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Failure Analysis and Fatigue Life Estimation of a Shaft of a Rotary Draw Bending Machine

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***Abstract*—**Human consumption of the Earth's resources increases the need for a sustainable development as an important ecological, social, and economic theme. Re-engineering of machine tools, in terms of design and failure analysis, is defined as steps performed on an obsolete machine to return it to a new machine with the warranty that matches the customer requirement. To understand the future fatigue behavior of the used machine components, it is important to investigate the possible causes of machine parts failure through design, surface, and material inspections. In this study, the failure modes of the shaft of the rotary draw bending machine are inspected. Furthermore, stress and deflection analysis of the shaft subjected to combined torsion and bending loads are carried out by an analytical method and compared with a finite element analysis method. The theoretical fatigue strength, correction factors, and fatigue life sustained by the shaft before damaged are estimated by creating a stress-cycle (S-N) diagram. In conclusion, it is seen that the shaft can work in the second life, but it needs some surface treatments to increase the reliability and fatigue life.

***Keywords*—**Failure analysis, fatigue life, FEM analysis, shaft, stress analysis.

I. Introduction

ROTERY draw bending (RDB) machines are utilized in the tubes forming process. Tubes can be used as pipes in factories, structural parts in aircraft and cars, or design elements for furniture. Some manufacturers of RDB machines produce new machines. In addition, they use re-engineering, remanufacturing, retrofit or rebuild as a technique to convert an outdated machine into a new and valuable machine for the second life utilization. The RDB machine consists of several main systems and main components. In this project, the shaft in swing arm and bend head system is analyzed as a case to study.

A shaft, which has a circular cross-section, is a rotating machine element used in mechanical equipment and machines to transmit rotary motion and power. Bearings, flywheels, gears, clutches, and other machine elements are usually mounted on the shaft, and help in the power transmission process. Generally, shafts do not have a uniform diameter, and they are stepped with shoulders where bearings, gears, or other components are mounted. In the shaft design analysis, it is important to ensure that the shaft geometry will satisfy the material strength requirements and shaft-supported element requirements. Stress analysis at specific points depends on the local geometry, while slope and deflection analysis relies on the overall dimensions of the shaft [1]-[4]. Stresses concentrate in shaft shoulders and key ways and depend on the local dimensions [2]. Shafts mostly work under the influence of fluctuated loads or combined torsion and bending loads. If a shaft supports a static load, the bending stresses are fully reversed and the torsion is steady [1], [5]. Even well-designed shafts, when subjected to repeated loads or overloads, will suffer from various kinds of mechanical failures.

Failure analysis is an indispensable tool that is used widely by industry sector to develop or improve the product design. The failures of machine elements are studied extensively by scientists to find methods in order to identify their causes and to prevent them from reoccurring. To determine the failure modes, analytical, experimental, and finite element analysis methods can be used [6]. Failure cause analysis requires complete information about the component geometry, material, load condition, work environment, and work constraints [3], [7].

Generally, shafts suffer from a transverse deflection as a beam and a torsion deflection as a torsion bar [5]. In addition, surface failure is a common failure mode of shafts [8]. Fatigue failure, due to recurrent load or overload, stress concentration, insufficient clearance and wrong bearing arrangement, is also possible to happen [3]. The main objective of this research is the failure analysis of a used shaft of an outdated rotary draw bending machine in order to prevent the occurrence of the failure in the second life utilization and to improve the performance of the machine.

II. Methodology

Shaft failure analysis and fatigue life estimation consisted of four steps. Firstly, material analysis and hardness tests were performed to identify the kind and strength of the material that the shaft made of. Brinell hardness tests were done on the shaft surface by using Wolpert tester under a load of 1830 N, with a 2.5 mm ball indenter. Material analysis test was carried out by using Spectro technology to identify the chemical composition of the shaft material. Secondly, an optical examination and surface roughness measurements on the shaft surface were conducted. With the optical examination by using an optical microscope, not only the surface condition, but some manufacturing data was identified. Roughness analysis was done, using a Mitutoyo measuring instrument to measure the current roughness and compare it with the design

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values. Thirdly, loads, stresses, maximum linear deflection, and maximum angular deflection were calculated and compared with the values obtained from the FEA method by using ABAQUS software. Finally, fatigue life estimation by creating a calculated stress-cycle diagram was made. At the end of this research, some surface treatments are recommended to ensure the reliable performance of the shaft in the second life cycle.

