SIMULIA

Understanding the Effects of Nonlinear Preloads on Engine System Dynamic Response using Abaqus/Standard

Abaqus Technology Brief

Summary

It is well understood that stress stiffening affects the dynamic response of a structure. Just as tightening a string changes the tones produced by a violin, tightening head bolts changes the dynamic response of an engine block assembly. While fundamental to nonlinear static stress analysis, stress stiffening effects are most often ignored when evaluating the steady-state dynamic response of engine assemblies.

Using Abaqus/Standard, this Technology Brief highlights the impact of various bolt loading assumptions on the flexibility and steady-state dynamic response of an engine head and block assembly. Including assembly load effects in the simulation workflow allows for more accurate prediction of system-level performance, such as engine radiated noise.

Background

The finite element method (FEM) is broadly used in the automotive industry for structural assessments across the entire design life of a vehicle. Prior to 1990, the bulk of powertrain finite element (FE) analyses utilized linear assumptions, which limited the ability to capture key physical features such as contact, gasket nonlinearity, advanced material laws, and nonlinear geometric effects.

Since the mid-1990s, there has been a progressive adoption of nonlinear analysis techniques within the engine and powertrain sealing & durability community. Today, the vast majority of major automotive manufacturers and engine suppliers utilize, to varying degrees, nonlinear assumptions within their structural FEM workflows. The importance of including the effects of nonlinearity in modal response has been acknowledged and reported [1], but for complex geometric structures it is difficult to account for such effects within the confines of a "traditional" linear code.

The simulation presented in this Technology Brief demonstrates the effect of bolt pre-tension on the forced frequency response of an engine assembly. The Abaqus/Standard automatic multi-level substructuring (AMS) eigensolver is used to extract the eigensolution of the engine assembly, which is then subsequently used in the steady-state dynamic analyses.

Analysis Approach

We will consider a V8 engine model representative of a typical sealing & durability analysis — with 3 million elements and 12 million structural degrees of freedom (Figure 1). The head and block are modeled as individual components, and are assembled with bolts. A representation of a gasket, consistent with simplified properties of a multi-layer steel (MLS) construction, lies between the head and block. Contact conditions are enforced between the individual components.

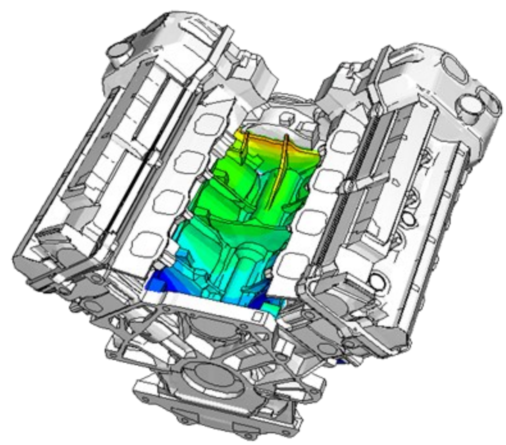

Figure 1: V8 engine assembly model

Key Abaqus/Standard Features and Benefits

- Advanced powertrain modeling capabilities to incorporate key physics such as contact, gasket nonlinearity, large deformation effects and nonlinear material behavior
- Automatic Multi-level Substructuring eigensolver based steady state dynamics for high performance forced response prediction
- Dynamic substructures in Abaqus/Standard account for pre-loading of bolt tension and other initial stress effects

Sources of nonlinearity

There are several potential sources of nonlinearity in the model, any of which could lead to stiffness changes and subsequent changes in the eigenmodes of the structure. For the purposes of this study, aluminum was treated as purely elastic, thus eliminating material plasticity as a source of nonlinearity. The remaining key features that could give rise to stiffness changes are contact and geometric nonlinearity (also commonly referred to as "stress stiffening" or "differential stiffening").

MLS gaskets used for head and block applications tend to have an initially low stiffness, followed by a "bottom out" stiffness which is often an order of magnitude higher. These gaskets can be modeled with varying degrees of complexity within the FE model. In this study, the gasket is simplified with a linear (constant) stiffness to allow for the isolation and investigation of other forms of nonlinearity. Given this assumption, it is expected that after initial assembly loading, the gasket allows the head and block to behave as a singular unit. This source of nonlinearity is thus dominated by contact stiffness — the two entities (head and block) transition from being individual entities to being a single unit (in terms of stiffness). Beyond the initial loading and contact resolution, there should be no progressive stiffening of the gasket (nor significant changes to contact), but there is the possibility of geometric nonlinearity ("stress stiffening") and progressive loading across the gasket interface.

