative rotation from high torque is unlikely,
- no internal dynamic forces arise because circumferential velocities are very low during differential gears meshing.

Gears of automobile differentials are loaded in low cycle fatigue. This means that the maximum stress occurs when the number of cycles is in the order of lower than 10^4 cycles. Load control is performed by modified procedures according to DIN 3990-T41, which is practically identical to ISO 6336, which is intended for the design of automobile gears.

For comparative calculations of allowable stress of differential gears, factors influencing additional load due to bending and contact can be neglected

TABLE 6: Estimated number of cycles for vehicle load spectrum

TABULKA 6: Odhady počtů cyklů spekter zatížení automobilů

	Vehicle type		
	Personal automobile	Van	Truck
Estimate of expected life L_D [km]	2.5×10^5	5×10^5	2×10^6
Mean value of rotation Δn_{si} [cycles/km]	$3 \div 5$ (measured)	$5 \div 10$ (measured)	$10 \div 20$ (estimated)
Total length of spectrum $N_C = L_D \times \Delta n_{si}$ [cycles]	$7.5 \times 10^5 \div 1.3 \times 10^6$	$2.5 \times 10^6 \div 5 \times 10^6$	$2 \times 10^7 \div 4 \times 10^7$
Total number of cycles at maximum load level in lifetime N_p [spectrum]	$7.5 \div 13$	$25 \div 50$	$200 \div 400$

$$K_F = K_A \times K_{Fv} \times K_{F\beta} = K_{Fa} = K_H = K_A \times K_{Hv} \times K_{H\beta} \times K_{Ha} = 1$$

To calculate the bending stress of an automobile differential gear it is appropriate to use the DIN 3390-3 method C, which considers the force on the head of the tooth according to Figure 8. Bending stress in gears with straight teeth will equal

FIGURE 8: Scheme of tooth load for the calculation of bending stress
OBRÁZEK 8: Schéma zatížení zubu pro výpočet napětí v ohybu

$$\sigma_F = \frac{F_T}{b_F \times m_n} \times Y_{Fa} \times Y_{Sa} \times Y_\epsilon$$

where

$$F_T = \frac{2000 \times T_D}{d} - \text{tangential load [N]}$$

T_D agreed torque according to differential type [Nm],
 b_F tooth width for calculation of the bending stress [mm]
 m_n normal module [mm]
 d reference diameter of drive differential element [mm]

Coefficients for the calculation of bending stress:

$$Y_{Fa} = 6 \times \frac{\frac{h_{Fa}}{m_n} \times \cos \alpha_{Fan}}{\left(\frac{S_{Fn}}{m_n}\right)^2 \times \cos \alpha_n} - \text{form factor},$$

$$Y_\epsilon = 0,25 + \frac{0,75}{\epsilon_\alpha} - \text{contact ratio factor, where } \epsilon_\alpha - \text{transverse contact ratio,}$$

$$Y_{Sa} = \left(1,2 + 0,13 \times \frac{S_{Fn}}{h_{Fa}}\right) \times \frac{S_{Fn}}{2 \times \rho_{Fn}} \left(\frac{1}{1,21 + \frac{2,3 h_{Fa}}{S_{Fn}}}\right)$$

– stress correction factor

Calculation of the limit contact stress (modified Hertzian relationship) to the tooth contact is given by the equation:

$$\sigma_H = Z_H \times Z_E \times Z_\epsilon \times \sqrt{\frac{F_t}{d \times b} \times \frac{u+1}{u}}$$

where

F_t tangential load identical with calculation of bending stress,
 d planet reference diameter (virtual gears for bevel gears) [mm],
 b tooth width for calculating the contact stress [mm],
 u gear ratio z_1/z_2 and virtual gears for bevel gears z_{v1}/z_{v2}

$$Z_H = \sqrt{\frac{2 \times \cos \alpha_{wt}}{\cos^2 \alpha_t \times \sin \alpha_{wt}}} - \text{zone factor},$$

$$Z_E = 190 \text{ [MPa]} - (\text{steel-steel}) - \text{elasticity factor},$$

$$Z_\epsilon = \sqrt{\frac{4 - \epsilon_\alpha}{3}} - \text{contact ratio factor}.$$

Determination of allowable bending stress and contact based on the fatigue properties is set out in the cited ISO and DIN standards. Gears of automotive differentials are made almost entirely of carburizing steels alloyed with Cr-Mo or Cr-Mn with medium quality (MQ). Fatigue properties in bending and contact can generally be described by the diagram in Figure 9 and data according to Table 6.

FIGURE 9: Fatigue properties of differentials
OBRÁZEK 9: Únavové vlastnosti ozubených kol diferenciálů

Parameters of S-N curves for gear materials 20MnCr5 (1.7147) and 20MoCr4 (1.7321) are directly determined and shown in the summary Table 6 according standard DIN 3990-41. Fatigue limit in bending s_{FE} is defined in the ISO standard in terms of amplitude, but standards DIN and ČSN indicate pulsating cycles where the following applies:

$$\sigma_{FEN} = 2 \times \sigma_{Flim}$$

Fatigue limit in bending and contact simply marked s_{lim} are shown in Table. 6.

$$\sigma_p = \sigma_{FEN} \times Y_{NT}$$

for bending,

$$\sigma_p = \sigma_{Hlim} \times Z_{NT}$$

for contact.

