AN INVESTIGATION OF CRACK INITIATION IN SPUR GEARS BASED ON FINITE ELEMENT DYNAMICS ANALYSIS: A REVIEW

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Abstract:
This review paper, the gear tooth flanks, contact fatigue is one of the main failure modes due to repeated compression and shear stress cycles. For the investigation of surface and subsurface stresses of gear teeth by using of hertzian theory and the finite element method. The number of loading cycles required for fatigue crack initiation is predicted using the Smith-Watson-Topper method. The effects of friction and speed on stress cycles and fatigue life are studied. Friction is found to shift the distributions of von misses stress, changes the extreme values of the shear stress cycle, and results in low fatigue life. The stresses near the engagement and recess areas are also found to be greater than the static contact conditions and thus result in low fatigue life, particularly at high speeds.

Key words: Spur Gear, Finite element dynamics analysis, crack initation.

1. Introduction:

The contact fatigue process can be divided into two main stages: initiation of micro-cracks and crack propagation. Initiation of fatigue cracks represents one of the most important stages in which cracks represents one of the most important stages in which cracks can be initiated either on the surface or at some distance under the surface depending on the contact conditions. The main aim of this study is to predict contact fatigue crack initiation resulting from high stresses or strains during the mesh process[1-2].
Recently, the finite element method has been increasingly used to simulate rolling and sliding contact, and considerably precise stress results are expected. The worst load condition is often regarded as the condition wherein one pair of teeth carries the full load, and the position in the area around the inner point of a single teeth pair engagement is simulated with the maximum value of contact pressure. However, Osman and Velex found that dynamic tooth loads are highly influential depending on speed range because at certain points on the tooth profile, they can induce fatigue damage, which would not occur at low speeds[3-4]. Given that the finite element dynamic contact is necessary to investigate stresses at various speeds. Once the stress loading cycles are established, the number of stress cycles required for a fatigue crack to appear can also be determined. Rolling contact fatigue is typically a multiaxial fatigue mechanism, for which several methods have been developed: examples of such methods include the Smith-Watson-Topper method and Coffin-Manson hypothesis.
In the present study, contact stress analysis and fatigue crack initiation prediction are applied to a pair of spur gears in a high-speed diesel engine. The stresses and numbers of loading cycles required to initiate fatigue cracks at various positions and speeds are investigated[5-9].

2. Crack initiation:

There are several methods can be applied to predict this crack initiation. Here Coffin-Manson method and Smith-Watson-Topper method used to predict crack initiation.

Coffin-Manson method:

The Coffin-Manson method is also referred to as the ε-N method for low cycle fatigue. Taking multiaxial strain condition into consideration, equivalent strains are applied, and the relationship between the strain increment $\Delta\varepsilon_{eff}$ and the number of loading cycles N_f is characterized using the following equation:

$$\frac{\Delta\varepsilon_{eff}}{2} = \frac{\Delta\varepsilon_{e_{eff}}}{2} + \frac{\Delta\varepsilon_{p_{eff}}}{2}$$
$$= \frac{\sigma}{E}(2\,N_f)^b + \varepsilon'_f(2N_f)^c, \qquad (1)$$

Where $\Delta\varepsilon_{e_{eff}}$ is the equivalent elastic strain increment,
$\Delta\varepsilon_{p_{eff}}$ is the equivalent plastic strain increment,
σ'_f is the fatigue strength coefficient,
ε'_f is the fatigue ductility coefficient,
b is the strength exponent, and
c is the fatigue ductility exponent.

If no plastic strain is observed, then Eq. is transformed into the Basquin's eq for high cycle fatigue conditions.

$$\sigma_{ae} = \sigma'_f (2N_f)^b \, , \qquad (2)$$

Where σ_{ae} is the amplititute of equivalent stress.

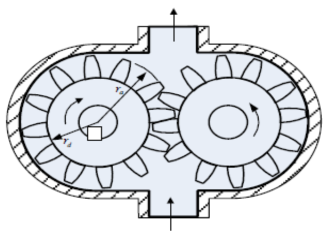
Smith-Watson-Topper method:

Smith used Basquin's equation for the maximum stress and multiplied it by the strain-life equation to obtain the SWT parameter, $\sigma_{max}\varepsilon_{a.}$

$$\sigma_{max}\frac{\Delta\varepsilon_1}{2} = \frac{{\sigma'_f}^2}{E}(2N_f)^{2b} + \sigma\varepsilon'_f(2N_f)^{b+c}, \qquad (3)$$

Where σ_{max} is the maximum normal stress on the critical plane.

