

Modeling and Analysis of Powertrain NVH with Focus on Growl Noise

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ABSTRACT

Superior NVH performance is a key focus in the development of new powertrains. In recent years, computer simulations have gained an increasing role in the design, development, and optimization of powertrain NVH at component and system levels. This paper presents the results of a study carried out on a 4-cylinder in-line spark-ignition engine with focus on growl noise. Growl is a low frequency noise (300-700 Hz) which is primarily perceived at moderate engine speeds (2000-3000 rpm) and light to moderate throttle tip-ins. For this purpose, a coupled and fully flexible multi-body dynamics model of the powertrain was developed. Structural components were reduced using component mode synthesis and used to determine dynamics loads at various engine speeds and loading conditions. A comparative NVH assessment of various crankshaft designs, engine configurations, and in- cylinder gas pressures was carried out. The main results include the crankshaft front-end and rear-end vibrations, bearing caps accelerations spectrum, and structure surface velocity levels in octave and 3rd octave bands. The correlation with experimental data was used to validate the analytical model. The analysis shows that a stiffer crankshaft results in a reduction of forced excitation transmitted to the bottom-end structure. The bearing beam stiffener also reduces bearing cap accelerations significantly. Both structural enhancements result in dramatically reduced growl noise.

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INTRODUCTION

Various parameters and design characteristics affect the powertrain sound quality. Cylinder gas pressure is the main excitation source of the powertrain systems. High peak cylinder gas pressure generates high excitation forces transmitted to the engine block structure resulting in high noise level. Cylinder to cylinder variations of gas pressure result in increased half order excitations which give a perception of undesirable sound quality. High pressure rise rates cause rapid variation of forces and impacts in joints with clearances. It is often necessary to reduce pressure rise rates, and thus sacrifice power and fuel economy as a compromise to maintain NVH performance within acceptable limits. The optimization of spark timing, exhaust gas recirculation (EGR) flow rate, and charge motion control valve position are widely known as means of controlling the peak, uniformity, and rise rate of cylinder pressure.

Lack of stiffness of the crankshaft supporting structure could be a significant contributing factor to the undesirable sound quality. Bearing beams with ferrous caps should provide good NVH characteristics due to high stiffness between bearing caps and isolation from the outside wall of the block. The bearing beam needs to be stiff in bending and prevent cap twisting and is helpful with either deep skirt or short skirt engines. Using cross bolted caps with stiff side joints is another commonly used solution predominantly for larger engines.

This paper presents an analytical investigation of a 4-cylinder in-line spark ignition engine with focus on growl noise. For this purpose a multibody dynamics model of the full powertrain was developed. The model consists of fully meshed powertrain structural components connected with specific joints and elasto-hydrodynamic bearings. The flexible bodies are reduced to a limited number of degrees of freedom (DOF) using component mode synthesis. A comparative assessment of various gas pressure calibrations,

crankshaft designs, and engine bottom end configurations was carried out.

MODEL DESCRIPTION

Figure 1 shows the finite element model of the I-4 powertrain including all relevant structural components and accessories. The powertrain structure consists of an aluminum engine block with separate balancer housing, cylinder head, ladder frame, front cover and intake manifold meshed using solid elements. The stamped metal-plastic-metal sandwich oil pan and cast iron exhaust manifold are meshed using shell elements. Valve cover to cylinder head connection includes a high damping rubber deck which is commonly used to decouple vibration path from the cover. Stiffness of the rubber deck elements is based on the rubber deck geometry and material properties and it is defined in Nastran using CELAS elements at bolt locations. The transmission housing is meshed using solid elements while the internal parts such as shafts, gears, and bearings are simplified as beam and concentrated mass elements. Engine mount brackets are fully meshed and auxiliaries are included in a simplified form to account for the correct distribution of mass and inertia.

Three different configurations of the bottom end were investigated as shown in Figure 2. The base engine consists of an aluminum ladder frame and cast iron bearing caps. Bearing caps are not connected to the ladder frame and, therefore, they have increased flexibility which could lead to an increased level of excitation transmitted to the bottom end. The second configuration consists of an aluminum bearing beam connecting main bearings 2, 3, and 4. In the third configuration, the bearing beam connects all main bearing caps. The bearing beam will increase the local stiffness of the bearing caps.

