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Nonlinear Dynamic Analysis to Extend the Value of Liquid Rocket Qualification Test

Mary Baker*
ATA Engineering, Inc., San Diego, California 92130
Frank X. Moehrle†
Pratt and Whitney Liquid Rocket Propulsion, West Palm Beach, Florida 33410-9600
and
Kyle C. Indermuehle‡ and Marcus D. Billings§
ATA Engineering, Inc., San Diego, California 92130

The goal of the methods and project example described is to use analysis to get more out of an engine qualification program. Nonlinearities are one of the reasons that testing is important in final validation of engine designs. An efficient method of modeling discrete nonlinearities was used to extend the understanding and value of both the sinusoidal vibration testing and the linear dynamic analysis that had been done before testing. In the example, discrete nonlinear elements are identified and separated from the overall engine finite element model. The nonlinear system is solved in a multibody dynamic program with the bulk of the engine represented by its modes. Final data recovery for the engine components is done within a linear, dynamic analysis program using the modal time histories from the multibody program.

Introduction

WITH the pressure to design higher performance and more reliable launch vehicles in less time and at lower cost, more efficient means are needed to qualify vehicle components for flight. Attempts to achieve improved efficiency by replacing tests with analysis have not proved to be feasible. However, analysis can be used in coordination with tests to get more out of each test. This paper describes a portion of a liquid rocket engine development program for which only sine vibration testing was used to qualify the engine for all boost-phase dynamic loads. Through analysis, this single type of testing was extended to cover many types of loads and many design concerns.

The test directly qualified many components of the engine for flight environments but also exposed important nonlinear responses that had not been addressed in the linear modal approach to dynamic loads analysis. Some type of nonlinear analysis was then needed to interpret the test data to determine what components had actually been qualified experimentally and what parts of the analysis for margins of safety were valid after consideration of the nonlinear behavior.

For most of the dynamics analysis for this engine development program it was necessary to use linear methods for margins of safety due to the large number of load cases and detailed design concerns. A full nonlinear analysis of this engine would have extended the modeling and analysis time and cost to impractical levels. Modal testing and qualification testing verified many aspects of the overall dynamics loads analysis. However, this testing also made it clear that nonlinear analysis was necessary to understand fully what had happened in the tests and to address specific nonlinear responses. This paper presents an efficient way to extend the dynamic loads analysis and limited testing of a liquid rocket engine to include selected nonlinear dynamic phenomena without greatly extending the overall loads analysis and qualification process.

Vehicle, Dynamic Loading, and Testing Overview

The engine to be qualified was the RL10B-2 (Fig. 1), the world's highest performance upper-stage engine. Figure 1 shows the nozzle in the stowed configuration, which is used for the boost phase of flight. During operation of this engine, the outer nozzle is deployed along ball screws using a belt drive system until the outer nozzle snaps into place at the bottom of the inner nozzle. The loading environments addressed in this paper were the boost-phase dynamic loads.

Extensive dynamic analysis had been done on the overall launch vehicle by the system-integrator dynamics staff to determine the boost-phase dynamic environments that were appropriate for this upper-stage engine. Detailed finite element models were prepared and used with these dynamic environments to produce stresses, deflections, and margins of safety to guide development. Of particular consideration has been the dynamics of the carbon–carbon deployable nozzle when stowed. (These studies are described in Refs. 1–3.) Sample models from these analyses are shown in Fig. 2. Most of this dynamic analysis consisted of linear modes-based analysis. Modal testing has been done to verify the models for key modes in the frequency range of interest. The environments included shock, transient, random, acoustic, and sinusoidal loads, which were applied as a forced response using the modes.

The qualification testing program for this stage consisted of acoustic and sine vibration testing. Analysis was used to relate flight environments to qualification-testing load levels. Actual level of stress and acceleration obtained during the qualification testing were evaluated through analysis to assess if critical areas of the engine were in fact qualified for flight through the testing and by the analytical margins obtained. For any area that was not qualified, component-testing programs were undertaken. (Reference 4 describes the component testing approaches.)

Assessment of Nonlinearities

There were several known nonlinearities in the engine response. In particular, the electromechanical actuators that control the nozzle direction and the lateral motion of the stowed engine are known to

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*Senior Vice President and Technical Director, 11995 El Camino Real, Suite 200. Member AIAA.