Loading

The bolt assembly loads are defined through pre-tension sections. For all bolts, a series of tension loads is applied in four separate analyses: (i) 0.1% of nominal load ("0.1%"), (ii) 50%, (iii)100%, and (iv) two times the nominal load ("200%"). These four loading conditions will provide a reasonable indicator of the degree of nonlinearity to which a "normal" engine may be subjected. The 0.1% case is artificially low, but allows one to confirm the assumptions of contact and gasket behavior under marginal loads. The other loads more accurately capture realistic behavior and loading profiles. Scatter in pre-load has been demonstrated in tests and has been noted to vary by a factor of two or three, depending on conditions and the bolt tightening processes employed [2,3]. While the 50% and 200% cases represent extreme values, such scatter is reasonable across real-world scenarios and allows one to understand the impact of this range on system response. To understand the effect of geometric nonlinearity, the model was analyzed with and without this capability enabled.

The bolt loads are applied in an initial non-linear static step. For each bolt loading scenario, an eigenvalue extraction was performed to compute all modes up to 9,000 Hz. A frequency response analysis was then performed out to 6,000 Hz. A vertical unit load was defined near the main journal bearing and responses were measured at various points. For the frequency response analysis, modal damping of 2% was used (which is within the range of commonly utilized assumptions for such a system).

Results and Discussion

For all loading conditions and assumptions, two sets of data were evaluated. The first set was associated with the engine cylinders. For each of the eight cylinders, a set of nodes associated with an upper, lower, and middle ring were used for output. The three rings of each cylinder allow us to evaluate "ovalization" and other mode shapes. These modes are of significant interest in assessing dynamic performance, and are also indicators of local stiffening changes. The second set of variables was the acceleration of various nodes on the outside of the cylinder bank. Not only are these accelerations indicators of system stiffness changes, they are also directly associated with radiated noise from the engine.

A set of trends was observed for all bolt loading scenarios: for frequencies below 3,000 Hz, there were negligible results differences, independent of the inclusion geometric nonlinearity. This would suggest that, for this model with simplified linear gasket stiffness, the dominant lower frequency effects are sufficiently captured once contact is established and the gasket transfers load. Above 3,000 Hz, there are noticeable but modest differences between the four loading scenarios when geometric nonlinearity is neglected (Figure 2). However, when geometric nonlinearity is included, there are substantial differences in the various responses. Calculated accelerations are dramatically different above 3,500 Hz (Figure 3).

Focusing on the frequency range of 3,000 – 6,000 Hz, it can be further seen that the four bolt loading conditions tend to form "bands" of increasing and decreasing responses—the 0.1% and 200% loads tend to form minima and/or maxima, while the 50% and 100% loads fall between them respectively. To further evaluate and understand these effects, we focus on the specific relationships between the 100% and 200% loading scenarios.

For both the 100% and 200% load scenarios, there were exactly 200 modes below 6,000 Hz. The previously mentioned engine cylinder node set was used to evaluate specific mode shapes and compare them by means of the Modal Assurance Criteria (MAC) [4]. The function of the MAC is to provide a measure of consistency, or similarity, between estimates of a modal vector. The MAC varies between 1 and 0, with 1 representing perfect consistency and 0 representing no consistency. The MAC in Abaqus can be calculated using an Abaqus/CAE plug-in [5]. Given two output database (`.odb`) files that contain displaced shape field output from frequency extractions (mode shapes), the Abaqus/CAE plug-in computes the MAC and

