Engine Cranktrain System Simulation and Validation

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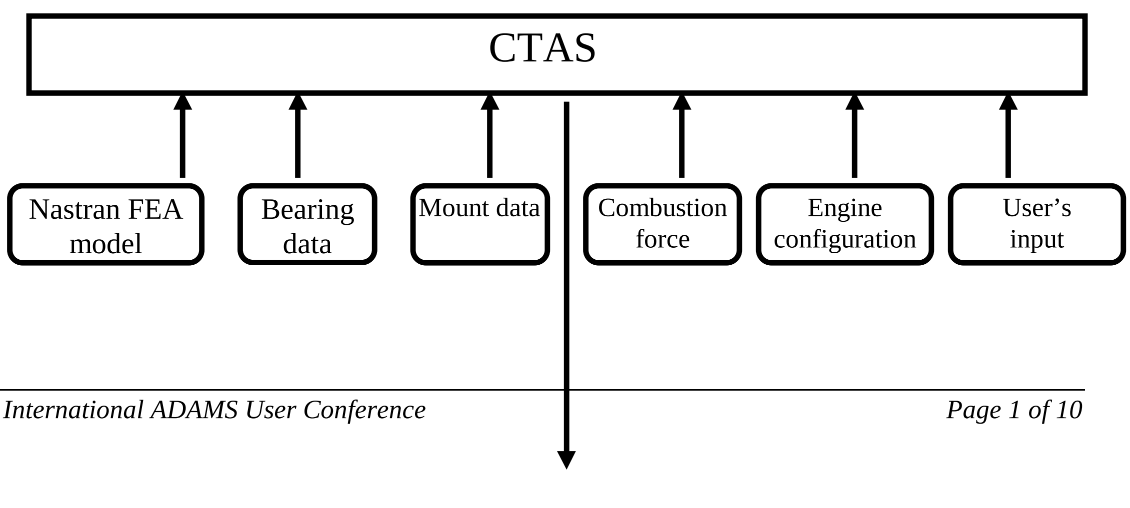
1. Summary

The purpose of this paper is to analyze and validate the dynamic response of engine cranktrain system. The system includes, a flexible cranktrain, a flexible block, six rigid pistons and connecting rods. An ADAMS based engine model was developed to study the dynamic response. Both 1-Dimensional and 3-Dimensional hydrodynamic bearing models were used to investigate the interactive effect between the crankshaft and block. The external excitations were gas pressures applied on the piston; A prescribed Motion applied on Flywheel were used for steady state engine simulation. For an example 3.8L-V6 engine considered, the forced responses from engine main bearings were examined. Experimental measurements from the test engine were used to confirm the theoretical prediction. The measurements of the block surface velocity and acceleration are in good agreement with the theoretical predications.

2. Introduction

A cranktrain system preprocessor tool developed by Ford in-house (see Appendix E. CTAS users manual) is used to generate input data for rigid or flexible engine models. User can select a configuration among four different combinations of crankshaft, block, and connecting rod in terms of flexibility of each component. For each flexible component, a Nastran FEA super-element model is first created and analyzed, the eigen values and eigen vectors are then imported to ADAMS to generate the flexible component. A template generated from CTAS contains all the information such as number of pistons, bank angle, bearing parameters, parts location, orientation, mass properties etc. ADAMS will create an engine model based on the CTAS template (see flow chart in Figure 1).

In this study, three engine models were created from CTAS engine templates. Model A included all 6 pistons and connecting rods without considering piston rings. Model B used 6 bob weights (as a replacement for the piston assemblies). Model C used 3 pistons and 3 bob weights. The engine models were driven with an external motor at a selected speed. The rod and piston were assumed rigid, and block and crankshaft were flexible. The output (acceleration and velocity) from bearing caps was employed to validate the engine models.



