Bending Strength Analysis of Involutes Spur Gears with Asymmetric Profile

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Abstract— Analysis estimating the variation of most bending stress and make contact with magnitude relation reckoning on tooth range and pressure angle of the drive facet, has been developed for uneven drives. The bending stress analyses are performed with the help of FEM for uneven and bilaterally symmetrical tooth. The strain results obtained by FEM analyses and calculable by the developed program are compared. It's been proved that uneven teeth have higher performance than each bilaterally symmetrical tooth it's been confirmed that, because the pressure angle on the drive facet will increase, the bending stress decreases and also the bending load capability will increase. It's been seen that, whereas the worth of most bending stress is dynamical, the situation of most bending stress remains constant in finite component analysis.

Key words: Uneven Gear, Bending Strength; Correct Stress; Safety Factor; Deformation

I. Introduction

Gear transmission is one among the foremost necessary mechanical transmissions in engineering systems, thus its dependability is crucial. Sufficiency in bending load carrying capability may be a significant issue, as regards the carburized or surface quality improved gears with terribly high surface fatigue strength, like plastic and form gears. There are a unit many ways in which to resolve the matter like heat treatments, rising tooth fillet surface quality, and employing a larger radius of cutters tip corner [1]. In our own way of skyrocketing the load capability of transmissions is to change the involutes pure mathematics. This has been a regular follow in subtle gear style for several years. The word describing these kinds of gear modifications is quite confusing with relevancy postscript modification profile shift, etc. a further alteration that's terribly seldom used is to create the gears uneven with totally different pressure angles for every facet of the tooth [2]. The aim of uneven tooth is to boost the performance of gears like increasing the load capability or reducing noise and vibration. Application of uneven tooth facet surfaces is ready to extend the load capability and sturdiness for the drive tooth facet [3]. Reducing bending stress levels whereby rising gear strength. A general side of drugs style with a special specializes in the uneven style and relates the current work to existing works. Geometric parameterization of the cutting implement and also the analytical form of a gear cut with this. Presents optimized styles of spur gears with totally different range of teeth [8].

II. Gear profile

- Symmetric Gear Profile
- Asymmetric Gear Profile

A. Bilaterally Symmetrical Profile

Bilaterally symmetrical gear tooth profile is common in use from the start of Gears. They're referred to as bilaterally symmetrical as they're having constant pressure angle on each facet of the gear tooth profile i.e. Drive facet and also the Coast facet.

- Drive facet of a tooth is that the side/ face of the tooth that comes to bear with the opposite gears tooth once transmission motion.
- Coast facet is that the opposite face of the drive facet of the tooth.

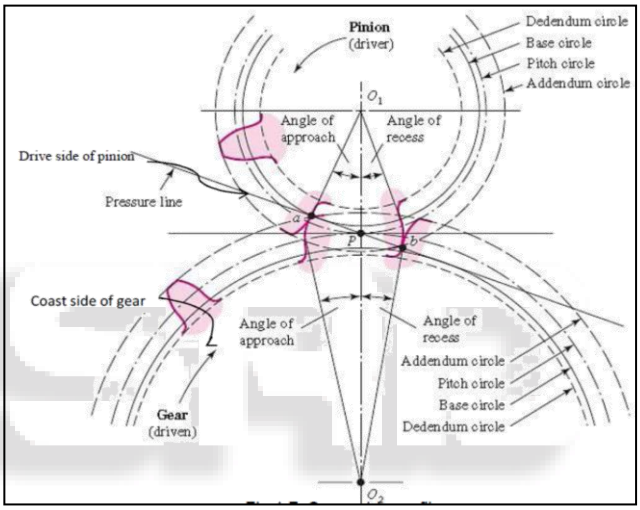


Fig. 1: Bilaterally Symmetrical Profile Gear

B. Uneven Profile

Uneven tooth profile is rare and unconventional gear tooth profile being employed to induce a lot of exactness drive with eliminating the defects and minimizing the possibility of failure. The two profiles (sides) of a gear tooth area unit functionally totally different for several gears.

