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CALCULATIONS OF CYLINDRICAL TOOTH WHEELS IN VIEW OF ISO 6336 STANDARD AND RECOMMENDATIONS OF CLASSIFICATION SOCIETY CN 41.2-DNV-GL

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Abstarct

In this study there are presented ISO 6336 standard and recommendations of classification society CN 41.2-DNV-GL regarding calculations of cylindrical toothed gears used in all oceanic ships and oil rigs. The calculation method presented in this study allows to perform calculations of the safety coefficient for stress at the tooth base S_F and for contact stress S_H. The calculations have been performed for gears with different geometric parameters and for three load spectra. The obtained results enable to determine the values of relative differences in δ_F and δ_H, which are: δ_F = 16,7% to 42,8% and δ_H = 10,4% to 22,6%.

Keywords : fatigue life, calculations of toothed gears, ISO 6336 standard

More important denotations:

K_A – application factor,
K_V – dynamic factor,
$K_{F\beta}$ – face load factor (root stress),
$K_{F\alpha}$ – transverse load factor (root stress),
K_γ – mesh load factor (takes into account the uneven distribution of the load between meshes for multiple transmission paths - $K_\gamma = 1+ (0,2/\theta)$,
θ – twist angle of shaft for maximum output torque,
F_t – (nominal) transverse tangential load at reference cylinder per mesh
b – facewidth,
m_n – normal module,
u – gear ratio ($u = z_2/z_1$),
Y_F – tooth form factor,
Y_S – stress correction factor
Y_β – helix angle factor (tooth root)
Y_{ST} – stress correction factor (for wheels of standard dimensions $Y_{ST} = 2$),
Y_{NT} – coefficient of fatigue for stress at the base of the tooth, ($Y_{NT}= 0.85$ for $N = 10^{10}$),
$Y_{\delta rel\ T}$ – relative coefficient of sensitivity to the notch action,
$Y_{Rrel\ T}$ – relative coefficient of the surface state,
Y_X – quantity coefficient for stress at the base of the tooth,
Y_M – mean stress influence factor
Y_N – fatigue strength for stress at the base of the tooth ($Y_N= 0.92$ for $N_L = 10^{10}$),
Y_c – depth coefficient of the toothed gear hardened surface,
$S_{F\ min}$ – required minimal safety coefficient for stress at the base of the tooth,

$S_{H\,min}$ – required minimal safety coefficient due to contact strength,
ρ_F – radius of transition curve in a dangerous cross-section,
$\sigma_{F\,lim}$ – agreed fatigue limit of the gear standard tooth base (ISO 6336-5),
σ_{FE} – local fatigue limit of the gear base (for toothed gears hardened inductively σ_{FE} = 0,7 HV+300),
$\sigma_{H\,lim}$ – agreed limit of contact fatigue at the side of the tooth, includes the impact of material, thermal machining and roughness of standard surfaces of model gears,
σ_{HG} – boundary contact stress,
$Z_{B,\,D}$ – coefficient of single pair tooth contact (wheel); this coefficient changes contact stress in the pole of the engagement into contact stress in the inner point of a single pair of tooth (wheel),
Z_H – thrust coefficient of the zone; includes curvatures of teeth sides in the pole of engagement and transformation of peripheral force on a pitch cylinder into peripheral force on the rolling cylinder,
Z_E – coefficient of elasticity: accounts for characteristic material properties: elasticity modules E1 and E2 and Poisson numbers v_1 and v_2,
Z_ε – tooth contact coefficient for calculation of contact stress; accounts for the impact of the active contact length,
Z_β – tooth line tilt angle coefficient; accounts for load fluctuations along the contact line,
Z_L – lubrication agent factor accounts for the impact of the lubricant viscosity,
Z_R – coefficient of roughness accounts for the impact of surface roughness,
Z_V – coefficient of speed accounts for the impact of peripheral speed on the rolling cylinder,
Z_W – coefficient of the material cold work hardening accounts for the impact of the gear unhardened surface cooperation with the cooperating wheel hardened surface,
Z_X – size factor for calculation of contact stress accounts for the impact of the tooth size on permitted contact stresses,
Z_{NT} – coefficient of fatigue life for calculation of contact stress; accounts for higher contact resistance for a limited number of cycles Z_N= 0.85for $N_L = 10^{10}$),
Z_N – coefficient of fatigue life for calculation of contact stress accounts for higher contact resistance for a limited number of cycles (Z_N= 0.92 for $N_L = 10^{10}$).

