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Design and Development of Re-Coiler Trolley Driven Shaft

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Abstract: The paper is about Design and development of re-coiler trolley driven shaft. The main objective of modification is to increases the life of the shaft and for continuous production in the JSW Steel Coated Pvt. Ltd. The re-coiler trolley shaft (continuous galvanizing line) use by the industry coil weight due to the shaft is break from its position where the load of coil more in order to eliminate this problem change in design of shaft or material of the shaft is needed the new model of shaft is developed in analytical and analysis is carried out using ANSYS14.0 by doing so the shaft capable to carry the weight of coil the analysis carried out in order to make sure that the stress developed within the block are within the permissible endurance limit.
Keywords: Fatigue life, torsion load shaft shoulder fillet parameter, diameter ratio, stress, ANSYS.

I. INTRODUCTION

A shaft is a rotating member usually of 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. 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.

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. The deflection and slope analyses cannot be made until the geometry of the entire shaft has been defined. Thus deflection is a function of the geometry everywhere, whereas the stress at a section of interest is a function of local geometry.

For this reason, shaft design allows a consideration of stress first. Then, after tentative values for the shaft dimensions have been established, the determination of the deflections and slopes can be made. 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. 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

II. PROPOSED WORK

A. Data Required

1) Shaft Stresses: Bending, torsion, and axial stresses may be present in both midrange and alternating components. Axial loads are usually comparatively very small at critical locations where bending and torsion dominate, for a rotating shaft with constant bending and torsion, the bending stresses completely reversed and the torsion is steady. The fluctuating stresses due to bending and torsion process for fatigue is highly dependent on stress concentrations. Stress concentrations for shoulders and keyways are dependent on size the stress analysis specifications that are not known the first time through the process. Fortunately, since these elements are usually of standard proportions, it is possible to estimate the stress concentration factors for initial design of the shaf
2) *Deflection Considerations:* Deflection analysis at even a single point of interest requires complete geometry information for the entire shaft. For this reason, it is desirable to design the dimensions at critical locations to handle the stresses, and fill in reasonable estimates for all other dimensions, before performing a deflection analysis. Deflection of the shaft, both linear and angular, should be checked at gears and bearings. Allowable deflections will depend on many factors, and bearing and gear catalogs should be used for guidance on allowable misalignment for specific bearings and gears. For shafts, where the deflections may be sought at a number of different points, integration using either singularity functions or numerical integration is practice

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3) *Overview Of Fatigue:* The majority of component designs involve parts subjected to fluctuating or cyclic loads. Such loading induces fluctuating or cyclic stresses that often result in failure by fatigue. About 95% of all structural failures occur through a fatigue mechanism. The damage done during the fatigue process is cumulative and generally unrecoverable, due to the following: it is nearly impossible to detect any progressive changes in material behaviour during the fatigue process, so failures often occur without warning. Periods of rest, with the fatigue stress removed, do not lead to any measurable healing or recovery. Fatigue, or metal fatigue, is the failure of a component as a result of cyclic stress. The failure occurs in three phases: crack initiation, crack propagation, and catastrophic overload failure. The duration of each of these three phases depends on many factors including fundamental raw material characteristics, magnitude and orientation of applied stresses, processing history, etc. Fatigue failures often result from applied stress levels significantly below those necessary to cause static failure.
4) *Endurance limit modifyied factor:*: It can be seen that the rotating beam specimen used in the laboratory to determine endurance limits or fatigue limits is prospered very carefully and tested under closely controlled conditions. It is unrealistic to expect the endurance limit of a mechanical element to match value obtained in the laboratory. Hence we modify the endurance limit of the material Seb by some modifying factors to obtain the endurance limit of a part Se.
5) *Factor for type of loading A:* The standard laboratory test is carried out under reversed bending loading For other type of load the modifying A is to be used.
6) *Reliability Factor Kg:* generally the machine are designed for a reliability of 50 percent to 99,99 percent depending on a application. Hence we used we proper value of Kr as per as requirement
7) *Temperature Factor K0:* The temperature all the machine properties of the material and that the existence of a static or mean load also induce creep in the material. Hence, if operating environment has a temp. greater than 70c the endurance limit modified by the factor k0
8) *Size Factor Kz:* the round routing beam gives endurance limit for a specimen usually having diameter of 7.5 or 12.5 mm. It turns out that the endurance limits of machine elements having a large size or a different shape, seldom approach the value found in the laboratory tests. This effect due to dimensions or shape is called the size of effect.
9) *Surface Finish factor Ks:* The surface of the rotating beam specimen is highly polished. Obviously most machine element do not have such a high quality finish. This modification factor depends upon the quality of the finish and the tensile strength of the material.
10) *Fatigue Stress Concentration Factor Kf:* Most mechanical parts have discontinuities which change the stress distribution. The stress concentration effect for static loading is taken care of by the theoretical stress concentration factor Kf as discussed

III. OBJECTIVE

A. To improve the fatigue life of driven shaft.
B. To find out failure parameter of driven shaft.
C. To design and develop the re-coiler trolley.
D. To increase the production rate.

