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Analysis of Stress and Bending Strength of Involutes Spur Gears with Fillet Asymmetric Profile

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***Abstract**:-* Analysis 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.

Keywords: Asymmetric gear; Bending strength; accurate stress; Safety factor; Deformation; Fillet

I. INTRODUCTION

Gear transmission is one of the most important mechanical transmissions in engineering systems, so its reliability is essential. Sufficiency in bending load carrying capacity is a serious problem, as regards the carburized or surface quality improved gears with very high surface fatigue strength, such as plastic and sintered gears. There are several ways to solve the problem such as heat treatments, improving tooth fillet surface quality, and using a larger radius of cutter's tip corner [1].

The purpose of this study is to determine bending load carrying capacities and contact conditions of asymmetric gear drives. Hence, a computer program, different from previous studies [1, 3, 6] which considers tooth number and drive side pressure angle, is developed. The asymmetric involutes tooth can be manufactured by the same process as in generating the symmetric involutes tooth. Asymmetric profile is achievable by adopting ac and ad values for the profiles of two sides of the rack. Depending on the special tooling, production cost of these gears increases. Therefore, the gears with asymmetric teeth should be considered for gear systems that require extreme performance like aerospace applications and for mass production transmissions where the share of the tooling cost per one gear is insignificant. The most promising application of asymmetric profiles seems to be in molded gears and powder gears [2].

This paper presents an analysis of the lateral extrusion of spur gear forms and a comparative evaluation of the methods of lateral extrusion and closed die forging of spur gear forms in terms of mechanical properties of the product [9].

II. GEAR PROFILE

A. Symmetric Profile:

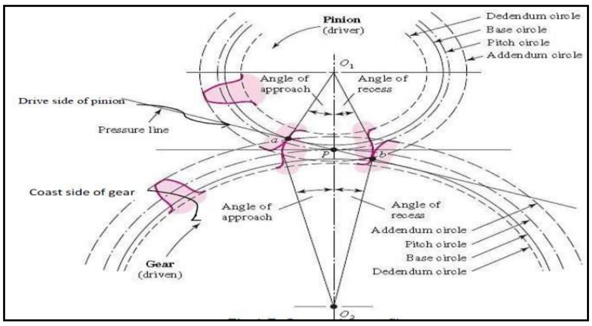


Fig. 1: Symmetric profile gear

Symmetric gear tooth profile is common in use from the beginning of Gears. They are called symmetric as they are having the same pressure angle on both side of the gear tooth profile i.e. Drive side and the Coast side.

B. Asymmetric Profile:

Asymmetric tooth profile is uncommon and unconventional gear tooth profile being used to get more precision drive with eliminating the defects and minimizing the chance of failure. The two profiles (sides) of a gear tooth are functionally different for many gears. The workload on one profile is significantly higher and is applied for longer periods of time than for the opposite one. The design of the asymmetric tooth shape reflects this functional difference.

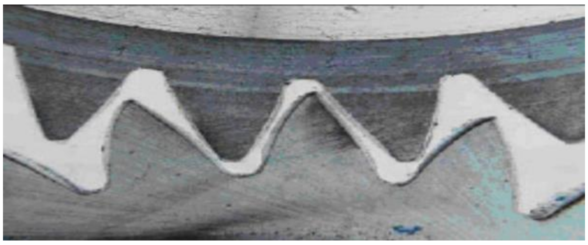


Fig. 2 (a): Asymmetric toothed gears in mesh

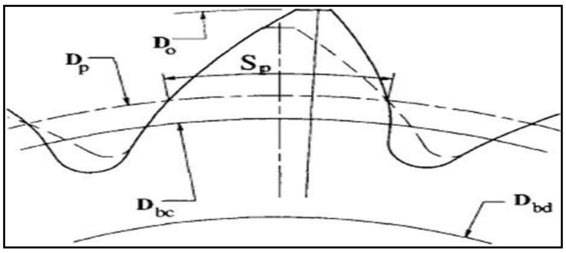


Fig. 2 (b): Showing asymmetric spur gear design with different base circles

The design intent of asymmetric gear teeth is to improve the performance of the primary contacting profile. The opposite profile is typically unloaded or lightly loaded during relatively short work periods. The degree of asymmetry and drive profile selection for these gears depends on the application.

Asymmetric gears simultaneously allow an increase in the transverse contact ratio and operating pressure angle beyond the conventional gear limits. Asymmetric gear profiles also make it possible to manage tooth stiffness and load sharing while keeping a desirable pressure angle and contact ratio on the drive profiles by changing the coast side profiles. This provides higher load capacity and lower noise and vibration levels compared with conventional symmetric gears.

