Analysis of Shaft by using Theoretical and FEA Approach for find failure with high torque

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***Abstract*—** Recent tread of technology requirement of automation, as part of industrial development which are development in automation. This research will work on analysis and optimization of shaft used in chemical plant for different loading and boundary condition to sustaining different kind of load it was static and dynamics for handling torque. The shaft analysis was checked by using theoretical and FEA approach to justified failure mode and checked design validation.

***Index Terms*—**Theoretical Approach, FEA, Shaft, Fatigue and Crack.

I. Introduction

Failure causes
In order to properly dimension the rotating system, the failure causes or modes have to be determined. Most failures are due to unpredictable factors and the cure is to create a robust design that can cope with the unknown. Predictable failures or lifetimes can be calculated although many parameters will vary and statistical considerations have to be taken into account.
Shaft
The shaft is designed to have an infinite life. This will be the case unless the shaft is overloaded or damaged.
Fatigue crack
The cause of a broken shaft is almost always fatigue. A crack starts at a stress concentration from a keyway or a sharp radius, or, in rare cases, from a material impurity. Flaws in the surface of a shaft, such as scratches, indents or corrosion, may also be the starting point of a fatigue crack. Loads that drive a crack are normally torsion loads from direct online starts or bending loads from the pulley or coupling end.

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II Calculation of Shaft
Force Analysis of shaft [1,2]
Design of shaft
Static Case: Analytical Solution
Motor capacity = 15 H.P.= 15 x 0.7457 = 11.186 kW
Motor speed (N) = 1000 rpm
Output speed (No) = 31.2 rpm

$$\omega = \frac{2\Pi N}{60} = \frac{2\ x\ \Pi\ x\ 31.2}{60}$$

$$\boldsymbol{\omega = 3.2673 rad/sec}$$

Torque (T) = Power / Speed = 11.186 x 1000 / 3.2673

$$\boldsymbol{T = 3423.6\ in\ Nm}$$

σ_{all} = 550 MPa
$\rho = 8000\ kg/m^3$
Self-weight = 21.15 kg
Spur weight = 75 Tone = 75000 kg

