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Force Analysis of Herring Bone Gears Used in Wind Mill Gear Box

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ABSTRACT: Generally, we know about the gear boxes are used to step-down & step-up process. In our project we consider about the windmill step-up gearboxes. Normally in this type of gear boxes we are using helical gears for step-up process. Instead of helical gears we are using herringbone gears. So the contact area may raise, the efficiency & effectiveness of the gear box will increase. Another main factor is life of the gearbox. It includes the life of internal parts like shafts, gears & bearings. Most considerable factor is gears, so we suppose to improve the life of the gears. For that we need to improve the hardness. Not only in improving hardness the change in material property is used for stress relieving also.

I. INTRODUCTION

1.1 GENERAL INTRODUCTON OF GEAR

Gearing is one of the most effective methods transmitting power and rotary motion from the source to its application with or without change of speed or direction. Gears will prevail as a critical machine element for transmitting power in future machines due to their high degree of reliability and compactness. The rapid development of heavy industries such as vehicle, shipbuilding and aircraft industries require advanced application of gear technology

A gearbox consists of a set of gears, shafts and bearings that are mounted in an enclosed lubricated housing. They are available in a broad range of sizes, capacities and speed ratios. Their function is to convert the input provided by the prime mover into an output with lower speed and corresponding higher torque. In this thesis, analysis of the characteristics of helical gears in a gearbox is studied using finite element analysis.

The crucial requirement of effective power transmission in various machines, automobiles, elevators, generators, etc.... has created an increasing demand for more accurate analysis of the characteristics of gear systems. For instance in automobile industry highly reliable and lightweight gears are essential. Further more the best way to minution of noise in engine requires the fabrication of silence gear system. Noise reduction in gear pairs is especially critical in the rapidly growing today's technology since the working environment is badly influenced by noise. The most successful way of gear noise reduction is attained by decreasing of vibration related with them. The reduction of noise by vibration control can be achieved through a research endeavor by an expert in the field.

Helical gears are currently being used increasingly as a power transmitting gear owing to their relatively smooth and silent operation, large load carrying capacity and higher operating speed. Designing highly loaded helical gears for power transmission systems that are good in strength and low level in noise necessitate suitable analysis methods that can easily be put into practice and also give useful information on contact and bending.

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1.2 HERRINGBONE GEAR

Herringbone gears use teeth with two opposing helices to transmit power and motion between parallel axes. The mirrored halves of the gear face have teeth that resemble an arrow head or the letter V. Some products have a machined center groove. Most herringbone gears are used with heavy machinery and large-diameter shafts. They apply equal force and friction to both gears and cancel the resulting thrust forces. Materials of construction for a herringbone gear include aluminum, brass, bronze, cast iron, carbon steel, hardened steel, and stainless steel. Herringbone gears are also made of plastics such as acetal, nylon, and polycarbonate. Gears that feature metal teeth with plastic inserts combine the strength and fit properties of a metal hub with the benefits of plastic teeth, which include quieter running and greater fit tolerances.

LITERATURE SURVEY

A herringbone gear, also known as a double helical gear,^[1] is a special type of [gear](#) which is a side to side (not face to face) combination of two helical gears of opposite hands.^[2] Unlike [helical gears](#) they can sustain axial load smoothly. From the top the helical grooves of this gear look like letter V.

Like the helical gear, they have the advantage of transferring power smoothly as multiple gear teeth engage and disengage simultaneously. Their advantage over the simple helical gear is that the side thrust of one half is counter-balanced by that of the other half, so that they can be used in a high-power gearbox without requiring an equally substantial [thrust bearing](#). Herringbone gears were thus an important step in the introduction of the [steam turbine](#) to marine propulsion.

They are used in heavy machinery. Herringbone gears are carefully made and of high precision, so they are more expensive than [spur gears](#) or [helical gears](#). Where the oppositely angled teeth meet in the middle of a herringbone gear, the alignment may be such that tooth tip meets tooth tip, or the alignment may be staggered, so that tooth tip meets tooth trough. The latter alignment is the unique defining characteristic of a Wuest type herringbone gear, named after its inventor.

Isay and Fong [2] applied the tooth contact analysis technique (TCA) and finite element method (FEM) to gear contact and stress analysis. In their study, a mathematical model for pinion and gear involute teeth is assumed. The geometry of the gears are described by parameters of manufacturing. Computer simulations of the conditions of gear meshing including the axes misalignment and center distance variation are performed. Their paper showed that the locations of bearing contact and contact pattern of mating tooth surfaces are determined by TCA techniques. The results of the TCA provide the location and the direction of applied loads for the computer aided FEM stress analysis, by applying the given mathematical model and TCA techniques.

