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Themo-mechanical analysis of crankcase

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ABSTRACT

A systematic, sensitivity based design process to perform durability simulation of a crankcase for a six cylinder four stroke truck diesel engine with a peak firing pressure (PFP) of 180bar is presented. Traditionally, a trial and error method is used when a design fails to meet the strength and the design criteria, which often conflict. This old approach not only is time and cost consuming but also does not provide insight into structural behavior. This proposed process uses modern CFD, Multibody and FEA technologies such as design sensitivity analysis, coolant flow analysis, graphical user interface, etc. It handles multi-discipline design criteria simultaneously and provides design engineers insight into structural responses for durability, and stiffness concerns and a means for systematic design review for its strength and quality improvement. Involving Identifying and considering PFP, inertia loads, thermal loads and assembly loads. This needs performing crankshaft dynamic and thermal analysis on multibody software. The obtained loads were then applied to the FE model and boundary conditions were applied according to the engine assembly.

Keywords— **Dynamics, Inertia, Multibody**

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I. INTRODUCTION

Fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading even when the nominal maximum stress values are less than the ultimate tensile stress limit of the material. Generally fatigue is classified into two categories low cycle fatigue and high cycle fatigue. In high cycle fatigue the cyclic plasticity are not taken into account. The term high cycle fatigue implies that the fatigue mechanisms of macroscopic cyclic plasticity are not present. Under high cycle, low stress fatigue situations prevails and the material deforms primarily elastically; the failure time or the number of the cycles to failure under such high cycle fatigue has been characterized in terms of the stress range. Fatigue can be developed in four stages. Crack nucleation; stage I crack growth; stage II crack growth; and Ultimate ductile failure. Factors that affect the fatigue-life are cyclic stress state, surface quality, material type, residual stresses, size and distribution of internal defects, direction of loading, grain size, environment and temperature.

Cylinder block is one of the most critical components which are subjected to high cycle fatigue. It is a complex part at the heart of an engine which adapts the cylinder head, crankcase, engine mounts, drive housing and engine ancillaries, with passages for coolants and lubricants. Engine blocks are usually made from cast iron or in modern engines, aluminium and magnesium are used.

The loads that are acting in the engine block are Combustion load, Piston side loads, balance shaft loads, main bearing loads, main bolt clamp load, thermal load and head bolt clamp load. These loads are considered for the durability analysis of the engine block. Durability analysis can be used to determine how long a component can survive in a given service environment. In a general case, durability refers to the ability of a component to function in the presence of defects for a given environment/ loading. Hence, the term durability analysis will be used to describe the analysis of a fatigue performance. A finite element method (abbreviated as FEM) is a numerical technique to obtain an approximate solution to a class of problems governed by elliptic partial differential equations. Such problems are called as boundary value problems as they consist of a partial differential equation and the boundary conditions. The finite element method converts the elliptic partial differential equation into a set of algebraic equations

which are easy to solve. It has 3 stages; Preprocessing, Analysis and Post processing.

II. OVERVIEW OF THE CRANKSHAFT SYSTEM MODEL

In its most general form, it couples the crankshaft structural dynamics, the main bearing hydrodynamic lubrication and the engine block stiffness using a system approach. Figure1. Shows the 3-d FEA model

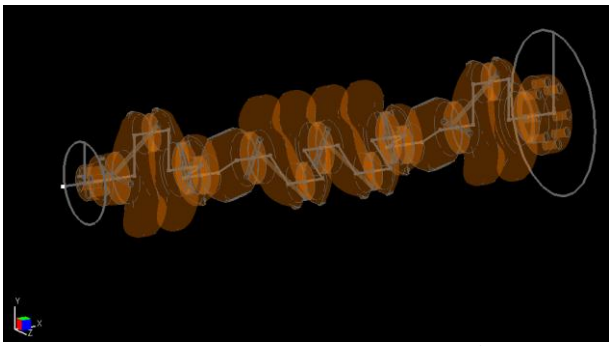


Figure 1: 3-d FEA model Crankshaft

A finite element mesh for the entire crankshaft is needed in order to calculate its structural dynamics response. The main bearing lubrication analysis is performed by solving the 2-D Reynold's equation for each main bearing using the finite element method. The flexibility of the engine block is represented by its stiffness at each bearing location. The main output from the analysis is the crankshaft dynamics response in terms of terms of displacements, velocities and accelerations at the user specified grid points, natural frequencies and mode shaped, and crankshaft dynamic stresses throughout the engine cycle. The main bearing loads (force and moments) and bearing performance parameters such as eccentricities, minimum film thickness and maximum film pressure are calculated and output for each bearing. The oil-film thickness and pressure distributions are also calculated at each crank angle throughout the whole engine cycle.