1. Introduction

The calculation method for cylindrical and bevel toothed gears according to DNV GL [2] is required for all oceanic ships and oil rigs. This method is significantly consistent with ISO 6336 calculation method [3] (method B). Differences involve including in calculations some phenomena by selection of given factors which are used for construction of a gear e.g. face load factor, factor of the toothed gear hardened surface depth.

The society Det Norske Veritas (abbr.: DNV GL) is a Norwich classification society founded in 1864 which now belongs to three largest classification societies in the world. The main goal of the society is protection of life, property and the environment by means of development of standards (norms and directives) for marine and electrical branches.

One of examples of standards prepared by Det Norske Veritas society is a classification notes CN 41.2-DNV-GL issued in 2003 – Calculation of Gear Rating for Marine Transmissions. This method is largely **based** on fatigue life calculations of gears defined in ISO 6336 parts: 1-5 [3]. Calculations of seizure safety factors are performed according to ISO/TR 13989 standard [4], and geometry of toothed gears corresponds to ISO 53 [5] for cylindrical gears and ISO 23509 [6] for bevel gears.

The aim of this study is to discuss a calculation method for toothed gears in terms of ISO 6336 standard and recommendations of classification society CN 41.2-DNV-GL.

The scope of this study includes calculations of stresses at the tooth base as well as contact stresses allowing to determine values of safety factors S_F and S_H.

2. Check calculations of cylindrical toothed gears

International standard ISO 6336 is the basis for check calculations of cylindrical toothed gears used in ship industry. Two calculation methods have been presented in this norm:

a) A method – used only in special cases where failures could have very serious consequences),

b) B method – commonly used in all kinds of industries due to its calculation accuracy. B method is particularly useful for numerical calculations and analysis of experimental tests results.

Check calculations for cylindrical toothed gears are also based on internal methods of a given classification unit, e.g. CN 41.2 DNV-GL. The calculation process according to B method of ISO 6336 standard assumes calculations of stress at the base of the tooth σ_F

$$\sigma_F = \frac{F_t}{b \cdot m_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \cdot K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \le \sigma_{FP} \tag{1}$$

Allowable stress at the tooth base σ_{FP}

$$\sigma_{FP} = \frac{\sigma_{F\lim} \cdot Y_{ST} \cdot Y_{NT}}{S_{F\min}} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X = \frac{\sigma_{FG}}{S_{F\min}} \tag{2}$$

Contact stress σ_H

$$\sigma_H = Z_{B,D} \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_t}{d_1 \cdot b} \frac{(u+1)}{u} K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha}} \le \sigma_{HP} \tag{3}$$

Allowable contact stress σ_{HP}

$$\sigma_{HP} = \frac{\sigma_{H\lim} Z_{NT}}{S_{H\min}} Z_L Z_V Z_R Z_W Z_X = \frac{\sigma_{HG}}{S_{H\min}} \tag{4}$$

Check calculations for cylindrical toothed gears according to standards CN 41.2-DNV-GL, involve determination of stresses at the base of tooth σ_F according to the formula

$$\sigma_F = \frac{F_t}{bm_n} \cdot Y_F \cdot Y_S \cdot Y_\beta \cdot K_A \cdot K_V \cdot K_\gamma \cdot K_{F\beta} \cdot K_{F\alpha} \tag{5}$$

The difference between formulas (1) and (5) is connected with application of load $K\gamma$ division in formula (5). It takes into consideration load distribution in two track gears, as well as non-uniform distribution of torque with application of many pinions and one passive wheel. Ratio of loading division is defined as a ratio of maximum real force acting on the tooth, including transmission errors, and the force acting on the tooth with uniform maximum loading. Simplified coefficient of load division is expressed by the formula:

$$K\gamma = 1 + \left(\frac{0{,}2}{\theta}\right) \tag{6}$$

The remaining coefficients in formula (5) are consistent with recommendations of ISO 6336 standard [2]. Stresses permitted at the base of tooth σ_{FP}

$$\sigma_{FP} = \frac{\sigma_{FE} \cdot Y_M \cdot Y_N}{S_{F\min}} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X \cdot Y_C = \frac{\sigma_{FG}}{S_{F\min}} \tag{7}$$