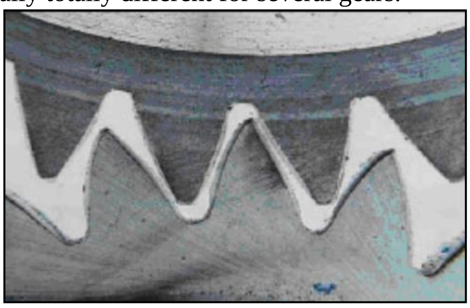


Fig. 2 (A): Uneven Toothed Gears in Mesh

The work on one profile is considerably higher and is applied for extended periods of your time than for the alternative one. The look of the uneven tooth form reflects this purposeful distinction.

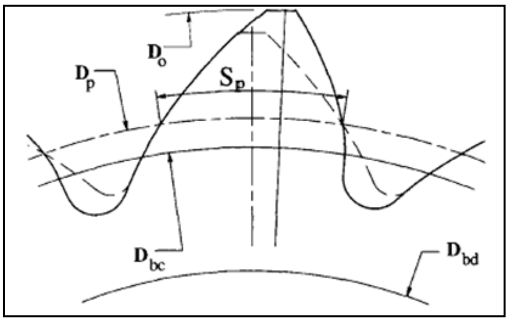


Fig. 2 (b): Showing Uneven Gear Wheel Style with Totally Different Base Circles

The design intent of uneven gear teeth is to boost the performance of the first contacting profile. The alternative profile is usually unloaded or gently loaded throughout comparatively short work periods. The degree of imbalance and drive profile choice for these gears depends on the appliance. The distinction between bilaterally symmetrical and uneven tooth is outlined by 2 involutes of 2 totally different base circles Dbd and Dbc. The common base tooth thickness doesn't exist within the uneven tooth. The circular distance (tooth thickness) Sp between involutes profiles is outlined at some reference circle diameter displaced person that ought to be larger than the biggest base diameter. Asymmetric gears at the same time enable a rise within the transversal contact magnitude relation and in operation pressure angle on the far side the standard gear limits. uneven gear profiles conjointly create it potential to manage tooth stiffness and cargo sharing whereas keeping a fascinating pressure angle and make contact with magnitude relation on the drive profiles by dynamical the coast facet profiles. This provides higher load capability and lower noise and vibration levels compared with typical bilaterally symmetrical gears. By the impact of those defects these four major failure modes geared systems occur:

1) Tooth bending fatigue,
2) Contact fatigue,
3) Surface wear and
4) Scoring.

III. Model Analysis

A modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine element. It may function a start line for one more, a lot of elaborate, dynamic analysis, like a transient dynamic analysis, a Fourier analysis, or a spectrometry. The natural frequencies and mode shapes area unit necessary parameters within the style of a structure for dynamic loading conditions. You'll conjointly perform a modal analysis on a pre-stressed structure, like a spinning rotary engine blade. If there's damping within the structure or machine element, the system becomes a damped modal analysis. For a damped modal system, the natural frequencies and mode shapes become advanced. For a rotating structure or machine element, the rotating mechanism effects ensuing from movement velocities area unit introduced into the modal system. These effects amendment the system's damping. The damping may be modified once bearings gift, that may be a common support used for rotating structure or machine element. The evolution of the natural frequencies with the movement rate is studied with the help of Campbell Diagram Chart Results. A modal analysis can be performed victimization the ANSYS, Samcef, or ABAQUS convergent thinker. Any variations area unit noted within the sections below. Rotor dynamic analysis isn't offered with the Samcef or ABAQUS convergent thinker.

A. Calculations for Symmetric Gear Profile

1) *From the Given Data*

a) Module

m = 3 mm
Tooth: N = 49 for gear
Pressure angle: $\alpha = 25^o$

b) Hence for Gear;

Pitch circle diameter D = m*N = 3*49 = 147 mm
Circular Pitch CP= π*m = 3.142 * 3 = 9.426 mm per teeth
Clearance c = CP/20 = 9.426/20 = 0.4713 mm
Addendum a = m = 3 mm
Dedendum d = a + c = 3+0.4713 = 3.4713
Tooth thickness CP/2 = 9.426/2 mm = 4.713 mm
Addendum diameter D_a = D + 2*a = 147 + 2*3 = 153mm
Dedendum diameter D_d = D – 2*d = 147 – 2*3.4713 = 140.057mm
Similarly for Pinion;
Number of teeth N = 27
Pitch circle dia D = m*N = 3*27 = 81 mm
Addendum diameter D_a = D + 2*a = 81 + 2*3 = 87mm
Dedendum diameter D_d = D – 2*d = 81 – 2*3.4713 = 74.057mm

