Comparative Fatigue Life Prediction of Spur Gear Under Fully Reversed Loading by Using Finite Element Analysis

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Abstract
This study here presents the fatigue life prediction of spur gears in mating condition based on finite element analysis under fully reversed load conditions. Gears being the vital components of any automobiles, power generation systems and in heavy machinery industries, they need to have good fatigue properties such as fatigue life, endurance limit and fatigue strength for better life and performance of the equipment or machinery. The objective of this study is comparative simulation on fully reversed loading conditions in the fatigue life analysis on general gear materials, SAE materials and concludes which suits for the better purpose of usage. The Finite Element Method (FEM) has been performed on the gear models to observe the distribution of stress and damage. Comparison was done between Aluminum alloy and SAE materials. The results under fully reversed fatigue loading conditions of the earlier mentioned materials are determined and are tabulated depending on the criteria. After a keen study of the results obtained under various conditions and effects the better material is recommended.

Keywords
Fatigue Life, Non Constant Amplitude Proportional Loading, Spur Gear Fem, Total Life Approach

I. Introduction
Gears are the most common means of transmitting power in the modern Mechanical engineering world. They vary from tiny size used in the watches to the large gears used in marine speed reducers; bridge lifting mechanism and others. They Form vital elements of main and ancillary mechanism in many machines such as Automobiles, tractors, metal cutting machine tools, rolling mills, hoisting and transmitting machinery and marine engines etc. The four major failure modes in gear systems are tooth bending fatigue, contact fatigue, surface wear and scoring. Here we’re concerned on complete reverse loading. 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. The initial crack can be formed due to various reasons. The most common reasons are short-term overload, material defects, defects due to mechanical or thermal treatment and material fatigue. The initial crack then propagates under impulsive loading until some critical length is reached, when a complete tooth breakage occurs. The service life of a gear with a crack in the tooth root can be determined experimentally or numerically (e.g. with finite element method). The fatigue life of components subjected to sinusoidal loading can be estimated by using cumulative damage theories.

II. Fatigue Analysis
The fatigue analysis is used to compute the fatigue life of a component and to predict the damage areas in components. Necessary inputs for the fatigue analysis are the material properties, loading history

A. Material Properties
ALSI4027 is the general material applied to spur gears for power transmission in automobiles. Materials like SAE1045-450-QT, SAE5160-825-QT has high tensile strength and yield strength compared to general material. Bandara and Ranjith did a research study in fatigue strength prediction formulae for steel [9]. Fatigue properties were calculated based on this study and plotted in (table-2). Metal composition for the materials were plotted in (Table 1)

Table 1: Materials Composition

<table>
<thead>
<tr>
<th>Material</th>
<th>Si %</th>
<th>S%</th>
<th>C%</th>
<th>Mn%</th>
<th>P%</th>
</tr>
</thead>
<tbody>
<tr>
<td>ALSI4027</td>
<td>0.34</td>
<td>0.04</td>
<td>0.25</td>
<td>0.70</td>
<td>0.04</td>
</tr>
<tr>
<td>SAE1045-450-QT</td>
<td>0.32</td>
<td>0.04</td>
<td>0.23</td>
<td>0.27</td>
<td>0.034</td>
</tr>
<tr>
<td>SAE160-825</td>
<td>0.35</td>
<td>0.039</td>
<td>0.40</td>
<td>0.90</td>
<td>0.04</td>
</tr>
</tbody>
</table>

B. Loading Histories
Loading is another major input for the finite element based fatigue analysis. Unlike static stress, which is analysed with calculations for a single stress state, fatigue damage occurs when stress at a point changes over time. There are essentially four classes of fatigue loading, with the ANSYS Fatigue Module currently supporting the first three:

• Constant amplitude, proportional loading.
• Constant amplitude, non-proportional loading.
• Non-constant amplitude, proportional loading.
• Non-constant amplitude, non-proportional loading.

Constant Amplitude, fully reversed loading within the fatigue Module uses a “quick counting” technique to substantially reduce...
runtime and memory we can see in (Fig. 20). Loading is of constant amplitude because only one set of FE stress results along with a loading ratio is required to calculate the alternating and mean values is the classic, “back of the envelope” calculation describing whether the load has a constant maximum value or continually varies with time. The loading ratio is defined as the ratio of the second load to the first load (LR = L2/L1). Loading is proportional since only one set of FE results are needed (principal stress axes do not change over time) Since loading is proportional, looking at a single set of FE results can identify critical fatigue locations for this constant amplitude fully reversed loading determined the stress life analysis we can obtain the solution for Goodman, Gerber equation in stress life analysis and SWT Morrow in strain life analysis.