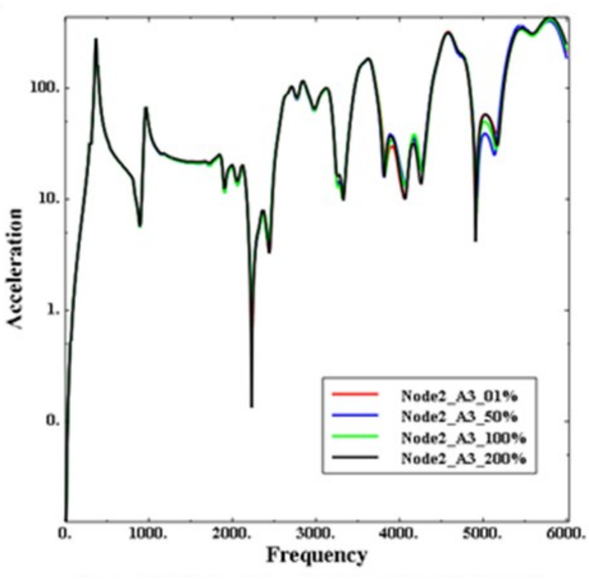

*Figure 2: Vertical acceleration (A3) from 0-6,000 Hz on the engine block. Nonlinear geometric effects **not** considered.*

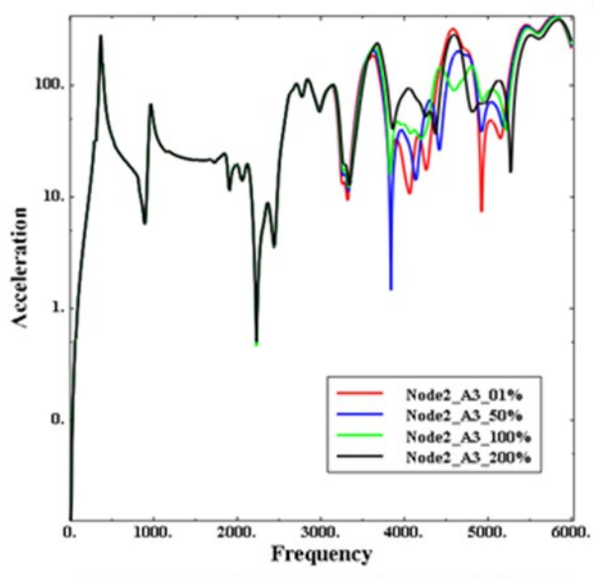

Figure 3: Vertical acceleration (A3) from 0-6,000 Hz on the engine block. Nonlinear geometric effects considered.

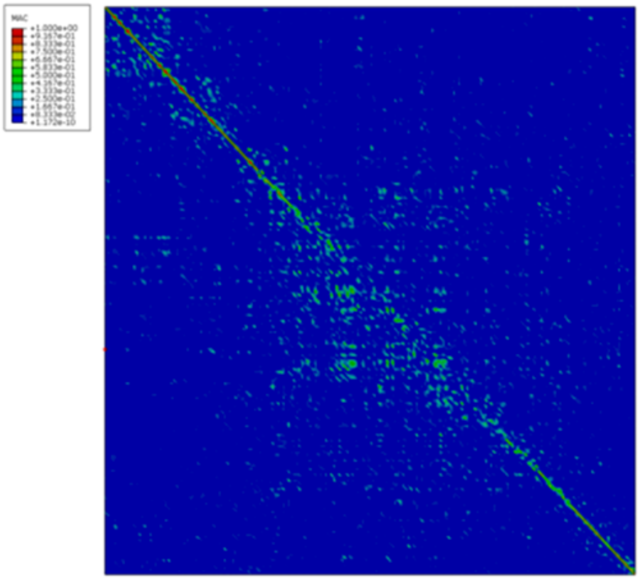

Figure 4: Modal Assurance Criteria, modes 1-200, for 100% vs. 200% bolt load scenario. Nonlinear geometric effects considered.

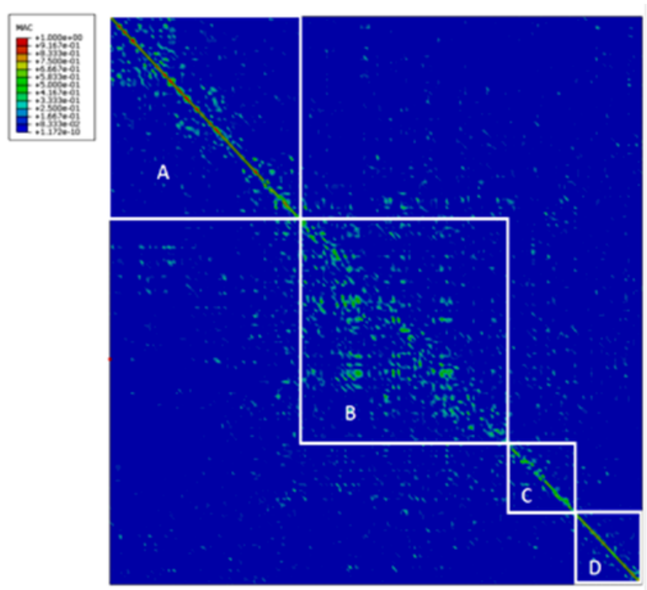

Figure 5: Modal Assurance Criteria, modes 1-200, for 100% vs. 200% bolt load scenario, annotated by regions of interest. Nonlinear geometric effects considered.

displays the results in bar graph or contour plot form.

