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Dynamic Stress Analysis of a Multi cylinder Two-stage Compressor Crankshaft

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Abstract

In this work, the dynamic stress analysis was carried out to change the crankshaft material from forged steel shaft (En19C) and steel casting (En18C) to spheroidal graphite cast iron (SG600/3) to reduce the initial investment cost for development of the new product. A dynamic simulation was conducted on a crankshaft from six cylinder two-stage oil less compressor. Finite element analysis was performed to obtain the stress magnitude at critical locations at worst load cases. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the compressor specification. The dynamic load calculation is done analytically by considering the reciprocating masses and its inertial loads in all cylinders. This load was applied to the FE model in ABAQUS, and boundary conditions were applied according to all loads acting on the crankshaft mounting .The stresses are calculated from FE model and plotted in Goodman diagram to calculate the factor of safety. Results achieved from aforementioned analysis can be used in fatigue life calculation and optimization of this component.

Keywords: Crankshaft, compressor, FE model.

Introduction

In reciprocating compressor transmission system consists of crankshaft, main bearings, connecting rod and coupling connected to the prime mover. In this design, the compressor crankshaft drives the compressor as well as it is used to transmit the power to other accessories like traction motor blower and radiator fan in diesel locomotives. Crankshaft is a large component with a complex geometry in the compressor transmission system, which converts the rotary motion of the shaft to reciprocating displacement of the piston with a four link mechanism. This analysis was conducted on a six-cylinder two-stage compressor.

Crankshaft experiences large forces from first and second stage cylinders during loading and unloading of the compressor. The power is transmitted to the crankshaft through coupling connected from locomotive main engine crankshaft. As the crank pin is rotated, the connecting rod converts the rotary motion to reciprocate the piston. The magnitude of the force depends on many factors which consists of crank radius, connecting rod dimensions, weight of the connecting rod, piston, piston rings, and pin. Gas forces in the top of the piston and inertia forces acting on the crankshaft cause two types of loading on the crankshaft structure; torsional load and bending load.

Due to the crankshaft geometry and compressor mechanism, the crankshaft fillet experiences a large stress range during its service life. Figure 1 shows a crankshaft in the engine block from side view. In this figure it can be seen that at the moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crack initiation, leading to fracture.

Payer et al.[1] developed a two-step technique to perform nonlinear transient analysis of crankshafts combining a beam-mass model and a solid element model. Using FEA, two major steps were used to calculate the transient stress behaviour of the crankshaft; the first step calculated time dependent deformations by a step-by-step integration using the newmark-betamethod.Using a rotating beam-mass-model of the crankshaft, a time dependent nonlinear oil film model and a model of the main bearing wall structure, the mass, damping and stiffness matrices were built at each time step and the equation system was solved by an iterative method. In the second step those transient deformations were enforced to a solid-element-model of the crankshaft to determine its time dependent stress behaviour.

The major advantage of using the two steps was reduction of CPU time for calculations. This is because the number of degrees of freedom for performing step one was low and therefore enabled an efficient solution. Furthermore, the stiffness matrix of the solid element model for step two needed only to be built up once.

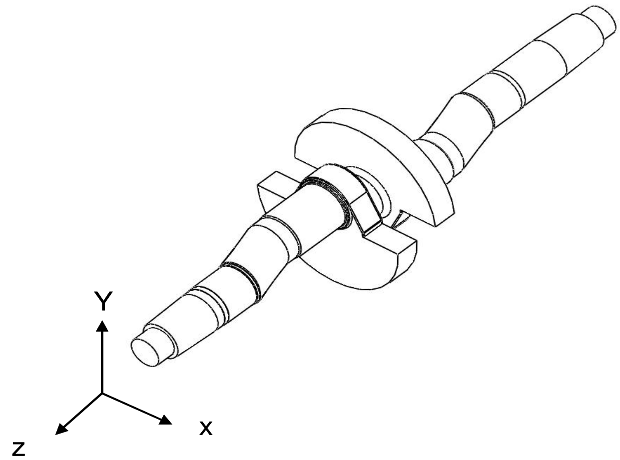

Figure-1
Crankshaft configuration

A geometrically restricted model of a light automotive crankshaft was studied by Borges et al.[2]. The geometry of the crankshaft was geometrically restricted due to limitations in the computer resources available to the authors. The FEM analysis was performed in ANSYS software and a three dimensional model made of Photo elastic material with the same boundary conditions was used to verify the results. This study was based on static load analysis and investigated loading at a specific crank angle. The FE model results showed uniform stress distribution over the crank, and the only region with high stress concentration was the fillet between the crank-pin bearing and the crank web.

