VOL. 18, NO. 3, FEBRUARY 2023 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

DESIGN AND FATIGUE ANALYSIS OF DRIVE SHAFT BY USING FINITE ELEMENT METHOD AND CATIA SOFTWARE

Aliyi Umer Ibrahim[1] and Bejai Naiker Nerash[2]
[1]Department of Production Engineering, Center for Research and Community Service, College of Engineering, Ethiopian Defence University, Ethiopia
[2]Bejaiethio Industrial and Engineering solutions, Ethiopia
E-Mail: aliyiumer@yahoo.com

ABSTRACT

The main objective of this research is to investigate the design and fatigue analysis of stresses & deflections of drive shaft subjected to combine bending & torsion by using Finite element (FEM) and (CATIA) software. Then checking for fatigue life as well as compare the results with analytical calculations to verify the accuracy of the results. The drive shaft is a critical component used in paper-converting machines. It carries a load of two vacuum rollers weighing around 1471N and rotates at 1000 rpm, also subjected to the reaction force. This shaft has key slots at the area of change in cross sections giving rise to localized stress concentration. As the pressure load on the shaft is found to be stressing the material well below its yield point stress, the High Cycle Fatigue analysis (HCF) is chosen and the estimated S-N diagram for the material of the shaft gave an estimated life equal to 20,934 cycles analytically. The Finite Element Modeling and analysis of the shaft performed using ANSYS resulted in a fatigue life of 20935 cycles corresponding to cumulative fatigue damage equal to 0.7. The ANSYS result for Maximum stress and Deflection are 41.89Mpa and 0.045 respectively. Similarly, Maximum stress and Deflection are 41Mpa and 0.039 respectively from the analytical results. As can be seen, the FEM-based analysis and the analytical results do match very well.

Keywords: fatigue analysis, shaft stress analysis, FEM analysis, shaft failure analysis.

1. INTRODUCTION

A shaft is a rotating member usually of a circular cross-section (solid or hollow), which is used to transmit power and rotational motion in machinery and mechanical equipment in various applications. Elements such as gears, pulleys (sheaves), flywheels, clutches, and sprockets are mounted on the shaft and are used to transmit power from the driving device (motor or engine) through a machine [1] [4]. In deciding on an approach to shaft sizing, it is necessary to realize that a stress analysis at a specific point on a shaft can be made using only the shaft geometry in the vicinity of that point. Thus the geometry of the entire shaft is not needed. In design, it is usually possible to locate the critical areas, size these to meet the strength requirements, and then size the rest of the shaft to meet the requirements of the shaft-supported elements [7].

Most shafts are subjected to fluctuating loads of combined bending and torsion with various degrees of stress concentration. For such shafts, the problem is fundamentally fatigue loading [3] [6]. Failures of such components and structures have engaged scientists and engineers extensively in an attempt to find their main causes and thereby offer methods to prevent such failures.

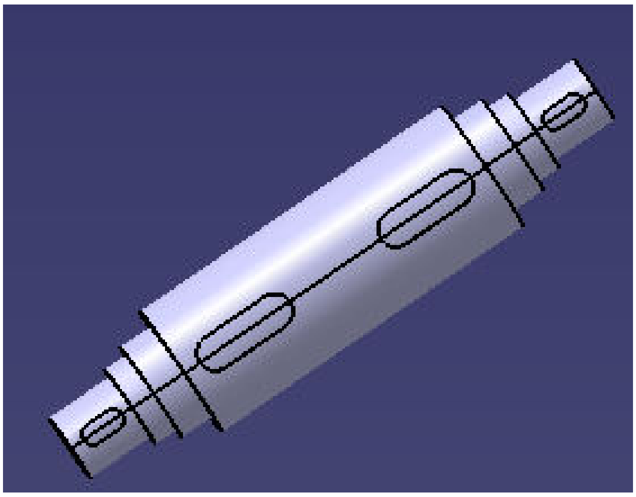

Figure-1. 3D diagram of the shaft modeled by catia.