B. Calculation for Asymmetric Gear Profile:

From the given data:
Module: m = 3 mm
Tooth: N = 49 for gear

1) *Hence for Gear;*

Pitch circle diameter D = m*N = 3*49 = 147 mm
Circular Pitch CP= π*m = 3.142 * 3 = 9.426 mm per teeth
Addendum diameter D_a = D + 2*a = 147 + 2*3 = 153mm
Dedendum diameter D_d = D – 2*d = 147 – 2*3.4713 = 140.057mm
Similarly for Pinion some data is common as in gear;
Number of teeth N = 27
Pitch circle dia D = m*N = 3*27 = 81 mm
Addendum diameter D_a = D + 2*a = 81 + 2*3 = 87mm
Dedendum diameter D_d = D – 2*d = 81 – 2*3.4713 = 74.057mm
Now the main part of designing of asymmetric profile is here:
"It must be noted that these gears are designed under the Direct Gear Design procedure aka DGD"
When designing the Gear & Pinion both we have a relation defining the Thickness of the tooth:
$\lambda = S_p/S_g$
Where;
λ = Thickness ratio, S_p= Centroid Thickness of Pinion, S_g= Centroid Thickness of Gear
Also the relation is engaged with tooth space.
$S_p = W_g$ & $S_g = W_p$
Where;
Wp = Tooth space of pinion,Wg = Tooth space of gear
$S_p + W_p = S_g + W_g = CP = 9.426$
Hence;

$S_p + S_g = 9.426$ mm

There is a relation between the Pitch circle diameter and base circle diameter with respect to the Pressure angle:

$D_b = D_p \cos \alpha_i$

Where;

D_b = base circle diameter

D_p = pitch circle diameter

α_i = pressure angle of { i = d (drive side) or c (coast side) }

$S_p = S_{pd} + S_{pc}$

Where;

S_{pd} = centroid tooth thickness of drive side

S_{pc} = centroid tooth thickness of coast side

$S_t = S_{td} + S_{tc}$

Where;

S_{td} = top land thickness of tooth of drive side

S_{tc} = top land thickness of tooth of coast side

$$s_{td} = \frac{d_t}{2}\left(\frac{s}{dp} + inv(\alpha_d) - inv(\alpha_{td})\right) \quad \text{.................(1)}$$

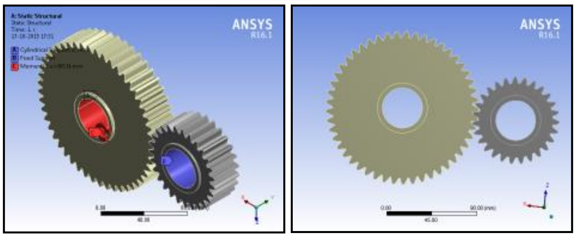
$$s_{tc} = \frac{d_t}{2}\left(\frac{s}{d_p} + inv(\alpha_c) - inv(\alpha_{tc})\right) \quad \text{..............................(2)}$$

IV. Boundary Conditions for Analysis

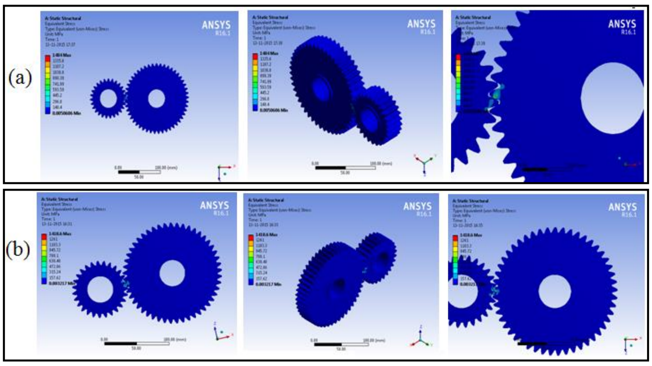
- Material used: Stainless steel
- Meshing: Automatically Generated
- Meshing Relevance centre: Medium
- Contact meshing with size of: 1mm element
- Support: GEAR- fixed on the inner side
- PINION- cylindrical support on the inner side
- Moment load: 300 N-m on pinion
- (Specifically the moment is given so as only the drive sides comes in contact in case of asymmetric profile spur gear drive)