IV. SHEAR FORCE AND BENDING MOMENT DIAGRAM

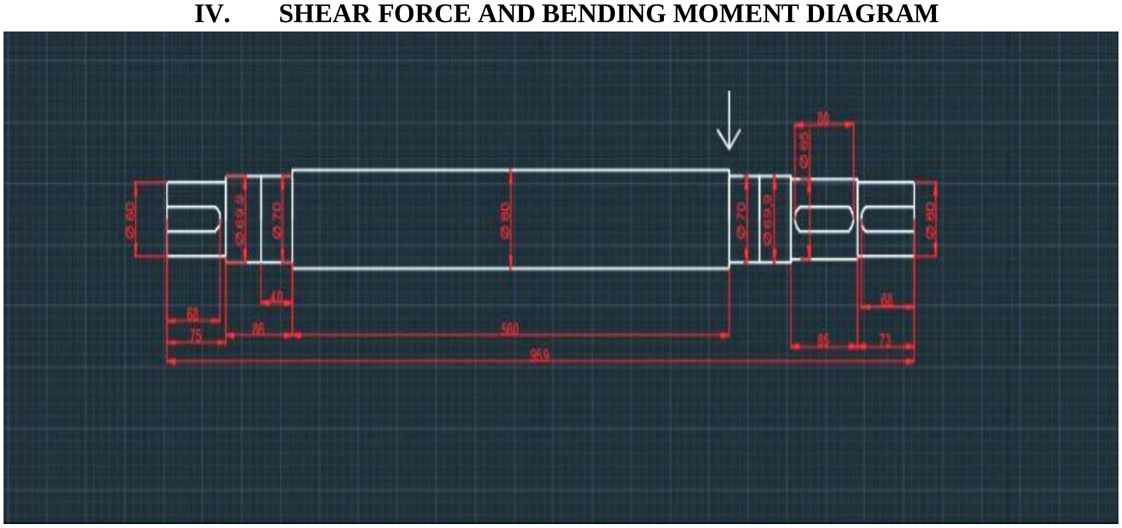

Fig.1: CAD Model of drive shaft

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V. DESIGN ANALYSIS

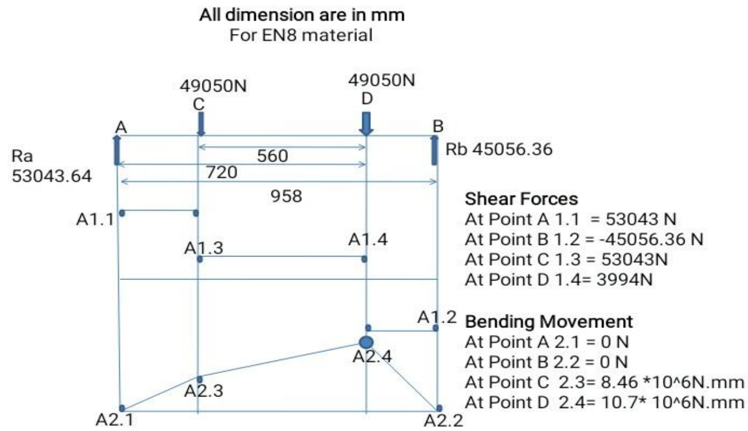


Fig 2: shear force and bending moment diagram

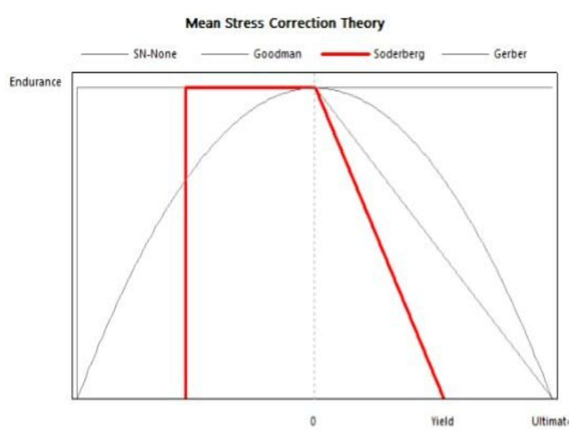
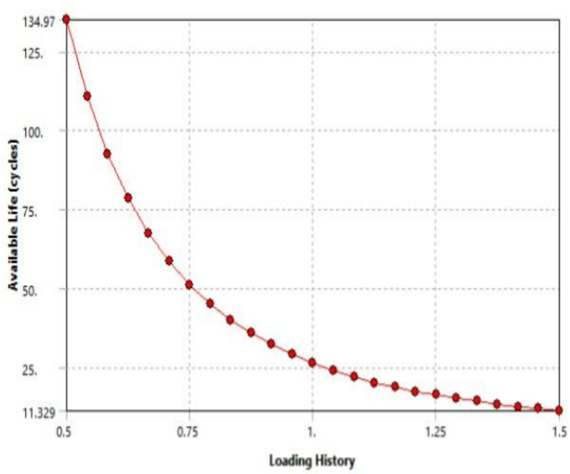


Fig 3: Mean Stress Correction Theory



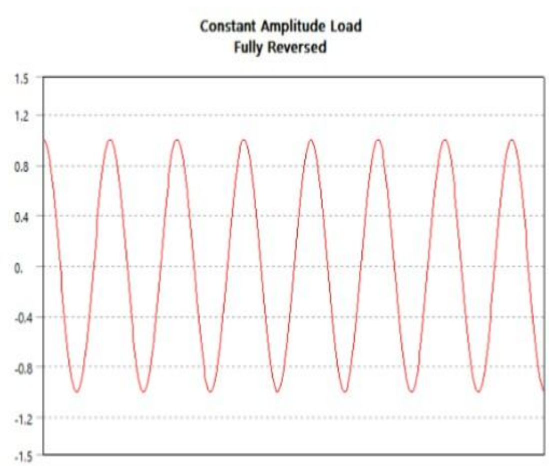


Fig 4: life cycle

fig 5: constant amplitude load for fully reverse

