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Research Article

Parametric investigation of the combined effect of Gear parameters on Tangential Force and Dynamic Tooth Load of 40 Ni2 Cr1 Mo 28 Steel Helical Gear

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Abstract

Global competitive business market had brought increasing awareness to optimize the gear design. Gears have the advantage over friction and belt drives in that they are positive drive, a feature which most of the machine tools require, since exact speed ratios are essential. The design and manufacture of precision cut gears, made from materials of high strength, have made it possible to produce gears which are capable of transmitting extremely large loads at extremely high circumferential speeds with very little noise, vibration and other undesirable aspects of gear drives. The present work is to focus on investigating the combined effect of gear parameters on tangential force and dynamic tooth load of helical gear.

Keywords: *Buckingham equation, Helical gear design, Optimal design, Tangential force, Dynamic tooth load.*

1. Introduction

The major concerns of gear analysis deals with the analysis of gear stresses, transmission errors, dynamic loads, noise, and failure of gear tooth, which are very useful for optimal design of gear set. In gear analysis, form input parameters which influence the output parameters viz. bending stress, compressive stress, tangential force, dynamic tooth load, wear tooth load, beam strength [Venkatesh et al. 2011], are of interest to researchers and manufacturers. Solid modeling of gear teeth [Huston et al. 1992] with application in design and manufacture is proposed. They have presented a new approach to modeling of gear tooth surfaces. A computer graphics solid modeling is used to simulate the tooth fabrication processes. This procedure is based on the principles of differential geometry that pertain to envelopes of curves and surfaces. The procedure is illustrated with modeling of spur, helical, bevel, spiral bevel and hypoid gear teeth. They have also proposed non standard tooth forms, to cams and to rolling element bearings. Tooth contact analysis technique (TCA) and finite element method (FEM) [Tsay and Fong 1991] to gear contact and stress analysis investigated. In this study, a mathematical model for pinion and gear involute teeth is assumed. The geometry of the gears is described by parameters of manufacturing. Computer simulations of the conditions of gear meshing including the axes misalignment and center distance variation were considered. 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 of type of gearing was investigated by Von-Mises stress contour distribution. 3D stress analysis of helical gear teeth for calculating root stresses [Rama Mohana Rao and Muthuveerappan 1993] analysis presented. Root stresses are evaluated for different positions of the contact line when it moves from the root to the tip. A parametric study was made by varying the face width and the helix angle to study their effect on the root stresses of the helical gears. Their study revealed that the three dimensional way of calculating the root stresses of the helical gear gives realistic results. Static analysis of composite helical gears using a three-dimensional finite element method [Vijayarangan and Ganeshan 1993] is presented. Conventional material helical gears can be replaced by helical gears made of composite materials. This results in light-weight helical gears, which add to the existing list of advantages of helical gears, such as smooth and silent operation and larger load carrying capacity, while at the same time maintaining higher strengths, which is an important requirement in applications like spacecraft and aircraft. They have compared the performance of Kevlar epoxy and graphite epoxy material helical gears with that of carbon steel helical gears using a three-dimensional finite element The rotor dynamics of a geared system in helical gear, lateral, torsional and axial vibrations [Choi.S.H et al. 1999] investigated in understanding the severe vibration problems that occurred on a 28-MW turbo

