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Multi-body dynamic stress analysis of a crankshaft for V8 engine

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Abstract : Comprehensive, accurate and inexpensive way of analyzing a crankshaft in terms of stress has been suggested. Putting deformable crankshaft in analysis, this method takes into account torsional vibration, journal bearing and gas pressure simultaneously giving transient stress values. In opposition to various precedential approaches, the method doesn't require separate rigid body dynamic analysis and stress analysis. In other words, one single multi-body dynamic analysis including crankshaft, piston, flywheel and torsional damper gives comprehensive stress values considering most of key factors influencing crankshaft stress. As additional results, main bearing reaction forces and journal center orbts were also investigated. A crankshaft for V8 heavy-duty diesel engine was analyzed and evaluated by suggested method.

Key words : Crankshaft, Dynamic analysis, Finite element method, Torsional vibration, Journal bearing

1. INTRODUCTION

A crankshaft is one of the most critically loaded parts in internal combustion engines, giving significant damage in case of its failure. Therefore, its precise evaluation is highly important to guarantee overall soundness especially for heavy-duty diesel engines of up to 5,000 horsepower. Moreover, it is also essential to have reasonable safety margin preventing excessive production cost to be competitive in fierce diesel engine industry. So far, versatile approaches has been tried to obtain realistic stress values to avoid extravagant safety margin, however, most of them turned out to be either inaccurate or too expensive to calculate precisely.

This study is intended to implement accurate yet inexpensive analysis method for general cranktrain consisting of crankshaft, piston, flywheel, connecting rod, pulley and torsional damper. Unlike other preceding methods[5), 9), 10), 14)], proposed approach doesn't involve separate rigid dynamic analysis and stress analysis, however, single dynamic stress analysis is carried out to consider vibration, periodic loading and stress at the same time.

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2. ENGINE AND CRANKSHAFT

The study has been done with worldwide popular 1,500 hp two-stroke diesel engine of eight-cylinder, Vee type and turbocharging. The engine has been in service such as offshore drilling, stationary power generation and locomotive, worldwide for a while without any major problems so that stress values from the analysis should be at sound level not likely to have any potential fracture or failure. Since the engine is Vee type, loading is quite complicated, which makes multi-body dynamic analysis more evitable to have accurate stress values.

The crankshaft is wholly made out of forged steel having its all fillets cold-rolled to ensure enhanced fatigue characteristics. Connecting rods are connected to crankpin by forked formation. To prevent excessive torsional vibration, viscous torsional damper is attached at free end. Main bearings and journals are numbered starting from free-end side to flywheel, 1 to 5 and 1 to 4 respectively. Firing order is 1-5-3-7-4-8-2-6 and the crankshaft rotates counterclockwise looking at flywheel.

3. MULTI-BODY DYNAMIC ANALYSIS

3.1 Finite element model

Dynamic cranktrain model shown in Fig. 1 consists of crankshaft, flywheel, piston, connecting rod, torsioanl damper, pulley and journal bearings. Since the crankshaft is the first concern, it is modeled as deformable part and the rest of the parts are all rigid. The crankshaft was meshed using tetra element with fine meshing on concerning areas such as crankpin fillet and journal fillet. Concentrated point mass is put in application to model flywheel, eight pistons, torsional damper and pulley. Each point mass is connected to the crankshaft by rigid kinematic link. Connecting rods are also modeled using point mass which is split and assigned in both piston center and crankpin center. Torsional damper is simulated by dashpot element with empirical damping coefficient.

Journal bearing is one of the most difficult components to model precisely. Accuracy and expense were taken into consideration at the same time to model journal bearing and, contact interaction, consequencly, with exponential pressure-overclosure relationship was used to simulate it. This approach is somewhat similar to the method of putting spring-damper element around journal bearing suggested by a couple of people including Reiner[11)]. Putting real radial clearance between contact surfaces, it is possible to allow six degrees of freedom of journal motion resulting in asymmetric pressure distribution along journal surface. In other words, journal misalignment can be effectively and inexpensively simulated in this way.