Comparison of formulas (2) and (7) indicates that in the method according to standards of CN 41.2-DNV-GL society, factors: Y_M, Y_N, and Y_C were used; mean stress influence factor Y_M regards variability of work connected, among others, with temporal change of the tooth loading direction (passive and active toothed gears) and is expressed by the following formula:

$$Y_M = \frac{1}{1 - R \cdot \left(\frac{1-M}{1+M}\right)} \tag{8}$$

The scope of coefficient R variability is: -1,2 ≤ R ≤ 0,5, whereas the variability range of M: 0,3 ≤ M ≤ 0,65. Accepting values of factors for surface hardened toothed gears (R = -1,2 , M = 0,4) the value of load change coefficient is Y_M = 0,66. Fatigue factor of life for stresses at the base of Y_N tooth depends on the number of work cycles N_L. The range of variability of the coefficient is given in Table 1.

Table 1. Variability range of value changes for fatigue life coefficient for stresses at the base of tooth Y_N

No.	Range of work cycles N_L	Fatigue life coefficient Y_N
1	2	3
1	$N_L > 3 \cdot 10^6$	$Y_N = 1$ or $Y_N = \left(\frac{3 \cdot 10^6}{N_L}\right)^{0,01}$ (for $N_L = 10^{10}$ cycles of value $Y_N = 0{,}92$)
2	$10^3 < N_L < 3 \cdot 10^6$	$Y_N = \left(\frac{3 \cdot 10^6}{N_L}\right)^{p}$, where: $p = 0{,}2876 \cdot \log\left(\frac{\sigma_{FPst}}{\sigma_{FP}}\right)$ σ_{FPst} - allowable stresses for $N_L = 10^3$ cykli σ_{FP} - allowable stresses for $N_L = 3 \cdot 10^6$ cykli
3	$N_L < 10^6$	$Y_N = \frac{\sigma_{FPst}}{\sigma_{FP}}$ σ_{FPst} - allowable stresses for $N_L = 10^3$ cycles σ_{FP} - allowable stresses for $N_L = 3 \cdot 10^6$ cycles

Factor of the depth of the hardened surface of the gear Y_c is calculated from the formula:

$$Yc = \frac{1 + \frac{3 \cdot t}{\rho_F + 0{,}2 m_n}}{\sigma_{FE}} \tag{9}$$

The variability range of factor Y_c depends on the outline for involute toothed gears of the transition curve radius in dangerous cross-section (being $0{,}2 \cdot m_n < \rho_F < 0{,}4\ m_n$) and the method of thermal machining and the surface hardness. The value of coefficient is given in Table 2.

Table 2. Range of the coefficient variability

Lp.	Surface hardening method		t
1	2	3	4
1	Carburization (case hardening)	t_{550}	640
		t_{400}	500
		t_{300}	380
2	Nitriding	t_{400}	500
3	Induction- or flame hardening	T_{Hvmin}	1,1 HV_{min}

The method included CN 41.2-DNV-GL involves calculations of contact stresses σ_H from the formula

$$\sigma_H = Z_{B,D} \cdot Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \sqrt{\frac{F_t}{d_1 \cdot b} \frac{(u+1)}{u} K_A \cdot K_V \cdot K_\gamma \cdot K_{H\beta} \cdot K_{H\alpha}} \leq \sigma_{HP} \tag{10}$$

Comparing formulas (3) and (10) it was found that in dependence (10) there occurs loading division ratio of K_γ, loading expressed by formula (6).

Permitted contact stresses σ_{HP}

$$\sigma_{HP} = \frac{\sigma_{H\lim} Z_N}{S_{H\min}} Z_L Z_V Z_R Z_W Z_X = \frac{\sigma_{HG}}{S_{H\min}} \tag{11}$$

Comparison of formulas (4) and (11) indicates that in CN 41.2-DNV-GL method fatigue life factor Z_N, was used which depends on the number of loading cycles N_L. The range of value variability of Z_N factor is shown in Table 3.