†Lead Space Propulsion Structural Engineer, Liquid Rocket Engineering, P.O. Box 109600.

‡Project Engineer, 11995 El Camino Real, Suite 200. Member AIAA.

§Engineer, 11995 El Camino Real, Suite 200. Member AIAA.

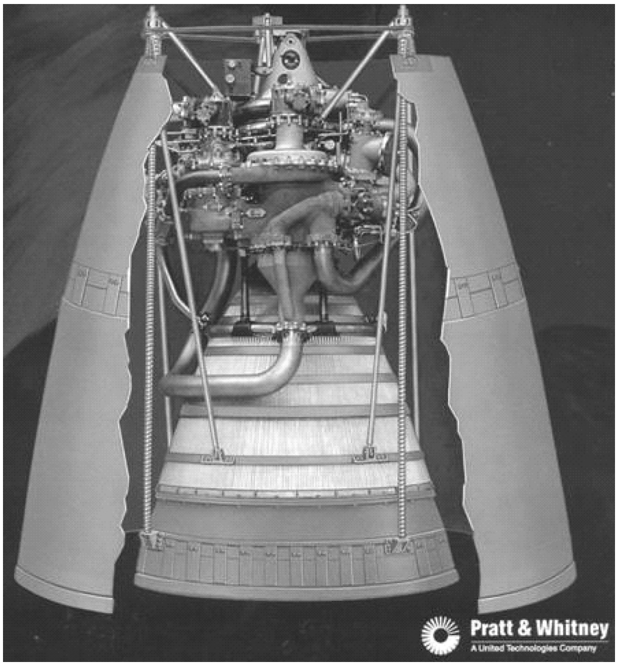


Fig. 1 Cutaway view of the RL10B-2 liquid rocket engine with the nozzle stowed for boost phase of flight.

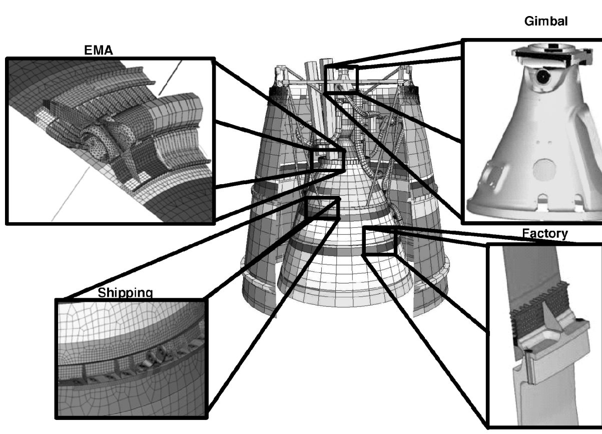


Fig. 2 Analytical models of the RL10B-2 engine used in determination of margins of safety.

have some free play, such as that shown in the force deflection plot in Fig. 3. (This curve illustrates the general type of data but not the actual values, which are proprietary.)

In addition, the belt drive for the deployment of the nozzle is observed to have some nonlinear behavior. Furthermore, some local load redistribution is expected due to peak loads. The initial approach to handling these nonlinearities was to make a linear approximation. Modal testing was used to verify the linear values used for selected excitation levels in the linear dynamic models.

For most of the engine components, this linear approach was quite effective. However, during the qualification testing, nonlinearities were observed. The first evidence of nonlinearities was the detailed transient waveform observed for the force and strain measurements during the test. Input consisted of sinusoidal acceleration sweeps over the frequency range of interest. Measured forces in the actuators showed a waveform that was very different in the positive and negative directions, indicating significant nonlinear behavior. Other evidence of nonlinear behavior was the shift in apparent frequency. Because of modal testing and test analysis correlation, the engine models matched modal test data very well in frequency. However, peaks in sine sweeps did not show responses at the expected frequencies. This nonlinear behavior required analysis support to determine what modes had actually been excited in the qualification program