The crankshaft assembly consists of the crankshaft itself, the torsional vibration damper (TVD), and the primary part of the torque converter (TC). The crankshaft assembly is represented as a single body in Excite (Figure 3). The primary part of the torque converter consists of a flexplate, housing, and pump. Since the current model is used to simulate the dynamometer configuration with the converter clutch in lock in position, the clutch and pistons are also included into the TC primary part. The secondary part is represented by the turbine. Half of the oil mass and inertia is included in the primary part while the other half is attached to the secondary part. The stator mass and inertia are part of the transmission shaft attached to the transmission. Three crankshaft designs are investigated. The baseline crankshaft has the main journal diameter of 52 mm and pin diameter of 48 mm. The second design has oversized pins while the third design has oversized pins and main journals.

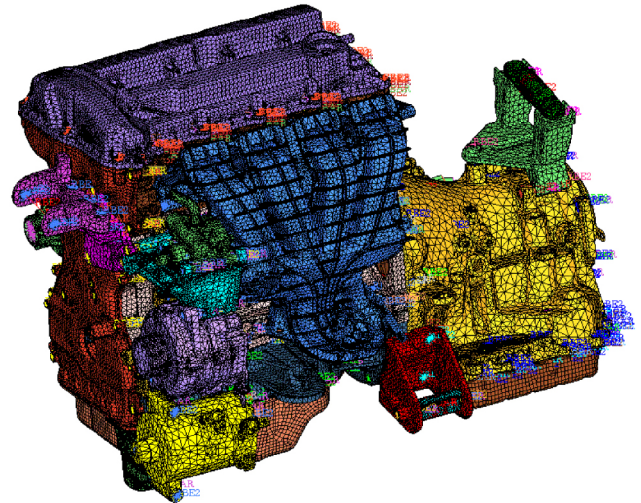


Figure 1. Finite Element mesh of the I-4 powertrain

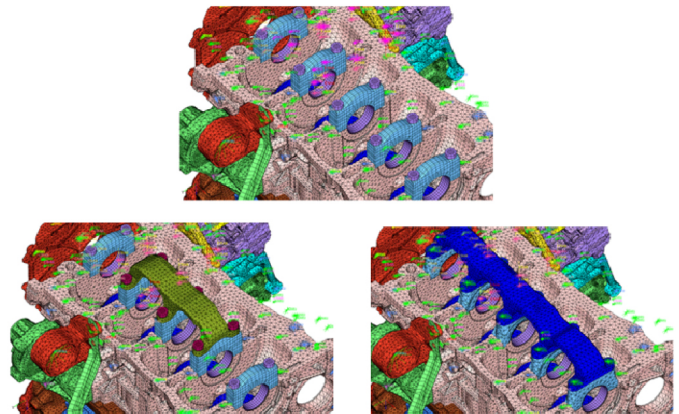


Figure 2. Engine bottom end design (a) free cast iron bearing caps, (b) aluminum bearing beams connecting main bearings 2, 3, and 4, (c) aluminum bearing beam connecting all bearing caps

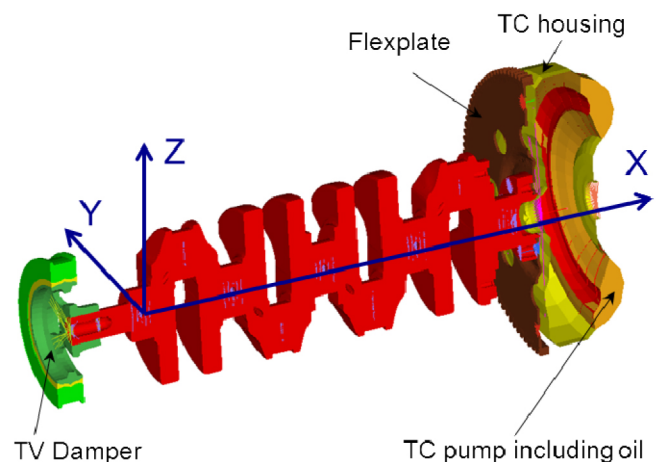


Figure 3. Crankshaft, TV damper, torque converter assembly

EXCITATION LOADS

The main excitation sources with a major impact on NVH within the powertrain are combustion gas loads, piston to liner contact forces, and valve train and timing drive loads. In the current analysis, measured gas pressure traces are used for each cylinder to take into account the cylinder-to-cylinder variation (Figure 4). Gas forces are applied at the connecting rod small end while the reaction forces are applied to the combustion chamber. Forces are calculated automatically based on the gas pressure, cranktrain kinematics, and geometry.