A three-dimensional stress analysis for this type of gearing was investigated by Von-Mises stress contour distribution. Vijayarajan and Ganesan [3] presented a static analysis of composite helical gears system using three dimensional finite element methods to study the displacements and stresses at various points on a helical gear tooth. The validity of their results of the FEM was tested by the root stress for C-45 steel material gear and comparing the result with obtained from conventional gear design equation. The paper presented also the evaluation of the performance of composite helical gears by comparison with that of the conventional carbon steel gear. It is observed from the result that composite materials can be used safely for power transmission helical gears but the face width has to be suitably increased.

Huston et al [4] discussed a new approach to modeling gear tooth surfaces. A computer graphics solid modeling procedure is used to simulate the tooth fabrication processes. This procedure is based on the principle of differential geometry that pertains to envelopes of curves and surfaces. The procedure is illustrated with the modeling of spur, helical, bevel, and spiral bevel and hypoid gear teeth. Applications in design and Manufacturing are discussed. Extensions to nonstandard tooth forms, to cams, and to rolling element bearings are proposed.

In 1992, Rao and Muthuveerappan [5] explained about the geometry of helical gears by simple mathematical equations, the load distribution for various positions of the contact line and the stress analysis of helical gears using the three dimensional

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finite element methods. A computer program has been developed for the stress analysis of the gear. Root stresses are evaluated for different positions of the contact line when it moves from the root to the tip. To check the validity of the developed program, the changes in the trend of the maximum root stress values at various places of the tooth along the face width were compared with the experimental results. A parametric study was made by varying the face width and the helix angle to study their effect on the root stresses of helical gears. Based on their investigation the effect of helix angle and face width on the root stresses of helical gears was clarified for different positions of the contact line.

II. GEAR DESIGN CALCULATION

2.1 GEAR CALCULATIONS

Data: $W = 250 \text{ kW}$; $n_1 = 40 \text{ rpm}$; $n_2 = 1500 \text{ rpm}$; Life 10^8 cycles.

s $Z_1 = 100$, $Z_2 = 16$. Helix angle of 20° and normal pressure angle $\phi_n = 20^\circ$ are taken for the gears and $b = 1.2 \text{ pa}$ is assumed. $i = z_2 / z_1 = 16 / 100 = 0.16$ $\omega = 2\pi n_1 / 60 = 4.188 \text{ rad/sec}$

$$\text{Torque } T = 1000W / \omega =$$

$$59683.103 \text{ N.m Torque on each half is } T_1 = 59683.103 / 2 = 29841.550 \text{ N.m} = 29841550 \text{ Nmm.}$$

The AGMA bending stress equation:

$$\sigma = F_t K_v K_o (0.93 K_m) / b m_n J$$

$$o p = \pi m = \pi m_n / \cos \psi = \pi m_n / \cos 20 = 3.343 m_n$$

$$Pa = p / \tan \psi. \text{ Assuming } b = 1.2 \text{ pa} = 1.2 p / \tan \psi = 1.2 \times 3.343 m_n / \tan 20^\circ = 11.021 m_n \text{ Ft} = 2T_1 / d_1 = 2T_1 / mZ_1 = 2T_1 \cos \psi / m_n$$

$$Z_1 = 2 \times 29841550 \times \cos 20^\circ / m_n \times 100$$

$$= 560837.686 / m_n \text{ N J for the pinion with teeth } Z_{v1} = Z_1 / \cos^3 \psi$$

$$= 100 / \cos^3 20^\circ = 120.515, \psi = 20^\circ \text{ is: } J = 0.47 \text{ from Fig. 3.1}$$

$$J \text{ multiplier for mating with } Z_{v2} = Z_2 / \cos^3 \psi$$

$$= 16 / \cos^3 20^\circ = 19.28, \text{ is } J = 0.93 \text{ from Fig. 4.2}$$

$$\text{For pinion } J = 0.56 \times 0.93 = 0.5208$$

$$J \text{ factor for the gear with teeth } Z_{v2} = 19.28$$

o

$$\text{and } \psi = 20 \text{ is } J = 0.48 \text{ from Fig. 12.6 } J \text{ multiplier for mating with } Z_{v1} = 120.515 \text{ is } J = 1.010 \text{ from Fig. 12.7 For gear } J = 0.48 \times 1.010 = 0.4848$$

$$K_o = \frac{[78 + (200V)]}{78} = 1.25 \text{ assumed since } V \text{ is not known.}$$

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$K_o = 1.25$ assuming uniform source of power and moderate shock from driven machinery, Table 12.1.

