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Design &Development of Triplex Pump Crankshaft Assembly – Core Shaft

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Abstract: *In this paper, study of a crankshaft employed in a reciprocating pumping of Oil & Gas sector called mud pump, is done for an offshore model. Mud pumps are used to circulate drilling fluid and mud under rated pressures. It has well service and oil production applications. The pump rotated at speeds of 120 to 150 rpm is taken into consideration. Considering the overall pump system; forces, moment and torque acting on a shaft is used to calculate the stresses induced on it. Stress analysis also carried out by using FEA and the results are compared with the calculated analytical values. Shaft is having varying cross sections due to this stress concentration is occurred at the stepped areas, keyways, shoulders, sharp corners etc. that causes fatigue failure of shaft. So, we have calculated stress concentration factor from which fatigue stress concentration factor is derived. Endurance limit using Modified Goodman Method, fatigue factor of safety and theoretical number of cycles sustained by the shaft before failure is estimated and is compared with results using FEA. The life of bearings is calculated as an empirical step to check on the first component failure in the entire system of the pump and as the life of the least durable part. The plausible solution to combat the fatigue failures in shafts is also discussed in this document.*

Keywords: *Crankshaft, Pump, Factor of Safety, Static, Fatigue, modified Goodman, Stress Concentration, FEA.*

I. Introduction

A crankshaft is a rotating member, usually of circular cross section, used to transmit power or motion in any mechanical system, say pumps. It provides the axis of rotation, or oscillation, of elements such as gears, pulleys, flywheels, cranks, sprockets, and the like and controls the geometry of their motion and is supported by bearings.A shaft is subjected to torsion and bending in combination. Generally shafts are not of uniform diameter but are stepped, keyways, sharp corners etc. The stress on the shaft at a particular point varies with rotation of shaft there by introducing fatigue. Even a perfect component when repeatedly subjected to loads of sufficient magnitude, will eventually propagate a fatigue crack in some highly stressed region, normally at the surface, until final fracture occurs.

There is really nothing unique about a shaft that requires any special treatment beyond the basic methods that is already developed. However, because of the ubiquity of the shaft in so many machine design applications, there is some advantage in giving the shaft and its design a closer inspection.

II. Literature Review

For study purposes of likewise cases, we have considered the crankshaft of a small well service pump that could fail in service. An investigation to be performed in order to determine the failure root cause and contribution factors. Investigation methods included visual examination, optical and scanning electron microscope analysis, chemical analysis of the material and mechanical tests. A finite element analysis was also performed to quantify the stress distribution in the shaft. It was concluded that the shaft failed due to fatigue and that the failure was caused by improper reconditioning of the shaft during routine overhaul.

The solution found was that the shaft failed as a result of fatigue. The cyclic load leading to fatigue failure was caused by the weight of the gearbox and motor being carried (partially) by the conveyor pulley shaft.

Fatigue failure is highly unlikely to have occurred without the contribution of the following two factors; one, an extremely sharp corner was machined at the shaft shoulder where its diameter changes and two, weld restoration of the shaft external surfaces caused a heat affected zone in the sharp corner at the shaft shoulder.

The mechanical design and material selection of the shaft is appropriate for its intended service. Failure can be attributed to improper repair/reconditioning of the shaft.

There are various means to increase the life of the crankshaft i.e.

A] Optimizing Geometry by a 4” THROUGH HOLE through the solid crankshaft with following purpose: -

a) Weight reduction b) Stress relief c) Fatigue relief; B] Stepped shaft corner radius INCREASED to delay crack growth initiation; C] Pre-stressing the entire shaft; D] Surface Treatments by Polishing (removes machining flaws etc.); E] Introducing compressive stresses (compensate for applied tensile stresses) into thin surface layer by “Shot Peening”-firing small shot into surface to be treated. Ion implantation, laser peening; F] Case Hardening -create

C-or N-rich outer layer in steels by atomic diffusion from the surface. Makes harder outer layer and also introduces compressive stresses.