2. METHODOLOGY

A detailed survey of published literature on the effects of fatigue life and the mechanism of a drive shaft is done. Information from the industries which are using drive shaft is gathered. The drive shafts is modeled using CATIA V5 R19 software. An estimated S-N (Stress-No. of cycles) diagram is constructed. Fatigue analysis on the drive shaft to estimate the fatigue of the drive shaft analytically and estimate the number of cycles of the drive shaft analytically from the fracture mechanics approach using MATLAB is performed. Finally, the FEM model of the drive shaft is made and analyzed and the results are compared.

3. ANALYTICAL ANALYSIS OF FATIGUE DRIVE SHAFT

The drive shaft is made of ferritic - pearlitic steel forging SAE 4340 and the mechanical properties of the metal and the constants used for the analysis are as follows:

Brinell hardness	(BHN) = 409
Ultimate tensile strength	(S_u= 1470Mpa)
Modulus of elasticity	E= 200GPa
Yield point	S_y= 827Mpa
Cyclic strain hardening exponent	n'= 0.15
Fatigue strength coefficient	σ'_f = 2000Mpa
Fatigue strength exponent	b = - 0.091
Fatigue ductility coefficient	$\varepsilon'_f = 0.48$
Fatigue ductility exponent	c = -.60
Material constant	C=6.8X10^{-12}
Minimum stress	σ_{min}=0
Paris exponent of the material	n or m=3
Dimensionless parameter	$Y or \alpha = 1.12$

Fatigue concentration factor for a drive shaft, K_f = 2.601.
Maximum working pressure acting on drive shaft (with each firing) =6700 bar or 670 MPa.
Poison's ratio= 0.3 and fracture toughness =135MPa$\sqrt{m}$

a) The S-N Curve

For the designer, it is critical that the relationship between applied stress and expected life be characterized so that fatigue life can be predicted [5]. One of the early methods for characterizing this relationship is the S-N curve (i.e., S = cyclic stress range, N = the number of cycles to failure).

The correction factor to the theoretical endurance limit [7] is given by:

Se = K_{load} .K_{size} .K_{surf}.K_{temp} .K_{reliab} .S'_e
Where S'_e=700 N/mm^2

The various strength reduction factors are:

Loading factor-For bending load, a strength reduction *load factor* k_{load} is equal to: k_{load}= 1.0

Size Effect-Strength-reduction size factor k_{size} needs to be applied to account for the fact that the larger fail at lower stresses due to the higher probability of a flaw being present in the larger stressed volume [1][3]. This may be approximately taken equal to 0.85 for ultimate stress of 1470 N/mm^2 from the standard graph.

Surface Effect-The specimen is polished to a mirror finish to preclude surface imperfections serving as stress risers.
The strength-reduction surface factor k_{surf} is given by [2]:
$k_{surf}=A * (\sigma_{ut})^{b}$

Assuming the surface finish as machined or cold-rolled would give A=4.51Mpa and b=-0.265
σ_{ut}= 1470 MPa.
k_{surf}= 4.51 × (1470)$^{-0.265}$
k_{surf}=0.653

Temperature effect-A strength-reduction *temperature factor* k_{temp}, which was suggested by Shigley and Mitchell, is as follows: k_{temp}= 1 for $T \leq 450^0$C
k_{temp}= 1 – 0.0058* (T – 450) for $450^0C \leq T \leq 550^0C$

Since the for operating temperature less than $T \leq 450^0$C, then k_{temp}= 1

Reliability-Thus, the strength-reduction *reliability factor* k_{reliab} =99.999 %

Now by applying all the strength-reduction factors on the uncorrected (theoretical) endurance limit value $S'e$, the corrected endurance limit Se for drive shaft may be obtained as:

s_e=1.0x0.85x0.653x0.659x700
$s_e = 256.05\ N/mm2.$

Now, considering a fatigue concentration factor for drive shaft, K_f = 2.601, the actual fatigue strength will be 256.05/2.601= 98.43 N/mm^2, N=10^6 cycles.

S_N= 0.9 S_{ut}=0.9x1470=1323 N/mm^2 at N=10^3 cycles.