Table 3. Variability range of fatigue lifei Z_N

Lp.	Range of loading cycle N_L	Fatigue life factor Z_N
1	2	3
1	$N_L \geq 5 \cdot 10^7$	$Z_N = 1$ or $Z_N = \left(\frac{5 \cdot 10^7}{N_L}\right)^{0,0157}$ (for $N_L = 10^{10}$ cycles of value $Z_N = 0,92$)
2	$10^5 < N_L < 5 \cdot 10^7$	$Z_N = \left(\frac{5 \cdot 10^7}{N_L}\right)^p$ $p = 0,37 \cdot \log\left(Z_{N10^5}\right)$
3	$N_L = 10^5$	$Z_N = Z_{N10^5} \frac{\sigma_{H10^5} Z_{X10^5} Z_{Wst}}{\sigma_{H\lim} Z_L Z_V Z_R Z_X Z_W}$
4	$10^3 < N_L < 10^5$	$Z_N = Z_{N10^5} \left(\frac{10^5}{N_L}\right)^p$ $p = 0,5 \cdot \log\left(\frac{Z_{N\,10^3}}{Z_{N\,10^5}}\right)$
5	$N_L \leq 10^3$	$Z_N = Z_{N10^3} \frac{\sigma_{H10^3} Z_{X10^3} Z_{Wst}}{\sigma_{H\lim} Z_L Z_V Z_R Z_X Z_W}$

The presented methods for check calculations of cylindrical toothed gears (according to ISO 6336 (B method) and CN 41.2-DNV-GL) indicate differences connected with factors: K_γ, Y_M, Y_N, Z_N and Y_C. The final result of calculations depends on them.

3. An example of calculations for cylindrical toothed gears according to ISO 6336 and CN 41.2-DNV-GL

3.1. Formulation of the calculation problem

Differences in calculation results for the method according to ISO 6336 and CN 41.2-DNV-GL will be presented on the basis of a one-degree gear with toothed cylindrical gears with parameters presented in Table 4.

The gears were assumed to be made of alloy steel 42CrMo4 subjected to thermal treatment involving induction (according to ISO 6336-5 standard recommendations) which allow to achieve hardness of the teeth surface 600 HV.

Loading of the toothed gear was described by a load spectrum which defines the shares of working conditions during service. Three variants of load spectrum were accepted during service. They are presented in Table 5.

Operation time L_h = 2460 and application coefficient K_A = 1 were assumed. The calculation example applies to a closed gear operating under conditions of oil lubrication for temperature 70ºC.

Table 4. Variants of parameters of gears accepted for calculations

Example of gear	Parameters of a toothed gear	Gear 1	Gear 2
Variant I	Number of teeth	$z_1 = 54$	$z_2 = 55$
	Normal module	m_n = 6,0 mm	
	Facewidth	b = 120,0 mm	
	Normal pressure angle	$\alpha_n = 20^o$	
	Helix angle	$\beta = 0^o$	
Variant II	Number of teeth	$z_1 = 27$	$z_2 = 82$
	Normal Module	m_n = 6,0 mm	
	Facewidth	b = 120,0 mm	
	Normal pressure angle	$\alpha_n = 20^o$	
	Helix angle	$\beta = 0^o$	
Variant III	Number of teeth	$z_1 = 15$	$z_2 = 93$
	Normal module	m_n = 6,0 mm	
	Facewidth	b = 120,0 mm	
	Normal pressure angle	$\alpha_n = 20^o$	
	Helix angle	$\beta = 0^o$	

Table 5. Parameters of load spectra of accepted calculations

Load spectrum		Share of percentage degrees in a spectrum	Power	Speed	Torque
Designation	No of alloy	%	kW	rev/min	Nm
1	2	3	4	5	6
Spectrum A	1	95	10,0	24,7	3866,4
	2	5	7,89	16,5	4566,6
	3	0	4,63	8,2	5392,3
Spectrum B	1	75	10,0	24,7	3866,4
	2	20	7,89	16,5	4566,6
	3	5	4,63	8,2	5392,3
Spectrum C	1	50	10,0	24,7	3866,4
	2	35	7,89	16,5	4566,6
	3	15	4,63	8,2	5392,3

3.2. Results of calculations of safety factors S_F and S_H

Calculations of safety factors for stresses at the base of the tooth and for contact stresses and for constant stresses were performed on the basis ISO 6336-5 standard and procedures CN 41.2-DNV-GL. The process of selection of parameters for a gear with parameters according to variant III and load spectrum B is presented in this study. The results are included in tables 6 and 7. The values of safety factors S_F and S_H for the remaining cases are presented in Tables 8 and 9.

