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Analysis of Drive Shaft of Packaging Machine

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ABSTRACT

Drive shaft of a packaging machine is one of the critical component , the failure of which in service must be avoided. A horizontal and a vertical electrical heater used for packaging are mounted on the shaft. The objective of this dissertation is to perform Fatigue analysis of drive shaft and find out stress of the shaft. A general finite element method (FEM) used for the stress and deformation analysis and also calculate failure calculation of shaft. MATH CAD software is selected as mathematical tool for the study. The study involves solid modeling, meshing of drive shaft and analysis with the solver. The validation will be carried out with available results obtain from experimental setup.

Keywords- **Factor of Safety, Fatigue, FEA, Goodman, Stress Concentration**

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I. INTRODUCTION

In SMD packaging machine (Side sealing mechanical double head machine) the drive shaft is mainly for transmission of motion from the servo motor to vertical and horizontal heater operate mechanism. i.e. it is transfer motion from point to another point. Drive shaft of SMD machine show in following fig.1.
Rotary motion of drive shaft convert into translatory motion by cam &follower and drive link. cam& follower mechanism present at top end of drive shaft and giving translatory motion to vertical heater. drive link present at bottom end of drive shaft. this drive link connected to crank drive shaft.It can be observed that a drive shaft is one of the most important components, which is responsible for the actual movement of the all machine component.

Functions of the Drive Shaft

- It must transmit torque from the Servo motor to the Drive link.
- The Drive shaft must be capable of rotation at high speed.
- During the operation it is necessary to transmit maximum torque developed by servo motor.
- The length of the drive shaft must also be capable while transmitting torque.

I. THEORY

A.Fatigue failure

In many engineering applications, the mechanical components are subjected to cyclic loadings the load may vary in magnitude and/or direction. The loads, which vary in magnitudes and/or direction with respect to time, are known as fatigue, and the variables stress induced in the comp is known as fluctuating stress. It has been observed that, when the mechanical comp is subjected to fluctuating loads, it fails at a stress considerably below the ultimate strength and quite frequently even below the yield strength. Such type of failure is known as fatigue failure.The most distinguishing characteristics of fatigue failure is that the stress are repeated a very large number of times. It has been found that the magnitude of the stress at which fatigue failure assures decreases as the number of stress cycles increases.

A fatigue failure occur in following way.

Stage I-is the initiation of one or more micro cracks due to cyclic plastic deformation followed by crystallographic propagation extending from two to five grains about the origin. Stage I cracks are not normally discernible to the naked eye.

Stage II -progresses from micro cracks to macro cracks forming a sudden, fast fracture. stage III - fracture can be brittle, ductile, or a combination of both. Quite often the beach mark.

B. Fatigue- life methods

The two major fatigue life methods used in design and analysis are,

1) stress-life method

The stress-life method, based on stress levels only, is the least accurate approach, especially for low-cycle applications. However, it is the most traditional method, since it is the easiest to implement for a wide range of design applications, has ample supporting data, and represents high-cycle applications adequately.

2)strain-life method

The strain-life method involves more detailed analysis of the plastic deformation at localized regions where the stresses and strains are considered for life estimates. This method is especially good for low-cycle fatigue applications. In applying this method, several idealizations must be compounded, and so some uncertainties will exist in the results. For this reason, it will be discussed only because of its value in adding to the understanding of the nature of fatigue.

C. Selection and use of failure theory

Here select the Distortion energy theory for fatigue failure analysis to find maximum stress values because there is combined loading of bending and torsion. Distortion energy theory is used when the factor of safety is to be held in close limits and the cause of failure of the component is being investigated. This theory predicts failure most accurately for ductile material. But design calculations involved in this theory are slightly complicated as compared with other theories of failure. The below equation is for design of shaft for fluctuating load calculated von-Mises stress.