From Underwood JH (1984) the S-N diagram is constructed based on the formula:

$S_{(N)}=aN^b$
Log $S_{(N)}$=loga+blogN
Log 1323=loga+3blog10, and log98.43 =loga+6blog10

Solving for 'a' and 'b' b=-0.376 and a=17782.475

Therefore the stress life equation will be:

S $_{(N)}$ =17782.475N$^{-0.376}$ for $10^3 \leq N \leq 10^6$

$$S_{(N)} = 17782.475N^{-0.376} \text{ for } 10^3 \leq N \leq 10^6 \quad (1)$$

b) Calculation of bending stress

The force acting on the shaft is F=pressure x are:

$$F = P \times A \quad (2)$$

= 670× 150 × 250
=25.125× 10^{6}N/mm^2

The amplitude and mean stress are calculated as follows:

σ_a= 581.59/2=290.5 N/mm^2 σ_{min}= 0
σ_m= 581.59/2=290.5 N/mm^2

To find the equivalent stress amplitude with 'zero' mean stress:

Considering the factor of safety equal to 1.5 then the ultimate design stress for the material of will be:

140 0/1.5=933.33 N/mm^2

VOL. 18, NO. 3, FEBRUARY 2023 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

The equivalent stress amplitude with 'zero' mean stress,

$$S_{(N)}= S_u[\frac{S_a}{S_u - S_m}] \quad (3)$$

$$= 933.33[\frac{290.5}{933.33-290.5}]$$

$=422.042\ N/mm^2$

The number of cycles of the drive shaft when the S_N =422.042 N/mm^2 will be calculated using equation (1):

$422.042=17782.475N^{-0.376}$

Therefore, N= 20934cycles.

Figure-2 shows the estimated stress versus number of cycle's diagram (S-N diagram) for the material of drive shaft.

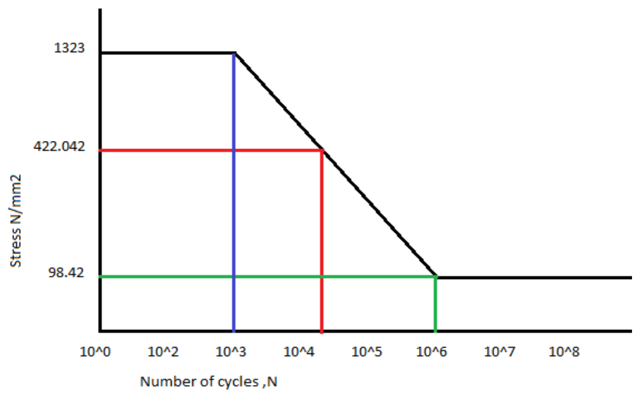

Figure-2. Stress versus number of cycles diagram (s-n diagram).

4. ANALYSIS OF FATIGUE BY USING FEM

ANSYS is a finite element software package that was first commercially available in 1970 (Swanson Analysis Systems, Inc.). Since then, ANSYS has been used by design engineers throughout the world for such engineering applications as structural, thermal, fluid, and electrical analyses. In this thesis, ANSYS is used as a computational tool for modeling and simulation of the drive shaft. For any fatigue life analysis there are always three inputs, analysis, and hence getting the result as illustrated in Figure-3.

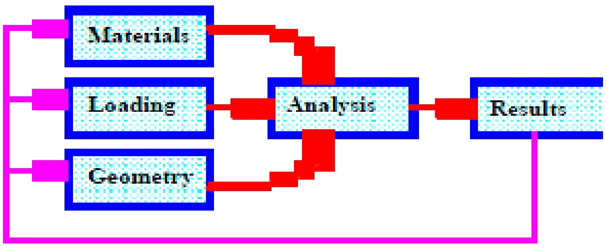

Figure-3. The 'fatigue'5 box trick.

a) The reasons to prefer the element type 'PLANE 182'

a) It is used for two-dimensional modeling of ax symmetric structures

b) It has compatible displacement shapes and is well-suited to model curved boundaries,

c) It provides more accurate results for mixed (quadrilateral-triangular) automatic meshes and can tolerate irregular shapes without as much loss of accuracy,

d) There is no product to restrict to this element.

b) Assumptions and restrictions as regards to PLANE182

The area of the element must be nonzero. The element must lie in a global X-Y plane and the Y-axis must be the axis of symmetry for ax symmetric analyses. An ax symmetric structure should be modeled in the +X quadrants. There are no product-specific restrictions for this element.

c) Finite element modeling

Proper modeling and analysis specifications are crucial to the success of any finite element analysis. The finite element model contains all the necessary data for each step of numerical simulation namely, geometry, elements, loads, boundary conditions, solution of the system of equations, visualization and output of results, etc. This section attempts to conceptualize and illustrate the procedure for building a complete model and then performing simulation for the shaft.