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set consisting of steam turbine, double helical gear and generator. The new dynamic model of the shaft line was based on the most accurate simulation of the static shaft lines, which are influenced by variable steam forces and load dependent gear forces. The gear forces determine the static shaft position in the bearing shell. Rigid disks and distributed springs were used for shaft line modeling. The tooth contact was modeled by distributed springs acting normally on the flank surfaces of both helices. A finite element method with distributed mass was used for lateral and torsional vibrations. It was coupled to a lumped mass model describing the axial vibrations. The forced vibrations due to unbalances and static transmission errors were calculated. The eigen value problem was solved by means of a stability analysis showing the special behavior of the coupled system examined. A model based upon a finite element procedure for analyzing the influence of tooth friction on spur and helical gear dynamics [Velex and Cahouet 2000] was proposed. The equations of motion are solved by combining a time-step integration method with several iterative algorithms aimed at satisfying normal and tangential contact conditions. Comparisons between simulated and measured quasi-static bearing forces are satisfactory and largely validate the theoretical developments. Their results reveal the potentially significant contribution of tooth friction to gear vibration and noise. Simulations are then extended to high speeds and the benefit of considering both transmission error and tooth friction excitations to achieve silent gears is discussed. A dynamic model which considers the effects of sliding friction, tooth shape deviations and time-varying possibly non-linear mesh stiffness in spur and helical gears has been described. Comparisons between measured and simulated bearing forces show that the proposed model correctly reproduces the influence of tooth friction on the geared system behavior. To compute the contact force and tooth deflection [Park and Yoo 2004] was presented. The contact forces between teeth are determined from the transmitted torque, and then the deformation overlap is calculated with the contact forces as boundary conditions. They extended the theory to three dimensional problems and implemented to helical gear pair. Study involving the tooth flank correction of power transmission helical gears with wide face width using a finite element based shaft deflection analysis program in conjunction with a numerical load distribution analysis procedure [Shan Chang et al. 2005] was discussed. The load distributions along the line of action of gear toot, the elastic deflections, transmission errors of gear pairs obtained by solving the equations of compatibility of displacement and equilibrium of forces. The authors also discussed the influence of tooth flank corrections (tip relief, root relief, load modification, end relief and their combinations) on gear stresses and transmission errors due to shaft deflections. This technique has the capability of modeling all significant geometric and elastic contributions due to tooth contact of the pair being analyzed as well as other gears mounted on the same shafts. Their results indicate that it is possible to optimize, at the design stage, the gear micro-geometry for minimum stresses and transmission errors without changing the gear macro-geometry. Nonlinear finite element contact mechanics model of a helical gear pairs [Wagaj and Kahraman 2002] was demonstrated. They have carried out detailed parametric study to quantify the changes in the contact and bearing stresses as a function of tooth profile, combined influence of modification parameters and torque transmitted on the maximum stresses is described. Results indicated that both bending and contact stresses are applied, potentially causing an underestimation of the actual stress values when the modifications are not included in stress calculations. Ross and Buckingham studied the prediction of tooth dynamic loads for designing gears [Alec strokes 1970] at higher speeds. A method for minimising the centre distance of a helical gear set of prescribed power capacity using a fixed ratio of face width to PCD and formulae contained within AGMA procedures [X.Li et al. 1996] reported. Although power capacity is dependent upon many factors, the method has used module, numbers of teeth, and helix angle as the significant variables. The procedure is based upon determining limiting values of AGMA geometry factors I and J which are derived in order to establish upper and lower boundaries for the search interval of centre distance. The method is effective in that the minimum centre distance is a global optimum design or near global optimum design when the local non-monotonicity affects the final result. Based on modeling approximations and assumptions, organizations like AGMA (American Gear Manufacturers Association) and BS (British Standards) provides standards for gear useful lifetime formulations. The literatures such as “gearing”, “High performance gear design ” and design data [Shigley et al. 1996] was useful in designing the helical gear according to AGMA standards.

2. Design methodology

The design of helical gear is almost similar to spur gear design with slight modifications in Lewis and Buckingham equations due to helix angle [Venkatesh et al. 2010]. According to Lewis equation, the Beam Strength of helical gear tooth is given by

$$F_b = [\sigma_b].b.\ \pi m_n \cdot y_v \tag{1}$$

Tangential force

$$F_t = M_t/d \tag{2}$$

Where, $[\sigma_b]$ = Allowable contact stress in kgf/cm^2

b = Face width of gear blank = $10m_n$,

m_n = Normal module which must be standardized.

M_t = Torque Transmitted

y_v = Lewis form factor which depends on the virtual number of teeth

$Z_v = Z_v = Z/\cos^3 \beta$

The dynamic load acting on helical gear tooth may be found out using Buckingham equation as

$$F_d = F_t + \frac{21v\left(Cb\cos^2\beta + F_t\right)\cos\beta}{21v + \sqrt{Cb\cos^2\beta + F_t}} \tag{3}$$

3. Results and discussion

The numerical analysis of the helical gear design is carried out by using MATLAB program. The MATLAB does its internal arithmetic in floating point precision using approximately 16 decimal digits, but the default display is only 5 decimal digits. Fig 1 shows the variation of dynamic tooth load and tangential load for different modules and different face width for Steel alloy. For the set of gear ratio (i) = 7 and helix angle (β) = 25 are kept constant. The face width of 41 is kept constant and the module is varied from 16 to 24. The corresponding Tangential force observed to decrease from 154640.8kgf to 103093.9kgf, Dynamic tooth load decrease from 156970.6kgf to 105955.8kgf. The face width of 43 is kept constant and the module is varied from 16 to 24. The corresponding Tangential force decrease from 154640.8kgf to 103093.9kgf, Dynamic tooth load decrease