Fig. 2: Fully Reversed Cycle

**III. Fatigue Analysis Algorithm**

Shaik Kalam and Abdul Hasan predicted fatigue analysis of spur gear[5], deals with the calculation of static and dynamic analysis, and fatigue life estimation of test gear, which is contacting master gear and assuming loading on the gear is random or constant amplitude by Finite Element package ANSYS. J.P.Karthik and Chaitanya determined fatigue analysis of truck wheel rim[2] under fully reversed loading and predicted the life. Based on these studies an integrated FE based durability analysis is considered a complete analysis of an entire component. Fatigue life can be estimated for every element in the finite element model and contour plots of life. Geometry information provided by FE results define how an applied load is provided by FE results for each load case applied independently. Data provided for the desired fatigue analysis method. The schematic algorithm of the integrated finite element based fatigue life prediction analysis is shown in (fig. 3)

**A. Computer Aided Design**

In the computer aided design the complete design of the gear with its specifications, like number of teeth, pitch diameter, type of gear have been taken into consideration. Basing on all the possible criteria’s various designs have been stimulated and an optimum design model is made for different materials considered and are subjected to fatigue loading for clear study.

**B. Finite Element Model and Analysis**

Later this model is subjected to various kinds of loads at various points of the design and then this model is sent to finite element analysis under the same loads and is analysed for various stresses. This analysis is so done to determine the various stresses that are acting at different points and at different positions for all the different gear materials taken into consideration while designing and finally a comparative study on all the kinds is made. Numerical techniques are necessary to simulate the physical behaviour and to evaluate the structural integrity of the different designs. The objective of the current study are to calculate the fatigue life of a spur gear using total life and crack initiation methods, to investigate the effect of mean stress on fatigue life and the probabilistic nature of fatigue on the S-N curve via the design criteria.

**C. Fatigue Analysis**

This stimulated model made by optimal designing after Finite Element Analysis is subjected to Fatigue loading for analysis of Fatigue failures. Initially the component i.e., the gear is subjected to Histories and then Reversed loads are acted upon it. When the gears are in mesh the Reversed loads are acted upon the gears and the fatigue life is calculated. At this condition the life of gears is found and if it’s in acceptable range it’s finalized and is sent for production. The mechanical properties for the materials are mentioned in (Table 2).

**Table 2: Mechanicaland Cyclic Properties of SAE1045-450-QT, SAE5160-825-QT and ALSi4027**

<table>
<thead>
<tr>
<th>Properties</th>
<th>Materials</th>
<th>SAE1045-450-QT</th>
<th>SAE5160-825-QT</th>
<th>ALSi4027</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield strength(Mpa)</td>
<td>1515</td>
<td>1070</td>
<td>325</td>
<td></td>
</tr>
<tr>
<td>Ultimate-tensile strength(Mpa)</td>
<td>1584</td>
<td>1550</td>
<td>515</td>
<td></td>
</tr>
<tr>
<td>Elastic modulus(Mpa)</td>
<td>207000</td>
<td>207000</td>
<td>210000</td>
<td></td>
</tr>
<tr>
<td>Fatigue strength coefficient(Sf)</td>
<td>2200</td>
<td>3047</td>
<td>1030</td>
<td></td>
</tr>
<tr>
<td>Fatigue strength exponent(b)</td>
<td>-0.08</td>
<td>-0.10</td>
<td>-0.083</td>
<td></td>
</tr>
<tr>
<td>Fatigue ductility exponent(c)</td>
<td>-0.69</td>
<td>-0.79</td>
<td>-0.722</td>
<td></td>
</tr>
<tr>
<td>Fatigue ductility coefficient(εf')</td>
<td>1.22</td>
<td>0.13</td>
<td>0.813</td>
<td></td>
</tr>
<tr>
<td>Cyclic-strain hardening exponent(n')</td>
<td>0.1</td>
<td>0.10</td>
<td>0.15</td>
<td></td>
</tr>
<tr>
<td>Cyclic strength coefficient(k)</td>
<td>2500</td>
<td>3498</td>
<td>1230</td>
<td></td>
</tr>
</tbody>
</table>

**Fig. 3: The Finite Element Based Fatigue Analysis Cycle**
D. Fatigue Analysis Methods

Fatigue analysis can be carried out one of three basic approaches i.e., the total life (stress-life) approach and crack propagation approach, the crack initiation approach. The total-life (stress-life) approach was first applied over a hundred years ago (Wohler, 1867) and considers nominal elastic stresses and how they are related to life. The crack-initiation approach considers elastic-plastic local stresses and how they are related to life. Crack-propagation or linear elastic fracture mechanics (LEFM) approach is used to predict how quickly pre-existing cracks grow and to estimate how many loading cycles are required to grow these to a critical size when catastrophic failure would occur. First two methods are used in this study and briefly discussed these two methods in the following sections.

