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Effect of Drive Side Pressure Angle on Load Carrying Capacity of Asymmetric Involute Helical Gear

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***Abstract** : Efficiency In this paper, the effect of drive side pressure angle on load carrying capacity of asymmetric helical gear is analyzed. The drive pressure angle of asymmetric gear tooth is varied from 20° to 40° in steps of 5° while the coast side pressure angle is kept constant (20°). Initially, the reduction in contact stress is analyzed and then the consequent increase in load carrying capacity is studied with increase in drive side pressure angle. Three dimensional model of gears are made on Solidworks®. The gears have been designed using AGMA standards. In each calculation the meshing parameters and boundary conditions are kept constant. It is found that as the drive side pressure angle increases, the contact stress on teeth reduces as compared to symmetric involute helical gear (20° drive and coast side pressure angle each). Maximum reduction in contact stress occurs in gear with drive side pressure angle of 40° with 25.4% reduction in stress. Due to maximum reduction in contact stress, the load carrying capacity of gear is increased by 34% in asymmetric gear with drive side pressure angle of 40°.*

***Keywords**: Asymmetric Helical Gear, Contact Stress, Drive pressure angle, Load Capacity, AGMA*

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1. INTRODUCTION

Helical gears are extensively used to transmit power by gears due to their relatively smooth operation and high load capacity. In many gear applications the gears rotate in only one direction and rarely in the reverse direction. Due to this, the pressure on loaded (drive) profile of tooth is significantly higher than that on unloaded (coast) profile of the tooth. The gear tooth design can be made more efficient by using different pressure angles on the drive and coast profile of gear tooth. Such gears with different drive and coast pressure angles are known as asymmetric gears.

The designing of asymmetric tooth gear was explained in great depth by Kapelevich [1]. He described designing of asymmetric tooth gearing, analytical and experimental comparison of symmetric and asymmetric tooth gears. Gawali et al. [2] analyzed the effect of coefficient of asymmetry on strength and contact ratio of different helical gear pairs. They further validated the results using digital strain gauge indicator mounted at the root of the gear.

Sondur et al. [3] described a method for investigation of bending stress at the critical section of asymmetric involute spur gear. They concluded that the stress was reduced by 9% which in turn can increase the load carrying capacity of gears. Anusha et al. [4] analyzed contact stress generated in symmetric helical gear by using AGMA and ANSYS. They studied the contact stress by varying pressure angle, helix angle and face width of the gear.

In the past, most of the analysis of asymmetric helical gears were focused on study of bending stress in gears but very little work has been done on study of contact stress on asymmetrical helical gear tooth. Since contact stress is one of major factor influencing pitting of gear tooth surface, thus, it is felt necessary to find out the effect of load side pressure angle on variation of contact stress in asymmetric helical gears. Further, it is important to study change in load carrying capacity of asymmetric helical gears as compared to symmetric helical gears, to ascertain advantage of asymmetric helical gears over symmetric helical gears.

2. GEAR DESIGN

Conventional involute spur gears are designed with symmetric involute tooth profiles. However the conditions of load and meshing are different for drive and coast profile of gear tooth. Application of asymmetric tooth side surfaces enables to increase the load capacity and durability for the drive tooth side. Design of Asymmetric Involute helical gear is governed by various fundamental equations as given [1].

2.1 Asymmetric Tooth Design

An Asymmetric Involute tooth is formed by two involutes with same Pitch diameter (D_p), but different pressure angle for drive and coast sides (φ_d , φ_c) and correspondingly different Base Circle diameters (D_{bd} , D_{bc}).

$$D_b = D_p * \cos(\varphi) \quad (1)$$

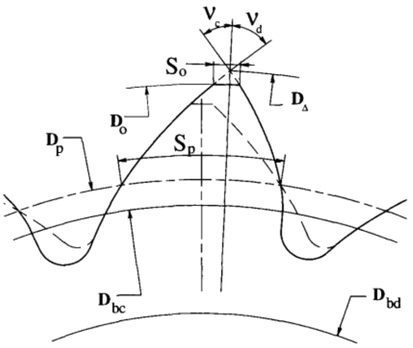

Figure 1. Asymmetric Involute tooth profile [1]

For generation of involute spur gear following parametric equations are used:

$$x(t) = D_p/2*(\cos \varphi)*[\cos(t)+ t*\sin(t)] \quad (2)$$

$$y(t) = D_p/2*(\cos \varphi)*[\sin(t)- t*\cos(t)] \quad (3)$$

where, t is angle parameter in radius;
x(t) is a function with respect to parameter t;
y(t) is a function with respect to parameter t

2.2 Gear Specifications

Various Parameters used while designing the Pinion and Gear are given in Table 1.