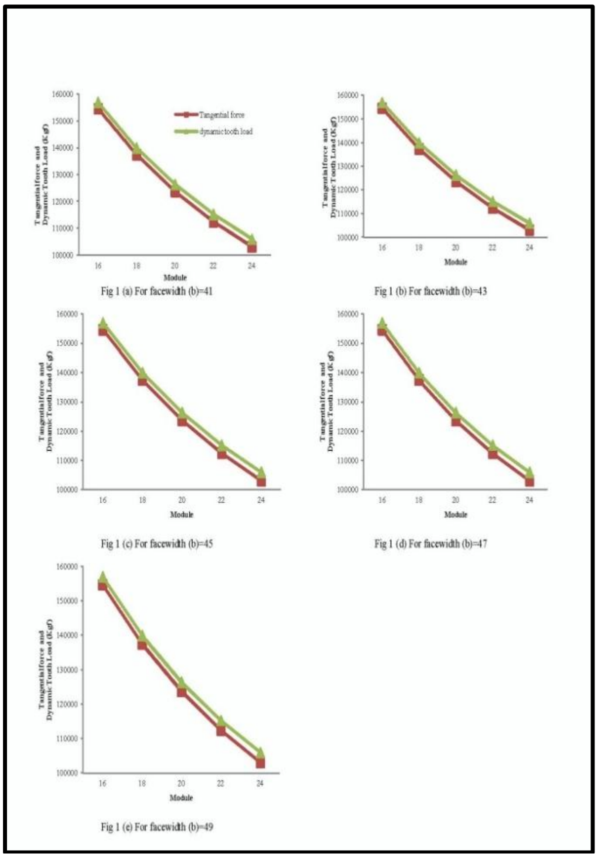

Fig 1 Effect of module on tangential force and dynamic tooth load

from 156974.2kgf to 105962.3kgf. The face width of 45 is kept constant and the module is varied from 16 to 24. The corresponding tangential force decrease from 154640.8kgf to 103093.9kgf, Dynamic tooth load observed to decrease from 156977.8 kgf to 105968.9kgf. The face width of 47 is kept constant and the module is varied from 16 to 24. The corresponding tangential force decrease from 154640.8kgf to 103093.9kgf, Dynamic tooth load decrease from 156981.5 kgf to 105975.4kgf. The face width of 49 is kept constant and the module is varied from 16 to 24.

The corresponding tangential force found to decrease from 154640.8kgf to 103093.9kgf, Dynamic tooth load decrease from 156985.1kgf to 105981.9kgf.

3.1 Optimum values

Optimum values of dynamic tooth load and tangential force to achieve low cost manufacturing for Steel alloy have been carried out.

3.1.1 The effect of gear ratio, face width, helix angle, module on Dynamic tooth load for steel alloy

The variation of Dynamic tooth load for different input variables (Venkatesh et al. 2011) are shown in figs. 2(a) – (d). The speed is kept constant. The fig 2(a) shows the relationship between Dynamic tooth load and gear ratio. The helix angle, face width, and module are kept constant. When gear ratio is increased from 4 to 8, the corresponding Dynamic tooth load remained constant. The fig 2(b) shows the relationship between Dynamic tooth load and Face width.