III. Analysis

The positive displacement pump generates very specific dynamic liquid flow and pressure conditions, but the response of the system to the pump dynamics is a result of the system operating conditions and specific piping arrangements.

There are three steady-state pressure conditions that affect system liquid dynamic responses: friction-generated pressure from the average liquid flow, vertical head from a column of liquid and pressure disturbances generated by the positive displacement pump: frictional pressure drop, acceleration from the liquid flow variation of the pump. And low-amplitude water hammer-type pressure disturbances that occur each time a pump valve opens or closes. These factors have direct and indirect impact on the crankshaft.

Currently the case revolves around the EH-1600 pump crankshaft, with following geometrical and material specifications to be used for validation.

r := 7 in = 177.8 mm …………………………………. Offset of connecting rod lobe on crankshaft
P1 := 50 psi = 344.74 kPa ………………………..…... Suction pressure
P2 := 5000 psi = 34.47 MPa …………………………. Discharge pressure
L1 := 45.0 in = 1143 mm …………………………….. Length of connecting Rod
D_p := 5.5in = 139.7 mm ……………………………… Diameter of Liner
a := 9.06in = 230.12 mm…………………..……… Locations of Main Bearing 1
b := 18in = 457.2 mm…………………...………… Locations of Eccentric 1
c := 12.69in = 322.33 mm………………...………… Locations of Eccentric 2
d := 9.06in = 230.12 mm………………...…..……… Locations of Eccentric 3
e := 12.81in = 325.37 mm……………………..…..… Locations of Main Bearing 2
C_d := 13.75in = 349.25 mm ………………………….. Diameter of Crank Shaft
SY := 68000psi = 468.84 MPa ………………………..Ultimate tensile strength of material
θ := 0 , 10 …. 360 …………………………………… Crankshaft angle

The CAD and the geometry of the EH-1600 triplex pump crankshaft is as follows:

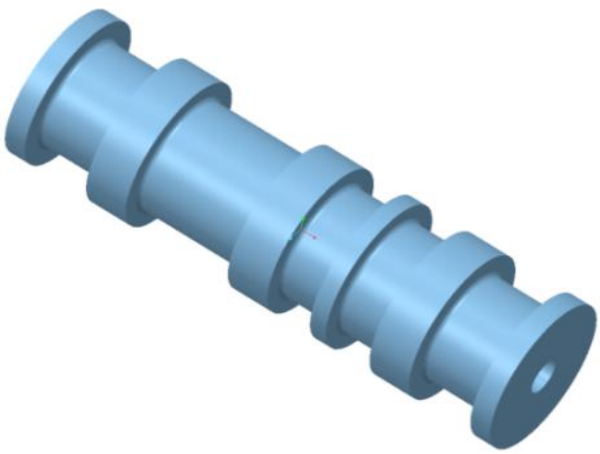

Figure 1 Crankshaft Core

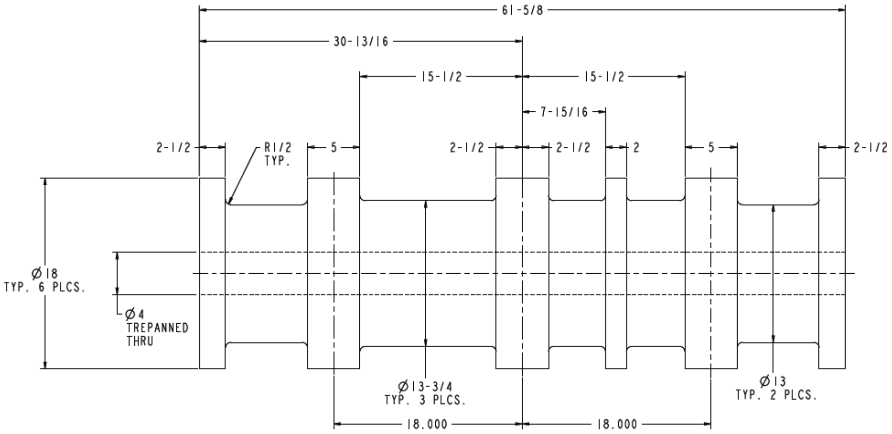

Figure 2 Crankshaft Geometry& Dimensions

The dynamically loaded eccentric-shaft connection that without macro relative movement between shaft and accessories are exposed to the danger of fatigue in the contact zone of welds and plain fatigue at the core (out of contact zone) at the same time. The zones that get affected are as follows:

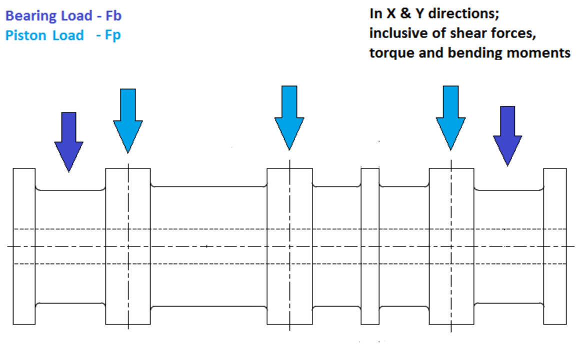

Figure 3 Piston and Bearing Load locations

Now based on these zones, the theoretical analysis is done, in a step by step approach; starting with static calculations to further it with the fatigue calculations.

IV. Analytical Calculations

For evaluating the static and fatigue calculations the set standardized formulations are applied of machine design.

All the calculations are finalized down to the basic von-Misces criterion:

$$\sigma 1 = \frac{1}{\sqrt{2}} \cdot \sqrt{\left(\sigma_{X1} - \sigma_{Y1}\right)^2 + \left(\sigma_{Y1} - \sigma_{Z1}\right)^2 + \left(\sigma_{Z1} - \sigma_{X1}\right)^2 + 6 \cdot \left[\left(\tau_{XY1}\right)^2 + \left(\tau_{YZ1}\right)^2 + \left(\tau_{XZ1}\right)^2\right]} \quad \text{. Eqn (1)}$$

The calculations are done at various angles of the crankshaft and the values of forces, moments, and stresses i.e. torsional, bending at the two following points were taken into consideration:-

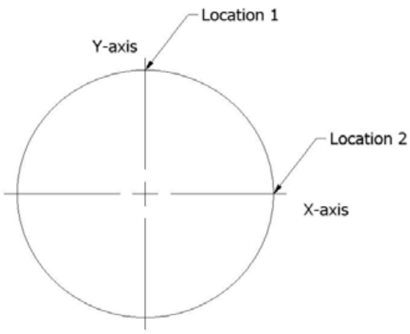

Figure 4 Locations of various stress calculations in crankshaft

The analytical values that were evaluated using the traditional Von-Misces criterion is as follows:

Sr. No.	Crank Angle θ (°)	Von Mises Stress	FOS	Von Mises Stress	FOS
		σ1	FOS for Location 1	σ2	FOS for Location 2
1	10	3523.32	19.3	2918.45	23.30
2	20	2995.59	22.7	2730.92	24.90
3	30	2518.52	27	2566.04	26.50
4	40	2369.34	28.7	2377.62	28.60
5	50	2656.25	25.6	2615.38	26.00
6	60	3119.27	21.8	2893.62	23.50
7	70	4892.09	13.9	5312.50	12.80
8	80	4625.85	14.7	5714.29	11.90
9	90	4071.86	16.7	6296.30	10.80
10	100	3953.49	17.2	6868.69	9.90
11	110	4171.78	16.3	7234.04	9.40
12	120	4276.73	15.9	7234.04	9.40
13	130	4223.60	16.1	6868.69	9.90
14	140	4415.58	15.4	6181.82	11.00
15	150	4415.58	15.4	5271.32	12.90
16	160	4121.21	16.5	4387.10	15.50
17	170	3560.21	19.1	3675.68	18.50
18	180	3076.92	22.1	3162.79	21.50