D. Material properties of EN8 material

The CAD model of the shaft with its components is shown in fig.4.The material of the shaft isEN8 steel. Mechanical properties are shown in Table I,

TABLE I
Material Properties of Shaft

Properties	Value
Density (ρ)	2810 Kg/m^3
Ultimate Tensile Strength (S_{ut})	660MPa
Tensile Yield Strength (S_{yt})	590MPa
Young’s Modulus (E)	210 GPa
Poisson’s Ratio	0.30

II. FATIGUE ANALYSIS BY ANALYTICALAPPROACH

In this section, calculated the von-Mises stress, factor of safety & Number of cycles(N).

A. Input Forces and Torque on shaft.

Total force exerted by spring on cam,

where

K = Spring rate

x_{max}= Max. deflection of spring

x_{min}= Min. deflection of spring

F = spring force

F_{max}=max. force on cam

F_{min}=min. force on cam

spring force= (spring rate) * (deflection of spring)

Max. force on cam by 2 spring,

$F_1 = 2 * K * x_{max}$

Min. force on cam by 2 spring,

$F_2 = 2 * K * x_{min}$

B. SFD & BMD diagram of shaft.

from SFD & BMD diagram calculate , maximum bending moment,minimum bending moment at different points are given in table II,

TABLE II
Specifications of drive shaft

R_A	Reaction at bearing on point ' A'	5424.6 N-mm
R_C	Reaction at bearing on point ' C '	911.26 N- mm
$(M_B)_{Max}$	Maximum Bending Moment atB	383633.9 N-mm
$(M_B)_{Min}$	Minimum Bending Moment at B	383633.9N-mm

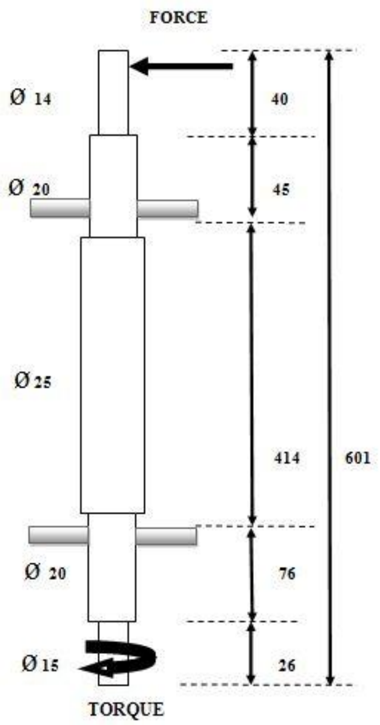

Fig.1Drive shaft

C. To find Resultant Mean stress (σ_m)and Resultant Amplitude stress(σ_a)

According to Distorsion energy theory

Resultant mean stress (σ_m)

$$\sigma_m = \sqrt{(\sigma bm)^2 + 3(\tau m)^2} \text{---------}(1)$$

Resultant stress amplitude (σ_a)

$$\sigma_a = \sqrt{(\sigma ba)^2 + 3(\tau a)^2}$$

D. To find Endurance limit (S_e)

Endurance limit of standard specimen

Se'= 0.5* Sut ------ for steel

Endurance limit of Mechanical component

S_e= K_a* K_b*K_c*K_d*K_e*K_g*Se'-------------(2)