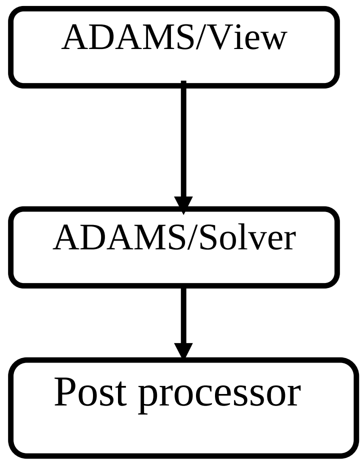


Figure 1. System flow chart

3. Cranktrain system definition

3.1 Models Description

a). *Engine model A with rod and pistons*

Engine block and crankshaft FEA model used in this study were validated previously in terms of natural frequency (it is not clear whether the two assemblies are validated in terms of eigen vector). In the crankshaft assembly, the dynamic damper and flex-plate were assumed rigid with an offset from crank axis as a replacement of unbalanced mass. Between piston and block, a cylindrical joint was used as a constraint ignoring the effect of piston rings and piston clearance. Universal joints were used at both ends of rigid connecting rod.

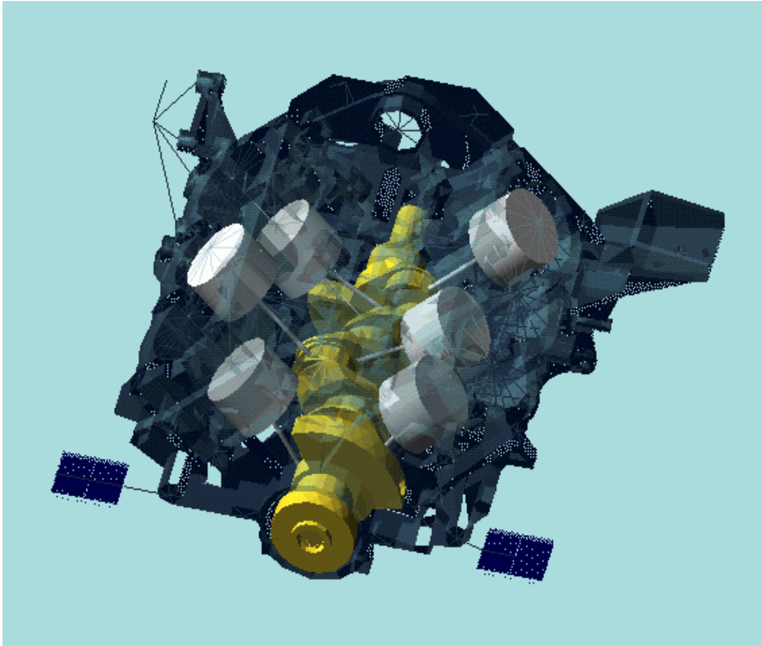


Figure 2. Engine model A: Full engine cranktrain assembly

b). *Engine model B: engine block, crankshaft, and six bob-weights assembly*
In bob weights model configuration, bob weights were used to represent the same effective rotational and translational inertia of piston-rod assemblies.

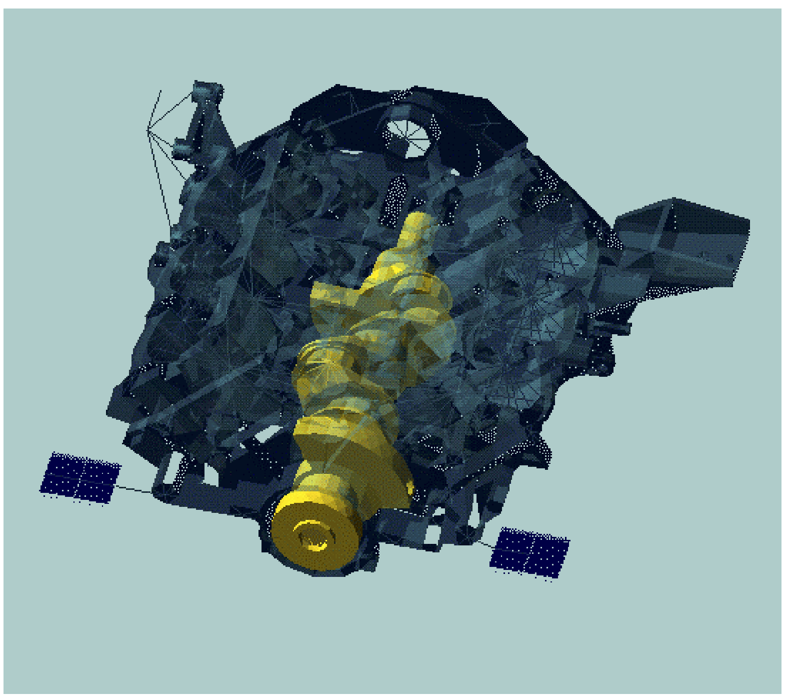


Figure 3. Engine model B with 6 bob weights

c). *Engine model C: engine block, crankshaft, 3 pistons/con rods, and 3 bob-weight assembly*
The 3 piston and rod assemblies at the right bank of the engine were replaced with three bob weights, while the 3 piston and rod assemblies at the left bank of the engine kept the same as model A.