3.2 Boundary condition

Gas pressure is major excitation force in cranktrain dynamic analysis. The pressures for each cylinder should be given in form of periodic function to consider torsional vibration which is probably the major cause for crankshaft failure. In particular, trigonometric form of gas pressure by Fourier series expansion is put into practice[1), 2)]. Firstly, real gas pressure data was acquired on normally operational engine for eight different speed levels so called notch(Fig. 2). Secondly, each pressure data was converted into Fourier series in trigonometric form of Eq. (1). Thirdly, coefficients of the series were put in analysis as load boundary condition for each cylinder. In this way, complex kinematic calculation which is especially tricky for a Vee type engine, can be eliminated in order to calculate boundary conditions.

$$f(t) = A_0 + \sum_{n=1}^{N} \left\{ A_n \cos(n\omega_0 t) + B_n \sin(n\omega_0 t) \right\} \qquad (1)$$

Even though an engine can run at any speed level, consideration of only a couple of discrete levels is proper for most of medium-speed diesel engines not having frequent rpm changes but keeping constant speed over majority of service period. For example, engine usually runs at maximum rated speed for 24 hours and 365 days continuously in case of stationary power generation. However, it might be necessary to gather much more pressure data with respect to different rpm if the same strategy is applied to automotive engines with frequent rpm changes.

While crank journals are free to move and rotate with respect to any axes, rigid bearing shells are fully constrained in all directions. Therefore, fully constrained bearing shell and contact interaction between journal and shell surfaces will finally define quasi-hydrodynamic motion of crank journals.

To simulate external torque load carried out by flywheel giving regulated rotational speed, the torque given by all gas forces should be balanced by exactly the same amount of torque, otherwise, constant growth or reduction of speed might happen. Under complex multi-body dynamic analysis like this, it is practically impossible to predict exact resulting torque value generated by all of gas pressures, and apply it on flywheel. Given gas pressure, corresponding constant rotational speed was used as the external loading instead of specific torque values.

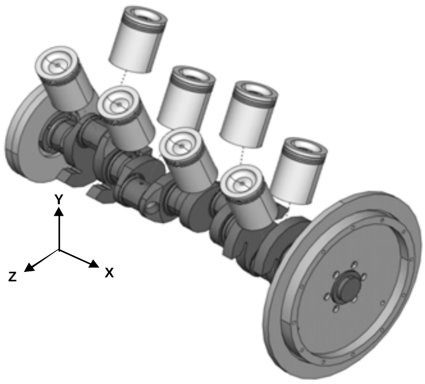

Fig. 1 Multi-body dynamic model of a cranktrain

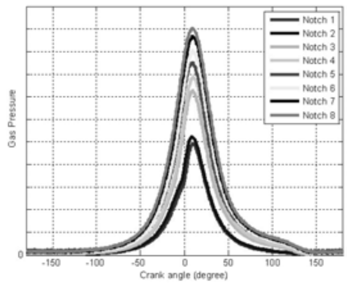

Fig. 2 Gas pressure for eight different notches

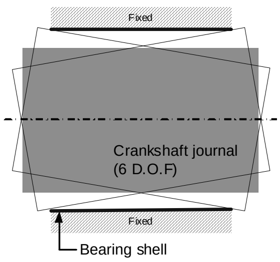

Fig. 3 Journal bearing with allowable misalignments

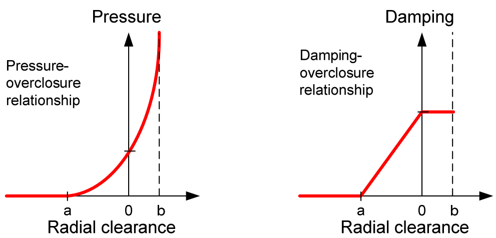

Fig. 4 Oil-film pressure and damping

Oil film around crank journal generates pressure and also damping at the same time. The pressure usually increases very stiff when eccentricity ratio gets near one. To simulate it, contact interaction with nonlinear pressure of exponential pressure-overclosure relationship is implemented between journal surface and bearing shell. Zero radial clearance pressure should be given either empirically or by either separate hydrodynamic analysis. The resulting eccentricity should be within allowable radial clearance at any moment.