$K_m = 1.4$ expecting $b = 225 \text{ mm}$ Accurate mountings, small bearing clearances, minimum deflection, precision gear For the pinion

$$\sigma_{b1} = F_t K_v K_o (0.93 K_m) /$$

$$b m_n J$$

$$= 560837.686 \times 1.25$$

$$\times 1.25 \times 0.93 \times 1.4 / 11.021 m_n^3 \times 0.5208$$

$$= 159025.294 / m_n^3$$

$$\text{For the gear } \sigma_{b2} = F_t K_v K_o (0.93 K_m) /$$

$$b m_n J$$

$$= 560837.686 \times 1.25 \times$$

$$1.25 \times 0.93 \times 1.4 / 11.021 m_n^3 \times 0.4848$$

$$= 213542.6291 / m_n^3$$

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The pinion & gear material is made from C45 steel with hardness 444 BHN and tensile strength $\sigma_{ut} = 1500$ MPa.

Corrected bending fatigue strength of the pinion: $\sigma_e = \sigma_e' k_L k_V k_S k_R k_T k_f k_m \sigma_e' = 0.5 \sigma_{ut} = 0.5 \times 1500 = 750 \text{ MPa}$

$k_L = 1.0$ for bending $k_V = 1.0$ for bending for $m \leq 10$ module, $k_S = 0.645$ for $\sigma_{ut} = 1500$ MPa $k_R = 0.897$ for 90% reliability $k_T = 1.0$ with Temp. $< 120^\circ\text{C}$, $k_f = 1.0$ $k_m = 1.33$ for $\sigma_{ut} = 1500 \text{ MPa}$ $\sigma_e = 750 \times 1 \times 1 \times 0.645 \times 1 \times 1 \times 0.897 \times 1 \times 1.33 = 577.118 \text{ MPa}$

Corrected bending fatigue strength of the gear:

$\sigma_e = \sigma_e' k_L k_V k_S k_R k_T k_f k_m$

$\sigma_e' = 0.35 \sigma_{ut} = 0.35 \times 1500 = 525 \text{ MPa}$ $k_L = 1.0$ for bending $k_V = 1.0$ for bending for $m \leq 10$ module, $k_S = 0.673$ for $\sigma_{ut} = 525 \text{ MPa}$ $k_R = 0.897$ for 90% reliability $k_T = 1.0$ with Temp. $< 120^\circ\text{C}$, $k_f = 1.0$ $k_m = 1.33$ for $\sigma_{ut} = 525 \text{ MPa}$ $\sigma_e = 525 \times 1 \times 1 \times 0.673 \times 1 \times 1 \times 1 \times 1.33 = 421.52 \text{ MPa}$

Permissible stress for the pinion in bending fatigue with factor of safety 1.6 for finite life gearing
 $[\sigma_b]_1 = \sigma_e / S_b = 577.118 / 1.6 = 360.69 \text{ MPa}$ Permissible stress for the pinion in bending fatigue with factor of safety 1.6,
 $[\sigma_b]_2 = \sigma_e / S_b = 421.52 / 1.6 = 263.45 \text{ MPa}$

For the pinion

$\sigma_{b1} = 159025.294 / m_n^3$
 $m_n^3 = 159025.294 / 360.69$
 $m_n = 7.22 \text{ mm}$

For the gear

$\sigma_{b1} = 213542.6291 / m_n^3$ $m_n^3 = 213542.6291 / 263.45$
 $m_n = 9.32 \text{ mm}$

Take a standard value of **10 mm**

$m = m_n / \cos 20^\circ = 10 / \cos 20^\circ = 10.641 \text{ mm} = 10 \text{ mm}$

$d_1 = m Z_1 = 10 \times 100 = 1000 \text{ mm}$ $d_2 = m Z_2 = 10 \times 16 = 160 \text{ mm}$ $p = 3.1416 m_n = 3.1416 \times 10 = 31.416 \text{ mm}$ $P_a = p / \tan \psi$
 $\psi = 11.021 / \tan 20^\circ = 110.21 \text{ mm}$

$b = 1.2 p_a = 1.2 \times 110.21 = 132.25 \text{ mm}$, take 135 mm $d_{a1} = d_1 + 2 m_n = 1000 + 2 \times 10 = 1020 \text{ mm}$ $d_{a2} = d_2 + 2 m_n = 160 + 2 \times 10 = 180 \text{ mm}$ Transverse pressure angle: $\tan \phi_n = \tan \phi \cos \psi$

$\Phi = \tan^{-1} (\tan \phi_n / \cos \psi) = \tan^{-1} (\tan 20^\circ / \cos 20^\circ) = 21.17^\circ$