III. Inspections

All the following inspections were performed by certified operators and tools, with referred to the best practices or suitable guidelines. Hardness test, optical examination, and surface roughness measurement were carried out in the laboratories of the Mechanical Engineering Department at the University of Siegen in Germany. Moreover, material analysis test by using Spectro technology was conducted in (SPECTRO Analytical Instruments GmbH) company in Germany.

A. Material Analysis

This investigation was carried out in order to identify the chemical composition of the material that the shaft is made of and to compare it to the standard metals database. The shaft material, based on the material analysis test, is a C15Pb Lead–alloyed case- hardening steel used for structural, machine elements, toothed wheels, joints, bushings, etc. Table I shows the nominal chemical composition of C15Pb steel, and the mechanical properties of the shaft material are reported in Table II.

TABLE I
Chemical Composition of C 15 Pb Steel

C%	Si%	Mn%	P%	S%	Pb%
0.12–0.18	≤0.40	0.30–0.80	≤0.045	≤0.045	0.15–0.30

TABLE II
Mechanical Properties of C 15 Pb Steel

Tensile strength Mpa	Yield strength Mpa	Brinell Hardness (HB)	Poisson's ratio	Rigidity's modulus GPa	Mass density Mg/m^3	Young's modulus MPa
500	385	143	0.28	80	7.8	200000

TABLE III
Results of Surface Roughness Measurement in μM

Generatrix	Ra Needle bearing	Ra Ball bearing	Ra Chain	Rz Needle bearing	Rz Ball bearing	Rz Chain
0^0	0.133	0.174	1.0495	4.639	1.730	16.246
60^0	0.105	0.194	1.6595	3.388	4.744	14.337
120^0	0.073	0.315	2.156	**1.346**	8.965	25.646
180^0	0.095	0.238	4.690	2.861	3.841	**41.653**
240^0	0.130	0.157	1.062	3.527	2.422	20.162
300^0	0.134	0.187	0.812	1.922	3.753	10.090

B. Hardness Test

To evaluate the current surface hardness and to compare it with the required surface hardness, Brinell hardness tests were made on the shaft surface. The test results confirm the results of the chemical composition analysis by Spectro technology and prove that the shaft material has a hardness of about 140-150 HB and an ultimate strength of about 500 MPa. Furthermore, the shaft surface, based on the hardness measurement results, needs a re-carburizing process to increase the surface hardness, wear resistance, fatigue strength and tensile strength, in order to match the required surface hardness of about 610 - 726 HB.

C. Optical Examination

Optical examination by using an optical microscope was carried out in order to specify what finishing process was used. A unique texture on the shaft surface is left by each finishing process, based on the used parameters such as tool type and material, feed rate and speed, cutting direction, machine power, etc. [9]. However, analogous surface finishing processes leave same texture with same roughness. As a result of this examination, the shaft surface was produced by turning process with typical roughness (Ra) of about $0.40 < Ra < 6.30$ μm. Moreover, the shaft shoulders where the bearings sit, according to the bearing manufacturer, have a specific roughness (Ra) which has to be ≤ 0.2 μm, meanings that these shoulders were produced by using a specific turning process with a specific roughness value.

D. Surface Roughness Measurement

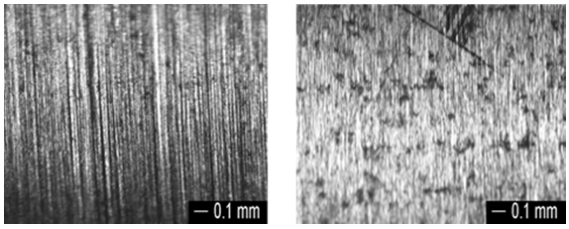

Fig. 1 Optical examination images

Surface roughness is a part of surface characteristics. It refers to the deviation of the measured surface from the ideal surface. In addition, it can be a root that initiates fatigue cracks [8]. Roughness measurements are necessary to know how a real surface will react to real working conditions. The shaft surface roughness was measured at the shaft shoulders where the bearings and chain sit, along six generatrices along the axis. The roughness value (Ra) on the chain location was

uniform, but on the ball bearings location was above the design value. On the other hand, the Rz value of the chain location refers to a high local roughness that indicates to a local surface pitting. The results are reported in Table III.