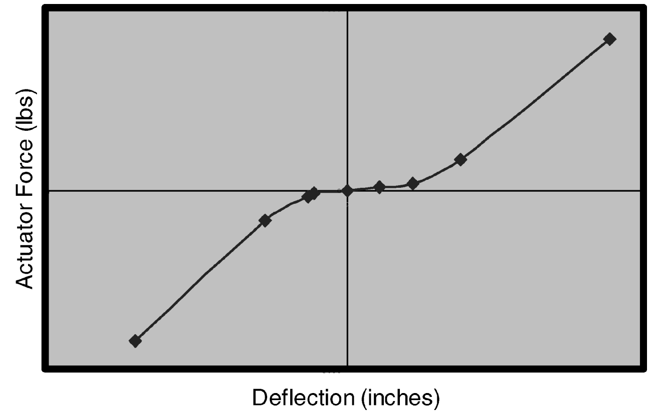


Fig. 3 Force deflection curve for nonlinear electromagnetic actuators. (Scales have been removed to avoid disclosing actual data.)

and whether or not the critical stress situations predicted for flight had been achieved in test. Of most interest to this paper were nonlinearities that made it difficult to tell which components were qualified and whether analytical margins were valid.

Approaches to Address Nonlinearities

Most of the dynamic loads analysis was performed using a modal approach. Modal models of components were assembled and solved to obtain modal models of the engine subsystem, as well as the stage and the whole launch vehicle. The modal models have many advantages. In particular, the solutions are very efficient so that very detailed models and many load cases can be processed. A further advantage is that the modal models can be verified by modal testing. Although the nonlinearities due to actuators, materials, or deformations can be included in the finite element model, modes can no longer be used once the models contain nonlinearities. Once nonlinear elements are included, the forced response must be performed using direct integration of the mass and stiffness matrices. In this particular case, the disadvantage of the direct stiffness integration was the loss of the test verification based on modes. Even more important, the processing of many load cases for a very detailed engine model appeared to be too time-consuming for the schedule.

For this reason, an alternative approach was used. The nonlinear elements of the finite element model were separated from the rest of the model. For example, the nonlinearities of the actuators were studied by treating the engine without the actuators as a linear model for which modes were obtained. These modes were than used to simulate the flexible dynamic behavior of the linear engine, which represented most of the detail of the engine. The nonlinear dynamic system was then simulated in a multibody dynamics code, which in this case was ADAMS®.[5] The actuators were represented by their nonlinear force deflection curves. The engine was included as a flexible component described by its modes. The forced response for the dynamic system was determined in ADAMS by applying the qualification vibration input at the gimbal plane of the flexible modal component. From this simulation, the dynamic response of the actuators and of the flexible component was obtained. This dynamic response included all boundary conditions on the flexible component, as well as the transient response of all of the modes within the flexible component. Once these dynamic boundary conditions and modal responses were obtained, they were passed back to the linear dynamic program to obtain detailed stress and deflection data for the physical nodes and elements within the modal component. [This overall capability is called ADAMS Flex and includes an interface for dynamics between the modal solver (I-DEAS® Model Solution), the dynamic simulation (ADAMS), and the forced response computation (I-DEAS Response Analysis)[6].] This process is shown schematically in Fig. 4.

This modeling approach is illustrated further in Fig. 5. The engine minus the actuators is a single flexible component in a multibody system model. The actuators are represented in the multibody system model as springs, which can have any force deflection curves such as that shown. The ADAMS simulation generates dynamic responses for the dynamic event. These responses include forces, deflection, velocity, and acceleration time histories at points on each component, which can be viewed in ADAMS postprocessing. It also generates deflection histories for the modes within the flexible components. By the transfer of these responses, including modal responses, back to I-DEAS Response Analysis, the full dynamic stresses and deflections can be recovered. Thus, the margin of safety calculations can be obtained for detailed results within the engine using the same test-verified finite element model with the same element formulations as were used for the rest of the dynamic-load recovery.

The dynamic modeling considerations for the engine modal component in this approach are described in the next section.