Material: EN8
Minimum diameter of shaft = d_{min} = 80mm

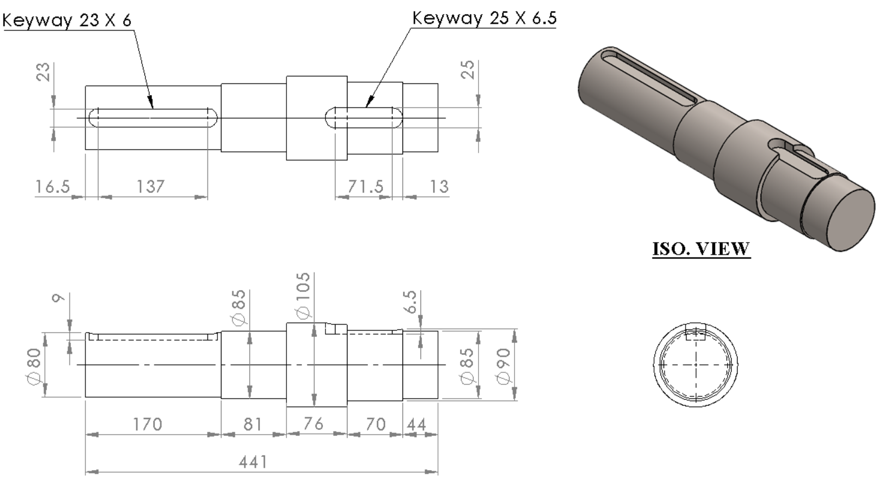

Fig. 1 Detail drawing of shaft

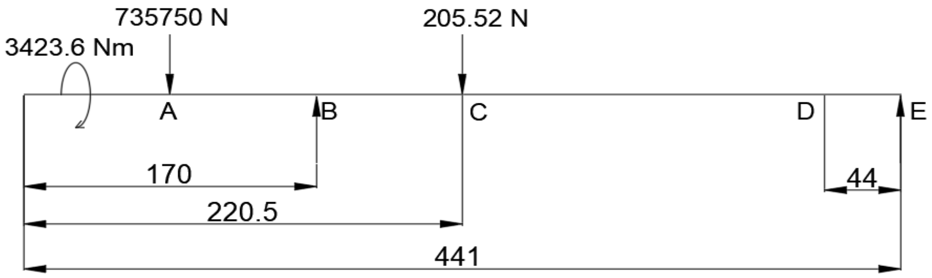

Fig. 2 Free Body Diagram

Free Body Diagram of shaft
T = 3423.6 Nm
T = 3423600 Nmm

Spur weight = 75000 x 9.81 = 735750 N
Self-weight (Shaft weight) = 20.95 x 9.81 = 205.52 N

To calculate reaction forces:
735750 + 205.52 = $R_B + R_E$

Taking moment @ R_B
(205.52 x 50.5) = R_E x 271
(10.379×10^3) = R_E x 271
R_E = 38.299 N

We know that,
$R_B + R_E$ = 735960
R_B + 38.299 = 735960
R_B = 735960 – 38.299
R_B = 735920 N

To plot Bending Moment Diagram (BMD)
BM_A = 735750 x 397 = 292090000 Nmm
BM_B = 0
BM_C = 205.52 x 220.5 = 45317 Nmm
BM_D = 0
BM_E = 0

FBD in SI Units

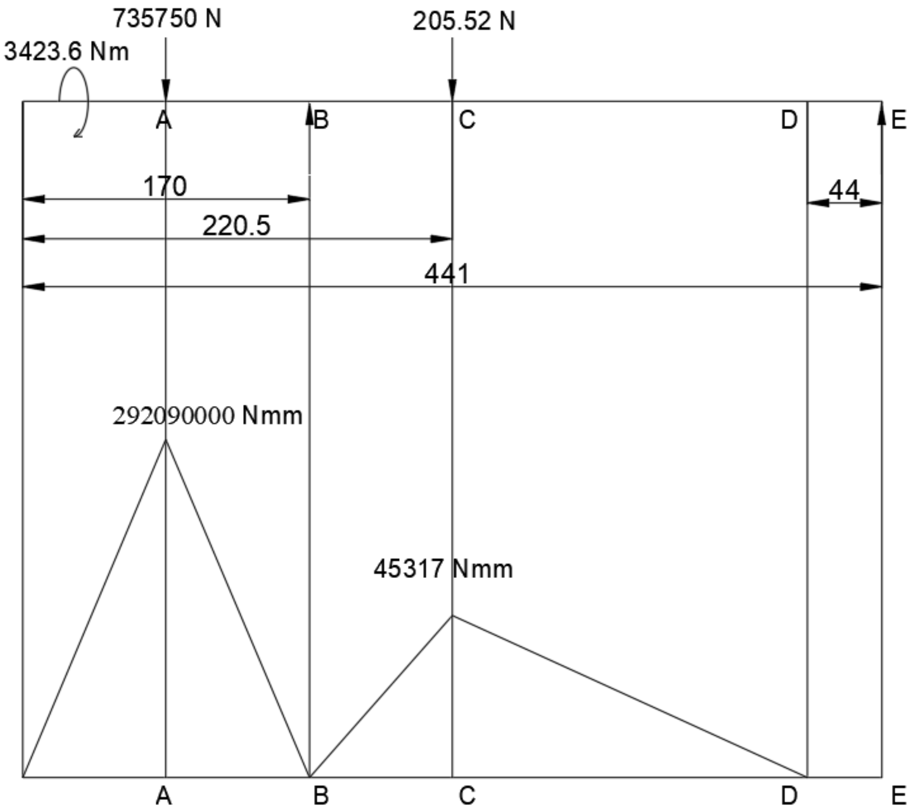

Fig. 3 Force Bending Diagram of shaft

From BMD

BM_{max} = 292090000 Nmm

To calculate equivalent Torque (T_e)

$$T_e = \sqrt{(k_b M_b)^2 + (k_t T_t)^2}$$

M_t = 3423600 Nmm

M_b = 292090000 Nmm

For given application and loading condition

k_t = 1.0, k_b = 1.5

$$T_e = \sqrt{(1.0\ x\ 292090000\ )^2 + (1.5\ x\ 3423600)^2}$$

$$T_e = \sqrt{85317000\ x 10^9 + 26372\ x\ 10^9}$$

$$T_e = \sqrt{85343\ x 10^{12}}$$

$$T_e = 292140000\ Nmm$$

Using torsion theory:

$$\tau_{max} = \frac{16\ T_e}{\Pi d^3}$$

There is keyway on shaft

$$\tau_{max} = 0.75\ x\ \frac{16\ x\ 292140000}{\Pi 80^3}$$

$$\boldsymbol{\tau_{max} = 2179.5\ N/mm^2}$$

Induced

For given material: EN 8

But

S_{ut} = 550 MPa

S_{yt} = 280 MPa

Assuming FOS = 1.5

$$\tau_{allowable} = \frac{280}{1.5} = 186.67\ MPa$$

$\tau_{\max induced} \gg \tau_{allowable}$

Hence shaft is safe for given loading condition.

III. Analysis of Shaft by Finite Element Analysis (FEA) Approach [3]

Solid works static structural toolbox is used to build the model and apply the boundary condition and then view the results and compare it to the analytical results. Numerical model of the cantilever beam has been built using Solidworks static structural tool. The results of the maximum value of Von Mises stress and the maximum deflection values were compared swith the analytical solution. Figure 4 shows the finite element result of Von Mises equivalent stress distribution through the beam length. The maximum value of Von Mises stress located at the support is **σ =111.6 MPa,** which is lower than the yield strength of the material of the shaft (EN 8) (138MPa). Figure 5 shows the finite element result of the shaft deflection, with maximum value of **0.0607 mm** at the free end of the simply supported beam. Analytical and finite element results are listed in Table 1, where these results are compatible with each other.