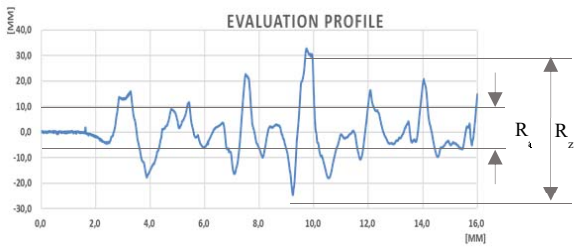

Fig. 2 Ra and Rz roughness values on the chain shoulder

IV. Analytical Calculations

For the strength assessment of the shaft material, the stresses have been calculated based on the ASME method. This standard is known as B106.1M-1985 for a machine shaft design, which assumes that the shafts commonly work under a fully reversed bending and a steady torque. Even if this method is an approximate approach, it is still a sufficient estimation to check the overall shaft situation [5]. In this study, the shaft connected to a hydraulic cylinder by a chain, and the maximum hydraulic cylinder force (F_B) is 25784.6 N. The diameter of the shaft shoulder (D) where the chain is mounted is 64.8 mm. The shaft torque (T) was calculated based on (1)

$$T = F_B \times \frac{D}{2} \tag{1}$$

In the forces analysis step, five points on the shaft are considered. These points are the needle bearing position (A), The chain position (B), the first ball bearing position (C), the second ball bearing position (D), and the key position (E). The reaction force (F) and the moment (M) at each point were calculated according to (2) and (3). The calculated results are reported in Fig. 3.

$$\Sigma F = 0 \tag{2}$$

$$\Sigma M = 0 \tag{3}$$

Von Mises stresses under combined bending and torsion loads were calculated according to (4), whereas under pure torsion loads were calculated according to (5).

$$\sigma_d = \sqrt{\left(\frac{32\,K_{fb}M_b}{\pi d^3}\right)^2 + 3\left(\frac{16T_m}{\pi d^3}\right)^2} \tag{4}$$

$$\sigma_d = \sqrt{3(\tau_{max})^2} \tag{5}$$

where $\sigma_d, M_b, K_{fb}, d,\ T_m\ and\ \tau_{max}$ are the von Mises stress, bending moment, fatigue stress concentration factor, shaft diameter, steady torsional moment, and maximum shear stress, respectively. K_{fb} is giving by (6)

$$K_{fb} = 1 + q(K_t - 1) \tag{6}$$

where K_t is the geometric stress concentration factor and q is the notch sensitivity factor given as:

$$q = 1 \div (1 + \sqrt{a}/\sqrt{r}) \tag{7}$$

The linear deflection and slope were calculated according to (8)-(10) by using numerical integration technique. Where M is the applied moment, E is the modulus of elasticity, and I is the cross section's area moment of inertia. M and I are functions of the shaft geometry at each point.

$$Moment\ function = \frac{M}{EI} \tag{8}$$

$$Slop(\theta) = \int \frac{M}{EI} d_x + C \tag{9}$$

$$Deflection(\delta) = \iint \frac{M}{EI} d_x + D + G \tag{10}$$

The angular deflection was calculated based on (11), where L is the shaft length, G is the shear modulus, J is the polar moment of inertia, and T is the torque.

$$\theta = \frac{TL}{GJ} \tag{11}$$

The results of the analytical approach are reported in Table IV.

V. Fatigue Failure Criteria

To estimate the fatigue failure criteria of the shaft material, the theoretical fatigue endurance limit $S_e^{'}$ was calculated as (12). S_{ut} is the ultimate tensile strength of the material.

$$S_e^{'} = 0.5 \times S_{ut} \tag{12}$$
$$S_e^{'} = 0.5 \times 500 = 250\ \text{MPa}$$

To determine the corrected fatigue endurance limit S_e, the correction factors to the theoretical endurance limit should be considered.

$$S_e = C_{load} \times C_{size} \times C_{surf} \times C_{temp} \times C_{reliability} \times S_e^{'} \tag{13}$$

$$C_{load} = 1\ (Bending\ case)$$
$$C_{size} = 1.189\ d^{-0.097}\ for\ 8mm < d \leq 250\ mm = 0.83$$
$$C_{surf} = 4.51 \times {S_{ut}}^{-0.265} = 0.84$$
$$C_{temp} = 1\ for\ T \leq 450\ C^0$$
$$C_{reliability} = 0.897$$
$$S_e = 156.34\ MPa$$

To estimate the fatigue life (N) of the shaft and for further fatigue behavior prediction, the stress-cycle (S-N) diagram was constructed. Tow strength values should be determined in

order to draw the S-N diagram. First, the material strength (S_m) at $N \leq 10^3$, and second, the material strength (S_e) at $N \geq 10^6$. The material strength at $N \leq 10^3$ is calculated according to (14), and the material strength at $N \geq 10^6$ is the corrected fatigue endurance limit calculated previously.

$$S_m = 0.9\, S_{ut}\ for\ bending \qquad (14)$$
$$S_m = 450\ MPa$$

The estimated S-N diagram is shown in Fig. 4. By using the typical industrial design factor (3), the shaft works under a maximum stress of about 357 MPa and for this stress, the shaft fatigue life is 4.524×10^3 cycles.