Sr. No.	Crank Angle θ (°)	Von Mises Stress	FOS	Von Mises Stress	FOS
		σ1	FOS for Location 1	σ2	FOS for Location 2
19	190	4625.85	14.7	4303.80	15.80
20	200	4303.80	15.8	4071.86	16.70
21	210	3885.71	17.5	4121.21	16.50
22	220	3820.22	17.8	4625.85	14.70
23	230	4473.68	15.2	5271.32	12.90
24	240	4927.54	13.8	5714.29	11.90
25	250	5151.52	13.2	6071.43	11.20
26	260	5190.84	13.1	6538.46	10.40
27	270	5000.00	13.6	6938.78	9.80
28	280	4689.66	14.5	7157.89	9.50
29	290	4387.10	15.5	7234.04	9.40
30	300	4358.97	15.6	7234.04	9.40
31	310	4689.66	14.5	7234.04	9.40
32	320	5074.63	13.4	7083.33	9.60
33	330	5354.33	12.7	6800.00	10.00
34	340	5528.46	12.3	6415.09	10.60
35	350	5440.00	12.5	5913.04	11.50
36	360	5190.84	13.1	5396.83	12.60

Graphically, these points are shown as follows:

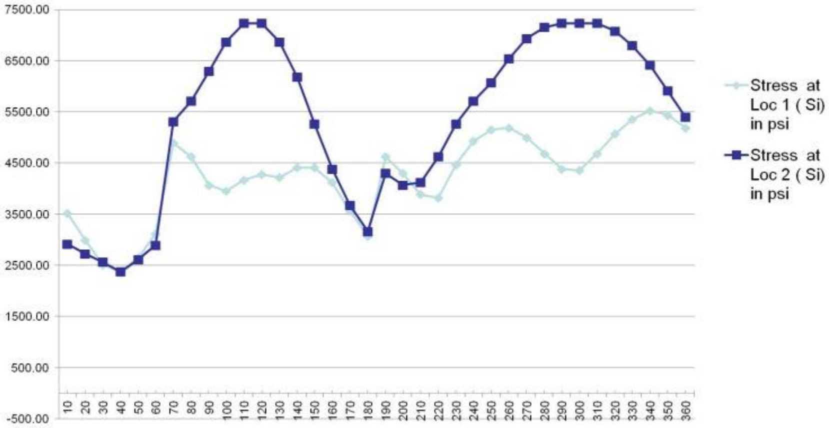

Figure 5 Data Plot of Stress induced at the Two Locations

In the following readings we can see that the crankshaft shows the weakest point at 310°, as the maximum stress is endured in that region, thus considering all the values determined at this location the fatigue analysis is further done.

The fatigue criterion used for this was as follows:

$$\sigma_a' = \left\{\left[(K_f)_{bending}(\sigma_a)_{bending} + (K_f)_{axial}\frac{(\sigma_a)_{axial}}{0.85}\right]^2 + 3\left[(K_{fs})_{torsion}(\tau_a)_{torsion}\right]^2\right\}^{1/2}$$

…. Eqn (2)

$$\sigma_m' = \left\{\left[(K_f)_{bending}(\sigma_m)_{bending} + (K_f)_{axial}(\sigma_m)_{axial}\right]^2 + 3\left[(K_{fs})_{torsion}(\tau_m)_{torsion}\right]^2\right\}^{1/2}$$

….Eqn (3)

The values arrived at are σMax = 7234psi (50394 kPa); σMin = 2566 psi (17911 kPa).

Fatigue calculations are done using modified Goodman theory as the crankshaft has stepped geometry and the fatigue calculations are done also on ANSYS is the evaluation done.