V. Symmetric & Asymmetric

A. Boundary Conditions



B. Equivalent Stress



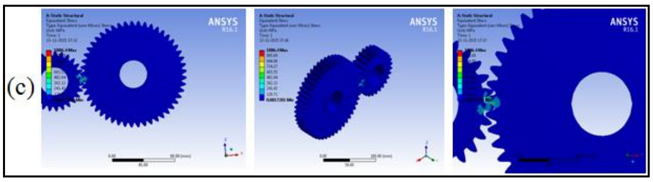


Fig. 3: (a) Symmetric Gear, (b) Asymmetric Gear & (c) Asymmetric Gear with Fillet

Total Deformation

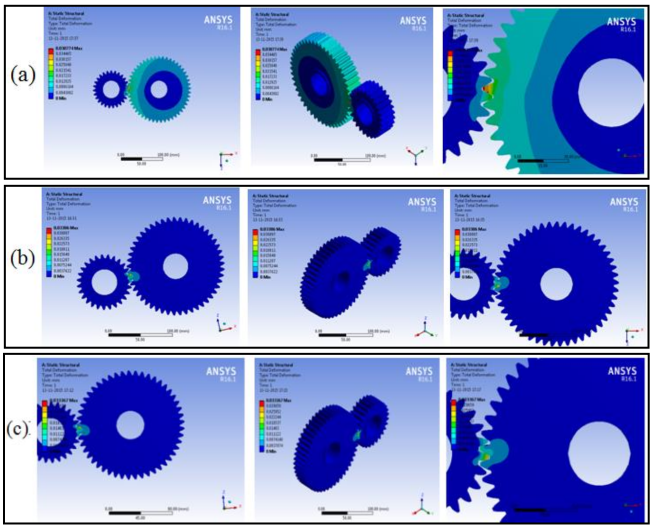


Fig. 4: (a) Symmetric Gear, (b) Asymmetric Gear & (c) Asymmetric Gear with Fillet

Safety Factor

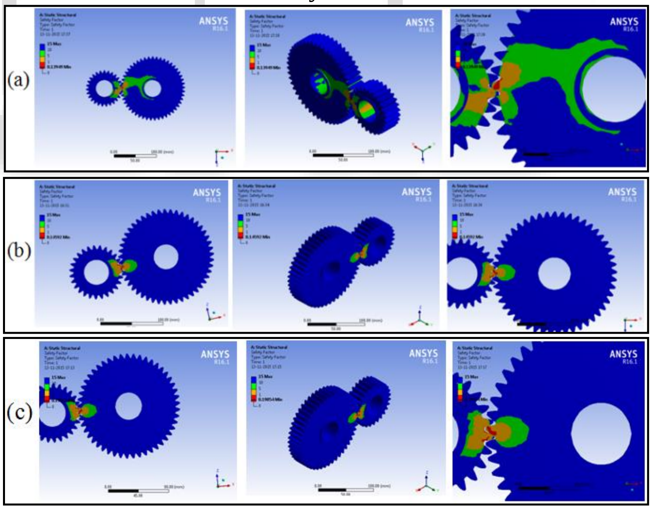


Fig. 5: (a) Symmetric Gear, (b) Asymmetric Gear & (c) Asymmetric Gear with Fillet

Mode shapes of gears

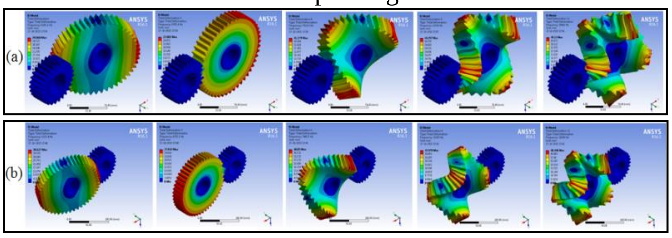
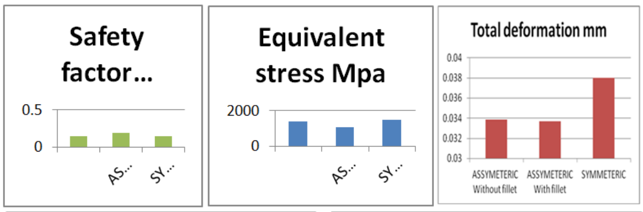