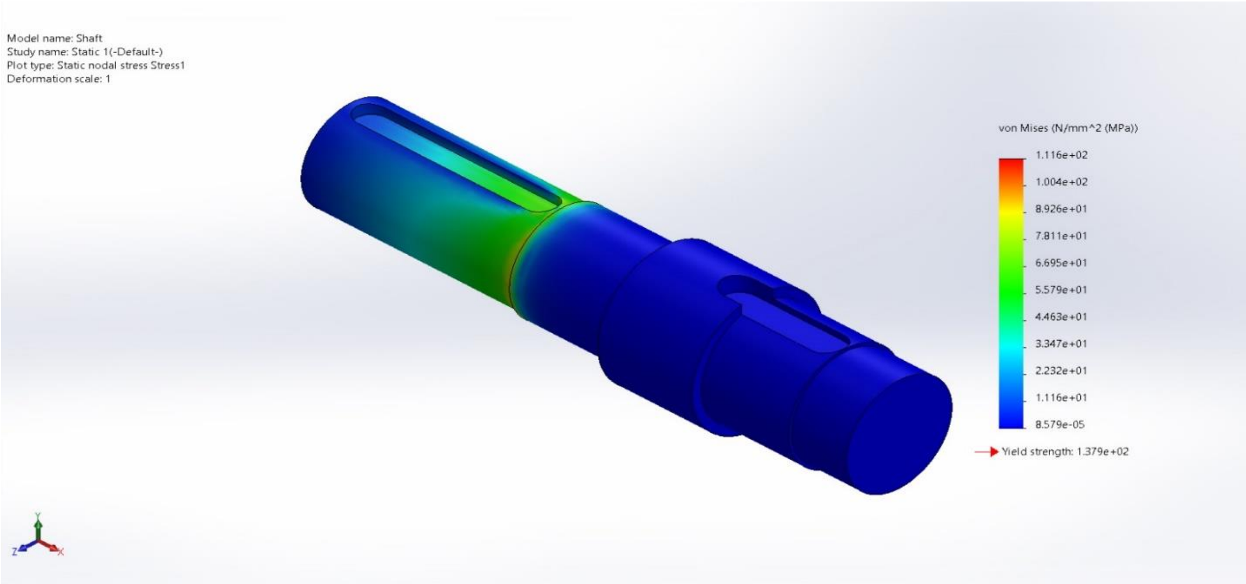

Fig. 4 Equivalent Stress Distribution

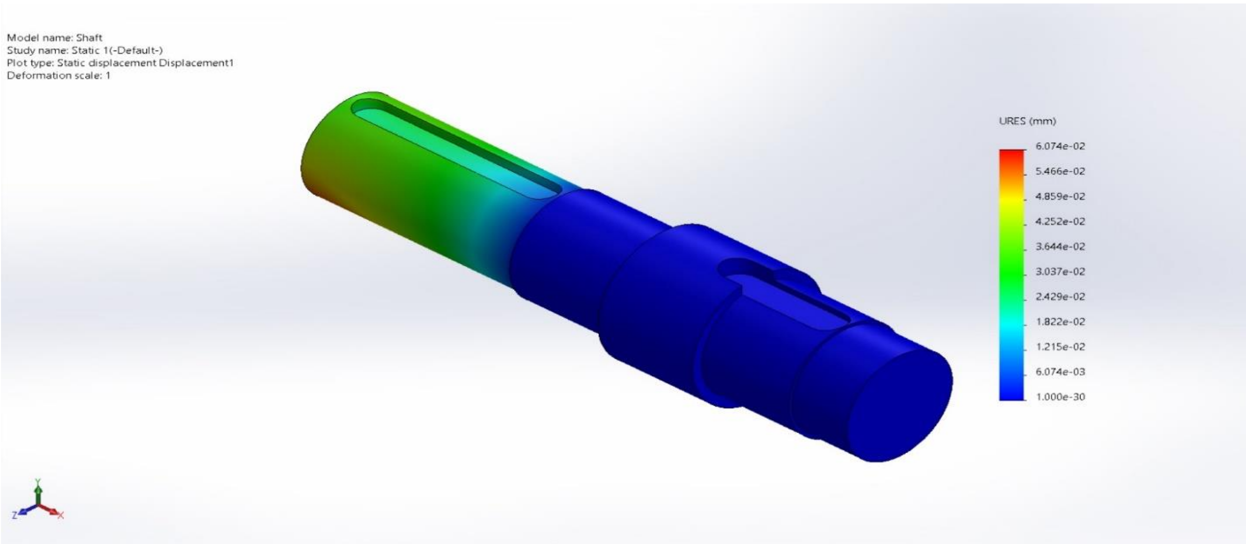

Fig. 6 Shaft Deflection

Table 1 Comparison theoretical and FEA results of shaft

	Analytical Result	FEA Result
Max Equivalent Stress	2179.5 MPa	111.6 MPa
Maximum Deflection	-	0.0607 mm

IV. Fatigue Analysis of Main Shaft

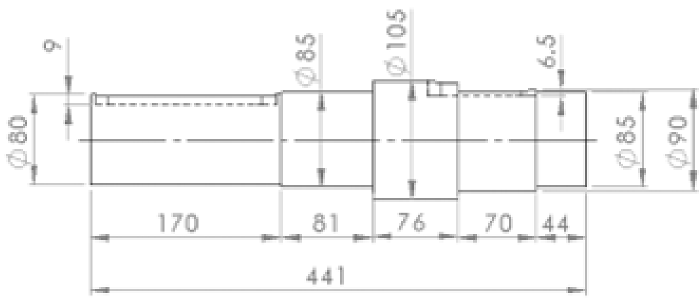

Fig. 7 Shaft Drawing

The calculation for the appropriate safety factor,
Data,
d=80mm
The material of shaft EN-8
Endurance strength (S_e) = 270 MPa
Ultimate tensile strength (S_{ut}) = 550 MPa
Yield strength (σ_y) = 465 MPa
The ASME elliptic is given by
Solid shaft data
The second moment of are

$$I = \frac{\pi\, d^4}{64}$$
$$I = \frac{\pi\, 80^4}{64}$$

$\boldsymbol{I = 2.0106\ X\ 10^6}$**mm⁴**

The polar second moment of area (mm^4)

$$J = \frac{\pi\, d^4}{32}$$
$$J = \frac{\pi\, 80^4}{32}$$

$\boldsymbol{J = 4.0212\ X\ 10^6}$**mm⁴**

c=distance from the neutral axis

$$c = \frac{d}{2} = \frac{80}{2}$$

$\boldsymbol{c = 40}$**mm**

From the static analysis result, we get the following data
M_a = Alternating bending moment = 292090000 Nmm
T_a = Alternating twisting moment or Torque = 3423600 Nmm
As per the ASME design code for the design of transmission shafting produced an approach but was suspended in 1995 by the ASME as improved understanding and more sophisticated methodologies have become available allowing more present modeling.