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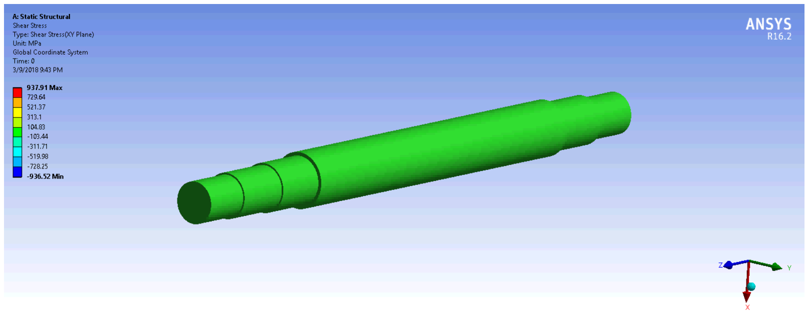


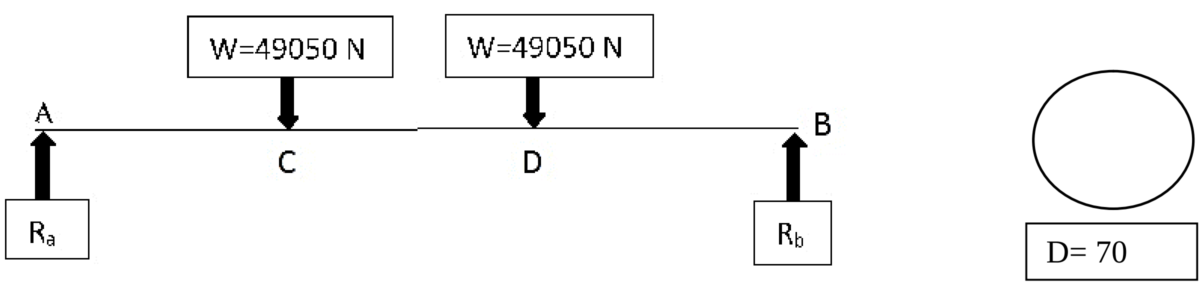
Fig 6. Shaft analysis of EN19
(SAE1080)

Properties Of Material

MATERIAL	EN8	EN42 J	EN19
Ultimate tensile strength	585Mpa	770 to 870Mpa	1300Mpa
Yield strength	210Mpa	460 to 520Mpa	1130Mpa
Endurance limit in reverse bending	260Mpa	435Mpa	647Mpa
Shear modulus	80Gpa	72Gpa	85Mpa
Young’s modulus	204Gpa	190GPa	206Mpa
Application	Axials, cranes, hooks seamless tube	Real axial, front axial	Transmission shaft