Fig. 6: Different Mode Shapes of Symmetric Gear (a) Symmetric Gears & (b) Asymmetric gears

Parameter	Unit	ASSYMETERIC Without fillet	ASSYMETERIC With fillet	SYMMETERIC
Equivalent stress	Mpa	1418.6	1086.4	1484
Total deformation	Mm	0.03386	0.03367	0.038
Safety factor	stress tool	0.14592	0.19054	0.13949



ASSYMETERIC		
Mode	Frequency (Hz)	Deformation
1	6133.4	38.627
2	6355.3	27.047
3	7466.5	40.85
4	12265	43.978
5	18290	48.446
6	18304	48.673

SYMMETERIC		
Mode	Frequency (Hz)	Deformation
1	6181.1	39.068
2	6365.4	27.082
3	7522.5	41.179
4	12376	44.297
5	18462	49.13
6	18481	49.038

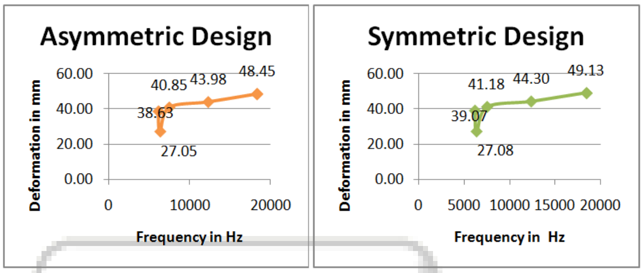


Fig. 7: Comparison of Full Scale Gear and Pinion Model Analysis

PROPERTIES		SYMMETRIC PROFILE SPUR GEAR	ASYMMETRIC PROFILE SPUR GEAR
EQUIVALENT VON- MISES STRESS (MPa)	CONTACT STRESS	1484	1086
Stress tools	Safety Factor	0.13	0.19
TOTAL DEFORMATION		0.038	0.033

This analysis was performed under the constraints and loads so that they simulate the actual designing conditions for the gears.

PROPERTIES		SYMMETRIC PROFILE SPUR GEAR	ASYMMETRIC PROFILE SPUR GEAR	ASYMMETRIC PROFILE SPUR GEAR (OPTIMIZED FILLET)
EQUIVALENT VON- MISES STRESS (MPa)	CONTACT STRESS	1484	1418.6	1086.4
Stress tools	Safety Factor	0.13	0.145	0.19
TOTAL DEFORMATION		0.038	0.03386	0.03367

Table 1: Comparison and Discussion on Optimized Fillet Asymmetric Profile Analysis

In this optimization of fillet of the asymmetric profile forms spur gear analysis we have used an iterative process to give an appropriate fillet to the asymmetric pinion so as to get the optimum and considerably practical results as compare to symmetric profile. After so many hit and trial we've reached to suggestible optimum valves for fillet generation. This fillet enhances the performance of the asymmetric profiled gear. The only problem with the previous design without fillet was its stress at the root of pinion tooth was higher than the symmetric one but this problem is eliminated here after iterative optimization even the contact stress are same as symmetric gears but the main maximum stress at root are significantly reduced so this is an acceptable design.

VI. Conclusion

Tooth model of involutes spur gears with asymmetric tooth has been developed. A computer program, estimating the variation of maximum bending stress and contact ratio depending on tooth number and pressure angle of the drive side, has been developed for asymmetric drives. The bending stress analyses have been performed with the aid of FEM for asymmetric and symmetric tooth. The stress results obtained by FEM analyses and estimated by the developed program have been compared. It has been proved that asymmetric teeth have better performance than both symmetric teeth. It has been confirmed that, as the pressure angle on the drive side increases, the bending stress decreases and the bending load capacity increases. It has been seen that, while the value of maximum bending stress is changing, the location of maximum bending stress remains the same in finite element analysis.

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