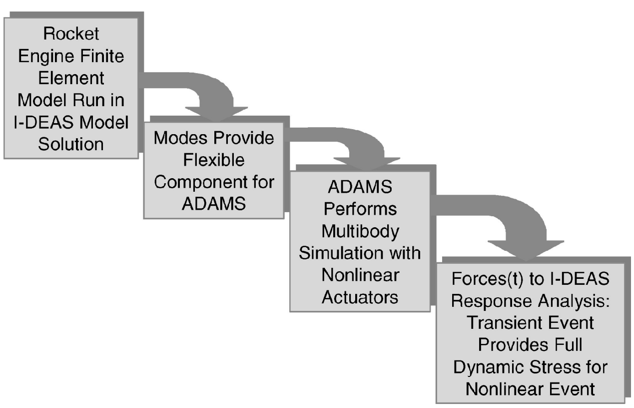
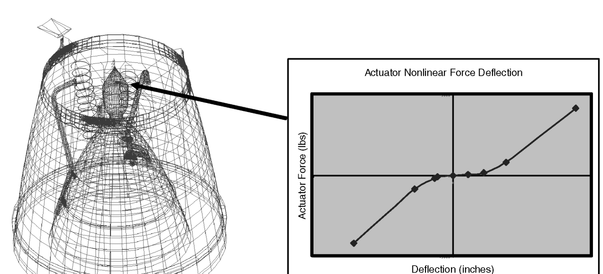


Fig. 4 ADAMS Flex and I-DEAS Response Analysis approach for simulating nonlinear dynamic response.



a) Full engine modeled by modes from finite element model without actuators

b) Actuators added as springs (linear or nonlinear) in multibody simulation

Fig. 5 Engine model shown as one component in the multibody simulation, which also includes a nonlinear spring.

Engine Modeling Considerations

The engine model without the actuators was solved for enough modes to cover the frequency range of interest. In I-DEAS the Superelement option was used for the modes solver, which provides modes that are fixed at all degrees of freedom that were identified as connections to other components in the ADAMS simulations. It also provides a constraint mode for each of these degrees of freedom, which including six at the gimbal plane, which is the point of excitation of the engine in the qualification test, and one each in the direction and location of attachment of each of two actuators. ADAMS expects flexible components to be represented by free–free modal components with six rigid-body degrees of freedom. I-DEAS, thus, does a second modal solve, this time using the fixed interface modal results from the first solve plus the constraint modes. The result is a component that is like a free–free component with six rigid-body modes. These results are written to a file called a modal neutral file (.mnf), which can be read by ADAMS.

Within ADAMS, the flexible component representing the engine was connected to a component that represents the shake table for which motion can be imposed. Springs, used in place of the actuators, spanned from the shaker table to the attach points on the engine at the angle of the actual actuators. The stiffness of the springs could be either the force-deflection curve measured for each actuator or the linearized value obtained from the modal test. Sine-sweep functions were generated that matched the qualification-test specification. These functions were used as imposed motion on the shaker table to mimic the qualification test.

Engine Simulation Results and Conclusions

Because the actual test data are proprietary, representative sine functions were generated that exhibited the same types of behavior from the spring values used, but the actual force data, including its distribution with frequency, are different.

The goal in these simulations was to understand the nonlinear effects to verify the analytically determined margins of safety. An even more important goal is to determine what components had been qualified. The decision on qualification included understanding the load and stress levels that different parts of the engine had experienced but also what modes had been fully excited during the test. The results reported here only cover the nonlinearity associated with the actuators and are limited in that the actual test data and actual forces and deflections cannot be shown. However, the approach and methods for using analysis to extend the usefulness of the sine test are illustrate by these sample results.

Figure 6 shows a sine-sweep transient analysis for both the linear spring and the nonlinear springs serving for the actuators. The two curves represent one actuator force vs time for a sweep up in frequency from 0 to 20 Hz in 20 s. Thus, a peak at 5 s indicates a mode in the linear model at 5 Hz and an apparent mode at 5 Hz in a nonlinear case. The two peaks in each curve represent the first and