Table 1: Parameters for Asymmetric Helical gear design

Parameters	**Value**
Material	Structural Steel
Module (m)	4.5 mm
Gear Ratio (G)	1
Number of teeth (N)	25
Top Land Thickness Coefficient (m_o)	0.016
Drive side Pressure Angle (φ_d)	20^o, 25^o, 30^o, 35^o, 40^o
Coast side Pressure Angle (φ_c)	20^o
Helix Angle (β)	15^o
Face Width (b)	27 mm
Addendum (a)	4.5 mm
Dedendum (d)	5.625 mm
Clearance (c)	1.125 mm

3. AGMA Contact Stress Equation

A very important parameter when designing a gear pair is the maximum surface contact stress that exists between two gear teeth in mesh, as it affects surface fatigue along with gear mesh losses. The fundamental Contact Stress (σ_c) formula for gear teeth is given in ANSI/AGMA 2001-D04 [9]

$$\sigma_c = C_p \sqrt{W_t K_o K_v K_s \frac{K_m}{d\,F} \frac{C_f}{I}} \quad (3)$$

4. Finite Element Method

The Finite Element Method (FEM) is a numerical technique used to perform Finite Element Analysis (FEA) of any given physical phenomenon. ANSYS Workbench is used for the elemental analysis of contact stress in gear. The analysis is performed on Static Structural component system in the Ansys Workbench.

The solid model is made on Solidworks® software and imported to Ansys Workbench as parasolid file. Fig. 2 to Fig. 6 show solid model of asymmetric helical gears with constant helix angle of 15° and coast side pressure angle of 20°. The drive side pressure angle is varied from 20° to 40° in steps of 5°.

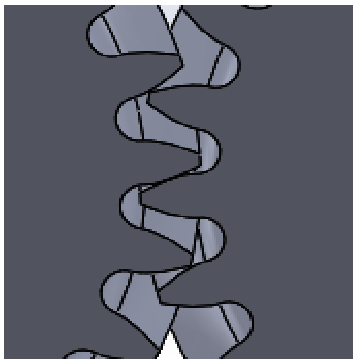

Figure 2. Helical gear model for $\varphi_d = 20°$, $\varphi_c = 20°$, β=15°

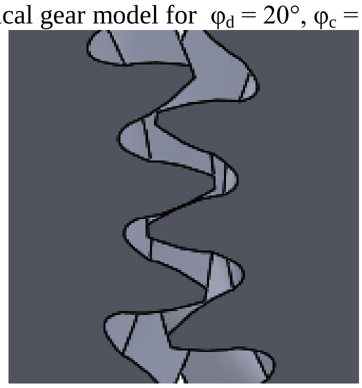

Figure 3. Helical gear model for $\varphi_d = 25°$, $\varphi_c = 20°$, β=15°

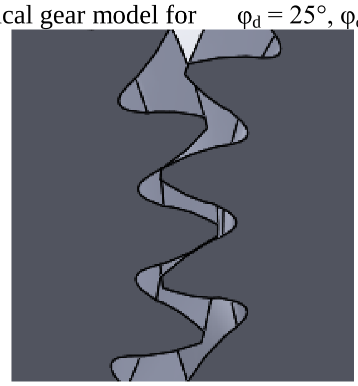

Figure 4. Helical gear model for $\varphi_d = 30°$, $\varphi_c = 20°$, β=15° (Symmetric gear)