TABLE 7: Fatigue characteristics of hardened gears

TABULKA 7: Únavové charakteristiky cementovaných ozubených kol

Common parameter of S-N curve	Bending		Contact	
	ISO 6336-5	DIN 3990-41	ISO 6336-5	DIN 3990-41
Vickers [HRC]	58÷64		58÷64	
σ_{lim} [MPa]	922 (1000)	920	1500	
σ_p [MPa]	-	1840 (2300)	-	2400
N_{lim}	-	3×10^6	-	5×10^7
Y_{NT}, Z_{NT}	-	2 (2.5)	-	1.6
N_p	-	10^3	-	10^5
q	-	11.5 (8.3)	-	13.2

Values in Table 6 apply for the probability of damage $P=1\%$. Values in brackets are for steels with higher carbon content (according to ISO standard smaller value applies for the strength in the core min. 850 MPa, the higher value is for strength in the core of min. 1000 MPa).

The safety of differential gears, which are loaded in low cycle fatigue, can be defined by common inequalities:

$$S \leq \frac{\sigma_{lim}}{\sigma_p}$$

for bending stress safety can be calculated from the equation:

$$S_F \leq \frac{\sigma_{FEN}}{\sigma_{FEN} \times Y_{NT}} = \frac{2 \times \sigma_{Flim}}{2 \times \sigma_{Flim} \times Y_{NT}} = \frac{1}{Y_F} = \frac{1}{2 \div 2,5} = 0,5 \div 0,4$$

and for contact:

$$S_H \leq \frac{\sigma_{Hlim}}{\sigma_{Hlim} \times Z_{NT}} = \frac{1}{Z_{NT}} = 0,625$$

The validity of these relationships is generally determined by the shape of the operating load spectrum. This spectrum is strongly dependent on operating conditions, type of vehicle and type of differential (axial, inter-axle). The hypothetical spectrum that was used to calculate the comparison differential gear life was created on the basis of measurements.

This calculation was performed for car applications listed in Table 5. Maximum stress σ_{max} was chosen at the upper limit of the highest class ($p=10$), which corresponds to the maximum level allowable static load σ_p according to Table 6. This is the highest load, if not allowing loading based on the yield strength. $\sigma_{min} = 0$ is chosen as the minimum stress of the spectrum in the lowest class ($p=1$). The numbers of spectrum cycles N_p and N_c are shown in Table 4. For minimum safety levels S_F and S_H the intensity of damage is listed in Table 8.

The intensity of damage D_H relates to the spectrum length L_p , which is given by an estimate of the life of the vehicle (Table 5).

The calculated lifetime is given by $L_{zv} = L_p / D_H$.

When assessing the resistance of contact and bending the shape of the S-N curve must be taken into account. This depends on the number of cycles N_{lim} and exponent q (Table 6). At the same load bending damage will be at a far lower number of cycles than the contact damage. Circumferential and slide velocities are so small that contact damage of the differential gears cannot be expected. The intensity of differential gear damage for those vehicles where the requirements for the total lifetime are in the range $(0.5 \div 1) \times 10^6$ km, is very low. The results of service life calculation of heavy trucks are broadly consistent with the required minimum safety levels.

CONCLUSION

The design of automobile differentials can be checked with sufficient reliability in terms of safety to fatigue limit. This assumption is confirmed by measurements and calculations. Contact safety ($S_H < 0.625$) is not to be observed on the basis of the comparative life calculations, so can be designed to select the value of safety:

$$S_{lim} = S_F = S_H > 0.5$$

Vehicles intended for mixed traffic and terrain are equipped with a mechanical differential lock. This prevents slippage of tires with reduced adhesion. A permanent differential lock causes the formation of parasitic torques, which increase the load, and it gives rise to higher requirements for the resistance of differentials. Adequate safety for different types of vehicles can be summarized as follows:

- personal automobile 4x2 with classical differentials without lock:
 $S_{lim} \doteq 0.5 - 0.6$
- vans and trucks for road use without differential lock:
 $S_{lim} \doteq 0.6 - 0.7$
- trucks and vehicles for off-road and all vehicles with differential locks:
 $S_{lim} > 0.7$.

TABLE 8: The calculation of lifetime for differential maximum load
TABULKA 8: Výpočet životnosti pro maximální zatížení diferenciálu

Vehicles		Personal automobile			Van		
Traffic		hilly			heavy		
Damage		F		H	F		H
Spectrum by Tables 5 and 6	σ_p	1840	2300	2400	1840	2300	2400
	N_p	13			50		
	N_c	1.3e6			5e6		
	S	1.3			1.3		
S-N curve parameters by Table 6	σ_{lim}	920	920	1500	920	920	1500
	N_{lim}	3e6	3e6	5e7	3e6	3e6	5e7
	q	11.5	8.3	13.2	11.5	8.3	13.2
required lifetime L_p [km]		250000			500000		
intensity of damage D_H (Haibach)		0.0369	0.0916	0.0002	0.1418	0.3523	0.0009
S_{min}		0.5	0.4	0.62	0.5	0.4	0.62

vehicle		truck					
traffic		combined off-road					
damage		F		H	F		H
spectrum by Tables 5 and 6	σ_p	1840	2300	2400	1840	2300	2400
	N_p	200			400		
	N_c	2e7			4e7		
	S	1.3			1.3		
S-N curve parameters by Table 6	σ_{lim}	920	920	1500	920	920	1500
	N_{lim}	3e6	3e6	5e7	3e6	3e6	5e7
	q	11.5	8.3	13.2	11.5	8.3	13.2
required lifetime L_p [km]		2000000					
intensity of damage D_H (Haibach)		0.567	1.4092	0.0035	1.134	2.8183	0.0069
S_{min}		0.5	0.4	0.62	0.5	0.4	0.62

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