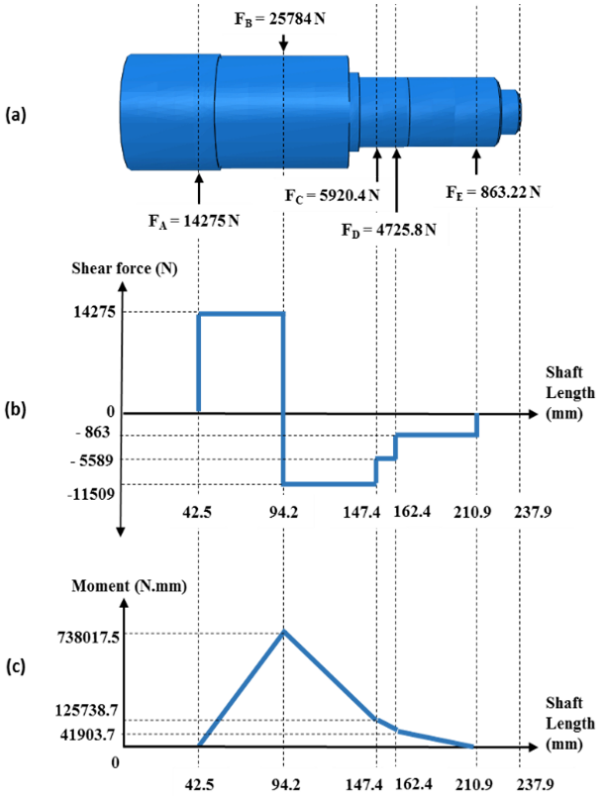

Fig. 3 (a) Free body diagram, (b) shear force diagram, (c) bending moment diagram

VI. Finite Element Method (FEM) Using ABAQUS Software

Finite element method (FEM) is a computational and powerful technique used to solve engineering problems. It is a method to predict how a physical object reacts to real boundary conditions such as torque, force, or vibration, to understand whether it will work safely or break. Mesh generation techniques are used in this method to divide a complex physical object into a net of small size elements by using a software program. It is an approach to automate the mechanical design by using modeling technology [1], [3], [4].

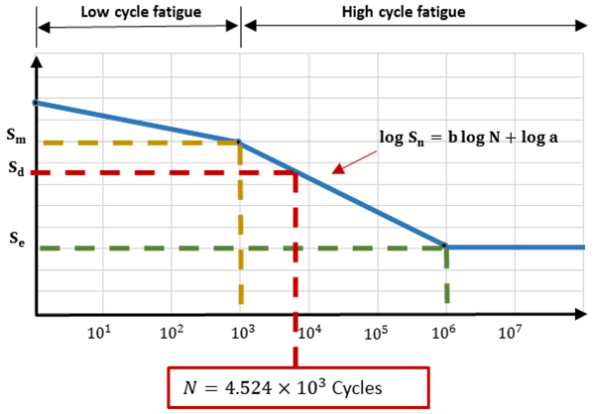

Fig. 4 Calculated stress-cycle (s-n) diagram

A. Shaft Modelling

To check the calculated stresses values and to examine the distribution of the stress, the most popular commercial software ABAQUS is employed for modeling the shaft. Boundary conditions under a maximum loading condition were analyzed, and Hex (C3D8R) elements were generated. At the shaft end where the key is mounted, six degrees of freedom (DoF) were constrained. The torque of about 835421 N-mm is applied on the shaft shoulder where the chain is located, by using a master node of a couple kinematic. Fig. 5 shows the CAD Model, mesh model, constraints and load condition.

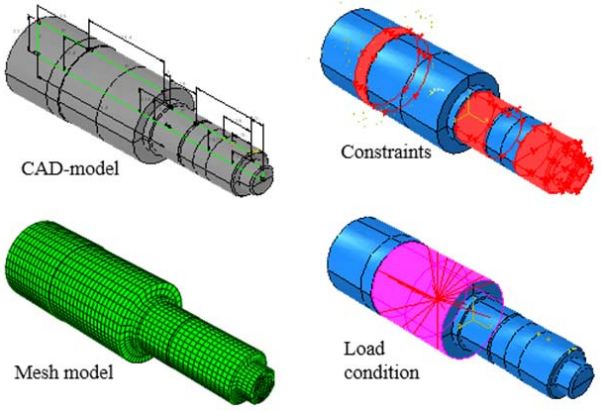

Fig. 5 Shaft modeling

B. Shaft Equivalent Stress

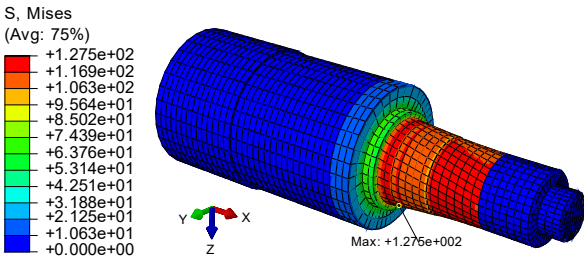

Fig. 6 Shaft equivalent stresses (von Mises)