C. Type of Failures:

By the effect of these defects these four major failure modes in gear systems occur:

– Tooth bending fatigue,
– Contact fatigue,
– Surface wear and
– Scoring.

Two kinds of teeth damage can occur on gears under repeated loading due to fatigue; namely the pitting of gear teeth flanks and tooth breakage in the tooth root. Tooth breakage is clearly the worst damage case, since the gear could have seriously hampered operating condition or even be destroyed. Because of this, the stress in the tooth should always be carefully studied in all practical gear application. The fatigue process leading to tooth breakage is divided into crack initiation and crack propagation period. However, the crack initiation period generally account for the most of service life, especially in high cycle fatigue.

To prevent these failures, we must prevent the defects explained before; we can eliminate these defects by:

1) Under cutting can be avoided by increasing the pressure angle.

2) Backlash and interference can be avoided by increasing the addendum ofmating gear.

3) Another way of increasing the load capacity of transmissions is to modify the involutes geometry. This has been a standard practice in sophisticated gear design for many years. The nomenclature MMdescribing these types of gear modifications can be quite confusing with reference to addendum modification or profile shift.

III. MODEL ANALYSIS

A modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component. It can also serve as a starting point for another, more detailed, dynamic analysis, such as a transient dynamic analysis, a harmonic analysis, or a spectrum analysis. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions. You can also perform a modal analysis on a pre-stressed structure, such as a spinning turbine blade. If there is damping in the structure or machine component, the system becomes a damped modal analysis. For a damped modal system, the natural frequencies and mode shapes become complex.

For a rotating structure or machine component, the gyroscopic effects resulting from rotational velocities are introduced into the modal system. These effects change the system's damping. The damping can also be changed when a bearing is present, which is a common support used for rotating structure or machine component. The evolution of the natural frequencies with the rotational velocity can be studied with the aid of Campbell Diagram Chart Results. A modal analysis can be performed using the ANSYS, Samcef, or ABAQUS solver. Any differences are noted in the sections below. Rotor dynamic analysis is not available with the Samcef or ABAQUS solver.

IV. STEP WISE MODELLING PROCEDURE

A. Sketching:

Using the base circle method of gear drawing with given data is used from the book “MACHINE DRAWING by P. S. Gill”

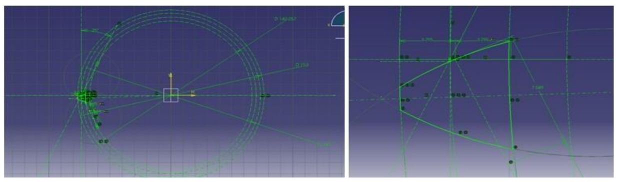
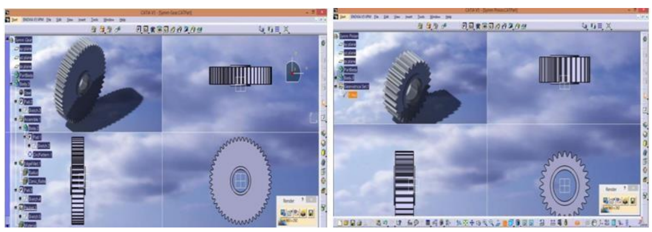


Fig. 4.1: Sketch of tooth profile of gear

B. Part modelling:

Using proper constrains and data book we create the part model in the Part designing module of “Catia”; as shown, Similarly the Pinion is also designed:



C. Assembly of Parts

For the sake of proper assembling of parts a shaft fixture is created on the Catia Part design module itself.

1) Shaft Fixture:

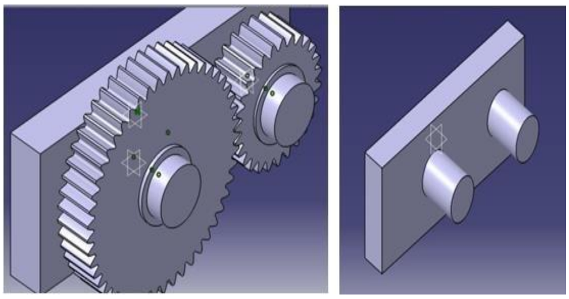
It consist of a fixed wall with two extruded shafts of the same diameter and the distance equal to the sum of the pitch circle radius of the gear and pinion, ie:

2) Distance between shafts:

X = Pitch circle dia of gear /2 + Pitch circle dia of pinion/2

= 147/2 + 81/2

= 114 mm



1) Shaft fixture for assembling the gears properly
2) Gear and Pinion Assembly

Final assembly is created by using proper constraints to define the proper contact between the teeth of gear and pinion so as to get reliable results in analysis.