VI. CALCULATIONS

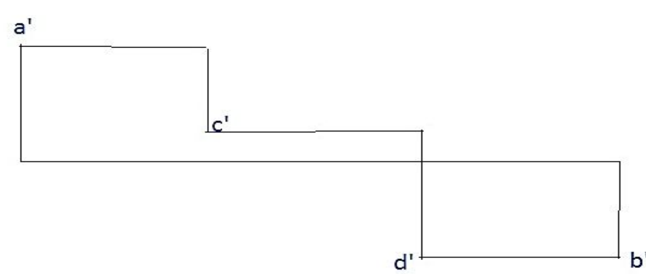
DESIGN OF TRANSMISSION SHAFT



Shear Force And Bending Moment

Reaction at each supported end
R_a = 53043.64 N
R_b =45056.36 N

At Point B = - 45056.36 N
At Point D = 3994 N
At Point C = 53043.64 N
At Point A = 53043.64 N



Bending Moment

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At Point A = 0 N.mm
At Point B = 0 N.mm
At Point C = 8.46×10^6 N.mm
At Point D =10.7×10^6 N.mm
Bending stress

$$\sigma_b = \frac{M}{Z}$$

$$\sigma_b = \frac{10.98\times10^6}{33673.94}$$

$$\sigma_b = 317.75\,\mathrm{N/mm^2}$$

Endurance Strength Limit

$$\sigma_e = k_{load}\times k_{size}\times k_{surface}\times k_{temp}\times k_{reliabilty}\times k_d\times k_{stress\ concentration}\times\sigma_{eb}$$

Where, $k_{load} = 1(\text{bending load})$

$$k_{size} = 1.24\,d^{-0.107} = 1.24\,80^{-0.107} = 0.78$$

$$k_{surface} = 4.51\,\sigma_{ut}^{-0.265} = 4.51\times 632^{-0.265} = 0.82$$

$$k_{temp} = 1 \le 450°C$$

$$k_{reliabilty} = 90\,\% = 0.90$$

$$k_d = \frac{1}{k_t} = \frac{1}{1+0.18(3-1)} = 0.73$$

$$k_{stress\ concentration} = 1.36$$

For Material EN8 (SAE=1040)

$$\sigma_e = 1\times0.78\times0.82\times1\times0.90\times0.73\times1.36\times260$$

$$\sigma_e = 148.58\ \mathrm{MPa}$$

Therefore $148.58\ \mathrm{MPa} \le 317.75\ \mathrm{MPa}$
Hence design unsafe

For Material EN42-j (SAE=1080)

$$\sigma_e = 1\times0.78\times0.82\times1\times0.90\times0.73\times1.36\times435$$

$$\sigma_e = 248.60\ \mathrm{MPa}$$

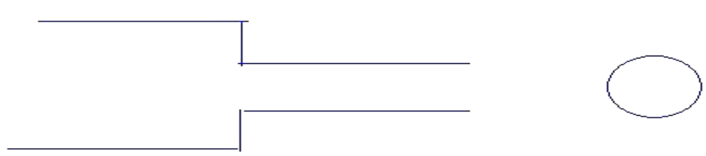
Therefore $248.60\mathrm{MPa} \le 317.75\ \mathrm{MPa}$
Hence design unsafe

For Material EN19 (SAE=4140)

$$\sigma_e = 1\times0.78\times0.82\times1\times0.90\times0.73\times1.36\times647$$

$$\sigma_e = 369.75\mathrm{MPa}$$

Therefore $369.75 \ge 317.75\ \mathrm{MPa}$
Hence design safe



D = 70 mm

Is subjected to both twisting and bending
Here, the shaft will be design based on twisting moment and bending moment
Since
Maximum bending moment on shaft = 10.7×10^6 N.mm