Load analysis: The crankshaft investigated in this study is shown in figure 1 and belongs to compressor with the configuration shown in table 1 and cylinder pressure versus crankshaft angle shown in figure 3. Although the pressure plot changes for different compressor speeds, the maximum pressure, which is much of our concern, does not change and the same graph could be used for different speeds. The geometries of the crankshaft, bearing assembly and connecting rod from CAD software, which provided the solid properties of the connecting rod such as moment of inertia and canter of gravity (CG). These data were used in to calculate inertial loads and total load on each connecting rod big end bearings against the crankshaft angle as shown in figure 4.

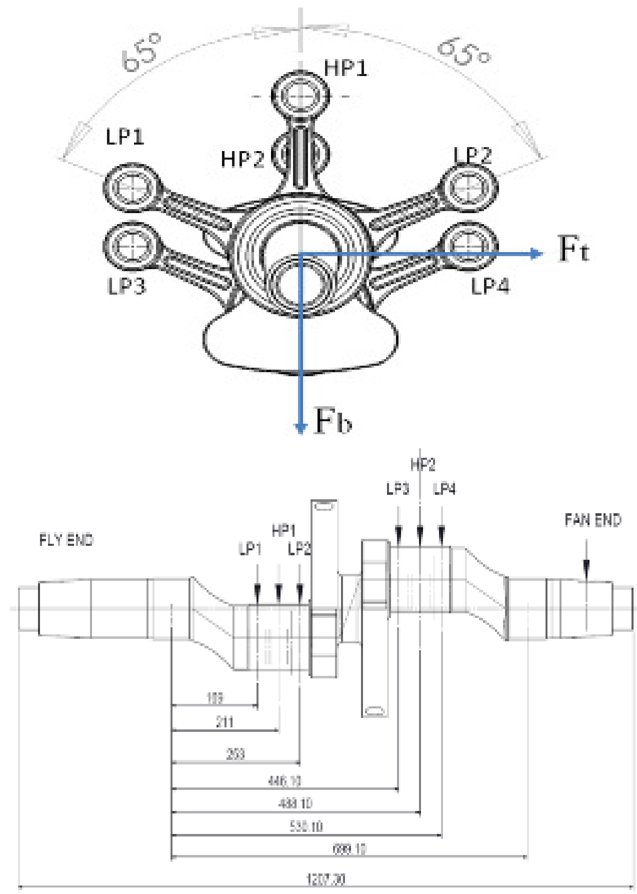

Figure-2
Cylinder orientation and its location

Table-1
Configuration of the compressor

Parameter	First stage	Second stage	units
Crank radius	4.3	4.3	cm
Diameter	15.0	11.0	cm
Max pressure	3.5	10.0	bar
Connecting rod length	27.0	27.0	cm
Distance of the CG of the connecting rod from crank end canter	7.5	7.5	cm
Mass moment of inertia of connecting rod	6.2×10^4	6.2×10^4	Kg mm^2
Mass of the connecting rod	2.65	2.70	kg
Mass of piston assembly	1.10	0.52	kg

Finite Element modelling: The finite element model of the crankshaft was developed using ABAQUS. Analysis was carried using Non linear quasi static approach and crankshaft geometry is meshed with second order tetrahedral elements. As a crankshaft is designed for very long life, stresses must be in the linear elastic range of the material. Therefore, all carried analysis are based on the linear properties of the crankshaft material. The meshed crankshaft with tetrahedral elements is shown in figure 5.

Crankshaft was analyzed as a part placed in between the bearing and the operational loads imposed as tangential and normal components. Gas pressure loads, fan torque and the inertial loads of the reciprocating piston assembly was considered in the simulation

The mesh of the finite element model used to evaluate the stiffness of the crankshaft is shown in figure 5. The same model was also used in the final step to evaluate the stress level in the crankshaft. The boundary conditions applied during the evaluation of the stiffness were null radial displacement in bearing regions.