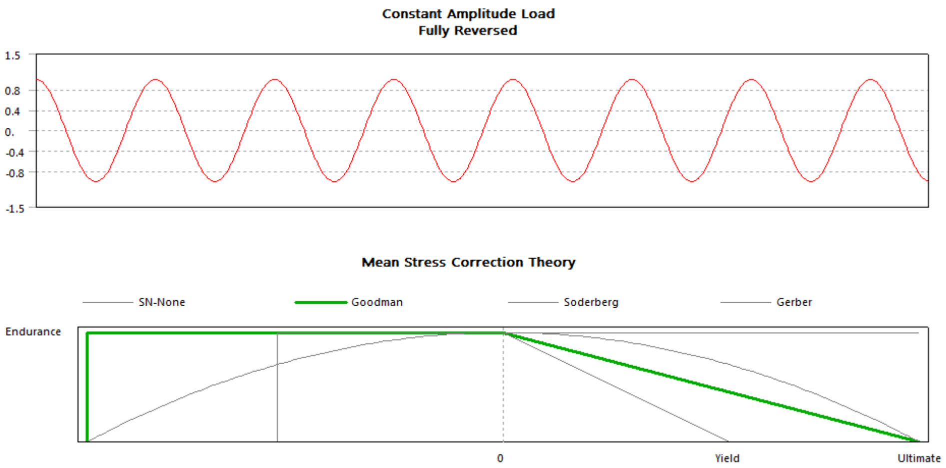

Figure 6 Fatigue Theory

V. FEA Results

The static analysis applying von-Misces stress theory helps us to arrive at a range for safe stress values that are observed while doing analytical calculations fall in the similar range that is found during FEA for the same.

The analytical values that are arrived at the maximum stress zone at 310° is 7234 psi and 11.49 and falls in the same range in ANSYS.

The following are the Stress and FOS values range developed in the FEA as against the stress and FOS in ANSYS:

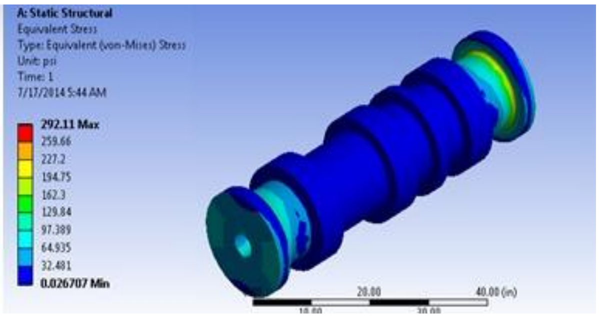

Figure 7 Static Equivalent stress on the crankshaft core

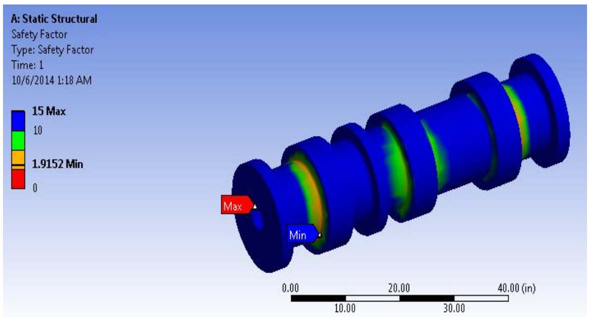

Figure 8 Static FOS range for the crankshaft core

The focus of fatigue in ANSYS is to provide useful information to the design engineer when fatigue failure may be a concern. Fatigue results can have a convergence attached. A stress-life approach has been adopted for conducting a fatigue analysis.

Following are the ANSYS results that were arrived at as against the theoretical values were that arrived with life value of 5.69 and FOS of 11.49 theoretically.

