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RESEARCH ARTICLE

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THEORETICAL STRESS ANALYSIS OF HELICAL GEAR (HG13)

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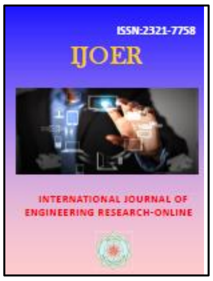

ABSTRACT

Objective of this paper is to dispense the information for calculating the bending and contact stresses acting on the helical gears which are used for marine applications. The marine engines are among heavy-duty machineries and are operated at very high speed which induces large stresses and deflections in the gears as well as in other rotating components. For the safe functioning of the engine, these stresses and deflections have to be minimized. There are various methods to find out stresses acting on a helical gear as AGMA, ISO-6336, finite element analysis and Photo-elasticity used extensively in industry for analysis purpose. This paper discusses ISO-6336 method is used for finding out the bending and contact stresses acting on the helical gear. It is proposed to focus on stress calculations and producing high accuracy gears.

INTRODUCTION

Gears are used for transmitting power between the shafts i.e. to transmit torque and angular velocity. The rapid development of industries such as vehicle, shipbuilding and aircraft require advanced application of gear technology. Customers prefer cars with highly efficient engine. This needed up a demand for quite power transmission. Automobile sectors are one of the largest manufacturers of gears. Higher reliability and lighter weight gears are necessary to make automobile light in weight as lighter automobiles continue to be in demand. The design of gear is a complex process. Generally, it needs large number of iterations and data sets. In many cases gear design is traditional and specified by different types of standards. B.Venkatesh [1] presented that the stresses were calculated for helical gear by using different materials. Kailash.C [2] focused on Lewis beam strength equation was used to finding out bending strength of helical gear. Yi-Cheng Chen[3] in their study stress analysis of a helical gear set with localized bearing contact have investigated the contact and the bending stresses of helical gear set with localized bearing contact by using finite element analysis S.Vijayaragan and N.Ganesan[4] carried out a static analysis of composite helical gears using three dimensional finite element methods to study the displacements and stresses at various points on a helical gear tooth. For determining the stresses at any stage during the design of gears helix angle and face width are important. Rao and Muthuveerappan [5] have explained the geometry of helical gears by simple mathematical equations. In helical gears there is a problem of failures at the root of the teeth because of the inadequate bending strength and surface pitting. This can be avoided or minimized by proper method and modification of the different gear parameters. In view of this the main purpose of this work is by using analytical approach and numerical approach to develop theoretical model of helical gear in mesh and to determine the effect of gear tooth stresses.

The main factors responsible for the failure of a gear set are bending stress and surface strength of a gear tooth. Therefore stress analysis becomes an important area of research which deals with minimization or reduction of the stresses and also with optimal design of gears. This work explains about the bending stress & contact stress of helical gears by using theoretical calculations.

Gear Design Data

Table I shows the parameters considered during the actual designing of helical gear HG13.The complexity in the design of helical gear increases with changes in pressure angle & helix angle. An alloy steel material is used with its specification EN353 for the designing of helical gear. The various constant values as per the reference tables are considered during the calculation of stresses induced in the helical gear.

Table 1: Dimensions Of Helical Gear:

Parameters	**Pinion**	**Gear**
Number of teeth	21	111
Pressure angle, normal	250	
Helix angle	150(RH)	150(LH)
Face width (mm)	53	
Normal module (mm)	4	
Material	Alloy Steel EN353	Alloy Steel EN353
Input speed (rpm) (A)	2000	………..
Input power (KW) (A)	193.382	………..
Diameter of pitch circle (mm)	86.96	459.66
Diameter of base circle (mm)	78.314	413.9505
Diameter of Addendum circle (mm)	94.9631	467.6626
Diameter of Dedendum circle (mm)	76.9631	449.6626
Young's Modulus (MPa)	200000	
Poisson's ratio	0.3	
Torque (N-m)	981000	

Procedure Of Stress Calculation For Helical Gear

An ISO 6336 methodology is used to calculate permissible & working (Actual) stress values for both bending & contact stresses. Most of the industries have adopted this design procedure since it gives high accuracy.

A. ***Contact Stress:***

1. Permissible contact Stress

$$\boldsymbol{\sigma_{Hp}} = \frac{\boldsymbol{\sigma_{Hlim} * Z_L * Z_l * Z_R * Z_S * Z_V * Z_W}}{\sqrt{\boldsymbol{K_R}}}$$

Where, σ_{Hlim} = Endurance limit for contact stress(N/mm^2)

$$\sigma_{Hlim} = A * x + B$$

Z_L = Life factor for contact stress, $L_N = 60 * n * L_H$

Z_l = Lubrication factor for contact stress,

$$\mathrm{Z_l} = \mathrm{C_{Zl}} + \mathrm{C_{Zl}}\frac{4(1-\mathrm{C_{Zl}})}{\left(1.2+\frac{80}{\vartheta_{50}}\right)^2},\ \mathrm{Z_1} = \mathrm{C_{Zl}} + \mathrm{C_{Zl}}\frac{4(1-\mathrm{C_{Zl}})}{\left(1.2+\frac{134}{\vartheta_{40}}\right)^2}$$