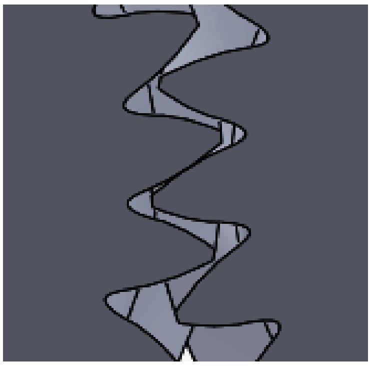

Figure 5. Helical gear model for $\varphi_d = 35°$, $\varphi_c = 20°$, $\beta=15°$

Figure 6. Helical gear model for $\varphi_d = 40°$, $\varphi_c = 20°$, $\beta=15°$

The meshing is done with medium relevance center and minimum edge length of 1.7mm. Total of 51,593 elements are generated with 83,059 nodes on the (20°/20°) symmetric gear with 15° helix angle. Suitable boundary conditions are given with moment load of 100 N-m and the results are calculated. The stress value is compared with theoretical AGMA stress value as shown in Table 2.

To calculate the change in load carrying capacity, the moment on pinion gear is increased till the stress value is equal to the stress induced in symmetrical (20°/20°) helical gear. The resultant load value is noted and compared with initial load to calculate the percentage increase in load carrying capacity.

5. Results and Discussion

To verify the solver settings and the boundary conditions, a test case is chosen as symmetric helical gear with 20°/20° pressure angle and 15° helix angle. The contact stress is calculated at the pitch point of gear using AGMA stress equation and then compared with the Finite Element Analysis (FEA) value. The magnitude of Contact Stress evaluated using ANSYS is found to be 393.26 MPa as compared to 393.78 MPa calculated using AGMA contact stress equation.

Table 2: Comparison of AGMA and FEA Contact Stress Values

Parameter	AGMA Value (MPa)	FEA Value (MPa)	Difference (%)
σ_C	393.78	393.26	0.13

Thus, both AGMA and FEA values are in close agreement with each other with deviation of 0.13%.

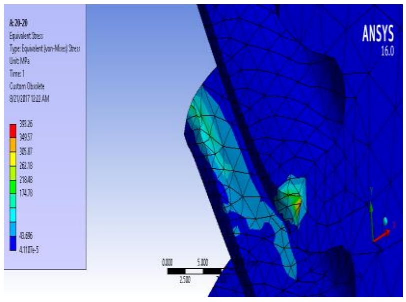

Figure 7. Contact Stress in 20°/20° Gear

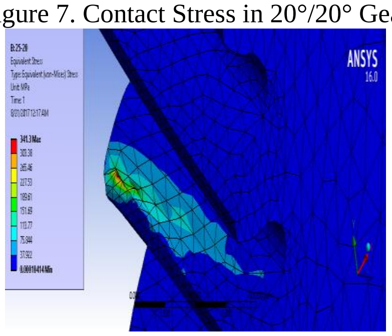

Figure 8. Contact Stress in 25°/20° Gear

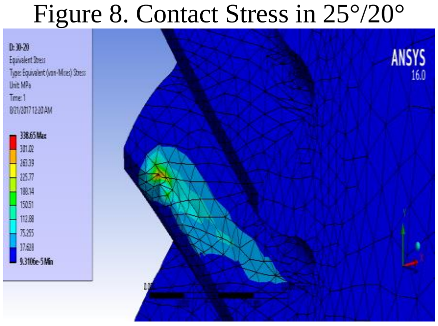

Figure 9. Contact Stress in 30°/20° Gear (Symmetric gear)