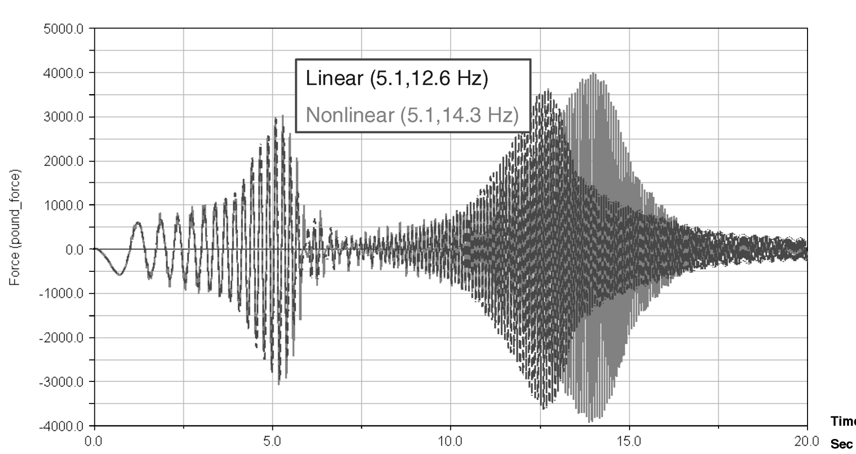


Fig. 6 Actuator force history for acceleration input at the gimbal plane for a sine sweep from 0 to 20 Hz.

second mode pairs of the engine. (Because of the symmetry of the engine, each mode is actually a mode pair in perpendicular directions. The actual mode or modes excited at each frequency depends on the direction of the input of the shake table.)

One of the objectives of this study was to understand whether the nonlinearity in the spring could explain observed shifts in natural frequencies. The first result of the nonlinear simulation shows no change for the first mode pair but an apparent mode increase from 12.6 to 14.3 Hz for the second mode pair. This result suggests that the first mode pair is controlled by stiffnesses other than the actuator springs, possibly one of the other nonlinearities that were identified.

To understand these results better, the excitation level was varied. In general, the qualification testing levels reach higher amplitudes than the flight levels and much higher amplitudes than modal testing. Looking at the response as a function of amplitude of excitation is, therefore, of interest. Figure 7 shows the same two cases with linear and nonlinear actuator springs used for a sweep over the same

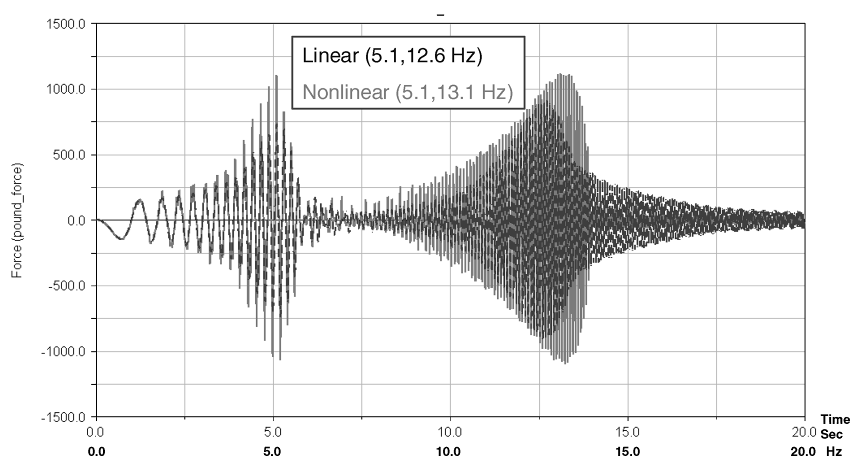


Fig. 7 Actuator force history for a sweep at one-quarter amplitude: no change in the apparent frequency for first mode pair but convergence of nonlinear second mode pair apparent frequency toward the linear frequency.