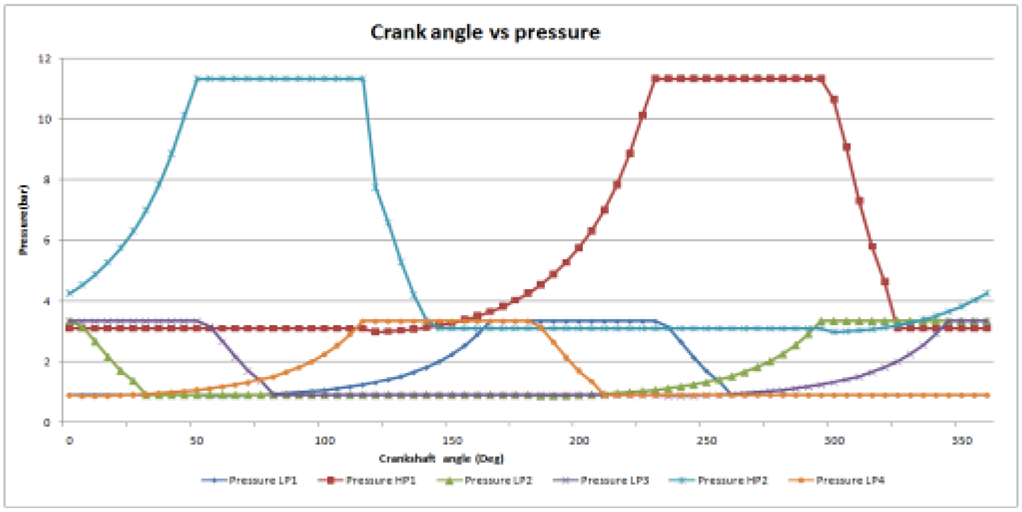

Figure-3
Cylinder pressure versus crankshaft angle

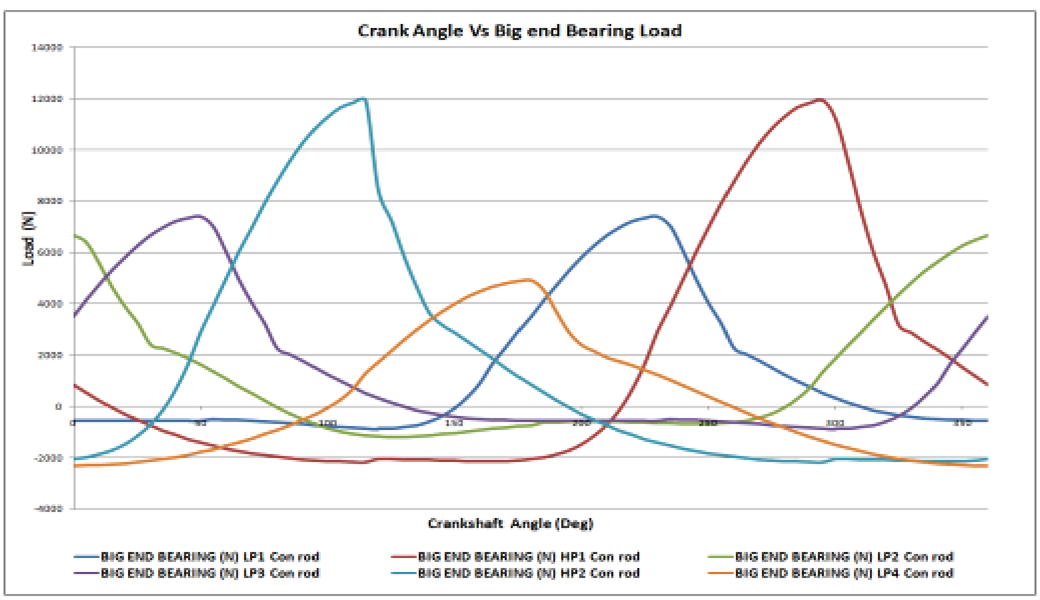

Figure-4
Big end bearing load versus crankshaft angle

Boundary conditions: Boundary conditions in the FE model were based on the compressor configuration. The mounting of this specific crankshaft is on two different bearings which results in different constraints in the boundary conditions. Two rigid rings used to simulate the bearing constraint in all degrees of freedom. This indicates that the surface cannot move in either direction and cannot rotate. Structural coupling elements used to constraint fly end in rotational degrees of freedom and fan load applied at the non drive end as torque. It should be noted that the analysis is based on dynamic loading, though each finite element analysis step is done in static equilibrium. The main advantage of this kind of analysis is more accurate estimation of the maximum and minimum loads. Design and analyzes of the crankshaft based on static loading can lead to very conservative results. In addition, as was shown in this section, the minimum load could be achieved only if the analysis of loading is carried out during the entire cycle.