V. THE THREE TEETH MODEL:

This model is prepared from the full scale model of the gear, according to many research papers we have reviewed; the three teeth section of a full scale model resembles the same picture and gives proper result in the analysis of stress.

Our aim to create and analyse this three teeth model independently is to get the independent results which can show the significance of the profile of the gear, i.e. the effect which the asymmetric profile gives to the gear in terms of its strength.

Following picture showing the definition and dimensions of the 3 teeth model generated from full scale model:

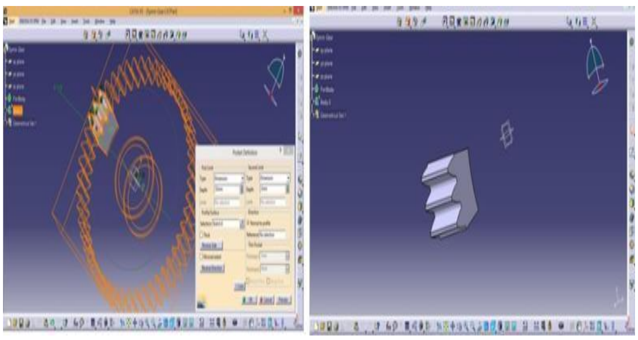
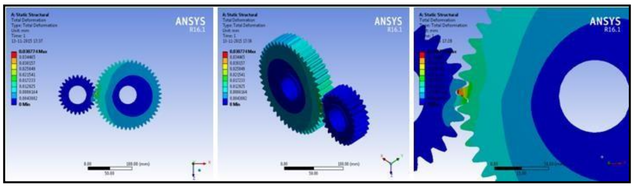
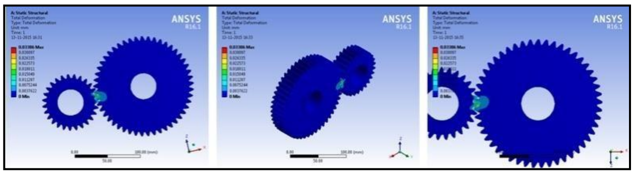


Fig. 4.4: showing the complete 3 teeth model of a symmetric profile spur gear

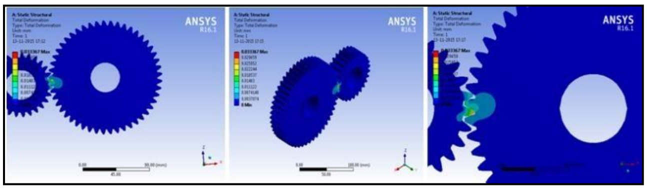
A. Total Deformation



(a)



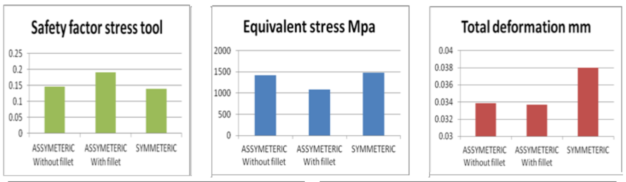
(b)



(c)

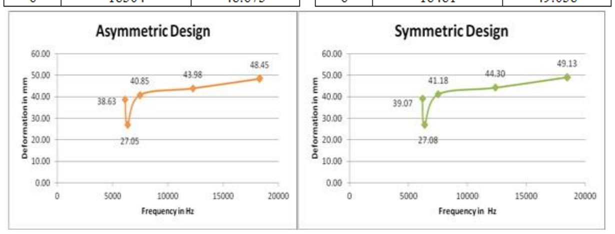
Fig. 4: (a) Symmetric gear, (b) Asymmetric gear & (c) Asymmetric gear with fillet

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



VI. COMPARISION 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. Here we can see that in case of asymmetric gear simulation the contact stress reduced significantly (approx. 200 MPa) but the stress at the root of pinion became higher, the reason may be because we didn't used the fillets in asymmetric gears. Hence there is the chance of improvement in it.

VII. COMPARISION AND DISCUSSION ON OPTIMIZED FILLET ASYMMETRIC PROFILE ANALYSIS

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

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.

VIII. CONCLUSIONS

According to this research, the following conclusions can be drawn:

1) 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.
2) 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.
3) It has been proved that asymmetric teeth have better performance than both symmetric teeth
4) It has been confirmed that, as the pressure angle on the drive side increases, the bending stress decreases and the bending load capacity increases.
5) 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.

IX. FUTURE WORK

Further numerical method investigations should be conducted on-

1) The transmission error for all types of gears for example: helical, spiral bevel and other gear tooth form,
2) Simulation of an oil film in contact zone.
3) Tooth bending fatigue test for evaluation of gear blank manufacturing using different methods like three point bending loading.

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