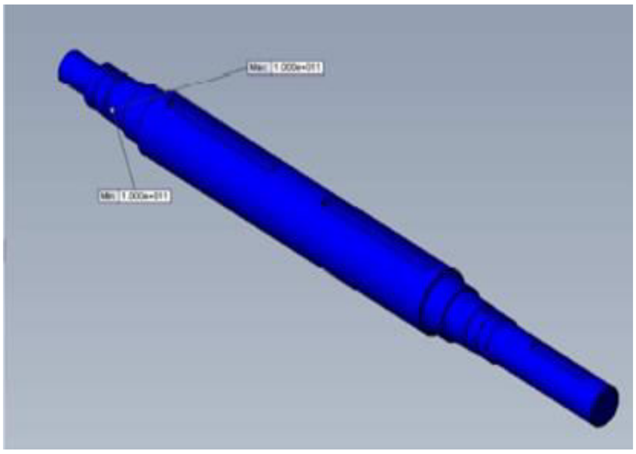

Figure-4. FEM study using simulation.

VOL. 18, NO. 3, FEBRUARY 2023 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

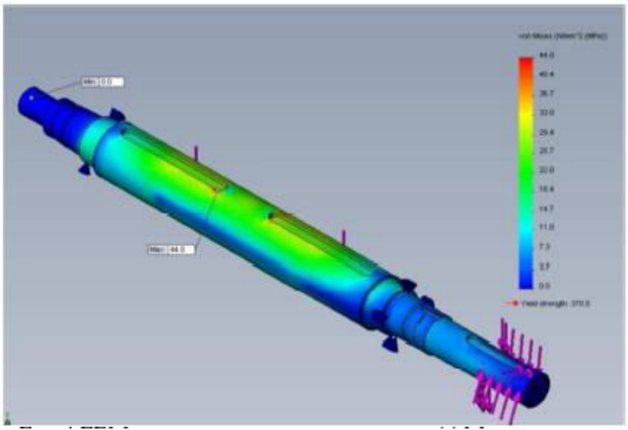

Figure-5. FEM Stress plot shows max stress of 41.89mpa at the centre.

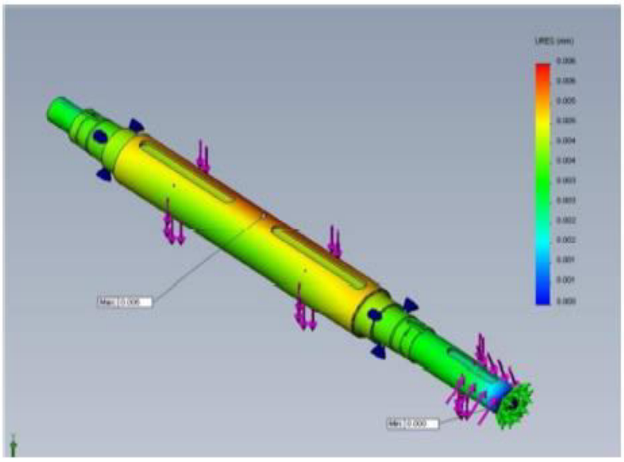

Figure-6. FEM deflection plot shows max deflection of 0.045 at the centre.

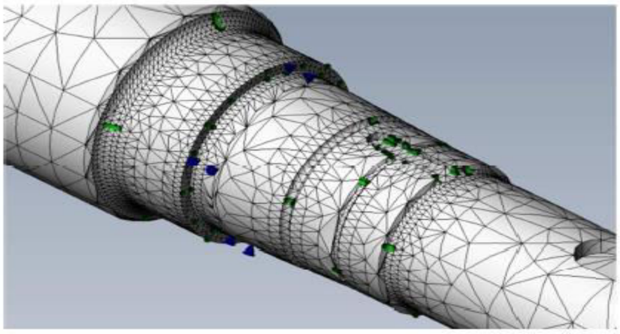

Figure-7. Close look at the mesh control at the critical junction's analysis.