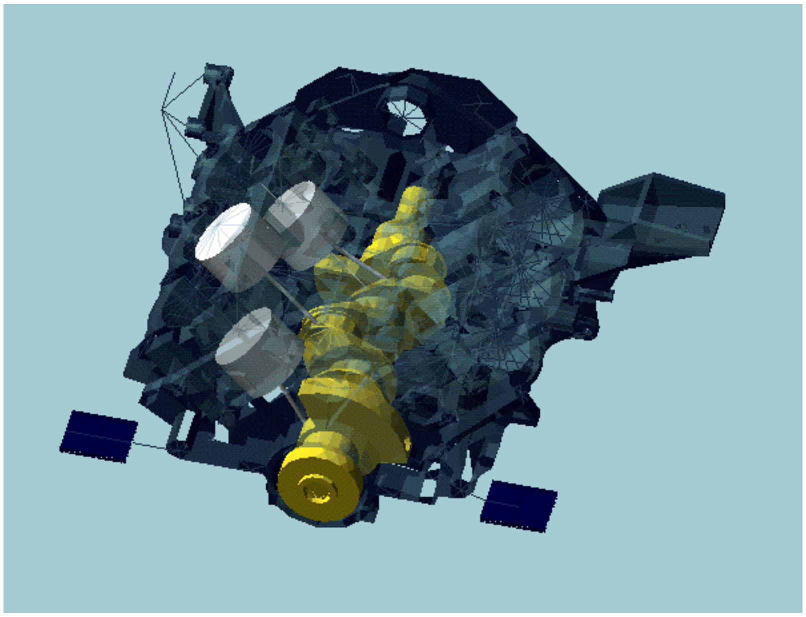


Figure 4. Engine model C with 3 bob weights and 3 pistons

3.2 Steady-State Condition Set-up
The simulations were performed at a constant speed driven by ADAMS MOTION function. The crankshaft rotate from zero speed to a specified RPM within 0.1 second. Steady-state runs of different RPM are listed as below.

a). *Engine model A with rod and pistons*
two simulations at 3000RPM and 4400 RPM respectively.
b). *Engine model B with bob weights*
two simulations at 3000RPM and 5800 RPM respectively.

3.3 Engine Mounts
Since engine mount data was not available, the stiffness and damping were assumed constant. Simple ADAMS bushing models were used for all the three cases. Two engine mounts were used in front (mount #1 and #2), and two transmission mounts were used in rear.