Z_R = Roughness factor for contact stress

Z_S = Size factor for contact stress

Z_V = Velocity or Speed factor for contact stress,

$$\mathrm{Z_V} = \mathrm{C_{ZV}} + \frac{2(1-\mathrm{C_{ZV}})}{\sqrt{0.8+\frac{32}{\vartheta}}}$$

Z_W = Work Hardening factor for contact stress,

ZW = 1 --------------- (130 < HB < 400)

K_R = Reliability factor for contact stress,

KR = 1 ----- (Corresponding to probability of failure of 1 in 100 at rated load & required life)

2. *Working (Actual) Contact Stress*

$$\sigma_H = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{bd_1}\left(\frac{u+1}{u}\right) K_A K_V K_{H\beta} K_{H\alpha}}$$

Where, σ_H = Working or actual contact stress (MPa)

Z_H = Zone factor for contact stress,

$$Z_H = \sqrt{\left(\frac{2cos\varphi_b}{cos^2\alpha_t}\right)\left(\frac{cos\alpha_{tw}}{sin\alpha_{tw}}\right)}$$

Z_E = Elasticity factor for contact stress

Z_ε = Contact ratio factor for contact stress,

$$\mathrm{Z_\varepsilon} = \sqrt{\frac{(4-\varepsilon_\alpha)(1-\varepsilon_\beta)}{3} + \left(\frac{\varepsilon_\beta}{\varepsilon_\alpha}\right)}$$

Z_β = Helix angle factor for contact stress, $Z_\beta = \sqrt{\cos\varphi}$

b = Face width

d_1 = Pitch circle diameter of pinion (mm)

u = Gear ratio (Z2/ Z1)

K_A = Application factor

K_V = Dynamic load factor, For helical gear with $\varepsilon_\beta \geq 1$

$$K_V = K_{V\beta}$$

$K_{H\beta}$ = Longitudinal load distribution factor for contact stress

$K_{H\alpha}$ = Transverse load distribution factor for contact stress,

$$K_{H\alpha} = K_{F\alpha}$$

F_t = Tangential force (nominal) at pitch circle appropriate to pinion or wheel calculations respectively (N),$F_t = {}^{P}/_{v}$

$$K_{H\alpha} = K_{F\alpha}$$

Factor of safety:

$$FS = \frac{\sigma_{HP}}{\sigma_H}$$

B. ***Bending Stress :***

1. *Permissible bending stress (σ_{FP})*

$$\sigma_{FP} = \left\{\frac{\sigma_{FE}}{K_R}\right\} * Y_N * Y_n * Y_R * Y_S$$

6_{FP}=permissible bending stress at the root of tooth (MPa or N/mm2)

K_R=reliability factor

Y_N=life factor for bending stress,$L_N = 60 * n * L_H$

Y_n=Notch sensitivity factor,$Yn = 0.04Y_k + 0.93$

Y_R=factor of relative surface roughness

Y_S=size factor for bending stress

6_{FE}=endurance limit of an un-notched specimen (MPa),

$$\sigma_E = A * x + B$$

2. *Working (Actual) bending stress (σ_F):*

$$\sigma_F = \left(\frac{F_t}{bm_n}\right) * Y_{Fa} * Y_k * Y_\varepsilon * Y_\beta * K_A * K_v * K_{F\beta} * K_{F\alpha}$$

6_F = working actual bending stress

F_t = tangential force

b = face width

m_n = normal module

Y_a = form factor

Y_k = stress conc. Factor

Y_ε = contact ratio factor for bending stress,$Y\varepsilon = 0.25 + \frac{0.75}{\varepsilon\alpha}$

Y_β = helix angle factor for bending stress,$Y_\beta = 1 - \varepsilon_\beta\left(\frac{\beta}{120}\right)$

K_A = application factor

$K_{F\beta}$ = longitudinal load distribution factor for bending stress

$K_{F\alpha}$ = transverse load distribution factor

ε_α = transverse contact ratio

Factor of safety

$$FS = \frac{\sigma_{FP}}{\sigma_F}$$

RESULTS AND DISCUSSION

This work primarily focuses on theoretical design procedure of helical gear used in marine applications. As per the actual working condition and material properties, Bending stresses and contact stresses are calculated. For helical gear used in industrial application, factor of safety is lies between 1.2 to 1.5.Calculated FOS mentioned in table shows within this range. Hence, design is safe in this case.

Table III: Calculated stresses & Factor of safety

Bending Stress (MPa)		Contact Stress (MPa)	
Permissible (σFP)	Working (Actual) (σF)	Permissible (σHP)	Working (Actual) (σH)
411.2336	285.75	1668.8980	1396.36
Factor of Safety = 1.439		Factor of Safety = 1.195	

Conclusions

1. Theoretical stresses are calculated by ISO 6336 gear design procedure.
2. In helical gear, due to initial point contact the bending stresses are induced at critical section (root of tooth) i.e. permissible & working (actual) stresses are estimated and discussed
3. In this critique, the factor of safety for contact &bending stresses for helical gear have been calculated such as 1.19 &1.43 respectively shows design is safe and used for industrial applications.

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