Design and Fatigue Analysis of Yaw Gear box Shaft (17CrNiMo6) Subjected to Pure Torsion

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Abstract: In material engineering it is important to determine the cause of the failure & prevention of the failure. In present day the failure of the machine component is about 90% of the failure is because of the fatigue. In present study the failure of the shaft in the yaw gear box is analyzed. The shaft is rotating part of the yaw drive is used to keep the rotor facing into the wind as the wind direction changes. It holds the maximum stress. In this case shaft of heat treated component with ultimate tensile strength Su= 1295Mpa is Designed in UG NX-10 and Analysis done with ANSYS18.1 (FEA) software & compared with the theoretical calculation. There are different methods which are used to predict fatigue life include stress life(S-N), strain Life (E-N) and Linear Elastic Fracture Mechanics (LEFM). In this project study, S-N approach is used to predict fatigue life for yaw gear box shaft.

Index Terms: Shaft, UG-NX10, FEA-ANSYS (18.2), SN-curve, Fatigue—life, Fatigue factor of safety, Fatigue damage.

I. INTRODUCTION
A shaft has a circular cross section & it is a rotating part of the yaw drive is used to keep the rotor facing into the wind as the wind direction changes. A shaft usually not a uniform cross section because it is mounted by the bearing, fly wheels, clutches & other machine elements are mounted on the shaft. In the present shaft of heat treated material (17CrNiMo6) is mounted by spur gear with pressure angle of 20° & supported by the two bearing (Roller bearing).

Stress Analysis
Stress in the Shaft In actual practice there are three kinds of stress are induced in it. a.) Shear stress by the transmission of the torque. b.) Bending stress by the force acting upon machine element, and weight of shaft itself. c.) Stress from both combined tensional and bending loads.

II. SHAFT MATERIAL COMPOSITION AND MECHANICAL PROPERTIES
Material is used in the shaft is Chrome–Nickel- Moly carburizing steel (17CrNiMo6)

Table 1 Chemical composition of 17CrNiMo6

<table>
<thead>
<tr>
<th>composition</th>
<th>percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>0.18</td>
</tr>
<tr>
<td>Silicon</td>
<td>0.20</td>
</tr>
<tr>
<td>Nickel</td>
<td>1.55</td>
</tr>
<tr>
<td>Manganese</td>
<td>0.70</td>
</tr>
<tr>
<td>Chromium</td>
<td>1.65</td>
</tr>
</tbody>
</table>

Mechanical properties of 17CrNiMo6

Tensile yield strength Sy =1050Mpa
Ultimate tensile strength $S_u = 1295\text{MPa}$
Young’s modulus of material $E = 207\text{GPa}$
Passion’s ratio $= 0.3$
Density $\delta = 7800\text{Kg/m}^3$

III. DESIGN AND CALCULATIONS

**Fatigue stress concentration factor**

When there is an existence of the irregularities or discontinuities, such as holes, grooves, or notches, in a part increases the theoretical stresses significantly in the immediate vicinity of the discontinuity defined as a stress concentration factor $K_t$ or $K_{ts}$.

Effective maximum stress is

\[ \sigma_{\text{max}} = K_f \sigma_{\text{nominal}} \]
\[ \tau_{\text{max}} = K_s \tau_{\text{nominal}} \]

Where $K_f$ is a reduced value of $K_t$ and is the nominal stress $\sigma_0$. The factor $K_f$ is commonly referred as fatigue stress-concentration factor and hence the subscript $f$. The resulting factor is defined by the equation

\[ K_f = \frac{\text{maximum stress in notched specimen}}{\text{Stress concentration notch free specimen}} \]

Effective maximum stress

\[ K_f = 1 + q (K_t - 1) \text{ or } K_{fs} = 1 + q (K_{ts} - 1) \]

Where $q = $ Notch sensitivity factor

\[ q = \frac{1}{1 + \sqrt{\frac{a}{r}}} \]

Where $\sqrt{a} = $ Neuber constant

$r = $ Fillet radius

Neuber constant is given by the equation

\[ a = 0.246 - 3.08 \times 10^{-3} S_u + 1.51 \times 10^{-5} S_u^2 - 2.67 \times 10^{-8} S_u^3 \]

\[ a = 0.190 - 2.51 \times 10^{-3} S_u + 1.35 \times 10^{-5} S_u^2 - 2.67 \times 10^{-8} S_u^3 \]

Applying $S_u = 1295\text{MPa}$

\[ q = 0.9486 \]

Where $K_t = $ Stress concentration factor for torsion.

\[ K_{fs} = 1 + 0.9486 (1.85 - 1) = 1.80631 \]

Generation of $S$-$N$ curve

Fatigue strength (Endurance strength $S_n$) of the shaft material was calculated as

\[ S_n = 0.5 \times UTS = 0.5 \times 1295 = 647.5\text{MPa} \]

Considering the corrections factors for endurance limit
Ka = surface finish factor = 0.5  
Kb = size factor = d>50 nm = 0.75  
Kc = Reliability factor (for 90% Reliability) = 0.897  
Kd = Load factor = 1  
Endurance limit Se = 0.5 Sn*Ka*Kb*Kc*Kd  
= 0.5 * 1295 * 0.5 * 0.75 * 0.897 * 1  
= 217.81 Mpa  