K_a= surface finish factor

K_b=size factor

K_c= load factor

K_d= Temperature factor

K_e=modifying factor for stress concentration

K_g=Reliability factor

E. To find Factor of safety(Nf).

Goodman Diagram

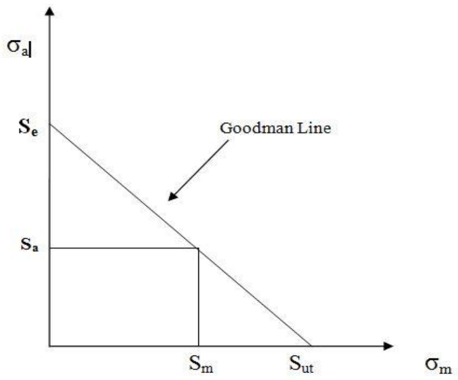

Fig.2Goodman diagram

$$\frac{1}{Nf} = \frac{\sigma_a{}'}{S_e} + \frac{\sigma_m{}'}{S_{ut}}$$

F. To findlife of shaft from S-N diagram.

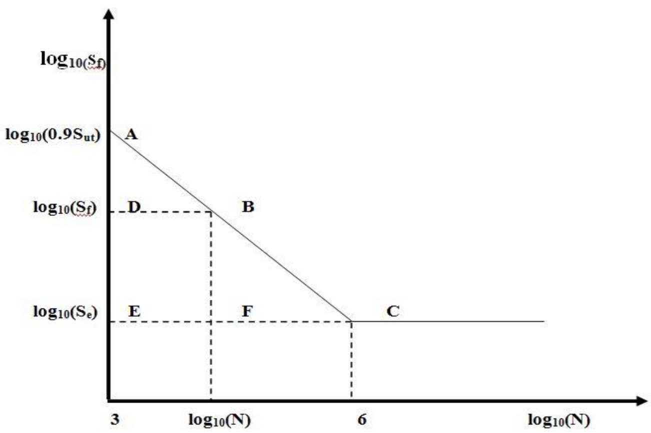

Fig.3S-N diagram

According to S-N diagram

⊿ FBC and ⊿ EBA

$$\frac{CF}{FB} = \frac{AE}{EB}$$

$$\frac{\log_{10}(S_f) - \log_{10}(S_e)}{6 - \log_{10}(N)} = \frac{\log_{10}(0.9S_{ut}) - \log_{10}(S_e)}{6 - 3}$$

--------------(3)

from above equation calculate fatigue life of shaft.

III. FATIGUE ANALYSIS BY FEA APPROACH

With the help of FEA analysis ,calculated the Von- Mises stress, factor of safety and life of shaft. Which is shown in bellow.