The maximum value of the equivalent stress is 127.5 MPa, which takes place in the cross-section of the shaft where the ball bearing was located, as shown in Fig. 6.

C. Shaft Shear Stresses

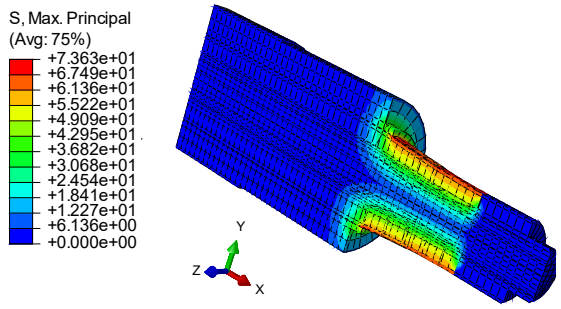

Fig. 7 Shaft shear stresses

The maximum shear stress is shown in Fig. 7, which appears in the cross-section of the shaft between the chain location, where the torque enters to the shaft, and the key location, where the torque goes out from the shaft.

D. Shaft Bending Deflection

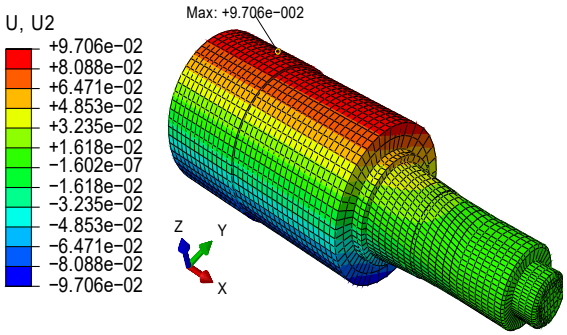

Fig. 8 Shaft bending deflection

The maximum value of the bending deflection is 0.09 mm, which is lower than the admissible linear deflection, at the cross-section of the shaft where a needle bearing is mounted.

E. Shaft Torsional Deflection

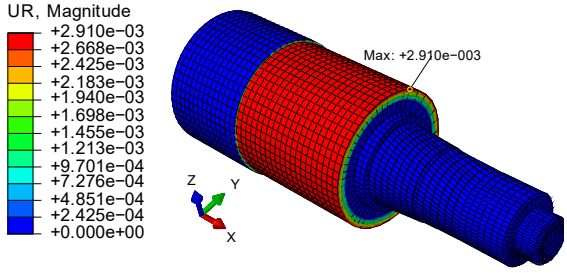

Fig. 9 Shaft torsional deflection

The maximum torsional deflection is 0.00291 rad, which is well below the permissible value of the maximum angular deflection.

The comparison results between the analytical approach and FEA approach are reported in Table IV.

TABLE IV
COMPARISON OF FEA RESULTS WITH ANALYTICAL RESULTS

Design specifications	Analytical model	FEA model
Max. equivalent stress	119.572 MPa	127.5 MPa
Max. bending deflection	0.05 mm	0.09mm
Max. shear stress	69.0357 MPa	73.63 MPa
Max. torsional deflection	0.175 deg	0.166 deg

VII. CONCLUSIONS

Shaft failure modes are analyzed in detail. Spectro analysis test is performed to determine the chemical composition and mechanical properties of the shaft material and it is found that the shaft material is C 15 Pb. From the surface roughness and hardness measurements, it can be deduced that the surface failure occurred due to the friction and wear on the mating surfaces. Forces, torques, and stresses are calculated by using an analytical approach and ABAQUS software. Both methods show that the stresses and deflections are nearly same and in the admissible range. The fatigue failure criteria are analyzed, the fatigue endurance limit was calculated, and the fatigue life of the shaft was estimated. In conclusion and based on the hardness and roughness inspection results, it is found that the shaft can be reused after applying a re-carburizing and refining processes on the surface to match the required surface hardness and roughness. Moreover, according to the analysis results, the shaft has a fatigue life of about 4.524×10^3 cycles.

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