3. Geometry and Stress Analysis:



Design parameter:

Discharge specification = 24.55litres/min

Inlet port of diameter = 32mm

Pressure angle = 20◦

Width = 38mm

No. of tooth = 12

Pitch circle diameter = 40 mm.

Circular pitch (P_c) = $\frac{\pi D}{T}$

Where D = diameter of the pitch circle

T = Number of teeth.

$= \frac{3.14*40}{12}$

= 10.466 mm

Module = $\frac{1}{P_c}$

$= \frac{1}{10.466}$

= 3.33

Center of distance (a) = $\frac{m\,(T_1+T_2\,)}{2}$

$= \frac{3.33\,(12+12\,)}{2}$

= 39.96 mm

3. Modeling of gear:

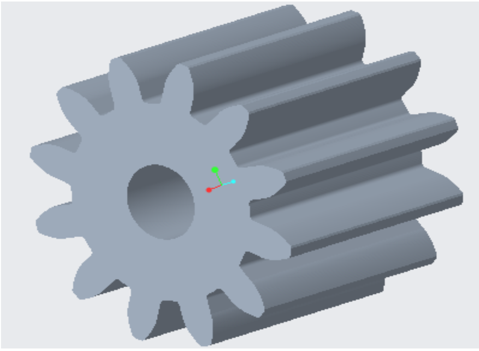


Fig.1 3-D model of spur gear

In this paper, Ansys for the analysis. We take pair of spur gear of gear pump at the speed 1400 rpm. The basic data of the spur gear geometry are listed in table1[6-15].

	Number of teeth	Module (mm)	Pressure Angle	Tooth width (mm)
Driving gear	12	3.33	20°	38
Driven gear	12	3.33	20°	38

Table-1 the basic data of the spur gear geometry.

The gear pair is meshed and when the rotation speed is applied on the surface of the shaft hole of the driving gear and the 405.353 N torque is applied on that at the driving gear teeth surface.