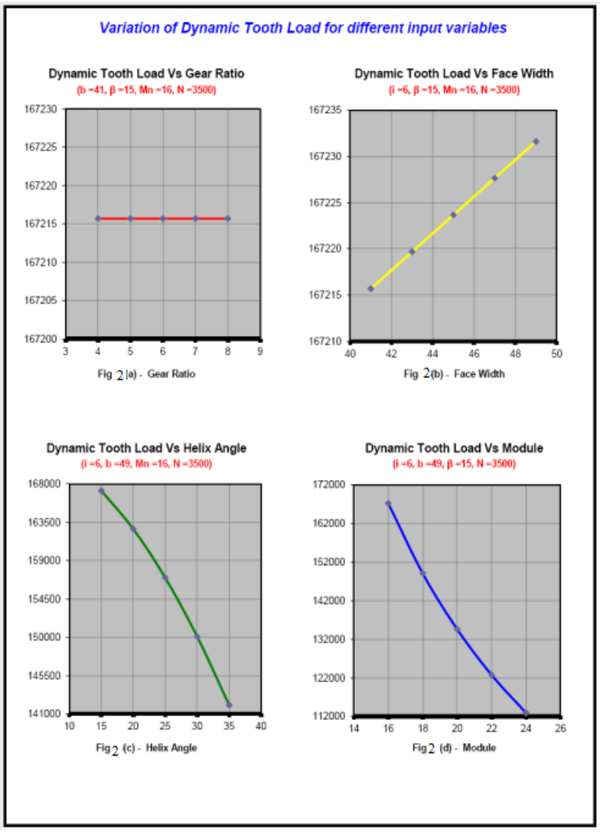

Fig 2 Variation of dynamic tooth load for different input variables

The Helix angles, gear ratio, module are kept constant. When face width is increased from 41 to 49, the corresponding Dynamic tooth load is increases from 167216 kgf to 167232 kgf. The fig 2(c) shows the relationship between Dynamic tooth load and Helix angle. The face width, gear ratio, corresponding to optimum value obtained earlier and module except Helix angle are kept constant. When helix angle is increased from 15^o to 35^o, the corresponding Dynamic tooth load decreased from 167232kgf to 142001kgf. The fig 2(d) shows the relationship between Dynamic tooth load and module. The value of face width, gear ratio, and Helix angle for maximum dynamic tooth load are kept constant. When module is increased from 16mm to 24mm, the corresponding Dynamic tooth load decreased from 167232kgf to 112868kgf. Thus the maximum dynamic tooth load 167232kgf is obtained for input parameters viz. gear ratio (i) = 6, helix angle (β) = 15^0 , face width (b) = 49 and Module (Mn) =16.

3.1.2. The effect of gear ratio, face width, helix angle, module on tangential force for steel alloy

The variations of Tangential force for different input variables are shown in figs. 3(a) – (d). The speed is kept constant. The Fig 3(a) shows the relationship between Tangential force and gear ratio. The helix angle, face width, and module are kept constant. When the gear ratio is increased from 4 to 8, the corresponding Tangential force remained constant. The fig 3(b) shows the relationship between Tangential force and Face width. The Helix angles, gear ratio, module are kept constant. When face width is increased from 41 to 49, the corresponding Tangential force remained constant. The fig 3(c) shows the relationship between Tangential force and Helix angle. The face width, gear ratio, corresponding to maximum value obtained earlier and module are kept constant. When Helix angle is increased from 15^o to 35^o, the corresponding Tangential force decreased from 164802kgf to 139785kgf. The fig 3(d) shows relationship between Tangential force and module. The value of face width, gear ratio, and helix angle for maximum tangential force are kept constant. When module is increased from 16mm to 24mm, the corresponding Tangential force found to decrease from 164802kgf to 109868kgf. Thus the maximum Tangential force 164802 kgf is obtained for input parameters viz. gear ratio (i) = 6, helix angle (β) = 15^0 , face width (b) = 41 and Module (Mn) =16.

3.2 Results of steel alloy material

Optimum parameters for maximum dynamic tooth load: The effect of gear ratio, face width, helix angle, and module on optimum dynamic tooth load is carried out [Venkatesh et al. 2011]. The helix angle, face width, speed and module are kept constant. When the gear ratio is increased, the corresponding dynamic tooth load remained constant. Keeping the helix angle, gear ratio, speed, module except face width are kept constant and for variation of face width, the dynamic tooth load remained constant. In the next step the face width, gear ratio, speed and module are kept constant and helix angle is increased, the corresponding dynamic tooth load is decreases. The helix angle 15^0, corresponding to maximum dynamic tooth load is taken for further optimization. The face width, gear ratio, speed, helix angle are kept constant and when module is varied, the corresponding dynamic tooth load decreases. The module 16mm, corresponding to maximum dynamic tooth load is taken constant. Optimum parameters for maximum tangential force: The effect of gear ratio, face width, helix angle and module on optimum tangential force is carried out. Keeping the helix angle, face width, speed and module as constant. When the gear ratio is increased, the corresponding tangential force remained constant. Keeping the helix angle, gear ratio, speed, module except face width is kept constant and for variation of the face width, the corresponding tangential force remained constant. In the next step the face width, gear ratio, speed and module are kept constant and helix angle is increased, the corresponding Tangential force decreased. The helix angle 15^0 corresponding maximum tangential force is taken for further optimization. The face widths, gear ratio, speed, helix angle are kept constant and module is varied, the corresponding tangential force found to decrease. The module 16mm, corresponding to maximum tangential force is taken as constant.