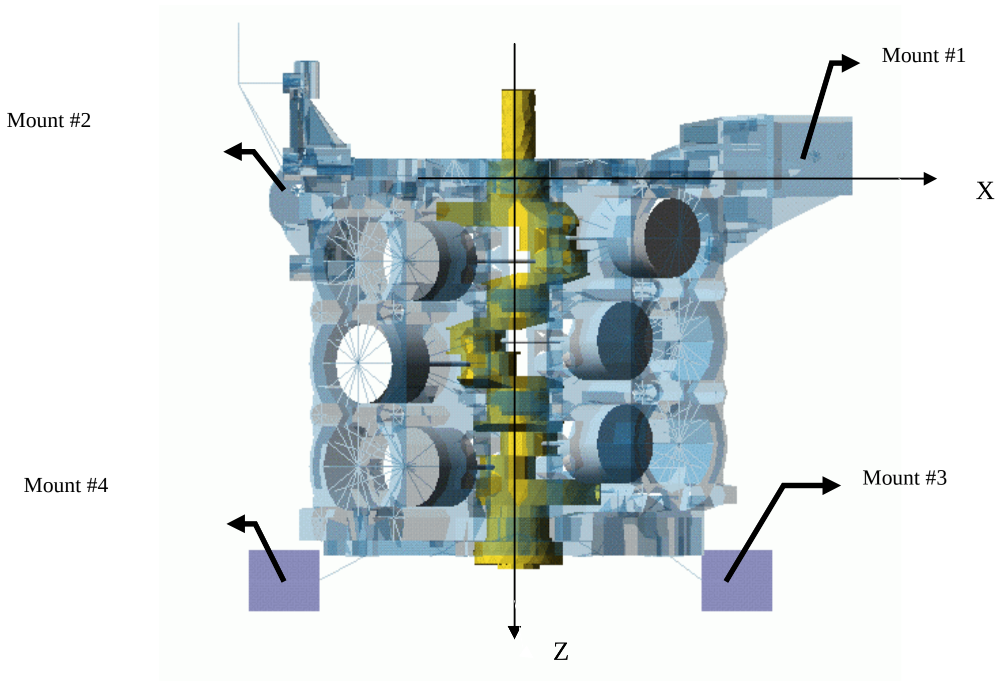


Figure 5. Engine mount number and locations (Top view)

3.4 Coordinate System

Global coordinate:

Z-axis parallel to the axis of the crankshaft, positive direction is from bearing #1 to #4.

Y-axis vertical from the ground, positive direction is up.

X-axis lateral direction is perpendicular to the crankshaft in horizontal plane. Positive direction is from the front to the back of the vehicle.

Local coordinate:

1. Crankshaft: Origin is at the center of main bearing #1. Z-axis is parallel to the axis of the crankshaft, positive direction is from bearing #1 to #4. Y-axis is vertical from the ground, positive direction is up. X-axis is lateral direction is perpendicular to the crankshaft in horizontal plane.
2. Block: Origin is at the center of main bearing #1. Orientation is the same as crankshaft.
3. Rod: Origin is at the center of crank pin. Z-axis is parallel to the axis of the crankshaft. Y-axis is point from the center of crank pin to the center of wrist pin.

3.5 1-D Bearing

1-D hydrodynamic bearing model used in this study was obtained from The Institute of Maschinenelemente, Germany. It provides three force components with zero moment output.

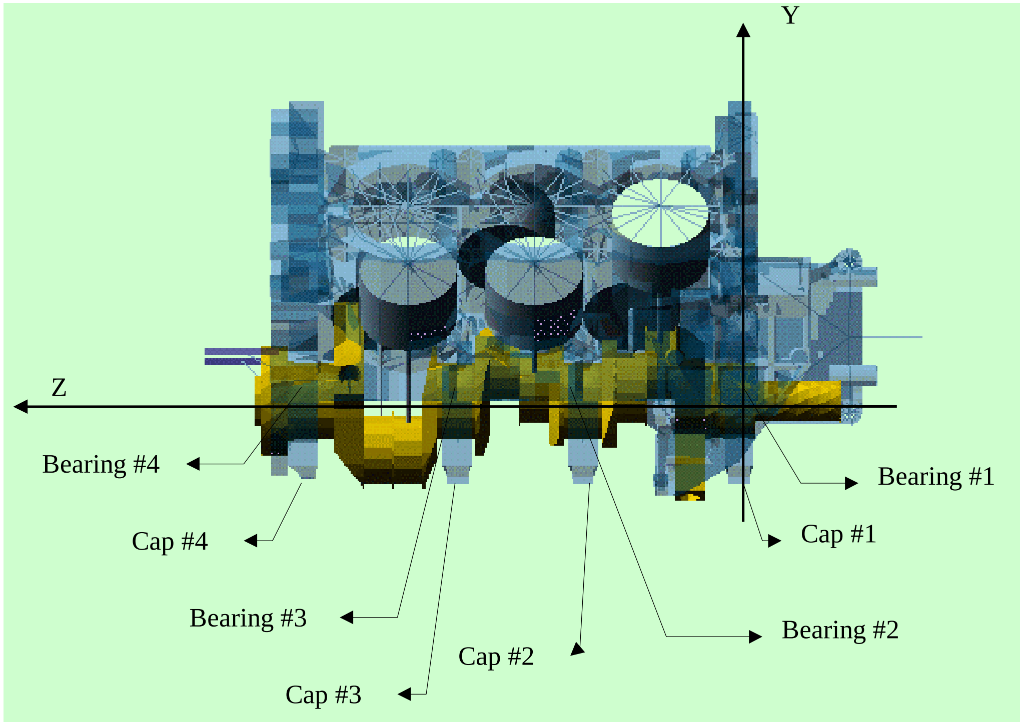


Figure 6. Bearing and Cap locations

3.6 3D bearing
3D hydrodynamic bearing model used in this study was obtained from professor John F. Booker. It is based on the finite element method. The bearing mesh was generated from Matlab interface. The bearing dimension and oil properties are described as follow:
Number of triangular elements = 480
Number of Nodes = 360