A two level dynamic sub-structuring is performed for the structural dynamic analysis of the crankshaft based on load-dependent Ritz vectors which are generated by a subspace algorithm. After the two dynamic reductions, the initial finite element model size is significantly reduced to very few generalized degrees of freedom which are efficiently integrated in time. The rotating crankshaft is properly coupled with the fixed compliant engine block. The block compliance is represented by a distributed linear elastic foundation at each main bearing location both in the vertical and horizontal planes. This representation accounts not only for the translational block stiffness but for the rotational block stiffness as well in both planes. As results, the bearing loads consist of reaction forces and reaction moments. The reaction moments introduces bearing journal misalignment which is normally neglected in traditional lubrication analysis.

The crankshaft structural analysis predicts the crankshaft dynamic response base on the finite-element method. A two level dynamic sub-structuring with special Ritz vector is preformed. Initially, a given three dimensional finite-element model of the crankshaft is divided into sub-structures. Each substructure is dynamically reduced using a set of load-dependent Rits vectors. Subsequently, all the substructures are assembled and a second dynamic reduction is performed using a new set of Rits vectors. A subspace algorithm is used to generate the load-dependent Rits vectors.

III. LOAD ANALYSIS

The crankshaft investigated in this study is shown in Figure 2 and belongs to an engine with the configuration shown in Table 1 and piston pressure versus crankshaft angle shown in Figure 3. Although the pressure plot changes for different engine speeds, the maximum pressure which is much of our concern does not change and the same graph could be used for different speeds [9]. The geometries of the crankshaft and connecting rod from the same engine were measured with the accuracy of 0.0025 mm (0.0001 in) and were drawn in the CAD software, which provided the solid properties of the connecting rod such as moment of inertia and center of gravity (CG). These data were used in ADAMS software to simulate the slider-crank mechanism. The dynamic analysis resulted in angular velocity and angular acceleration of the connecting rod and forces between the crankshaft and the connecting rod.

Engine	
No of Cylinders	6
Type of Engine	Inline
Maximum gas pressure	180bar
Firing Order	6-2-4-1-5-3
Connecting rod length	207mm
Connecting rod mass	2.115Kg
Izz of connecting rod about the center of gravity	15000kg.mm2
Distance of C.G. of connecting rod from crank end center	57.95mm
Mass of the piston assembly	1.263Kg

Table 1: Configuration of the engine to which the crankshaft belongs

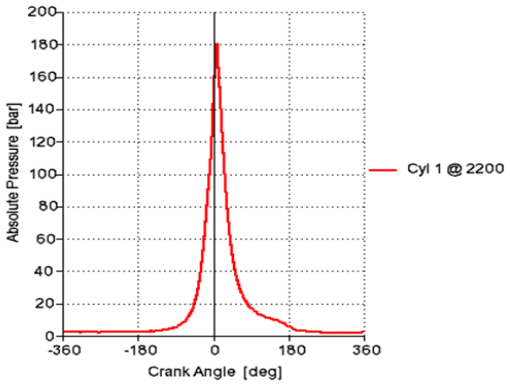


Figure 2: Piston pressure versus crankshaft angle diagram used to calculate forces at the connecting rod ends

There are two different load sources acting on the crankshaft. Inertia of rotating components (e.g. connecting rod) applies forces to the crankshaft and this force increases

with the increase of engine speed. This force is directly related to the rotating speed and acceleration of rotating components.

Variation of angular acceleration and angular velocity of the connecting rod for the engine speed of 3600 rpm is shown in Figure 3. The second load source is the force applied to the crankshaft due to gas combustion in the cylinder. The slider-crank mechanism transports the pressure applied to the upper part of the slider to the joint between crankshaft and connecting rod. This transmitted load depends on the dimensions of the mechanism.