A. CAD model of drive shaft

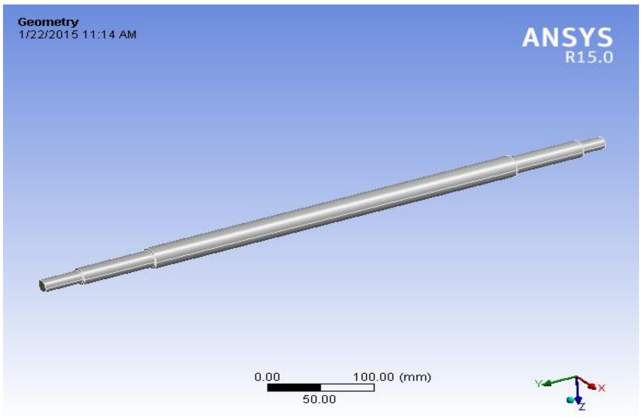

Fig.4CAD model of drive shaft

B. Meshing of drive shaft

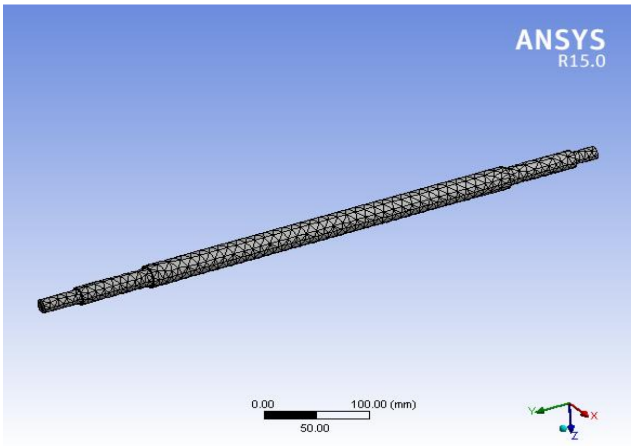

Fig.5Mesh model of drive shaft

C. Constraints of drive shaft

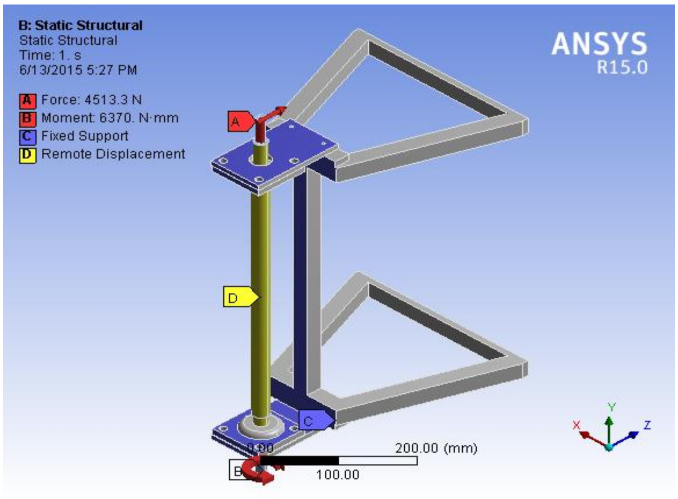

Fig.6Constraints of drive shaft

D. Results of drive shaft

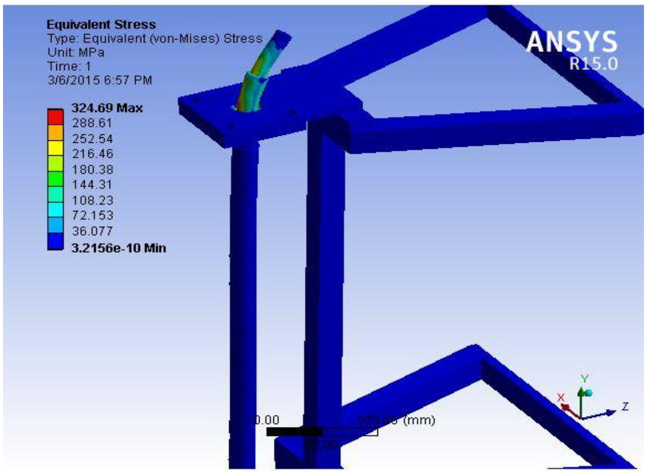

Fig.7Von-Mises stress of drive shaft

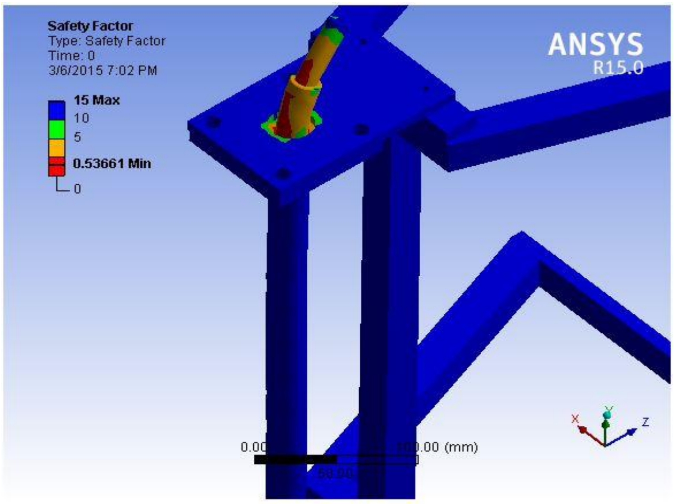

Fig.8Factor of safety of drive shaft

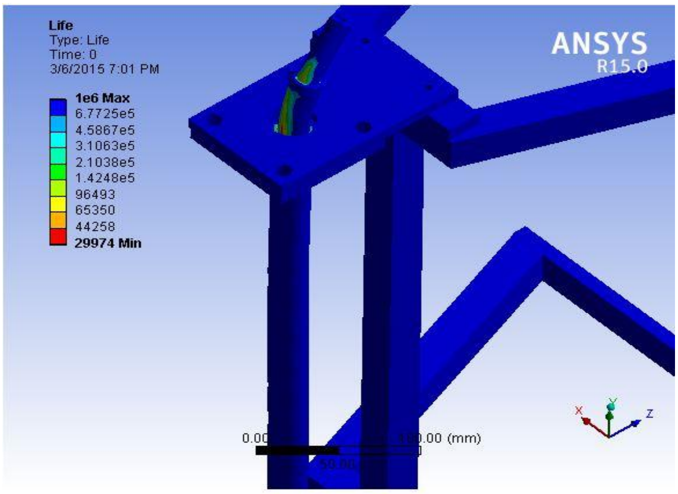

Fig.9Lifeof drive shaft

IV. RESULTS AND DISCUSSION

From the above Analytical and FEA analysis , compared results

TABLE III
Comparison of Analytical and FEA

Analytical approach	FEA approach
σm = 333.049 MPa	**σm = 324.69 MPa**
Nf = 0.785	**Nf = 0.536**
N = 48796 Cycles	N = 44258 Cycles

V. EXPERIMENTAL SETUP

Fatigue analysisshall be performed in Autocluster lab at pimprichichwad, Pune to find out fatigue life of drive shaft. following is setup of fatigue machine shown in cad model and also prepared fixture which is used for holding shaft, which used in fatigue testing machine.

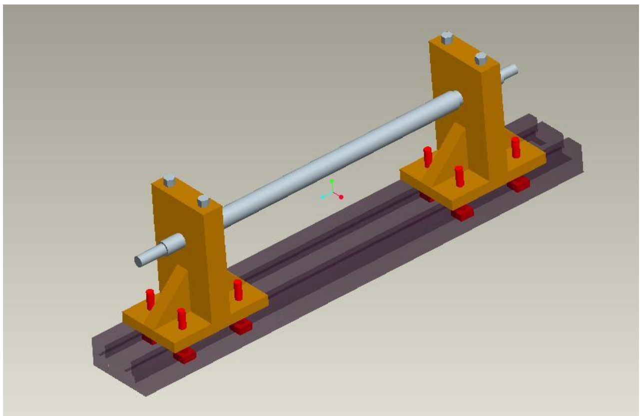

Fig.10Fixture of shaft