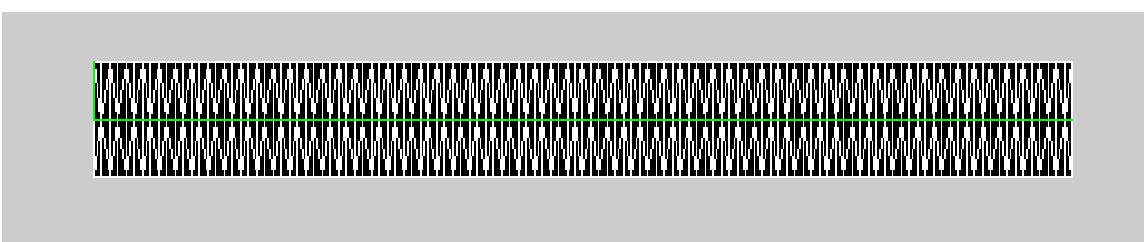


Figure 7. 3D bearing finite element mesh

3.7 Flexible Crankshaft Parameters
Flexible crankshaft was once validated with natural frequencies consistently lower than measurement (about 5 % lower). In this study, flexible plate, dynamic damper, and pulley were modeled as lumped mass with a small offset from crankshaft axis.

4. Experimental test stand setup

Engine block and crankshaft assemblies used in this study were originally in the previous study. Prior to the crank motoring test, the engine block, crankshaft, flexible plate and dynamic damper were sent for reconditioning. The crankshaft assembly was balanced as a unit. New pistons connecting rod assembly were used. Bob weights were used to represent the same effective rotational mass of piston assemblies. Engine block was mounted to the ground at four locations, two on pulley side (standard locations) using standard engine bracket and the others at the transmission side with custom L-shape brackets (Figure 8).