Forces applied to the crankshaft cause bending and torsion. Figure 2 demonstrates the positive directions and local axis on the contact surface with the connecting rod. Figure 4 shows the variations of bending and torsion loads and the magnitude of the total force applied to the crankshaft as a function of crankshaft angle for the engine speed of 2200 rpm. The maximum load which happens at (9',129', 249' for 1st three main bearings) degrees is where combustion takes place, at this moment the acting force on the crankshaft is just bending load since the direction of the force is exactly toward the centre of the crank radius (i.e. Fy = 0 in Figure 1). This maximum load situation happens in all types of engines with a slight difference in the crank angle. In addition, most analysis done on engines with more cylinders (e.g. 4, 6, and 8) is on a portion of the crankshaft that consists of two main journal bearings, two crank webs, and a connecting rod pin journal. Therefore, analysis done for this single cylinder engine can be extended to larger engines.

In many studies the torsional load is neglected for the load analysis of the crankshaft, and this is because torsional load is less than 10 percent of the bending load [10]. In this specific engine with its dynamic loading, it is shown in the next sections that torsional load has no effect on the range of von Mises stress at the critical location. The main reason of torsional load not having much effect on the stress range is that the maxima of bending and torsional loading happen at different times (see Figure 4). In addition, when the peak of the bending load takes place the magnitude of torsional load is zero.

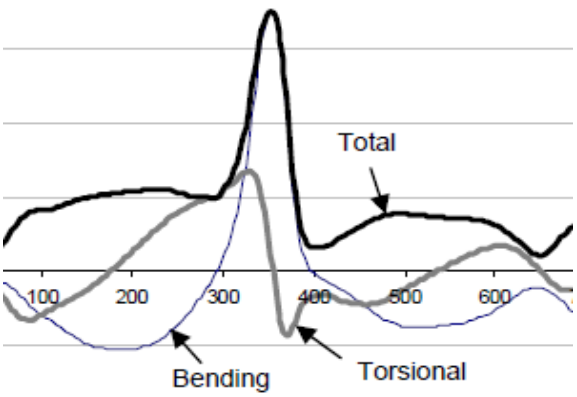


Figure 3: Bending, torsional, and the resultant force at the connecting rod bearing at the engine speed of 2200rpm

Results for this crankshaft at the maximum engine speed of 2200 rpm have been plotted for all the 7 main bearings are plotted (only three first main bearings are shown) Figure 5, 6, 7 shows the variation of the force acting on main bearings. Two components of the force are plotted, one along horizontal, Fx and the other normal to it, Fz.

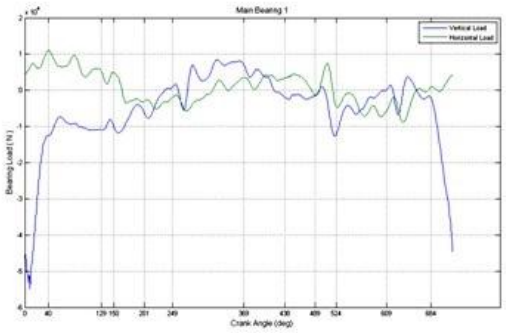


Figure 4: Loads on 1st main bearing

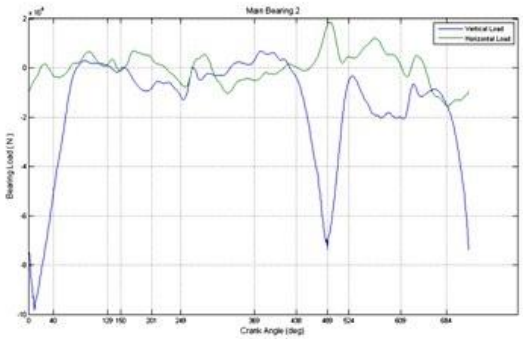


Figure 5: Loads on 1st main bearing

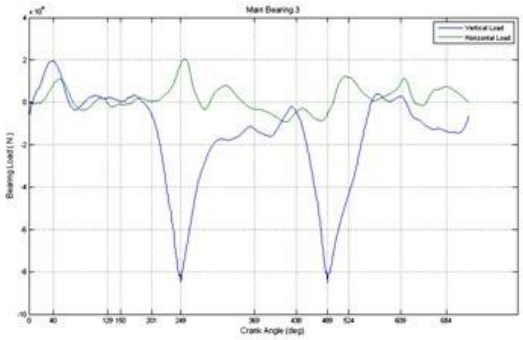


Figure 6: Loads on 1st main bearing

1. Maximum bearing load of 98.62 KN is observed at main journal bearing no 2 (from cranknose end)
2. Maximum torsional displacement (crankshaft twist) observed is 1.31 deg
3. Minimum oil film thickness is 2.5 microns