_S-N Curve_  
Tensile yield strength Sy = 1050 Mpa  
Ultimate tensile strength Su = 1295 Mpa  
Reliability = 90%  
0.9 Su = 0.9 * 1295 = 1165.5 Mpa  
log₁₀(0.9 Su) = 3.0665  
log₁₀(0.9 Se) = 2.34  
log₁₀(0.9 Sf) = 3.02  

_S-N CURVE_  

\[ EF = \frac{DB \times AE}{(6-3)(3.06-3.02) / (3.06-2.34)} = 0.18 \]  
\[ \log N_{10} = 3 + EF \times 3 + 0.18 = 3.18 \]  
\[ N = 1513.56 \text{ Cycles} \]  

_Shift Shear stresses_  
Yaw gear box shaft is subjected to a torque of 50000 N-m. The maximum shear stress is shown in which appears in the cross-section of the shaft \( \tau_{max} \) with the fatigue stress concentration \( k_{fs} = 1.8 \)
\[ \tau_{max} = 16T \times 1.8 \]
\[ \Pi d^3 = 16 \times 5000000 \times 1.8 \]
\[ \Pi 85^3 = 746.37 \text{ Mpa} \]

**Equivalent alternating stress or Equivalent von-misses stress**

In a fatigue Life Stress analysis, one always needs to be query of an SN curve to relate the fatigue life to the stress state. Thus of the “equivalent alternating stress” is the stress used to query the fatigue SN curve after accounting for fatigue loading type, mean stress effects, multi axial effects and any other factors in the fatigue analysis. Thus in a fatigue analysis, the equivalent alternating stress can be thought of as the fast calculated quantity before determining the fatigue life. The maximum value of the equivalent stress is 1527.4 Mpa, which takes place in the cross-section of the shaft where the Roller-bearing was located.

Equivalent von-mises stress or alternating stress \( \sigma_v \)

\[ \sigma_v = \sqrt{3 \tau_{max}^2} \]
\[ = 1292.75 \text{ Mpa} \]

**Factor of safety of fatigue life**

Factor of safety of fatigue life = \( \frac{\text{Endurance stress}}{\text{Equivalent alternating stress}} \)

\[ = \frac{217.8}{1292.75} \]
\[ = 0.17 \]

**Fatigue damage**

Fatigue damage is defined as the design life divided by the available life

Fatigue damage = \( \frac{\text{Design life}}{\text{Available life}} \)

\[ = \frac{10^6}{1513.56} \]
\[ = 660.6 \]

**IV. DESIGN OF YAW GEAR BOX SHAFT IN UG NX 10**

![Fig 3: Sketch of yaw gear box shaft](image)
V. ANALYSIS OF YAW GEAR BOX SHAFT

Meshing of shaft

Basic idea of FEM is to perform calculations at limited number of points called nodes and interpolate the results for entire domain using interpolation functions. Any continuous object has infinite degrees of freedom and such problems cannot be solved using this method. So this method reduces the degrees of freedom from infinite to finite by making assumptions and by discretization/meshing in other terms creating nodes and elements. Hexagonal mesh is used, relevance center is fine & elements size is 3.14mm created the nodes 10483 and elements 5839.

Loads acting on shaft

Stepped shaft is subjected to a torque of 50000N-m at the right end of shaft.
Shaft Shear stresses

Figure 7: Shaft Shear Stress in Mpa

Equivalent alternating stress or Equivalent von-misses stress

Figure 8: Equivalent alternating stress in Mpa

Factor of safety of fatigue life

Figure 9: Factor of safety
Number of cycles

![Figure 10: Showing the fatigue life](image)

Fatigue damage

![Figure 11: Fatigue damage](image)

V. RESULTS AND DISCUSSION

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Analytical results</th>
<th>FEM results</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Shear stress in Mpa</td>
<td>746.37</td>
<td>881.82</td>
</tr>
<tr>
<td>2. Equivalent alternating stress in Mpa</td>
<td>1292.75</td>
<td>1527.4</td>
</tr>
<tr>
<td>3. Factor of safety</td>
<td>0.17</td>
<td>0.06</td>
</tr>
<tr>
<td>4. Fatigue life cycles</td>
<td>1513.56</td>
<td>832.29</td>
</tr>
<tr>
<td>5. Fatigue damage</td>
<td>660.6</td>
<td>1000</td>
</tr>
</tbody>
</table>

VI. CONCLUSIONS

Fatigue Failure analysis of the shaft is investigated in detail. Force acting on the bearing due to the torque is determined. Endurance limit & fatigue factor of safety is calculated. Fatigue life of the shaft is estimated & fatigue damage is calculated. Forces and stresses are calculated by using an analytical approach and ANSYS software. Both methods show that the stresses and fatigue life are nearly same and in the admissible range.

1. From the static analysis it was observed that maximum stress is located at the change in the cross-section area of the shaft it is found 1527 Mpa.
2. Fatigue life is found to be 832 cycles.
3. Factor of safety is 0.06<1 which means the design is not safety.
4. Fatigue damage at the given life is 1000
5. Stress concentration is more where there is a change in the cross-section of the shaft.

REFERENCES