3.3 Analysis result

Totally eight analyses of eight different speeds were carried out independently. Firstly, main bearing reaction forces were investigated for each main bearing with respect to vertical and horizontal axes (Fig. 5 to Fig. 9). The reaction forces used to be observed in order to be applied in boundary condition especially when only one or part of throw was subject to finite element analysis[10]. Law and Haddock[3] suggested transmissibility influence coefficients to calculate main bearing reaction forces, however, this study revealed that the coefficients were not accurate enough to get the forces, and multi-body dynamic analysis should be executed to have precise reaction forces influenced by both torsional vibration and gas pressure. One interesting finding is that middle bearings have relatively higher reaction forces in magnitude than the first and the last main bearings right next to pulley and flywheel respectively. In particular, the center main bearing had the largest vertical and horizontal reaction forces among five bearings. Therefore, the fillets on the center journal should be given careful attention when they are evaluated in terms of stress or fatigue due to the highest forces applied.

Since torsional vibration is known to be a dominant cause of crankshaft failure[1], it should be examined thoroughly. Out of the multi-body dynamic analysis, torsional amplitude at significant nodes was extracted as in Fig. 10 to Fig. 15. Assuming zero torsional vibration at flywheel due to relatively large mass moment of inertia, center node of pulley which is the farthest part from flywheel, had the largest torsional amplitude of +0.34° to -0.10°. The torsional amplitude of each crankpin, that is, torsional vibration decreased as the location became closer to flywheel.

