Lifecycle simulation in the automotive industry

SIEMENS

White Paper

Advances in dynamic response with NX Nastran

Accelerated product development cycles are just one symptom of an extremely competitive automotive marketplace. Companies have been driven to embrace and extend virtual product development processes, replacing time intensive physical tests with the use of upfront performance simulation.

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Introduction

Accelerated product development cycles represent just one symptom of an extremely competitive automotive marketplace. Companies have been driven to embrace and extend virtual product development processes, replacing time intensive physical tests with use of upfront performance simulation.

The dynamic response of body and powertrain affects every area of customer satisfaction. Engineers are required to run simulations over very wide frequency ranges with models exhibiting extreme problem sizes. Speed and solution robustness are critical. Technically speaking, modal solution approaches are much preferred because of efficiency. Methods of generating the modal spaces are the subject of constant improvements, and the physical response quality may now be evaluated by the engineer.

Siemens PLM Softwares' Lifecycle Simulation vision and strategy recognizes the value of pervasive simulation throughout the product lifecycle. Performance simulation is carried out throughout the lifecycle during concept and design studies, manufacturing and testing, through operation of the product, to maintenance and disposal. For example, simulation during automotive design often involves, among others, the rotation of some flexible components, external dynamic excitation of structures, and the mechanism type motion of some parts of a complex structure. These are accommodated by rotor dynamic modal frequency response analysis, an alternative enforced motion computational strategy, and an interface to external multi-body simulation tools.

Manufacturing and testing phases are also increasingly important. Some manufacturing processes require a detailed physical simulation. A "simple" example from the world of civil engineering is a road bridge. In many cases bridges are not completely stable during the build process and are hence the focus of extensive simulation studies. The automotive world expends enormous efforts to simulate manufacturing processes and build sequencing as part of quality programs. Many industries have an extensive history of using simulation in support of test programs and also of enhancing simulation models with stiffness and other data derived from tests.

Similar considerations are valid for the maintenance and disposal phases of product lifecycles. Maintenance solutions regaining the damaged physical integrity of a structure, frequently needed in the aerospace industry, are also subject of extensive computer simulations.

The main focus of simulation was always on the operational phase of products; however, nowadays the operational scenarios traverse a much wider range of physical phenomena. In the automotive industry, for example, these scenarios include the incorporation of rotating components into the classical vibration analysis, the simulation of a wide spectrum of external excitations, and the coupled execution of multi-body analysis.

Very large problem sizes are the result of the desire to increase the fidelity of the physics simulation, and advanced computational methods are needed to support these ever increasing sizes of the finite element models of lifecycle simulation scenarios.

Model reduction technique

In the computational methods area we see a renaissance of the modal solution method, currently often called the model reduction technique. Modal dynamic solutions in industrial finite element analysis have a long history that began in the 1960s.[1] Their original emergence was to help overcome the very serious limitations of the computer hardware of the days. Due to the engineering intuition required to make the methods useful, they stayed mainly a specialist's tool for decades. With the emergence of high performance computers in the 1980's and their ability to analyze very large finite element models, it seemed that the technology had its days numbered.

Meshing technology advances in the past two decades and the emergence of lifecycle simulations, however, resulted in an explosion of problem sizes, contributing to renewed interest in the technology. The technology is summarized in the following. The equation of motion of a free structure after applying the boundary conditions and eliminating the constraints is:

1. $$M_{ff}\ddot{u}_f + B_{ff}\dot{u}_f + K_{ff}u_f = P_f$$

The matrices M, B, and K are the finite element mass, damping, and stiffness matrices, and the subscript f refers to the free partition of degrees of freedom of the model. The vector P contains the forces applied to the model and the vector u is the dynamic response; both of these are potentially frequency or time dependent.

Converting equation (1) to modal space, the equation of motion transforms to:

2. $$m_{hh}\ddot{\xi}_h + b_{hh}\dot{\xi}_h + k_{hh}\xi_h = \Phi^T_{fh}P_f$$

where

3. $$u_f = \Phi_{fh}\xi_h$$

$$m_{hh} = \Phi^T_{fh}M_{ff}\Phi_{fh}$$

$$k_{hh} = K^T_{fh}K_{ff}\Phi_{fh}$$

$$b_{hh} = \Phi^T_{fh}B_{ff}\Phi_{fh}$$

The matrices m and k are diagonal modal matrices, and b is diagonal in the case where it is proportional to either m or k (modal damping). The computational advantages of solving the problem posed in equation (2), as opposed to equation (1), are obvious when one considers the fact that the f-size may be on the order of tens of millions while the h-size is at most on the order of ten thousand. The eigenvectors facilitating this reduction are spanning a modal space of size h.