Table 6.Results of safety calculations for stresses at the base of the tooth

Lp.	Parameter	Values of parameters			
		ISO 6336 method B		CN 41.2 DNV GL	
		$z_1 = 15$	$z_2 = 93$	$z_1 = 15$	$z_2 = 93$
1	2	4	5	6	7
1	Addendum modfication coefficient x	0,39	-0,10	0,39	-0,10
2	tooth form factor Y_F	1,31	1,50	1,31	1,50
3	stress correction factor Y_S	2,12	1,99	2,12	1,99
4	Coefficient of the angle of tooth line tilt Y_{bet}	1,000		1,000	
5	Width of toothed rim b_{eff}, mm	120,0		120,0	
6	Nominal stresses at the base of the tooth σ_F, MPa	356,1	380,9	356,1	380,9
7	Relative coefficient of sensitivity to notch operation $Y_{d\ rel\ T}$	0,998		0,998	
8	Relative coefficient of surface state coefficient $Y_{R\ rel\ T}$	0,957	0,965	0,957	0,965
9	Coefficient of size for stresses at the base of the tooth Y_x	0,990	0,992	0,990	
10	Coefficient EHT Y_C	-	-	0,985	0,992
11	Coefficient of fatigue life for stresses at the base of the tooth Y_{Nt}	0,996	1,205	0,998	1,235
12	Product of coefficient ($Y_{d\ rel\ T}$, $Y_{R\ rel\ T}$, Y_x, Y_{Nt})	0,942	1,151	0,943	1,180
13	Coefficient of load change Y_M	-		0,900	
14	Coefficients of stress correction Y_{st}	2,0		-	
15	Agreed limit of fatigue at the base of the tooth σ_{FE}, MPa	799,2		720,0	
16	Boundary stresses at the base of the tooth σ_{FG}, MPa	752,51	920,03	601,89	764,54
17	Required safety factor $S_{F\ min}$	1,40		1,40	
18	Admissible stresses at the tooth base, MPa	537,5	657,2	436,9	546,1
19	**Stress safety factor at the base of the tooth S_F**	**2,218**	**2,454**	**1,822**	**2,040**

Table 7. Results of calculations for safety coefficient for contact stresses

No.	Parameter	Values of parameters			
		ISO 6336 methods B		CN 41.2 DNV GL	
		$z_1 = 15$	$z_2 = 93$	$z_1 = 15$	$z_2 = 93$
1	2	4	5	6	7
1	Coefficient of the stress zone Z_H	2,441		2,441	
2	Coefficient of elasticity Z_E	189,8		189,8	
3	Coefficient approach Z_{eps}	0,910		0,910	
4	Coefficient of angle of the tooth line tilt Z_{bet}	1,000		1,000	
5	Facewidth b_{eff}, mm	120,0		120,0	
6	Nominal contact stress σ_{HO}, MPa	1281,1		1281,1	
7	Surface stress on a division diameter σ_{Hw}, MPa	1331,8		1331,8	
8	Coefficient of one pair teeth approach $Z_{B,\ D}$	1,04	1,00	1,04	1,00
9	Surface stress σ_{HB}, σ_{HD} , MPa	1381,3	1331,8	1381,3	1331,8
10	Coefficient of lubricant Z_L	1,012	1,006	0,961	0,980
11	Coefficient of velocity Z_V	0,964	0,982	0,964	0,982
12	Coefficient of roughness Z_R	0,983	0,992	0,983	0,992
13	Coefficient of material strain hardening Z_W	1,000		1,000	
14	Fatigue life coefficient Z_{NT}	1,219	1,399	1,221	1,408
15	Product of coefficients (Z_L, Z_V, Z_R, Z_{Nt})	1,169	1,371	1,171	1,379
16	Size factor Z_x	1,000		1,000	
17	Agreed boundary of fatigue limit of the tooth side $\sigma_{H\ lim}$, MPa	1316,0	1316,0	1200,0	1200,0
18	Agreed surface stresses σ_{HP}, MPa	1538,1	1803,8	1334,4	1613,8
19	Boundary contact stress σ_{HG}, MPa	1538,1	1803,8	1334,4	1613,8
20	Required contact stress safety factor $S_{H\ min}$	1,00	1,00	1,00	1,00
21	**Contact stress safety factor S_H**	**1,138**	**1,384**	**0,991**	**1,197**