$d_{b1} = d_1 \cos \phi = 1000 \cos 21.17^\circ = 939.69 \text{ mm}$

$d_{b2} = d_2 \cos \phi = 160 \cos 21.17^\circ = 150.35 \text{ mm}$ $C = 0.5(d_1 + d_2) = 0.5(1000 + 160) = 580 \text{ mm}$

$V = 0.5 \omega d_1 = 0.5 \times 4.188 \times 1000 \times 10^{-3} = 2.094 \text{ m/s}$

$F_t = 2 T_1 / d_1 = 2 \times 59683.103 / 1000 = 56083.76 \text{ N}$

Contact stress on the gears is given by $\sigma = C_p \sqrt{[(F_t \cos \psi) K_v K_o (0.93 K_m)] / b d l (0.95 CR)}$

$C_p = 191 \text{ (MPa)}^{0.5}$ for steel pinion & gear $l = (\sin \psi \cos \psi) i / 2(i+1) = (\sin 21.17^\circ \cos 21.17^\circ) 0.16 / 2(0.16+1) = 0.0232$
Contact ratio is given by

$CR_t = (\sqrt{[(r_1 + a)^2 + r_{b1}^2]} + \sqrt{[(r_2 + a)^2 + r_{b2}^2]} - (r_1 + r_2) \sin \psi) / \pi m \cos \psi$

Using standard tooth system with $a = 1 m_n$,

CR_t

$CR_t = (\sqrt{[(510+10)^2 + 469.845^2]} + \sqrt{[(90+10)^2 + 75.175^2]} - (510+90) \sin 21.17) / (\pi 10.641 \times \cos 21.17) = 4.2065$

$K = \frac{[78 + (200 \times 2.094)]}{1} = 1.123$

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$$K_v = 1.123 \quad K_o = 1.25 \quad K_m = 1.5$$

$$\sigma_H = C_p \sqrt{[(F_t \cos \psi K_v K_o (0.93 K_m)) / b d l] (0.95 CR)}$$

$$= 984.914 \text{ MPa}$$

Contact fatigue strength of pinion is $\sigma_{sf} = \sigma_{sf}' K_L K_H K_R K_T$

σ_{sf}' = surface fatigue strength of the material = 2.8 (BHN) – 69 From Table

$$4.5$$

$$= 2.8 \times 444 - 69$$

$$= 1174.2 \text{ MPa}$$

2.3 HELICAL GEAR – SURFACE FATIGUE STRENGTH

Table 3.5 Surface fatigue strength σ_{sf}' (MPa) for metallic spur gears (10^7 cycle life, 99% reliability and temperature < 120°C)

$$K_L = 0.9 \quad \text{for } 10^8 \text{ cycles from Fig.4.6}$$

$$K_H = 1.0 \quad \text{for } K = 444/444 = 1 \text{ \& } i=4$$

$$K_R = 1.0 \quad \text{for 99\% reliability } K_T = 1.0 \quad \text{assuming temp. < } 120^\circ\text{C}$$

$$\sigma_{sf} = \sigma_{sf}' K_L K_H K_R K_T = 1174.2 \times 0.9 \times 1.0 \times 1 \times 1$$

$$= 1056.78 \text{ MPa}$$

K_H K = Brinell hardness ratio of pinion and gear, $K_H = 1.0$ for values of K below 1.2

K_T = temperature factor,
= 1 for $T \leq 120^\circ\text{C}$ based on Lubricant temperature.
Above 120°C , it is less than 1 to be taken from AGMA standards.

2.2 HELICAL GEAR – ALLOWABLE SURFACE FATIGUE STRESS (AGMA)

Allowable surface fatigue stress for design is given by $[\sigma_H] = \sigma_{sf} / s_H$

Design equation is: $\sigma_H \leq [\sigma_H]$

$$\text{For gear: } \sigma_{sf}' = 0.95[2.8(Bhn)-69] = 0.95[2.8 \times 444 - 69] = 1174.2 \text{ MPa}$$