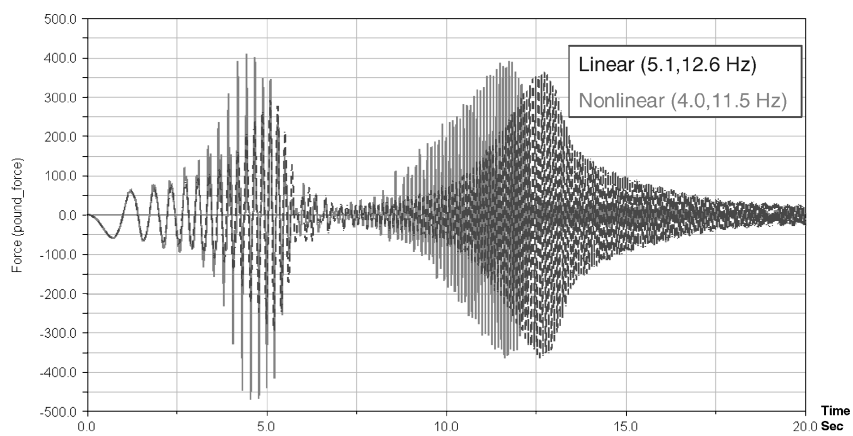


Fig. 8 At 1/10th amplitude, actuator force nonlinear apparent frequency shifts below linear frequencies for both mode pairs.

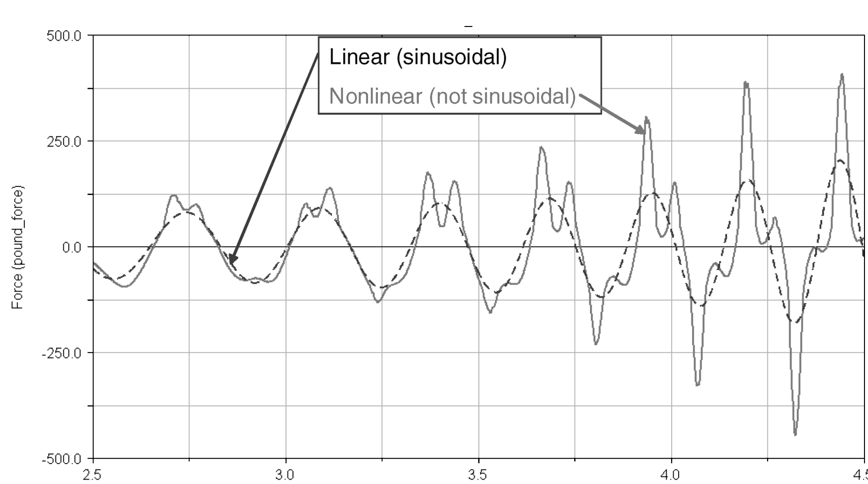


Fig. 9 Nonlinear actuators result in force history that is not sinusoidal in segment of force history between first and second mode pairs.

frequencies and time. The difference is the amplitude of excitation is multiplied by 0.25 at all frequencies. Note that the apparent frequency of the second mode pair has shifted closer to the linear apparent mode frequency for the second pair. There is still no change in the frequency of the first mode pair.

To generate Fig. 8, the amplitude was reduced to 1/10th of the original level. The result is that, for the nonlinear case, both the first mode pair and second mode pair shifted down in apparent frequency below the linear mode frequency.

The other symptom of nonlinearities that was observed in the test was the response waveform that was not sinusoidal and very different in the positive and negative directions. A closer look at the nonlinear results in Fig. 8 shows the same sort of waveform in this nonlinear analysis as was observed in the qualification test. This expanded view of a time slice is shown in Fig. 9.

This type of sudden jump in amplitude is characteristic of a strain-hardening (or softening) spring as explained by Thomson.[7] Equation (1) shows the order differential equation for a dynamic system with a stiffness that increases with deflection:

$$\ddot{x} + \omega_n^2 x + \mu x^3 = F \cos \omega t \tag{1}$$

The parameters μ and ω_n can be adjusted to achieve the shape of the curve and frequency behavior desired. Figure 10 shows the general shape of the locus of amplitudes vs frequency that satisfy this equation. Figure 10 shows why the nonlinear springs cause sudden jumps in amplitude as the frequency of motion changes. It also shows why the apparent frequency or frequency of peak amplitude differs for positive changes in frequency vs negative changes in frequency. Sweeping down along this curve results in a sudden jump up from one part of the curve to another, resulting in a peak amplitude at the jump at F_{n1}. Sweeping up in frequency results in a sudden jump down in amplitude to another stable point shown in the curve at F_{n2}.

Based on this explanation, the test results from sweeping up should differ from sweeping down. Figure 11 shows the same condition as Fig. 8. The two curves represent sweeps up from 0 to 20 Hz and down from 20 to 0 Hz, both with the nonlinear actuators.