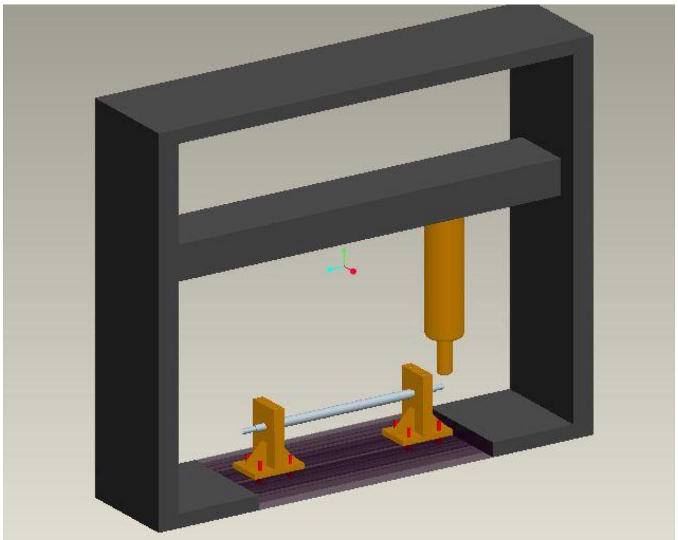

Fig.11fatigue testing machine setup

VI. Conclusions

The fatigue life prediction is performed based on finite element analysis and analytical method.Using the fluctuating loading , the fatigue life of the drive shaft has been predicted. This study can help to understand the behavior of the drive shaft and improve the fatigue life of the drive shaft using FEA tools. It will help to reduce cost, critical speed and times in research and development of new product. Combined loading (Bending and torsion) problem solved for calculating stresses, endurance limit, factor of safety and life of shaft by both mathematical as well as software solution.Thus, it was observed from the results that for combined loading both materials came out with comparable results.

Also, it is clear from above results, Von-Mises stress value by analytical approach, which are nearly same by using FEA approach having minimum difference in both results which is acceptable range. The number of life cycles are calculated by using Goodman method from S-N construction and by using FEA also matched.

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