IJRAME
ISSN (ONLINE): 2321-3051
INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING

Theoretical and Finite Element Analysis of Load Carrying Capacity of Asymmetric Involute Spur Gears.

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Abstract

The paper presents a method for investigating the bending stress at the critical section of "Asymmetric Involute spur Gear". The method, ISO/TC- 60 has been used to theoretically calculate bending stress at the critical section of this Gear. The determination of the tooth form factor, stress concentration factor, critical section parameters and contact ratio has been accomplished for each set of gear. The gears with different pressure angle have been modelled by using "CATIA V-5 R15"software.The results obtained by theoretical method have been verified by using "ANSYS 12.0 software". The comparative analysis of bending stress at critical section has been carried out.

***Keywords**:* Asymmetric Involute Spur Gear; Critical Section; Tooth form factor; Stress concentration factor.

1. INTRODUCTION

Gearing is one of the most critical components in a mechanical power transmission system. Gears are used in all applications where power transfer is required such as automobiles, industrial equipments, air planes and marine vessels. In an automobile industry, lighter weight gears with high reliability are necessary as lighter weight automobiles are continued in demand. Most of the gears used in power transmission system are spur and helical gears. [1]

Sufficiency in bending load carrying capacity is a serious problem. It has been observed that most of the failures due to bending stress are at critical section of a gear tooth (Fig.1) [2].

Fig.1

Fig.2

There are different ways to improve bending strength of gears such as providing circular fillet rather than trochoidal fillet at the root of a gear [3], heat treatments, improving surface quality, and using a cutter with large tip radius.

Another way to improve the load carrying capacity is to improve an involute geometry. The tooth form has left-right symmetry in symmetric Involute gear and the same performance can be obtained at forward and backward motion. The concept of asymmetric gear is based on the notion that both forward and backward motions are not always expected in practically used gear units for power transmission. Since most of gears may be expected to reverse operation occasionally, the tooth coast sides are still required to be conjugate, but with only rudimentary performance expectations [3]. This has the effect of producing the teeth which are thicker at the base and more pointed at the tip than standard gear tooth and are subsequently more resistant to bending. Asymmetric gear teeth help to provide extremely low weight to power ratio of a gear box (0.04 per meter) [1]. The main advantage of asymmetric gears is contact stress reduction, resulting higher torque density. Another important advantage is possibly to design the coast flanks and fillet independently from the drive flanks, managing tooth stiffness and load sharing while keeping a desirable pressure angle and contact ratio on the drive profiles. This allows reducing gear noise and vibration level. [4]

From the extensive review of the published research work on Asymmetric Gears, it is seen that the main issues in the design, development and manufacturing of Asymmetric Involute Spur Gears are:

1) Mathematical modelling and theoretical and experimental analysis are the ticklish problems.
2) Current trends in engineering globalization require research to revisit various normalized standards to determine their common fundamentals and those approaches need to identify "best practices" in industries [5].
3) There is a trend to gear customization and the interchangability is not very critical anymore and low tooling cost is not very important.
4) It is necessary to fine tune the ad-hoc procedure used in earlier studies for conducting bending stress analysis on asymmetric gear [4].

Considering all these issues, the theoretical and Finite Element Analysis of" Asymmetric Involute Spur Gear" has been carried out.

2. Tooth Bending Stress of an Asymmetric Involute Spur Gear

Since Asymmetric Involute Spur Gear is a nonstandard gear, no standard method has been developed to determine the bending stress of asymmetric gear. Pedrero et.al. [6] have developed a systematic method for bending stress analysis of asymmetric spur gear based on the recommendations of technical committee ISO/TC- 60. These standards are based on following assumptions. [2]

1. The compressive stress produced by radial component of the load can be neglected.
2. The critical section of the tooth is defined by the point of tangency with a root fillet of a line rotated 30 deg. From the tooth centre line.
3. The tooth load acts at the tip of tooth only.