$$K_L = 0.97 \quad \text{for } 2.5 \times 10^7 \text{ cycles from Fig.4.6}$$

$$K_H = 1.0 \quad \text{for } K = 444/444 = 1.0 \text{ \& } i=4$$

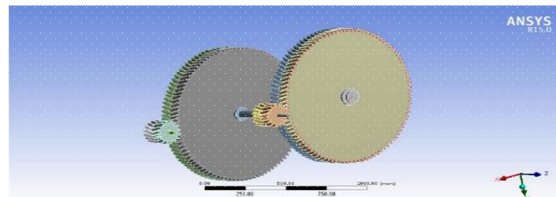


Fig 3.1 Herring bone Gear arrangements

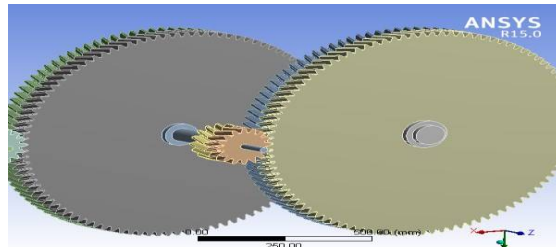


Fig 3.2 Herring bone Gear model

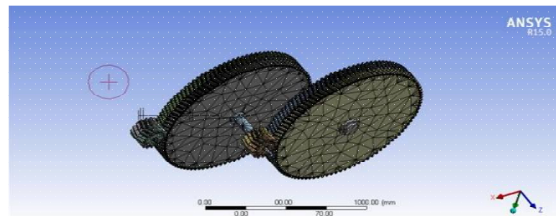


Fig 3.3 Meshing

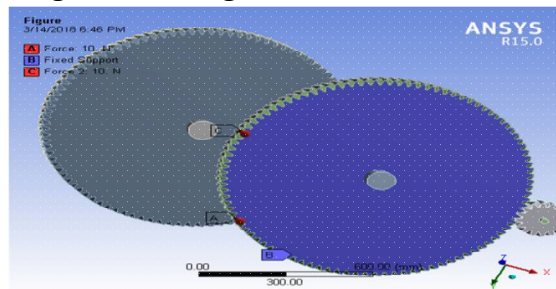


Fig 3.4 Force acting

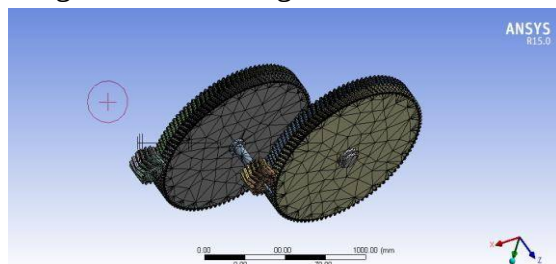


Fig 3.5 Shape finder

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$K_R = 1.0$ for 99% reliability $K_T = 1.0$ assuming temp. $< 120^\circ\text{C}$ $\sigma_{sf} = \sigma_{sf}' K_L K_H K_R K_T = 1174.2 \times 0.97 \times 1.0 \times 1 \times 1 = 1056.78 \text{ MPa}$

Factor of safety for the pinion against pitting: $s_{H1} = \sigma_{sf} / \sigma_H = 1056.78 / 984.914 = 1.072$

Factor of safety for gear against pitting: $s_{H2} = \sigma_{sf} / \sigma_H = 1056.78 / 984.914 = 1.072$

III. SHAPE OPTIMIZATION ANALYSIS

3.1 MODELING OF THE GEAR

S.No	Helical Gear Box	Herring Bone Gear Box
1.	Material – Chrome vanadium	Material – Nickel chromium
2.	Ultimate Tensile Strength = 13700 kgf / cm ²	Ultimate Tensile Strength = 15500 kgf / cm ²
3.	Produce Axial Force	Axial Force is Zero
4.	Life of Gears Theoretical – 32,000 Hours Actual – 29,000 Hours	Life of Gears Theoretical – 44,000 Hours Actual – 40,000 Hours (Assume)
5.	Rockwell hardness of material – 41	Rockwell hardness of material – 57
6.	Cost of gear – Rs 68,000	Cost of gear – Rs 72,000

IV. CONCLUSION

Based on the result from the contact stress analysis the hardness of the gear tooth profile can be improved to resist pitting failure: a phenomena in which a small particle are removed from the surface of the tooth that is because of the high contact stresses that are present between mating teeth.

By changing the gear type

- Elimination of the axial force is takes place.
 - The efficiency and effectiveness of the gear box is also improved. By changing the gear material
- The life factor of the internal parts which are used in the gear box was improved. Because of the improved hardness By considering these three factors in our concern, the herringbone gears will be the best choice for wind mill gearboxes

4.1 RECOMMENDATION AND FUTURE WORK

This thesis paper can be an interest for researchers, instructors and postgraduate students who have great enthusiasm to work more on gears. It may give enlightenment about the characteristics of involute herringbone gears and evoke pervious works of various bodies that are involved in gears research and production. Further more this study contribute to a better gear design, assist technological

institutions and all those who are interested in involute herringbone gears. More work can be done to improve this study and to obtain better output.

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