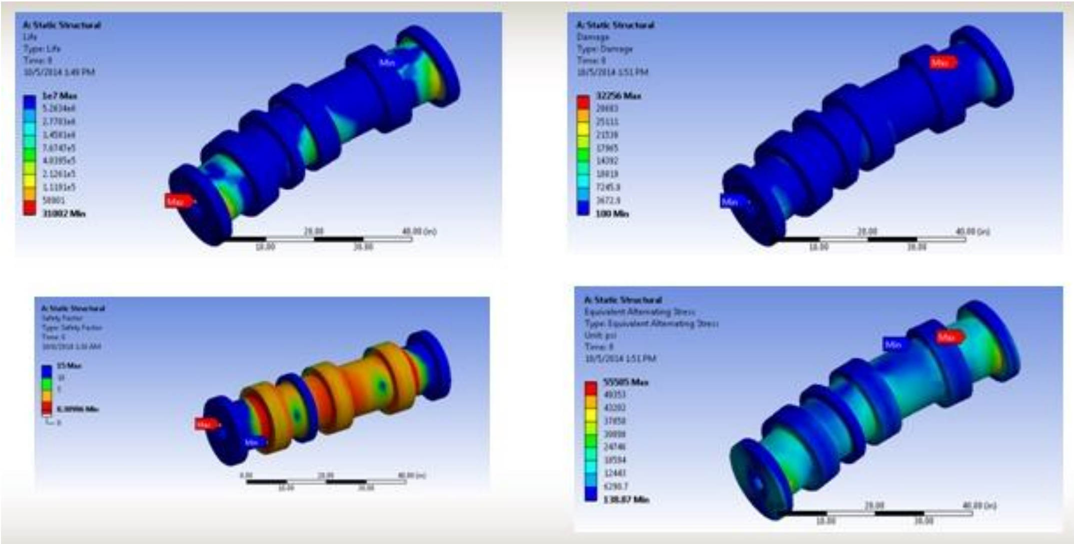

Figure 9Fatigue Results for Infinite Life, Safety Factor, Damage & Alternating stress

In this document, we have also evaluated the value of the life of the bearing using the empirical formulation:

$$L_{10} := \left(\frac{16667}{n}\right)\cdot\left(\frac{C_r}{P}\right)^{\frac{10}{3}}$$

…. Eqn (4)

For taper roller bearings, it is evaluated to be 1.15 to 1.5 ~ 2 years of the end and the intermediate bearings.

VI. Results

From the above calculations, catered to both static and fatigue criterions, following are the values derived.

Factors	Theoretical	Ansys Applied/ Derived
Life (L10)	**7.39 x 10^11 cycles (infinite cycle theory)**	**1 x 10^7 cycles (safe-infinite cycle)**
FOS	**11.95 (static)** **11.4** **(fatigue – mod- Goodman theory)** **5.69 (combined)**	**12 (static)** **5 (fatigue)**

As per calculations it is been found that,
Life of Bearings = 1.15 to 1.5 = ~ 2 years (approx.)
Life of the Crankshaft core (N) = 32+ = ~ 33 years (approx.)

This proves that for any eccentric shaft it is mandatory to evaluate which location is at the weakest or which component can fail the earliest while undergoing multiple loading or maybe combined loading.

VII. Conclusion

In this paper, we have evaluated the life of the crankshaft core by static and fatigue analysis and compared it with ANSYS results and the values have fallen in the safe range and the components can be considered as safe design.

The term 'design' shall apply to parameters or features of the equipment supplied by the manufacturer. Many other supporting documents have also been provided to the client as and when demanded for.

To add on, we have also catered to compliance for EH-1600, as the equipment (including auxiliaries), but excluding normal maintenance and wear parts as identified as they shall be designed and constructed for a minimum service life of 20 years and at least 3 years of uninterrupted operation. It is recognized that these requirements are design criteria, and that service or duty severity, mis-operation, or improper maintenance can result in a machine failing to meet these criteria.

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