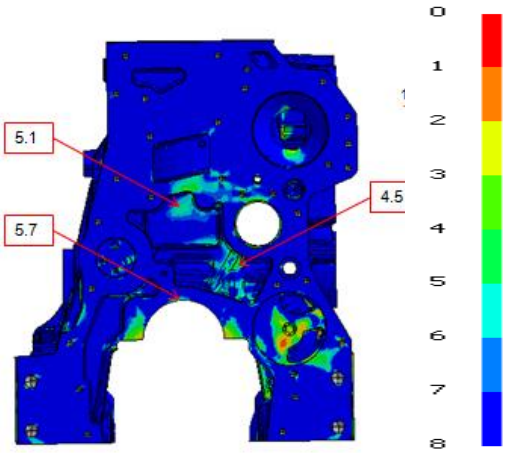
IV. DISCUSSION

Looking at the bearing load plots obtained from the crankshaft hydrodynamic analysis on on all the seven main bearings shells it is observed that, the load is maximum, at its combustion crank angles (9, 129, 249) degree, from which it is well understood that there will be highest value of the bending load acting on the crankshaft and on to the block through main bearings. So for any of the further crankshaft-block interaction analysis, loads acting at the firing crank(peak firing) angle and other minimum bearing loads (crank angle selected from the bearing load plot) can be considered to be the two worst cases and the design analysis can be performed.

V.GENERAL METHODOLOGY

The general procedure to carry out the durability analysis of the cylinder block considers engine operating loads every at the initial firing angle and at every subsequent 5 or 10 degrees of crankshaft rotation which results in either 144 or 72 load steps in addition to the room temperature and steady state hot assembly, so to avoid unnecessary solution and loading model building its enough if we can identify the critical max and min loads and reduce the solution time required for the durability simulation of the engine block.

The engine assembly simulation process is a sequential Thermo-mechanical analysis using a commercial analysis package like Abaqus. The metal temperature distribution in the finite element engine assembly model is obtained from thermal analysis. The coolant, oil and gas side boundary conditions obtained from various computational fluid dynamic simulations predicting operating conditions form the inputs for the thermal analysis. They are Gas side boundary condition, Coolant side boundary condition, Oil side boundary conditions. Post metal temperature predications, sequential mechanical analysis are followed. The assembly loads which include various bolt pre-tension loads and press-fits. After the room temperature and steady state hot assembly simulations are completed, it is followed with engine operating load simulations. The loading considered are cylinder firing loads, main bearing loads and piston side loads. These loads are applied in sequential analysis load steps to map the total behavior representing two revolutions of crankshaft. The resulting stress history obtained from the finite element simulation is used to carry out durability analysis. The fatigue analysis of the engine block is carried out using commercial software like Fe-Safe. The typical results of bearing panel factor of safety distribution on two bearing panels are shown below.



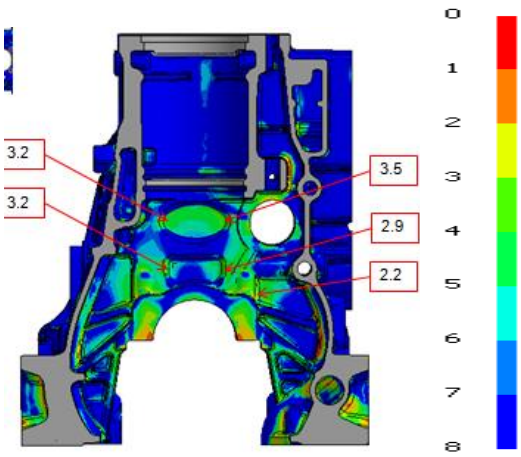


Figure 7: Fatigue Safety Factors

VI.CONCLUSION

The following conclusions could be drawn from this study:

1. 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.
2. There are two different load sources in an engine; inertia and combustion. These two load source cause both bending and torsional load on the crankshaft.
3. The maximum load occurs at the crank angle of 355 degrees for this specific engine. At this angle only bending load is applied to the crankshaft.
4. Considering torsional load in the overall dynamic loading conditions has no effect on von Mises stress at the critically stressed location. The effect of torsion on the stress range is also relatively small at other locations undergoing torsional load. Therefore, the crankshaft analysis could be simplified to applying only bending load.
5. Superposition of FEM analysis results from two perpendicular loads is an efficient and simple method of achieving stresses at different loading conditions according to forces applied to the crankshaft in dynamic analysis.

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