One other effect of interest is the preload in the actuators due to gravity. From the shape of the load deflection curve, a preload would be expected to make a difference, particularly at lower amplitudes. A gravity preload was considered in the simulation for an excitation level of 1/10th amplitude. The difference with and without the preload is shown in Fig. 12.

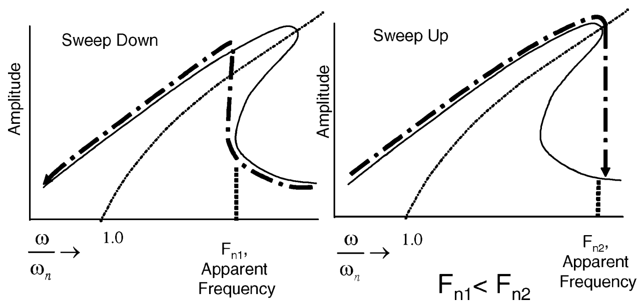


Fig. 10 Locus of amplitudes vs frequency for strain-hardening spring with apparent frequency differences for sweeps in different directions.

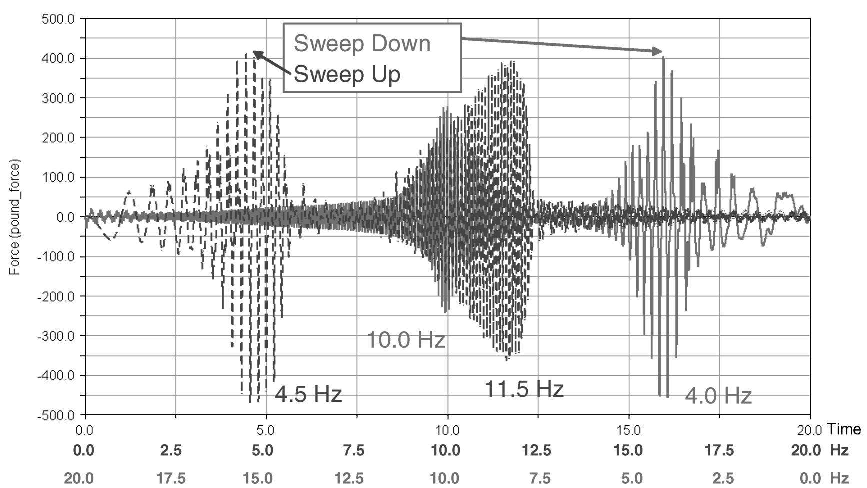


Fig. 11 Comparison of sweeping up to sweeping down in frequency: first mode pair shifts from 4.5 to 4.0 Hz and second mode pair shifts from 11.5 to 10.0 Hz.

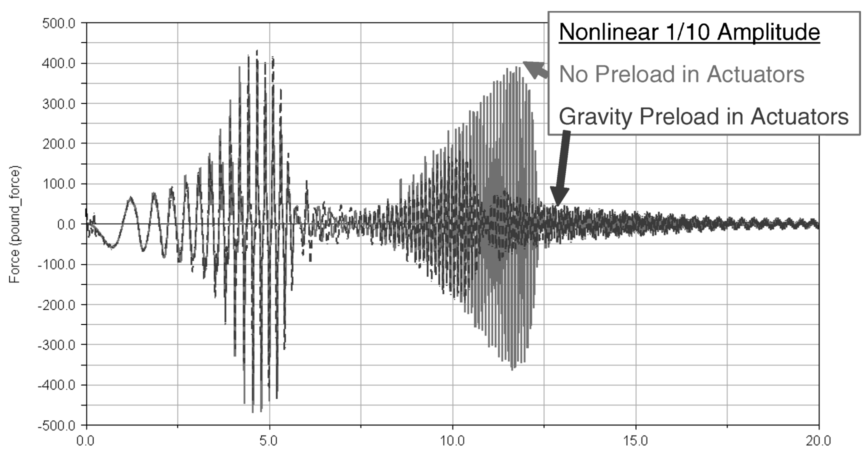


Fig. 12 Gravity preload in the actuators has marked effect for lower amplitude of excitation, in second mode pair but not first mode pair.