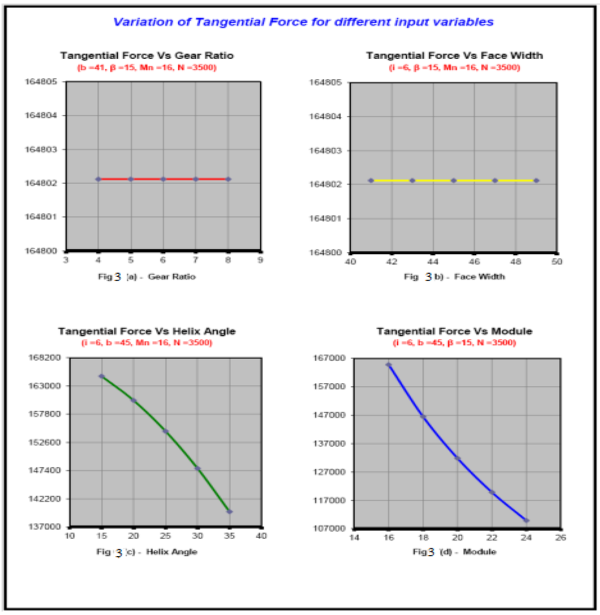

Fig 3 Variation of tangential force for different input variables

Conclusions

Current trends in engineering globalization require results to comply with various normalized standards to determine their common fundamentals and those approaches needed to identify best practices in industries. This can lead to various benefits including reduction in redundancies, cost containment related to adjustments between manufacturers for missing part interchangeability, and performance due to incompatibility of different standards. From this study, investigated the effect of tangential force and dynamic tooth load on the optimum design of helical gears.

References

B.Venkatesh, V.Kamala, A.M.K.Prasad, (2011) Parametric Approach to Analysis of helical gear for High Speed Marine Applications, *International Journal on Mechanical & Automobile Engineering*, Volume 17, Issue 1, pp 13-20.

Ch.Ramamohana Rao, G.Muthuveerappan, (1993) Finite Element modeling and stress analysis of helical gear Teeth. *Journal of computers and structures* Vol. 49, No. 6 pp.1095-1106.

B.Venkatesh, V.Kamala, A.M.K.Prasad,(2010) Design, Modeling & Manufacturing of helical gear, *International Journal of Applied Engineering Research,* Vol 1,No 1, pp 103 -114.

P.Wagaj, A.Kahraman, (2002) Impact of Tooth Profile Modifications on the Transmission error Excitation of Helical Gear Pairs, *6th Biennial Conference on Engineering Systems Design and Analysis*, Istanbul-Turkey.

B.Venkatesh, V.Kamala, A.M.K.Prasad, (2011) Parametric Approach to Analysis of high Speed Helical Gear with 40 Ni2 Cr1 Mo28 Steel for Marine Applications, *International Journal of RES Engineeria*, Vol.1, Issue 4, pp. 6-10.

S.Vijayarangan, N.Ganesan,(1993) A Static Analysis of Composite Helical Gears using Three-dimensional Finite Element Method, *Journal of computers & structures*, 49, pp.253-268.

S.H. Choi, J. Glienicke, D.C. Han, K.Urlichs, (1999) Dynamic Gear Loads due to coupled lateral, Torsional and Axial Vibrations in a helical Geared System, *Journal of vibration and acoustics*, Vol 121/141.

P.Velex, V.Cahouet, (2000) Experimental and numerical Investigations on the Influence of Tooth Friction in spur and Helical gear Dynamics, *ASME Journal of Mechanical Design*, Vol. 122, pp.515-522.

S.J.Park, W.Yoo, (2004) Deformation Overlap in the Design of spur and Helical Gear pair, *Journal of Finite Elements in Analysis and Design*, 40, pp.1361-1378

Shan Chang, R.H.Donald, J. Harianto, (2005) Tooth Flank corrections of Wide Face width Helical Gears that Account for Shaft Deflection, *Gear Technology*.

X.Li, G.R.Symmons, Cockerham, (1996) Optimal design of involute Profile Helical Gears, *Journal of Mechanism and Machine Theory*, Vol. 31 No 6, pp. 717-728.

Alec strokes,(1970) High performance of gear design.

J.E.Shigley, C.R.Mischke, (1996) Standard hand book of Machine Design, Mc Graw-Hill, USA.