Torque on shaft = 504×10^3 N.mm

A. Shear Stress Due To Torque In N.Mm

$$\tau_{max} = \frac{T}{Z_p}$$

$$where\ T = 504 \times 10^3$$

$$Z_p = \frac{\pi D^3}{16}$$

$$Z_p = \frac{\pi \times 70^3}{16}$$

Z_p =67347.89 N.mm^3

$$\tau = \frac{504 \times 10^3}{67347.89}$$

$$\tau = 7.48\ N/mm^2$$

shear stress due to bending

$$\tau = \frac{4F}{3A}$$

$$= \frac{16 \times 49050}{3\pi \times 70^2}$$

$$\tau = 16.99\ N/mm^2$$

Now, Shear stress

$$\tau_{max} = \frac{1}{2} \times [\sqrt{\sigma_b{}^2 + 4(\tau)^2}]$$

$$\tau_{max} = \frac{1}{2} \times [\sqrt{317.75^2 + 4(7.48)^2}]$$

$$\tau_{max} = 150.05\ N/mm^2$$

Here,

1. Checking Diameter For torsional

$$T_e = \sqrt{M^2 + T^2}$$

$$T_e = \sqrt{(10.7 \times 10^5)^2 + (504 \times 10^3)^2}$$

$$T_e = 10.71 \times 10^6\ N.mm$$

MATERIAL	τ	With factor of safety =2	Without factor of safety	D (with f.s)	D (without f.s)	Below 60 mm safe
EN 08	210	$T_e = \frac{\pi}{16} \times \frac{\tau}{2} \times d^3$ $10.71 \times 10^6 = \frac{\pi}{16} \times \frac{210}{2} \times d^3$	$T_e = \frac{\pi}{16} \times \tau \times d^3$ $10.71 \times 10^6 = \frac{\pi}{16} \times 210 \times d^3$	D=80.39	D=63.60	Unsafe
EN42-J	480	$T_e = \frac{\pi}{16} \times \frac{\tau}{2} \times d^3$ $10.71 \times 10^6 = \frac{\pi}{16} \times \frac{480}{2} \times d^3$	$T_e = \frac{\pi}{16} \times \tau \times d^3$ $10.71 \times 10^6 = \frac{\pi}{16} \times 480 \times d^3$	D=61.02	D=48.43	safe
EN19	1130	$T_e = \frac{\pi}{16} \times \frac{\tau}{2} \times d^3$ $10.71 \times 10^6 = \frac{\pi}{16} \times \frac{1130}{2} \times d^3$	$T_e = \frac{\pi}{16} \times \tau \times d^3$ $10.71 \times 10^5 = \frac{\pi}{16} \times 1130 \times d^3$	D=45.87	D=36.41	safe

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B. Checking diameters due to bending.

$M_{eq} = \frac{1}{2} \times [M + \sqrt{M^2 + T^2}]$

$M_{eq} = \frac{1}{2} \times [10.70 \times 10^6 + \sqrt{(10.7 \times 10^6)^2 + (504 \times 10^3)^2}]$

$M_{eq} = 10.71 \times 10^6 \text{ N.m}$

For, EN8 MATERIAL

$$\sigma_b = \frac{M}{Z}$$

$EN8 \rightarrow 260 \text{ N/mm}^2$

$$\sigma_b = \frac{10.71 \times 10^6}{\frac{\pi}{32} \times 70^3}$$

$\sigma_b - 318.05 \text{ N/mm}^2$

Therefore, $318.05 \text{ N/mm}^2 \geq 210 \text{ N/mm}^2$

ence design is unsafe

For, EN42 J MATERIAL

$$\sigma_b = \frac{M}{Z}$$

$EN8 \rightarrow 260 \text{ N/mm}^2$

$$\sigma_b = \frac{10.71 \times 10^6}{\frac{\pi}{32} \times 70^3}$$

$\sigma_b - 318.05 \text{ N/mm}^2$

Therefore, $318.05 \text{N/mm}^2 \leq 435 \text{ N/mm}^2$

Hence design is safe

For, EN19 MATERIAL

$$\sigma_b = \frac{M}{Z}$$

$EN8 \rightarrow 260 \text{ N/mm}^2$

$$\sigma_b = \frac{10.71 \times 10^6}{\frac{\pi}{32} \times 70^3}$$

$\sigma_b - 318.05 \text{ N/mm}^2$

Therefore, $318.05 \text{ N/mm}^2 \leq 647 \text{ N/mm}^2$

Hence design is safe

VII. CONCLUSIONS

Sr .No	Parameter	Analytical (EN8)	Analytical (EN42-J)	Analytical (EN19)
01	Max stress in mpa	317.75 Mpa	480 Mpa	1130 Mpa
02	Endurance Limit	148 Mpa	248.60 Mpa	367.75 Mpa

Sr.No	Parameter	ANSYS (EN8)	ANSYS (EN19)
01	Max stress in Mpa	656.67 Mpa	1899.7 Mpa
02	Endurance Limit	163.94 Mpa	177.61 Mpa

By observing the causes of failure are material of the shaft. So, to overcome these failure, we used to predictive maintenance, change of material & design modification.
By using ANSYS software we analyse the stress, life cycle, endurance limit, on the basis of that we will made changes in and also change the material of shaft according to ANSYS report

VIII. ACKNOWLEDGMENT

A. Accomplishment of any work involves many people. We take this opportunity to express our appreciation and thanks to all the people who have contributed directly or indirectly to make this project a success.
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