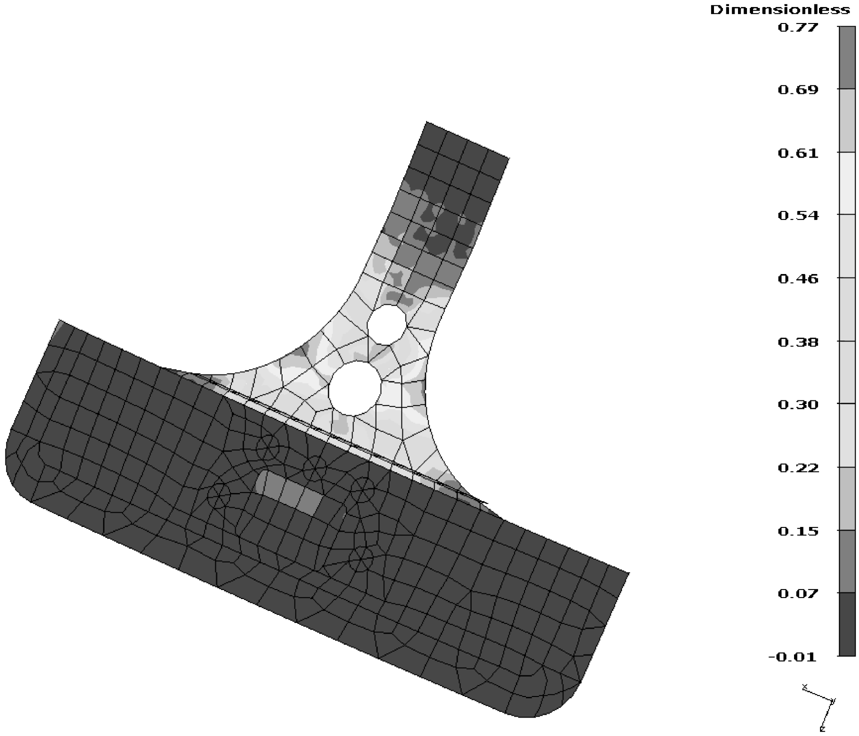


Fig. 13 Peak stress for one component over entire event history for nonlinear case normalized to an allowable value.

Dynamic Stress Analysis

Once the dynamic response was better understood, it was still of interest to recover loads and stresses for engine detailed components for the more accurate nonlinear cases. I-DEAS can read the results files from ADAMS results, which include modal histories. The ADAMS results become a dynamic event in I-DEAS Response Analysis. From this event, full stress, load, and deflection histories can be obtained. At any point in time, a full contour plot of results can be obtained for any or all components. Alternatively, the peak results for all time at each node or element can be saved as one data set such as that shown for one component in Fig. 13. Figure 13 represents the peak stresses over the entire nonlinear event normalized to an allowable value.

The peak value shown is 0.77 for the nonlinear event. When the linear event was run in the same way, the normalized peak stress had nearly the same contours with a peak of 0.82. Thus, for this component and this loading event, the nonlinear case resulted in a slight lowering of the peak stress value and a small increase in margin.

Results and Discussion

The nonlinear simulations allowed the following incremental levels of understanding. The nonsinusoidal waveform and the shifts in frequency at different levels could be explained by the nonlinear spring. However, the actuator nonlinearities had a greater effect on the second mode pair than the first mode. This suggested that there was another nonlinear structural element that would need to be included to predict fully some of the observed effects on the first mode pair.

The ability to simulate the nonlinear behavior and explain most of the dynamic responses by discrete nonlinear elements helped to validate the overall linear approach to dynamic loads analysis. The isolation of the nonlinearities and the use of a modal component to represent interaction between nonlinear elements and the rest of the engine provided time savings to the overall load-analysis process. The same elements and element formulations that were used in the linear coupled-loads analysis could be used for loads and stress recovery for these cases involving nonlinear elements.

Summary

The analysis approach provided an efficient way to interpret what had actually occurred during the test. Analysis was needed to determine what aspects of the engine specification had been met by testing and what components had reached flight levels during the test. It also opened a path to qualification through a combined analysis and test program for those components that needed further scrutiny without introducing a whole new set of elements, modeling approaches, simulation tools, or testing methods.

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T. Collins
Associate Editor