The fatigue total-life (S-N) approach is usually used for the life prediction of components subjected to high cycle fatigue, where stresses are mainly elastic. This approach emphasizes nominal stresses rather than local stresses. It uses the material stress-life curve and employs fatigue notch factors to account for stress concentrations, empirical modification factors for surface finish effects and analytical equations such as modified Goodman and Gerber equations are given below:

\[ \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_u} = 1 \]  
\[ \frac{\sigma_m^*}{S_e} + (\frac{\sigma_m}{S_u}) = 1 \]

(1)  
(2)

Fig. 4: Representation of These Mean Stress Correction Methods

Where \( \sigma_a \), \( S_e \), \( \sigma_m \) and \( S_u \) are the alternating stress in the presence of mean stress, alternating stress for equivalent completely reversed loading, the mean stress and the ultimate tensile strength, respectively. The typical representation of these mean stress correction method shown in (fig 4). The Basquin (1910) showed that alternating stress verses number of cycles to failure (S-N) in finite life region could be represented as a log-log linear relationship. Basquin equation was then used to obtain the fatigue life using the material properties listed in (Table 2). S-N approach uses to estimate the fatigue life for combined loading by determining an equivalent axial stress using one of the common failure criteria such as Tresca, Von-Mises, or maximum principal stress. The S-N equation is mathematically given by:

\[ S_e = \sigma_f^b (2N_f)^{b} \]

(3)

IV. Boundary Conditions and Loading

A. Modelling

Equation ally driven spur gears was drawn in CATIA V5. By considering following parameters. Gear mate assembly was done:

- No of teeth: 25
- Pressure angle: 20°
- Module: 0.035m
- Pitch circle: 0.04m

B. Meshing

Fine meshing of tetrahedron type is done to get the accurate results of contact stress. Total nodes: 130158, Total elements: 27192
C. Boundary Condition

Tangential load of 2315 N is applied at the point of contact during the mating of the two gears. Frictionless support and the moment of 194.46 N-m to the gear and moment of 76.395 N-m to the pinion in opposite direction is given.

V. Results and Discussion

The life of gear was calculated using ANSYS software by considering suitable boundary conditions.

Fig. 9: Equivalent Stresses of the Gear

The von-misses equivalent stresses (fig. 9) are used for subsequent fatigue life analysis and comparisons. However, in table 3, it can be seen that when using the loading sequences are predominantly tensile in the nature; the Goodman approach is more conservative. [figs:10,11,12] represents damage values of gear based on stress analysis. However, in [Table 3] it can be seen that when using the loading sequences are predominantly tensile in the nature; the Goodman approach is more conservative.

For the above analysis figure of Spur Gear of SAE5160-825QT, the damage value under fully revered loading is (0.253m)

For the above analysis figure of Spur Gear of SAE1045-450-QT, the damage value under fully revered loading is (0.245mm)

For the above analysis figure of Spur Gear of ALSI4027, the damage value under fully revered loading is (0.2678m)

The three-dimensional cycle histogram and corresponding damage histogram for materials using fully reversed loading histories is shown in the (figs.13&14) given below. (Fig.13) shows the results of the rain flow cycle count for the component. It can be seen that a lot of cycles with a low stress range and fewer with a high range. The height of each tower represents the number of cycles at that particular stress range and mean. Each tower is used to obtain damage on the S-N curve and damage is summed over all towers. (Fig. 14) shows that lower stress ranges produced zero damage. It is also showed that the high stress ranges were found to give the most of the damage and a fairly wide damage distribution at the higher ranges which mean that it cannot point to a single
event causing damage. Most realistic service situations involve nonzero mean stresses, it is, therefore, very important to know the influence that mean stress has on the fatigue process so that the fully reversed (zero mean stress) laboratory data are usefully employed in the assignment of real situations.

VI. Fatigue Analysis of Gears Using Total Life Approach

Fatigue analysis were conducted by using ANSYS software by considering the boundary conditions.

For the above analysis figure of Spur Gear of SAE1045-450-QT, the minimum value under Goodman condition is \(4.019 \times 10^5\) sec which can be considered (fig. 15).

For the above analysis figure of Spur Gear of ALSI4027, the minimum value under fully reversed loading is \(3.733 \times 10^5\) sec which is more conservative. SAE1045-450-QT, under Goodman condition is \(4.019 \times 10^5\) which can be significant (fig. 15).

From the (Figs. 10, 11 & 12) ALSI4027 was subjected to more damage compare to sae materials. SAE1045-450-QT subject to least damage.

VIII. Future Scope

An important aspect of the fatigue process is plastic deformation. Fatigue cracks initiate from the plastic straining in localized regions significant localized plastic deformation is often present, therefore, cyclic strain-controlled fatigue method could better characterize the fatigue behaviour of materials. The cyclic strain controlled fatigue particularly in notched members where the significant localized plastic deformation is often present. In the crack initiation approach the plastic strain is directly measured and quantified. The total-life approach does not account for plastic strain. In strain-life when the load history contains large over loads, significant plastic deformation can exist, particularly at stress concentrations and the load sequence effects can be significant.
In these cases, the crack initiation approach is generally superior to the total life approach for fatigue life prediction analysis. However, when the load levels are relatively low such that the resulting strains are mainly elastic, the crack initiation and total life approaches usually result in similar predictions.

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References