5. RESULTS AND DISCUSSIONS

The analytical and numerical results obtained on the fatigue and fracture analysis of the shaft are presented here.

As the nature of load on the shaft is of low magnitude type the High Cycle Fatigue analysis (HCF) is appropriate and the estimated S-N diagram for the material of the shaft gave an estimated life equal to 20, 934 cycles. The Finite Element Modeling and analysis of the shaft performed using ANSYS resulted in a fatigue life of 20445 cycles corresponding to cumulative fatigue damage equal to 0.7. The ANSYS result for Maximum stress and Deflection are 41.89Mpa and 0.045 respectively. Similarly, Maximum stress and Deflection are 41Mpa and 0.039 respectively from the analytical results.

Table-1. Comparison of results.

Parameter	Analytical results	FEM results
Max Stress in Mpa	41Mpa	41.89Mpa
Deflection in mm	0.039	0.045
Fatigue Life in cycles	20445	20445
Fatigue Damage	0.65	0.7

As can be seen the FEM based fracture analysis results and the analytical results do match very well.

6. CONCLUSIONS

The fatigue and fracture analysis performed on the drive shaft both by analytical means and by Finite Element Method analysis using ANSYS resulted in a fatigue life estimate which is very close to each other. Due to the involvement of several approximations and assumptions in the analytical analysis as regards the fatigue life estimate the method can be used as an initial estimate for the fatigue life of the shaft. However in the FEM analysis, as the load can be applied directly on the shaft, the obtained estimate for the fatigue life is more accurate. The knowledge gained as regards the fatigue life estimation may be extended to the analysis of similar other components which are subjected to fatigue loads.

Based on FEM results it is found that max stresses are found at the bearing junction because of the change of cross section and small radius at the junction. The geometry of the shaft is modified by introducing a step at both sides of the shaft before bearing junction and giving max. Radius, so that stress concentration could be minimized at the bearing junction.

RECOMMENDATION

The best information on the fatigue analysis under cyclic loading will come from actual testing. But no experimental testing was carried out on the drive shaft. Therefore, Experimental validation can be taken as one of the future tasks.

NOMENCLATURE

σ_a Alternating stresses (N/mm^2)
A Amplitude ratio (Unit less)
σ_{max} Local maximum stress (N/mm^2)
b Constant defined at boundary condition(Unit less)
σ_m Local mean stress (N/mm^2)
S_e Corrected endurance limit (N/mm^2)
σ_{max} Maximum stress (N/mm^2)
S_f Corrected fatigue limit (N/mm^2)

B Exponent used for various surface finishes (Unit less)
K_f Fatigue concentration factor (Unit less)
N Number of cycles

REFERENCES

[1] A. P. Parker. 1999. Prediction of fatigue life of the AA6061-T6-80HF aluminum alloy at most critical location. Journal of Pressure Vessel Technology. 121: 430-436.

[2] Beka Hailu Abebe. 1981. Fatigue Life Assessment of Diesel Engine Pump Part Subjected to Constant and Variable Amplitude Loading. Master Thesis and Ethiopia.

[3] E. Troiano. 1993. Fracture Analysis of 155-mm M185/M284 Rings S/N 1659, S/N 2101, and S/N 1623. Memorandum for Record, Benet Laboratories, Watervuet, NY.

[4] 2002. Fatigue Failures. Failure Analysis and Prevention, Vol 11, ASM Handbook, AS International.

[5] Fracture and Fatigue Control in Structures; Application of Fracture Mechanics. (3rded). ASTM, MNL41.

[6] J. H. Underwood. 1984. Fatigue Life Analysis and Tensile Overload Effects with High Strength Steel Notched Specimens. Materials Research Society Symposium Proceedings, Vol. 22, Elsevier Science Publishing, London. pp. 209-214.

[7] M. R. Mitchell. 1996. Fundamentals of Modern Fatigue Analysis for Design, Fatigue and Fracture, Vol. 19, ASM Handbook, ASM International.