Diameter for solid shaft

$$d = \left[\frac{32\, n_s}{\pi}\sqrt{(\frac{M}{\sigma_e})^2 + \frac{3}{4}(\frac{T}{\sigma_y})^2}\right]^{1/3}$$

As per design procedure select appropriate factor of safety
Consider,
n_s = 3 to 4 for well-known material under uncertain condition load, stress, and environment

$$\boldsymbol{n_s = 4}$$

$$d = \left[\frac{32\ X\ 4}{\pi}\sqrt{(\frac{292090000}{270})^2 + \frac{3}{4}(\frac{3423600}{465})^2}\right]^{1/3}$$
$$d = \left[40.76\ X\sqrt{(1.1703\ X\ 10^{12}) + (40.656\ X\ 10^6)}\right]^{1/3}$$
$$d = [40.76\ X\ 1.0818\ X\ 10^6\ ]^{1/3}$$
$$d = [44094\ X\ 10^3\ ]^{1/3}$$
$$d = 353.29$$

$\boldsymbol{d = 353}$ **mm**

But practically takes **d = 80 mm**

The endurance limit σ_e

Mischke (1987) had determined the relation between the endurance limit of test specimens and the ultimate tensile strength of the material.

$\sigma_e' = 0.504\ \sigma_{uts}$ for $\sigma_{uts} < 1400\ MPa$

$$\sigma_e' = 0.504\ X\ 550$$

$\boldsymbol{\sigma_e' = 277.2}$**MPa**

The surface finish factor is given by the equation

$$K_a = a\ \sigma_{uts}^{b}$$

Consider machining or cold drawing or ground go it.

$$\boldsymbol{a = 1.58\ MPa}$$
$$\boldsymbol{b = -0.085}$$
$$K_a = 1.58\ X\ 550^{-0.085}$$
$$\boldsymbol{K_a = 0.924}$$

Size factor (K_b)

For, d > 50mm, $K_b = 1.85\ d^{-0.1133}$

$$K_b = 1.85\ (80)^{-0.1133}$$
$$\boldsymbol{K_b = 1.1260}$$

The reliability factor (K_c)

K_c value is taken from Table

Consider shaft nominal reliability of 0.5

So,

$$\boldsymbol{K_c = 1.0}$$

The temperature factor (K_d)

As consideration

$$\boldsymbol{K_d = 1}$$
$$K_f = 1 + q\ (K_t - 1)$$

or

$$K_{fs} = 1 + q_{shear}\ (K_{ts} - 1)$$

Where, as design procedure,

$$\boldsymbol{K_t = 2.14}$$
$$\boldsymbol{K_{ts} = 3}$$

For notch sensitivity

$\boldsymbol{K_f = K_t}$ and $\boldsymbol{K_{fs} = K_{ts}}$

Miscellaneous factor K_g

$$\boldsymbol{K_9 = 1}$$

Required,

$$\sigma_e = K_a\ K_b\ K_c\ K_d\ K_e\ K_g\ \sigma_e'$$
$$\sigma_e = 0.924\ X\ 1.1260\ X\ 1\ X\ 1\ X\ 1\ X\ 1\ X\ 277.2$$

$\boldsymbol{\sigma_e = 280.41}$**MPa > 270 MPa**

Which not in within the limit. Design is not safe.

Shaft will be fail due to fatigue.

V. CONCLUSION

The result shows theorical calculation of shaft with respect to standard experimental data, it was safe as per Structural analysis But Fatigue analysis, it was failed up to standard cycle time.

VI. ACKNOWLEDGMENT

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REFERENCES

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[2] Somanth Kolgiri, "Static and Dynamic analysis for Rotor Shaft of Electric Motor", Master of Engineering Thesis.

[3] Solidworks 2018 CAD tool for Simulation of shaft.