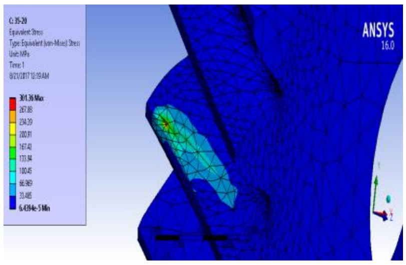

Figure 10. Contact Stress in 35°/20° Gear

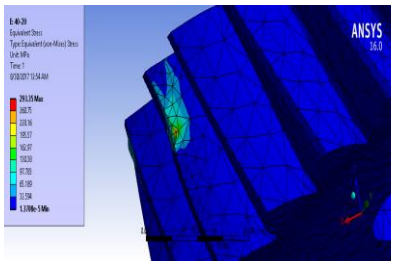

Figure 11. Contact Stress in 40°/20° Gear

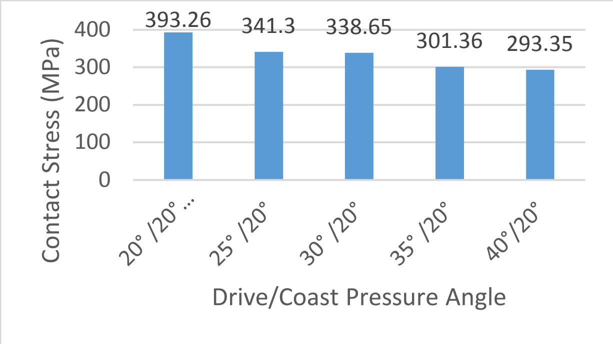

Figure 12. Variation of Contact Stress with Drive Side Pressure Angle

Table 3: Percentage Increase in Load Capacity of Gear

	Contact Stress (MPa)	Percentage reduction in contact stress (%)
20°/20° (Symmetric)	393.26	-
25°/20°	341.3	13.21
30°/20°	338.65	13.88
35°/20°	301.36	23.37
40°/20°	293.35	25.4

Fig. 7 to Fig. 11 show variation in contact stress with drive side pressure angle as obtained using finite element analysis. The contact stress continuously reduces with increase in drive side pressure angle. Fig. 12 further depicts the variation in contact stress using bar graph. The percentage reduction in contact stress on gear tooth for different values of drive side pressure angle is shown in Table 3. It is clearly seen that maximum reduction in contact stress as compared to that in symmetrical helical gear occurs in gear with drive side pressure angle of 40° with 25.4% reduction in contact stress.

Table 4: Percentage Increase in Load Capacity of Gear

	Initial Load (N-m)	Final Load (N-m)	% Increase in load
20°/20° (Symmetric)	100	100	-
25°/20°	100	115.3	15.3
30°/20°	100	116.3	16.3
35°/20°	100	130.5	30.5
40°/20°	100	134	34

Table 4 shows percentage increase in load carrying capacity of asymmetric helical gears. All asymmetric tooth profiles show considerable increase in load capacity however, the gear with 40° drive side pressure angle shows maximum increase (34%) in load capacity amongst all gears. This increase in load carrying capacity of gear is a direct result of reduction in contact stress due to increase in drive side pressure angle.

6. Conclusion

In the light of above discussions, the following conclusions are drawn –

- Contact stress in asymmetric helical gears decreases with increase in Drive side pressure angle.
- Maximum reduction in contact stress occurs in gear with drive side pressure angle of 40° with 25.4% reduction in stress.
- Load carrying capacity of gears increases remarkably with increase in dive side pressure angle.
- Gear with drive pressure angle of 40° shows 34% increase in load carrying capacity which is maximum among all.
- By increasing the drive side pressure angle in asymmetric gears, the value of contact stress on gear tooth reduces hence increasing the load carrying capacity of gears.

Thus, by using asymmetric gears in place of conventional symmetric gears in industries, load on gear can be increased to a large extent for the same amount of stress induced in them, or conversely, for the same load application the size of gear can be reduced by using asymmetric gears in place of symmetric gears.

7. Nomenclature

D_{bd} - Base circle diameter on drive side
W_t – Transmitted Load
D_{bc} - Base circle diameter on coast side
K_o -Overload Factor
D_p - Pitch circle diameter
K_v -Dynamic factor
D_o – Outside circle Diameter
K_s -Size Factor
$D_\wedge$ - Tip circle diameter
K_m -Load Distribution Factor
S_o - Tooth tip thickness
C_f -Surface Condition Factor
S_p – Tooth Thickness at Pitch diameter
F- Face width
ν_c, ν_d - Profile angle in the intersection point
I – Geometry Factor for pitting resistance
σ_c – Contact Stress
d- operating pitch diameter of pinion
C_p –Elastic Coefficient

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