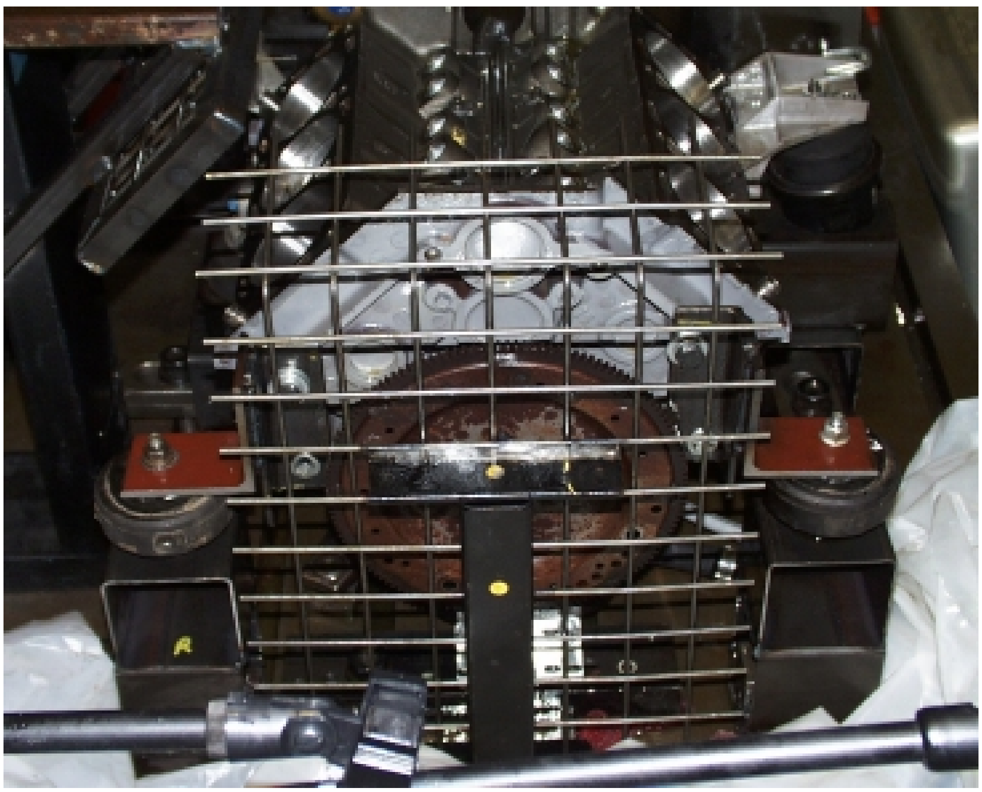


Figure 8. Two modified mounts at the transmission site

An AC motor with belt and pulley arrangement was employed to drive the engine (Figure 9). The pulley ratio was 1.2:1 where diameter of the motor pulley (D=9 in) is larger than the crank (D=7.5 in). By having different size pulleys, the motor vibration orders did not contaminate the engine orders. The distance from pulley to pulley was 22 in. The motor has 20HP at rated speed of 1800 RPM with a maximum speed at no load of 6500 RPM.

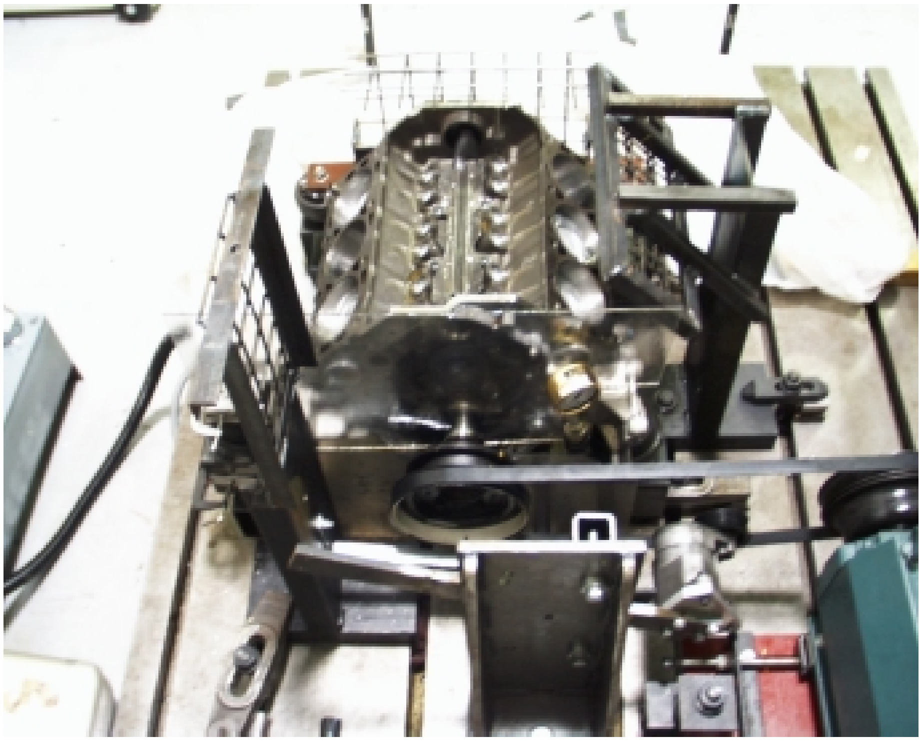


Figure 9. The crank-motoring set-up

The measurements were performed as a continuous speed sweep and several steady state speeds.
A. *Set-up A with Rod/Pin/Piston Assemblies*
   0 RPM - 4500 RPM speed sweep with 3 repeat runs.
B. *Set-up B with bob weights*
   0 RPM - 6000 RPM speed sweep with 3 repeat runs.
C. *Set-up C with 3 bob weights and 3 pistons*
   0 RPM - 5500 RPM speed sweep with 3 repeat runs.

5. Simulation results

Appendix A shows the plots of cap velocity and acceleration of engine model A (6 piston model at speed 4400 rpm). Figure A-1 shows the cap velocity, and Figure A-2 shows the cap acceleration. There are two curves in each plot. Red color curve is the measurement result, and blue color curve is the analysis result.