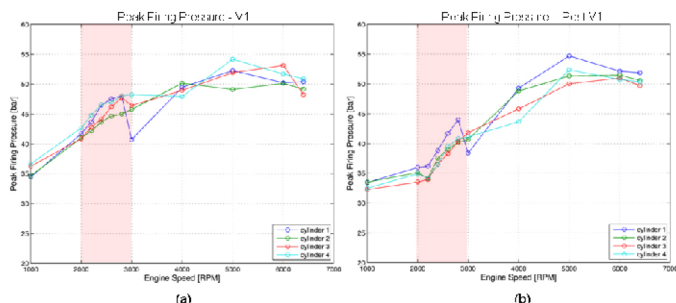


Figure 4. Cylinder gas peak firing pressure at part load:
(a) standard calibration, (b) spark retard calibration

The piston impact excitations are derived for separate calculations using a single piston model in EXCITE Piston and Rings software. The model considers only kinematic excitation due to the cranktrain motion. Forces are included in the thrust and anti-thrust planes of the liner. The calculation is carried out considering forces due to gas load, inertia, friction, and elasticity of the piston as well as deformed contour of the liner. The impact forces are due to piston oscillations, tilting, and radial motion in the clearance between piston and liner. The impact forces are applied at thrust and anti-thrust sides over the height of each cylinder liner.

EXCITE enables the calculation of the coupled vibrations of the crankshaft in torsional, bending, and axial directions due to gas and inertia forces. The gyroscopic effect of the torque converter and counterweights are considered since rotating crankshaft will have a deformed shape at each time instant. Forces are transferred to the crankcase through bearings including oil film based on Elasto-hydrodynamic (EHD) theory and housing flexibility. The current investigation is performed using gas, inertia loads, and piston - liner impact but neglects the influence of timing drive system. This is acceptable for frequencies below 1 kHz [1].

MODAL ANALYSIS - DEEP DIVE INTO GROWL CONTRIBUTORS

Multiple powertrain local modes have frequencies in the growl range, thus, force response analysis based on multi-body dynamic simulations is necessary to investigate interactions among components. The powertrain mount brackets natural frequencies are low. Brackets have to be designed such that their natural frequency is higher than the dominant engine orders.

Modal analysis of the crankshaft assembly shows two bending modes around 400 Hz dominated by the damper wobbling (modes 13 and 14) as shown in Figure 5. These modes were initially suspected as root cause for the crankshaft axial resonance peak observed experimentally. However, analysis results will show that damper wobbling modes have little contribution to the crankshaft axial vibrations. Instead, the first, second, and higher order bending modes are responsible for the crankshaft front end vibrations. The redesign of the crankshaft front end will not eliminate the 400 Hz resonance at the bearing caps.

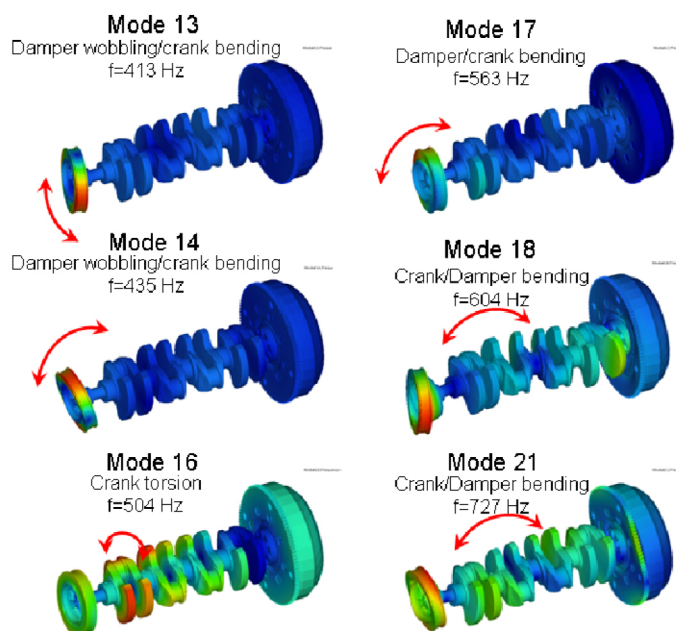


Figure 5. Free normal modes of the crankshaft assembly

Bending and torsional stiffness of the crankshaft increases if larger diameter mains or pins are used. It is expected that a stiffer crankshaft will have higher natural frequencies. In the case of automatic transmission, the crankshaft rear end flange is attached to the flexplate characterized by a low bending stiffness. Larger mains and pins results in 6% to 8% change of the natural frequencies of the crankshaft assembly.