As the dynamic loading condition is analyzed, only two main loading conditions are applied to the surface of the crankpin bearing due to gas forces in the low pressure and high pressure cylinders along with other loads like fan torque and inertial loads. These two loads are perpendicular to each other and their directions are shown in figure 7 as Fx and Fy. Since the contact surface between connecting rod and crankpin bearing does not carry tension, Fx and Fy can also act in the opposite direction to those shown in figure 7. There are four load cases, one is at maximum bending load occurs when HP1 at TDC and HP2 at BDC and HP2 at TDC and HP1 at BDC and second worst case occurs at maximum torsional load when β=8.5 degree happens to be the position of maximum cumulative loads of thrust and tangential loads. The values are shown in table 2.

Results and Discussion

Some locations on the geometry were considered for depicting the stress history. These locations were selected according to the results of FE analysis and all the selected elements are located on different parts of the fillet areas due to the high stress concentrations at these locations. Selected locations are labeled in figure 8 and maximum and minimum principle stress are extracted from the from the stress plot .The mean and amplitude stresses are calculated to plot the Goodman diagram and endurance strength of each material considered as 35% of its ultimate strength for 10^7 cycles to calculate the safety factor. In this design, the symmetrical arrangement of cylinder indicates reflects in stress magnitude and direction as shown in Table.