6. Discussions

The velocity responses on the four bearing caps have good agreement with the measurements. The results on bearing cap3 and cap4 matched better than the results on bearing cap1 and cap2. One reason for that is lack of some the data including engine mount property. Other factors which may contribute to the results are itemized below.

(1) Flywheel

In the analysis, a rigid flywheel was used attaching to the rear end of the flexible crankshaft. That simplification did not affect the torsional vibration, but it will has influences to the bending and axial translation deformations at bearing cap3 and cap4 due to lack of some mode of the flexible plate. A relatively detailed finite element flexible plate model is desired in order to get better results at the rear end of the engine.

(2) Rear engine mount bracket

Another contribution to the cap3 and cap4 results come from the simplified L-shaped engine mount brackets (no data or FEA model available) at the transmission side. Current rear brackets are a simple beam element in block FEA model. The stiffness was calculated based on the size of the rectangle cross section which is a rough estimation.

(3) Frequency dependent engine mount

Engine mounts are assumed as linear spring and damper elements in this analysis. Within a frequency range 0~500 Hz, a typical engine mount stiffness can vary from 100 N/mm to 500 N/mm. A linear spring mount is far from enough when simulating a engine vibration. The MDI frequency dependent bushing element is recommended for future study.

(4) Piston-cylinder contact

A cylindrical joint was used in the simulation between piston and cylinder. A damping coefficient of 0.5 along piston axis was assumed. To achieved a more accurate result, loads from detailed piston slap analysis need to be imported into this model.

(5) 3-D bearing

3-D bearing has more modeling detail including cavitation pressure, boundary conditions, and moment output, the results of 3-D bearing model is better than that of 1-D bearing, especially in high frequency domain. The setback is that it takes much longer CPU time.

Appendix. Cap velocity and acceleration of engine model A(6 piston) at speed 4400 rpm.

Figure A-1. Bearing cap velocity in y direction

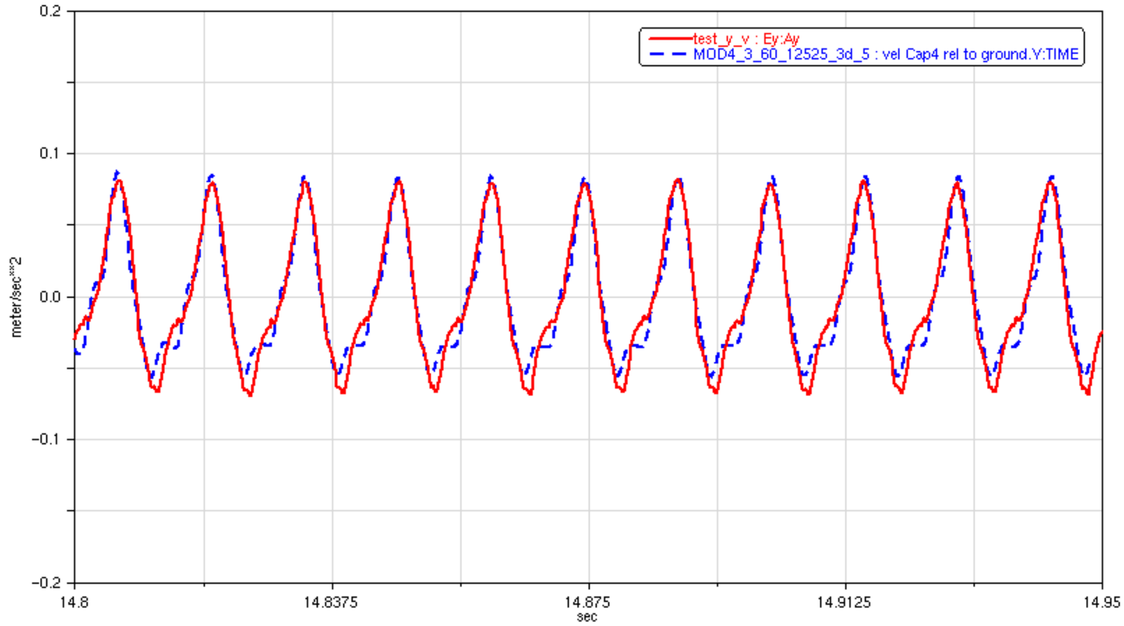


Figure A-2. Bearing cap acceleration in y direction