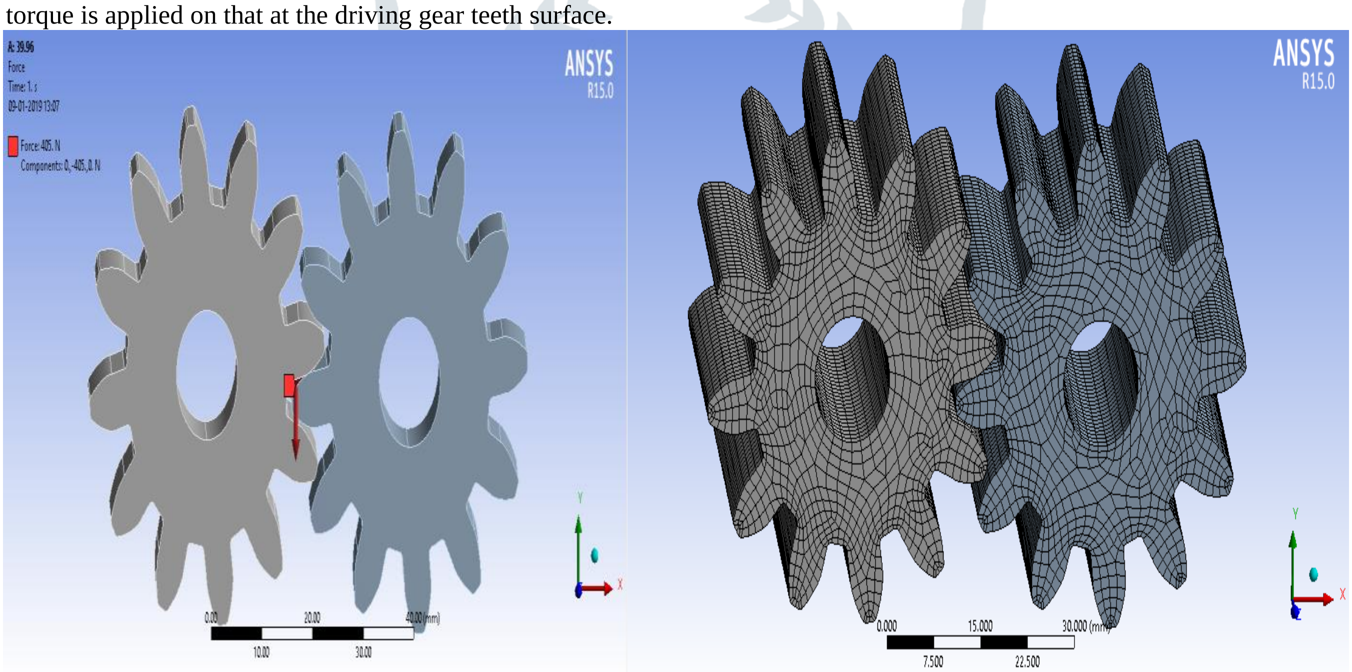


Fig.2 (a) spur gear assembly with of contact region, (b) Meshed gear pair.

The Von Mises equivalent stress distribution of one contact tooth pair meshing near the pitch circle at 1400 rpm. Is shown fig.3.

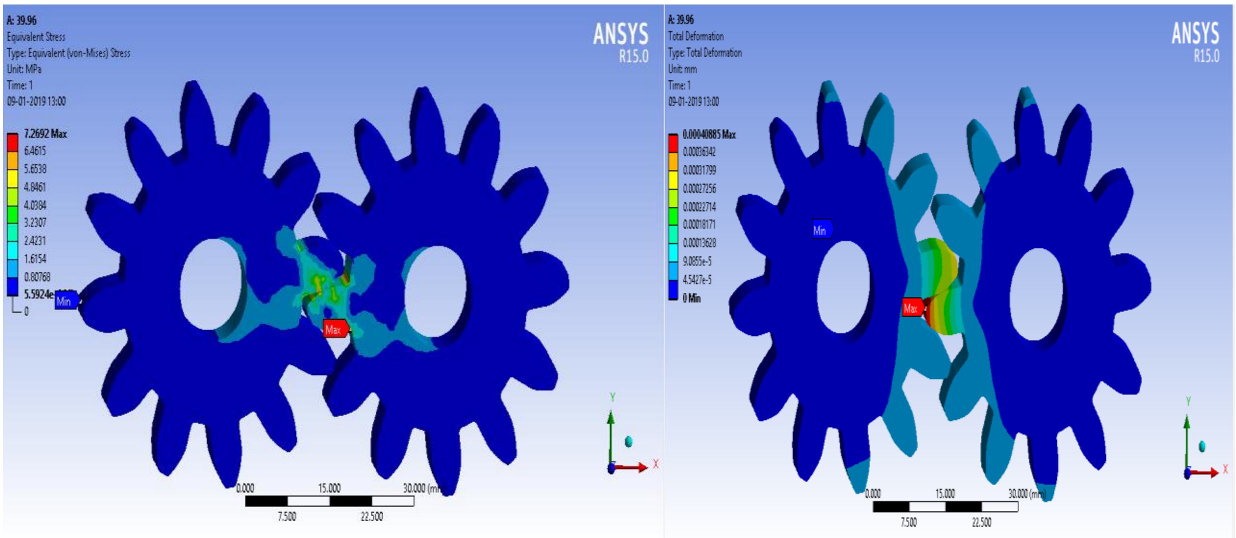


Fig.2 (a) Von Mises equivalent stress distribution, (b) Total deformation of spur gear.

Conclusion:

Based on the results, we use the SWT method and subsequently calculate the numbers of load cycles for the crack initiation of points on and under the contact surface at 1400 rpm. When the friction coefficients μ = 0; μ = 0.1 and μ = 0.3. And the fatigue life of pair of spur gear is 1.e+009 cycles. If we convert into the days so it is 496 days.

Furthermore, we will also taken into consideration investigated the total contact fatigue process including both crack initiation and crack propagation.

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