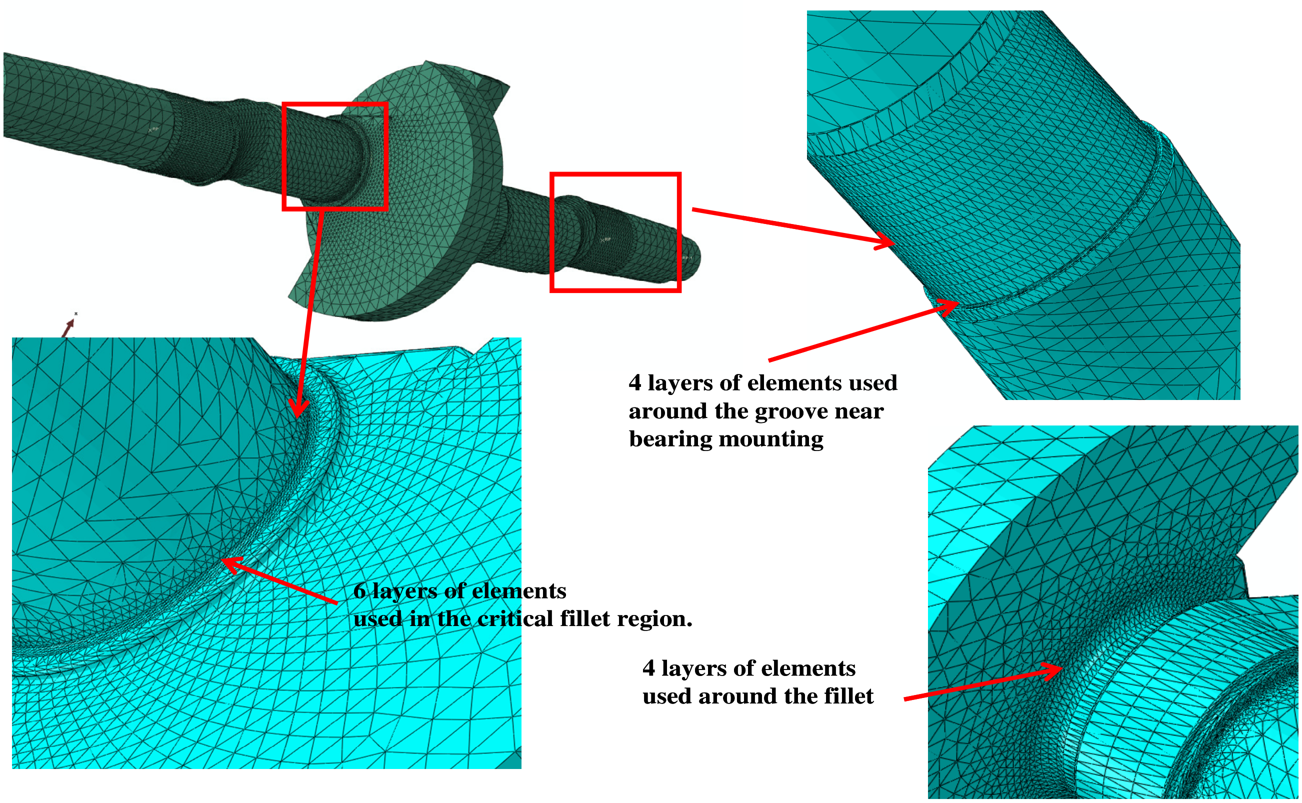

Figure-5
FE model of the crankshaft with fine mesh in fillet areas

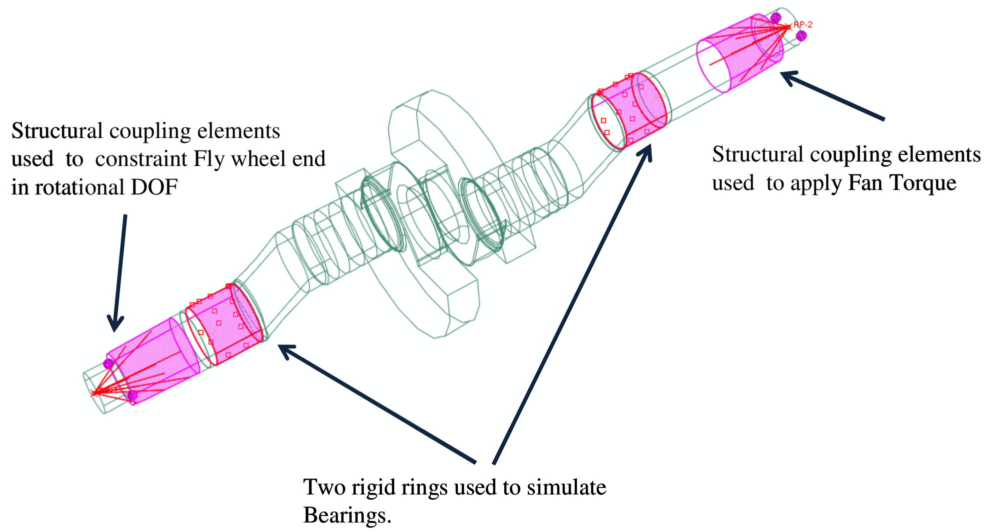

Figure-6
Boundary conditions applied in FEA model

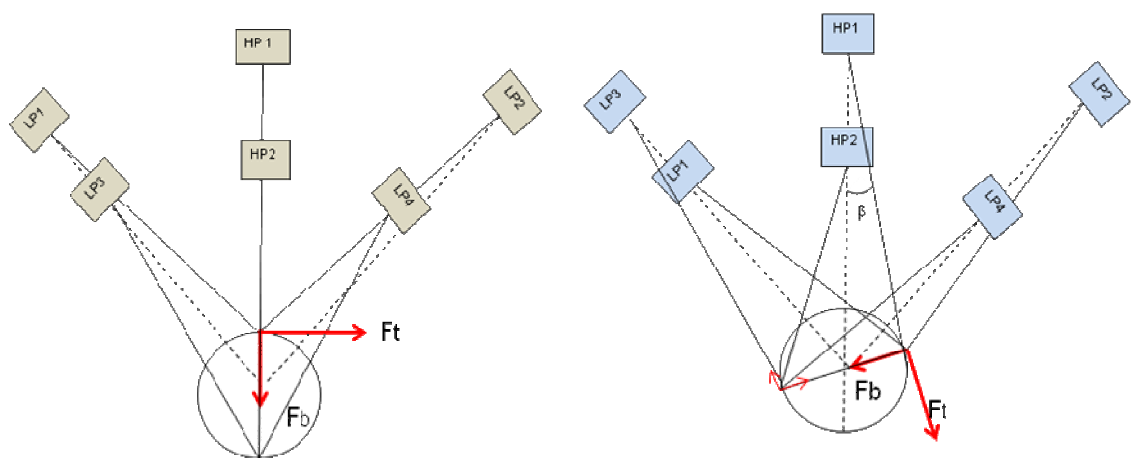

Figure-7
Bending and torsional load directions

Table-2
Load cases

Load Cases	Bending (Y) torsional loads (X)	Load (N)					
		LP1	**HP1**	**LP2**	**IP3**	**HP2**	**LP4**
Load case: 1 HP1, β=0 deg	Load in X axis	1915.8	0.0	-5079.1	-1298.0	0.0	1958.4
	Load in Y axis	5713	1-888.7	1514.5	-855.8	-2138.1	-1291.1
Load case: 2 HP2, β=0 deg	Load in X axis	1298.0	00	-16691	-1915.8	-11888.7	4708.2
	Load in Y axis	8558	2138.1	1100.4	-571.3	0.0	-1403.9
Load case: 3 HP1, β=8.4 deg	Load in X axis	0.0	-10730.8	-565.5	0.0	2058.6	1963.1
	Load in Y axis	7357.0	3199.8	-608.3	-527.2	-1357.2	1270.5
Load case: 4 HP2, β=8.4 deg	Load in X axis	0.0	-2058.6	-2565.5	0.0	10730.8	1633.3
	Load in Y axis	5271	1357.2	-1660.4	-7357.0	-3199.8	1757.0

Table-2
Stress results

Load Case	Maximum and Minimum Principle Stress at Critical Locations							
	Location - A		Location - B		Location - C		Location - D	
	Max (MPa)	Min (MPa)	Max (MPa)	Min (MPa)	Max (MPa)	Min (MPa)	Max (MPa)	Min (MPa)
1	41	2.5	0.32	-15	27	-1	-1	-21
2	-0.6	13	39	3	-0.2	-18	26	-0.35
3	39	1.7	10.5	-27	31.3	1	17	-30.9
4	-0.6	-11.1	34	1.6	1.02	-12.5	41.3	-7.7