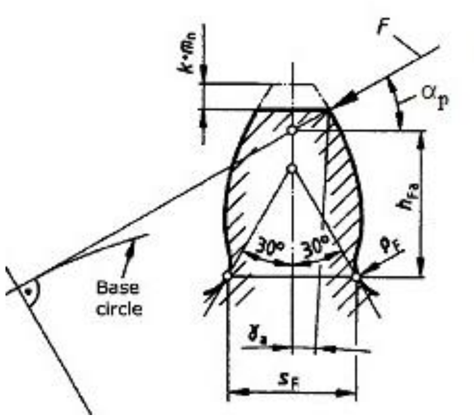


Fig. 2 Tooth Model for bending stress. [1]

According to the DIN 3990, the maximum tooth bending stress is given by

$$\sigma_{F0} = \frac{F_t}{b.m_n} . Y_{Fp} . Y_{Sp} \quad [6]$$

Where,

F_t : The tangential gear load (N)

b : The face width (mm)

Y_{Fp} : The tooth form factor

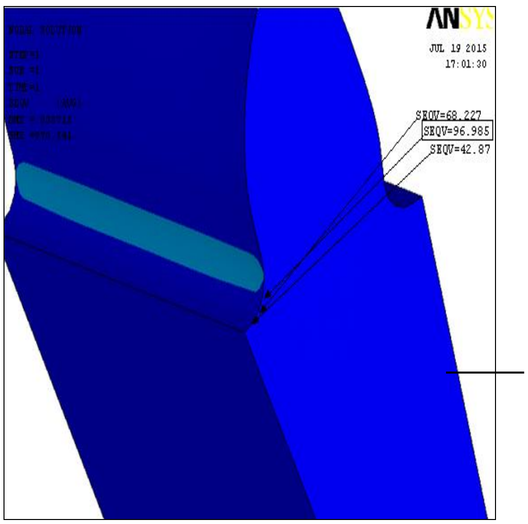
Y_{Sp} : The stress concentration factor.

m_n: Normal module (mm)

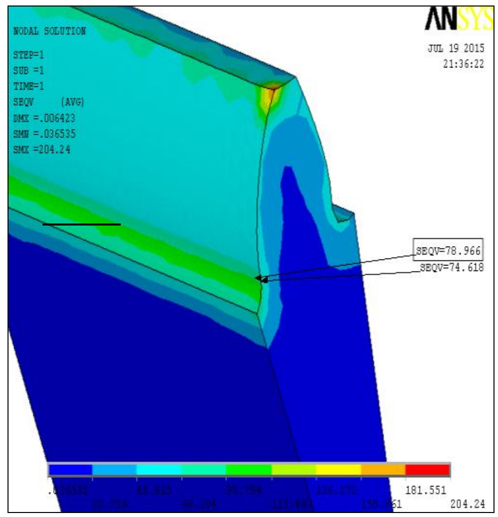
As discussed earlier, the three sets of gears with different pressure angles have been considered for analysis. The results obtained by theoretical method and F.E. analysis have been compared. These results are shown in a tabular form (Table.2). Table.1shows the input considered for analysis.

P	F_t	m_n	Z_p	Z_g	h_{ao}	B	N_p	Material
10KW	1579 N	3 mm	28	42	1.25	25mm	1440 rpm	Steel

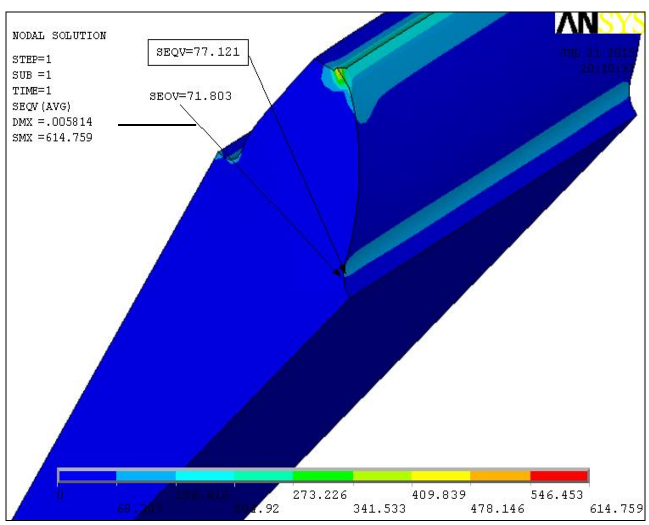
Table. 1



$\alpha_d = 20^0 / \alpha_c = 20^0$



$\alpha_d = 25^0 / \alpha_c = 20^0$



$\alpha_d = 30^0 / \alpha_c = 20^0$

Fig. 3 Bending Stress Analysis by using FEM

Tooth Bending Stress [N/mm2]				Contact Ratio
Pressure Angle in Deg. Drive side/ Coast side	According to DIN 3990	By using FEM	% Deviation	
20/20	87.03	96.985	9.3	1.68
25/20	82.73	78.966	4.54	1.49
30/20	79.12	77.121	2.52	1.36