Evaluating the MAC of the 100% vs. 200% cases (Figure 4), it is clear that there is a region in the middle of the mode range for which the mode shapes are substantially different. Not only is there a lack of diagonal terms, but there are no sets of modes that show significant correlation with any other modes. To further understand the degree of disconnect in the two mode sets, the MAC data can be broken into four regions (Figure 5):

A: 0 – 3,800 Hz: This region shows very high correlation, with the MAC generally > 0.95 and frequencies from the two models consistently within 0.25% of each other. Nearly all of the modes have a MAC of 0.99-1.00 until 3,600 Hz, at which point the MAC values still remain above 0.90 (indicating high correlation).

B: 3,800 – 5,000 Hz: After a quick degradation between 3,600 – 3,900 Hz, the two models show virtually no correlation. The MAC for all modes in this range is generally below 0.6, with only four pairs of modes (out of 82) showing correlation above 0.7. Many respective modes in this range have near-zero correlation.

C: 5,000 – 5,500 Hz: This frequency range transitions from no correlation to an area of relatively good correlation, with most modes exhibiting MAC values above 0.75. While this represents a modest MAC value, it suggests enough correlation to infer modal similarity. For these similar eigenvectors, the associated frequencies are generally within 0.5% of each other.

D: 5,500 – 6,000 Hz: After the transition of C, the modes return to quite high MAC values, with most mode pairs in this region exhibiting a MAC above 0.90. For these modes, the respective frequencies are generally within 0.2%.

Individual modes of the various models were evaluated, with special attention paid to the cylinder liners. As would be implied by the MAC values, there was clear discrepancy in mode shapes for different loading scenarios when geometric nonlinearity was considered. However, when geometric nonlinearity was omitted, the models behaved quite consistently, with only modest differences in response and with consistently high MAC values.

For the area of high deviation between the different models (3,800-5,500 Hz), the cylinders tend to be undergoing some combination of 2nd order cylindrical modes ("ovalization") and 3rd order cylindrical modes ("triangular modes"). Upon visual inspection, it is clear that the 100% and 200% load scenarios result in quite distinct oval and triangular modes within the cylinder liners. It appears that stress stiffening in the region of the cylinders is quite significant in accounting for higher-frequency structural vibration, both of the cylinders themselves and of the system as a whole.

Conclusions

These results suggest that including geometric nonlinearity in engine assembly models is particularly important if one wishes to more accurately capture higher-frequency linear dynamic responses. This observation is noteworthy because in general practice, engine durability workflows tend to neglect this aspect. For stresses, strains, and static stiffnesses (of primary interest to durability groups), it has been generally found that geometric nonlinearity only modestly impacts the results for most passenger vehicle engines. However, based on the results presented here, its impact for higher-frequency calculations is considerable.

References

1. T. Ghosh, "Component Mode Synthesis of Structures with Geometric Stiffening in MSC/Nastran," Proceedings MSC 1999 Aerospace Users' Conference, 1999.
2. J.C. Blake and H.J. Kurtz, "The Uncertainties of Measuring Fastener Preload," Machine Design, vol. 37, Sept. 30, 1965, pp 128 – 131.
3. S.A. Nassar, PhD, University Research & Engineering Consultants, private communication.
4. R.J. Allemang, D.L. Brown, "A Correlation Coefficient for Modal Vector Analysis," Proceedings, International Modal Analysis Conference, pp. 110-116, 1982.
5. DS Knowledge Base Article: QA00000008771 (www.3ds.com/support/knowledge-base)

SIMULIA References

For additional information on the Abaqus capabilities referred to in this document please see the following Abaqus 6.13 documentation references:

- Analysis User's Guide
  - "Prescribed assembly loads," Section 34.5
  - "Natural frequency extraction," Section 6.3.5

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