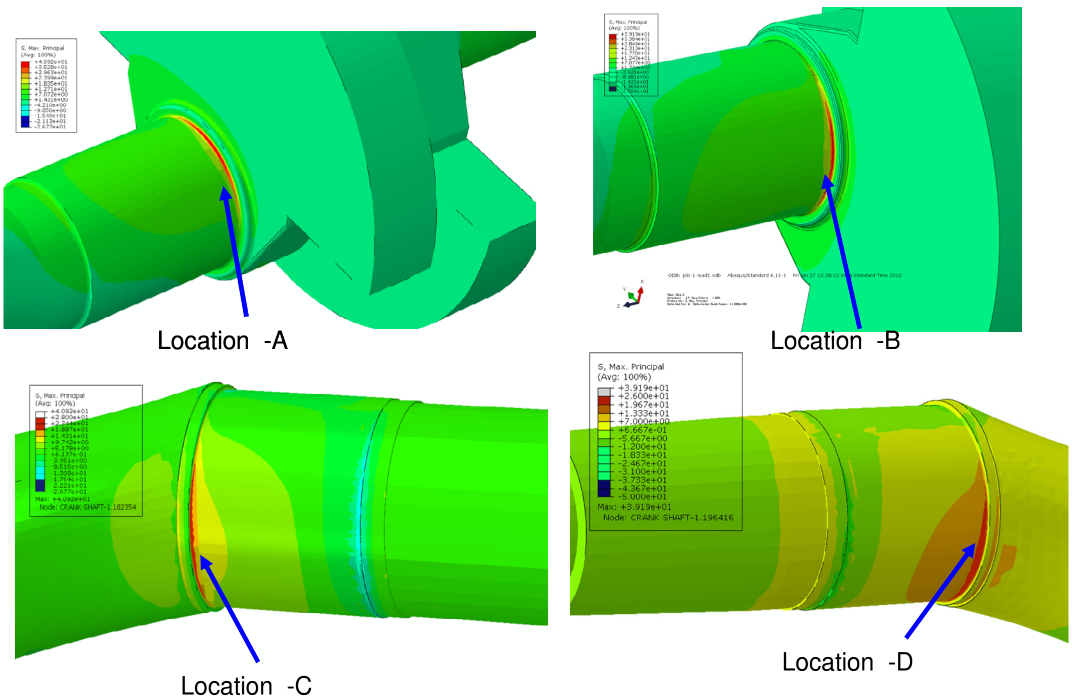

Figure-8
Goodman diagram and SN curve for three materials

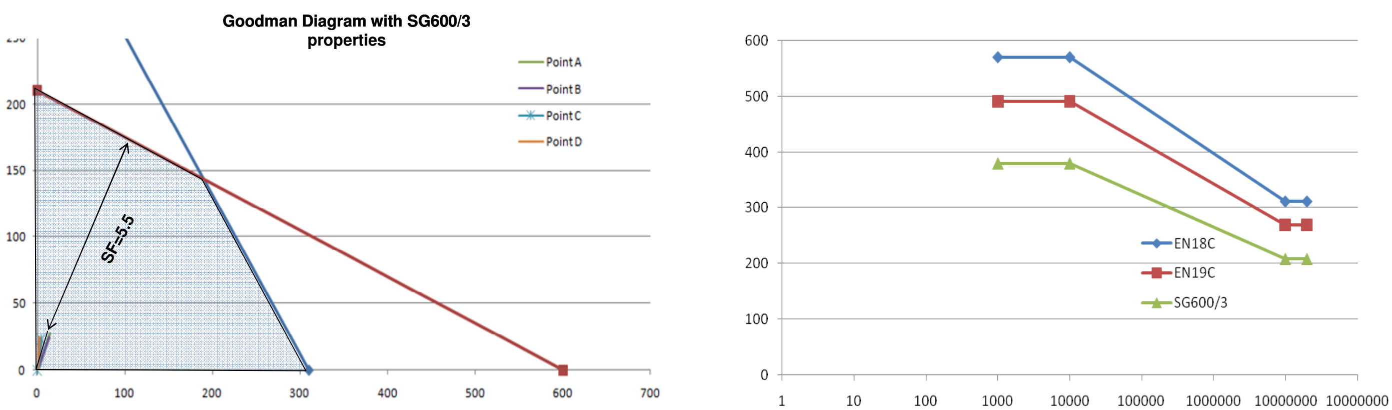

Figure-9
Goodman diagram and SN curve for three materials

Conclusion

Multi cylinder two-stage compressor load pattern is established to carry out the FE analysis. Dynamic loading analysis of the crankshaft results in more realistic stresses whereas static analysis provides an overestimate results. Accurate stresses are critical input to fatigue analysis and optimization of the crankshaft. There are two different load sources in compressor crankshaft, one is the gas forces and inertia of the reciprocating masses. These two load source cause both bending and torsional load on the crankshaft. Critical locations on the crankshaft geometry are all located on the fillet areas because of high stress gradients in these locations, which result in high stress concentration factors. The stresses are within the endurance strength and safety margin of 5.5 with SG 600/3 material. Therefore, we can conclude that, this material can used with casting process for development of crankshafts in compressor applications to bring down the initial investment cost for new product development.

References

1. Farzin H. Montazersadgh and Ali Fatemi, Dynamic Load and Stress Analysis of a Crankshaft, SAE Technical Paper No, 2007 -01-0258, Society of Automotive Engineers **(2007)**

2. Payar E., Kainz A., and Fiedler G.A., Fatigue Analysis of Crankshafts Using Nonlinear Transient Simulation Techniques, SAE Technical Paper No. 950709, Society of Automotive Engineers **(1995)**

3. Borges A.C.C., Oliveira L.C., and Neto P.S., Stress Distribution in a Crankshaft Crank Using a Geometrucally Restricted Finite Element Model, SAE Technical Paper No. 2002-01-2183, Society of Automotive Engineers **(2002)**

4. Becerra J.A., Jimenez F.J., Torres M., Sanchez D.T. and Carvajal E., Failure analysis of reciprocating compressor crankshafts, *Engineering Failure Analysis,* 18 **(2011)**

5. John B. Heywood., the Two Stroke cycle Engine, Society of automotive engineers, Inc.

6. Shigley Joseph E., Mechanical Engineering Design,3rd edition, McGraw-Hill **(1977)**