Table. 2

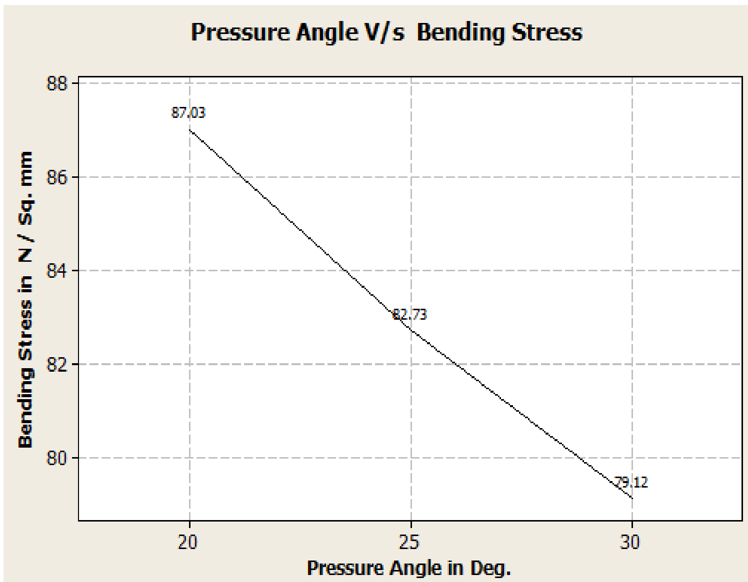


Fig. 4

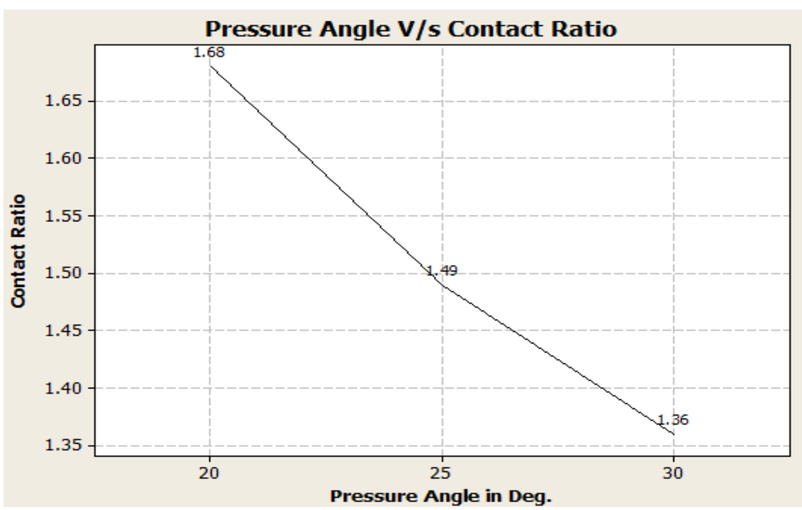


Fig. 5

3. Conclusion

According to the work carried out, following conclusions can be made:

1. The bending stress at the critical section of the tooth is reduced by about 9% there by increasing the load carrying capacity using asymmetric gear when drive side pressure angle is increased from 20^0 to 30^0. (Fig. 4)
2. The Contact ratio is reduced to 1.36 from 1.68 when the drive side pressure angle is increased from 20^0 to 30^0. But it is above the threshold value which is 1.2. (Fig. 5)
3. It is observed that the location of the maximum bending stress remains constant even though we increase the drive side pressure angle from 20^0 up to 30^0.

Nomenclature

α_c: Coastal side pressure angle.

α_d: Drive side pressure angle.

m_n: Normal module (mm) .

P: Power transmitted (kW).

h_{ao}: Tool addendum factor.

N_p: Pinion speed in rpm.

Z_p: No. Of teeth on pinion.

Z_g: No. Of teeth on gear.

Y_{Sp} : The stress concentration factor.

Y_{Fp} : The tooth form factor.

S_f: Thickness of the critical section in mm.

b: Width of the gear (mm).

F_t: Tangential force (N).

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