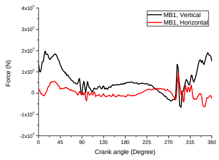

Fig. 5 Reaction force, MB1

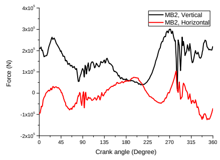

Fig. 6 Reaction force, MB2

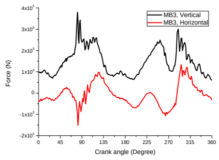

Fig. 7 Reaction force, MB3

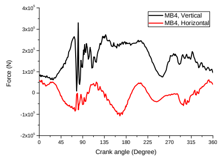

Fig. 8 Reaction force, MB4

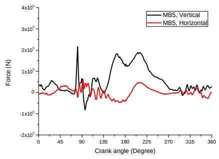

Fig. 9 Reaction force, MB5

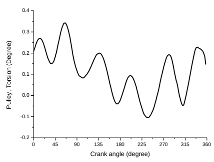

Fig. 10 Torsional amp., Pulley

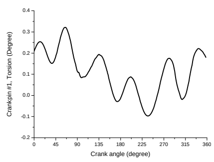

Fig. 11 Torsional amp., Pin #1

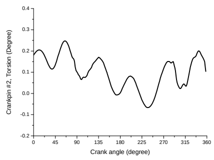

Fig. 12 Torsional amp., Pin #2

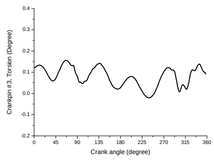

Fig. 13 Torsional amp., Pin #3

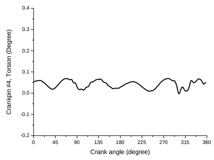

Fig. 14 Torsional amp., Pin #4

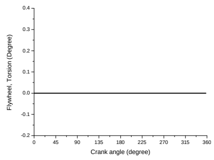

Fig. 15 Torsional amp., Flywheel

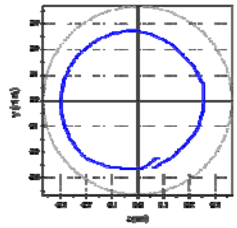

Fig. 16 Journal #1 center orbit

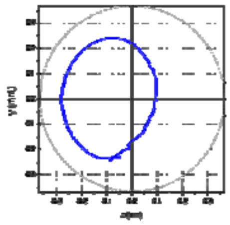

Fig. 17 Journal #2 center orbit

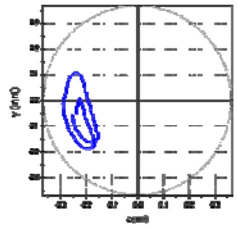

Fig. 18 Journal #3 center orbit

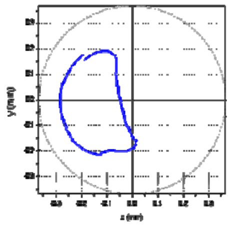

Fig. 19 Journal #4 center orbit

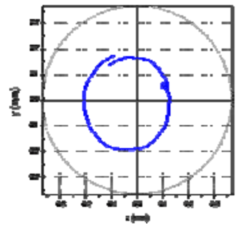

Fig. 20 Journal #5 center orbit

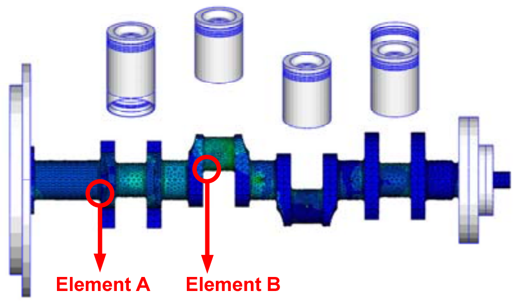

Fig. 21 Two locations of stress history

Journal center orbits for one engine cycle were plotted to see characteristics of five journal bearings(Fig. 16 to Fig. 20). In comparison to journal #3, the middle journal, the other four orbits showed symmetric tendency of circular orbit. Though they had different radiuses of orbiting circle, the first(Fig. 16) and the last(Fig. 20) journals showed quite circular orbit having center of orbit aligned with rotational axis. On the contrary, the middle journal(Fig. 18) showed very eccentric motion possibly due to large horizontal and vertical reaction forces applied on it. After taking the reaction forces and journal center orbits in consideration, it can be stated that the middle journal functions as a pivot where left and right half of the crankshaft wobble around.

Equivalent stress is investigated for all elements of the crankshaft over one engine cycle. One of the greatest advantages of multi-body dynamic analysis with full crankshaft is that it is possible to examine stress on any locations of the crankshaft. Quarter-throw[12], half-throw[6] or one throw[10] analysis are all limited to see only one particular throw though it is practically very difficult to pinpoint which throw is the most critically loaded. According to this study, the middle journal was the most highly stressed and fillet on fifth journal right next to flywheel was secondly stressed. High stress on the middle journal fillets may have its cause based on large reaction forces on it. Fig. 22 and Fig 23 are stress history of two elements on journal fillet and crankpin fillet respectively depicted in Fig. 21. Generally, overall stress increased proportionally to engine rpm, however, higher stress at lower rpm was also found occasionally. Especially, in terms of the highest stress, 830 rpm for journal fillet and 651 rpm for crankpin fillet showed the greatest values over higher speed levels. As a matter of fact, just a small amount of difference in stress was found among different engine rpm once the engine ran over 489 rpm.

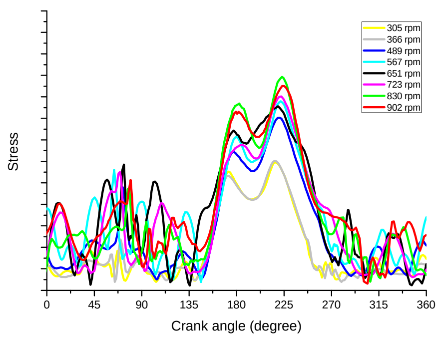

Fig. 22 Stress history of element A on crank journal

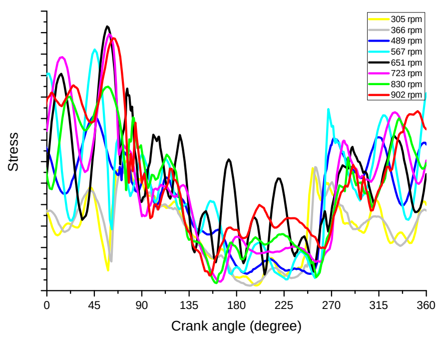

Fig. 23 Stress history of element B on crankpin

4. CONCLUSION

Comprehensive multi-body dynamic stress analysis was implemented applied to cranktrain of medium-speed diesel engine of 1,500 horsepower. Simultaneous consideration of vibration, gas force excitation and quasi-hydrodynamic journal bearing turned out to be inexpensive way to obtain accurate stress values having competitive advantage in terms of expense over separate dynamic and stress analysis. Hydrodynamic journal bearing was efficiently modeled using contact interaction with exponential pressure-overclosure relationship giving significant amount of reduction in computing time over hard contact approach. The analysis results showed that the farthest crank journal from flywheel has the highest torsional vibration and eccentricity. Journal fillet right next to flywheel had the highest stress.

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