Table 8. Values of safety for stresses at the base of the tooth S_F

		Kind of gear											
		Variant I				Variant II				Variant III			
		ISO 6336		CN 41.2		ISO 6336		CN 41.2		ISO 6336		CN 41.2	
		z_1=54	z_2=55	z_1=54	z_2=55	z_1=27	z_2=82	z_1=27	z_2=82	z_1=15	z_2=93	z_1=15	z_2=93
Spectrum	A	6,256	6,045	5,064	4,874	4,274	4,430	3,466	3,623	2,258	2,561	1,837	2,127
	B	6,238	6,022	5,058	4,866	4,229	4,288	3,453	3,521	2,218	2,454	1,822	2,040
	C	6,197	5,966	5,043	4,845	4,082	4,136	3,390	3,412	2,095	2,350	1,731	1,951

Table 9. Values of safety coefficient due to contact stress S_H

		Kind of gear											
		Variant I				Variant II				Variant III			
		ISO 6336		CN 41.2		ISO 6336		CN 41.2		ISO 6336		CN 41.2	
		z_1=54	z_2=55	z_1=54	z_2=55	z_1=27	z_2=82	z_1=27	z_2=82	z_1=15	z_2=93	z_1=15	z_2=93
Spectrum	A	3,029	3,021	2,619	2,624	2,049	2,252	1,775	1,987	1,154	1,403	1,001	1,211
	B	2,994	2,986	2,596	2,601	2,027	2,228	1,764	1,974	1,138	1,384	0,991	1,197
	C	2,958	2,949	2,570	2,575	2,001	2,200	1,749	1,958	1,120	1,363	0,979	1,182

The value of safety factor for stresses at the tooth base S_F and contact stresses S_H allow to determine the value of relative difference expressed by the formula:

$$\delta_F = \frac{S_{F(ISO)} - S_{F(CN\,41.2)}}{S_{F(ISO)}} \cdot 100\% \tag{12}$$

$$\delta_H = \frac{S_{H(ISO)} - S_{H(CN\,41.2)}}{S_{H(ISO)}} \cdot 100\% \tag{13}$$

Results of the calculations are presented in Tables 10 and 11. The values of relative difference δ_F for stresses at the base of the tooth S_F show that values $S_{F,}$ determined by the method according to ISO 6336, are higher than the values determined by the method according to CN 41.2-DNV-GL in the range from 16.9% to 19.4%, depending on the load spectrum and geometric parameters of the gear. The values of coefficients Y_N, Y_M, Y_C K_γ the occurred differences.

Table 10. Values of relative difference δ_F

		Case of gear					
		Variant I		Variant II		Variant III	
		z_1=54	z_2=55	z_1=27	z_2=82	z_1=15	z_2=93
Spectrum	A	19,0	19,4	18,9	18,2	18,6	16,9
	B	18,9	19,2	18,4	17,9	17,9	16,9
	C	18,6	18,8	16,9	17,5	17,4	17,0

Table 11. Values of relative difference δ_H

		Case of gear					
		Variant I		Variant II		Variant III	
		z_1=54	z_2=55	z_1=27	z_2=82	z_1=15	z_2=93
Spectrum	A	13,5	13,1	13,4	11,8	13,3	13,7
	B	13,3	12,9	13,0	11,4	12,9	13,5
	C	13,1	12,7	12,6	11,0	12,6	13,3

The value of safety coefficients for contact S_H determined according to ISO 6336 norm are higher than the values determined by the method according to CN 41.2-DNV-GL. Relative differences δ_H are included in the range 11,0% to 13,7%. Parameters Z_{NT} and K_{Na} have an influence on the value difference.

4. Conclusions

Lower values of safety factors S_F and S_H were determined on the basis of calculations for the method according to CN 41.2-DNV-GL as compared to the method according to ISO 6336. It was proved that the values of S_F and S_H depend mostly on the gear transmission and to a smaller degree on the load spectrum (Tab. 8, Tab. 9).The calculation method used for a design of cylindrical toothed gears has a large influence on their geometric characteristics. It is due to the material cyclical properties and determination of fatigue strength for a given number of work cycles. The calculations of relative differences δ_F and δ_H showed smaller differences in the values of safety coefficients on contact stresses S_H as compared to the value of the values of stresses at the base of the tooth S_F.

The presented calculations of cylindrical toothed gears, based on ISO 6336 standard and recommendations of classification society CN 41.2-DNV-GL, in some industrial applications are not sufficient. The selection of geometrical characteristics for gears to be made by designers employed in leading companies of the ship building industry is based on procedures developed in result of experiences gained during implementation of many construction projects.

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