Modal space computations

The main component of modal solution techniques is an efficient and robust computation of the modal space. The modal space computation is based on the normal modes problem of:

4. $$M_{ff}\Phi_{fh}\Lambda_{hh} = K_{ff}\Phi_{fh}$$

The industry leading solution is the Lanczos method, also of historical origin,[2] but with very modern implementation techniques.[3] It is an iterative process, written in compact form as:

5. $$Q_{j+1}B_{j+1} = (K_{ff} - sM_{ff})^{-1} M_{ff} Q_j - Q_j A_j - Q_{j-1} B_j^T$$

The Q matrices are blocks of Lanczos vectors that will be the basis of computation for eigenvectors. The A and B matrices are components of an intermediate block tridiagonal matrix, from which the eigenvalues are computed. The s value is a shift manifesting a spectral transformation and j is the iteration counter.

The prominent role of the Lanczos method in industrial applications is due to several important formulation, numerical, and computational characteristics. The most salient of the formulation characteristics of the Lanczos method are that:

- it is using the input matrices only in an operator form, facilitating the solution of very large problems,
- it solves generalized, linear, or quadratic (two or three matrix) eigenvalue problems of structural dynamics, and
- it may be used with spectral transformation, resulting in accurately finding the eigenvalues of rather wide frequency ranges of interest.

Fundamental numerical aspects contributing to the industrial success of the Lanczos method are:

- obtaining accurate results with only semi-orthogonality (only to the level of the square root of the computer's floating point operation accuracy),
- maintaining orthogonality with selective orthogonalization to prevent the repeated appearance of already converged eigenvectors, and
- accurately finding multiple eigenvalues, a practical occurrence in structural dynamics due to structural symmetry.

Finally, the Lanczos method enables the use of special computational technologies adherent to the prevailing computational environments of clusters of workstations, such as:

- geometric domain decomposition by automatically subdividing the geometry based on the connectivity information obtained from the finite element model,
- frequency domain decomposition with a systematic introduction of a series of intermediate frequency boundaries in the given frequency range, and
- their hierarchically applied distributed memory parallel implementation enabling the feasible use of even up to 64 to 128 processor nodes.

The Lanczos method is still undergoing further research and improvements. For some new numerical improvement directions, see the References section.[4] There are also solution techniques available for the approximate computation of the modal space, based on the automated component modal synthesis. While these methods are very fast, they are prone to missing components of the modal space. Hence their application is limited to the daily design cycle and the interior of optimization cycles, but not for the final validation of lifecycle scenarios.

Modal solution metrics

Specific metrics used to measure the completeness of the modal space and the accuracy of the modal solution constitute the third important component of modal solution computations. The two main groups of these metrics measure either the modal solution quality or the modal space completeness.

In the first group the most important components are:

- modal participation factors,
- modal effective mass,
- modal strain and kinetic energies, and
- modal contributions.

The modal space completeness is evaluated and possibly improved by:

- mode selection mechanisms,
- residual flexibility computations,
- mode space augmentation, and
- mode acceleration.

The residual flexibility of the structure considering a given modal space may be computed by the following expression:[5]

6. $$Z^r = K^{-1}_{ff} - \Phi_{fh} \Lambda^{-1}_{hh} \Phi^T_{fh}$$

Considering the time or frequency independent component G of the dynamic load P of the system, the modal space (after some orthogonalization) may be augmented:

7. $$\Phi_{f(h+l)} = \lfloor \Phi_{fh} Z^r G_f ; \rfloor \; P_f = G_f H_f(t)$$

In the case when no dynamic loads exist, the engineer may define virtual loads to aid this process. The advantage of applying such an augmentation is well pronounced when there are high amplitude excitations at low frequencies in the dynamic loads.

Dynamic analysis of rotating components

Dynamic analysis of rotating components has long been an instrumental element of aerospace applications, and it is becoming important in the auto industry as well. Analyzing car components rotating with high speeds, such as flywheels and tires, is possible with this technology.

The free vibrations of a flexible rotor are computed from the following complex eigenvalue analysis problem:

8. $$(\lambda^2 m + i\lambda (b + \Omega c) + (k + \Omega c))\varphi = 0$$

Here m, b, k are the aforementioned modal mass, damping and stiffness matrices. The c and h matrix components of the equilibrium equation are proportional to a sequence of rotation speeds (denoted by capital omega) supplied by the analyst. The c is the modal gyroscopic matrix, containing moments due to nodal rotations, and the h is the circulatory (internal structural damping) matrix.

They manifest the rotating phenomenon in connection with the conventional finite element model of the structure. The above modal complex eigenvalue problem is solved in a loop over the rotor speeds. The resulting eigenvectors are the whirl mode shapes and the eigenvalues are the whirl modes.

The eigenvalues of this quadratic eigenvalue problem appear in complex conjugate pairs: $\lambda=\omega\zeta(+2j)$ with natural frequency f, where $\omega=\pi 2f$ and the viscous damping coefficient is ζ. The natural frequencies are dependent on the rotor speed. Only the solutions with positive natural frequencies are considered.

The Campbell diagram, shown in Figure 1, is a method of presenting and interpreting the rotor dynamics results. In the diagram, the natural frequencies (vertical axis) are plotted as the function of the rotor speed (horizontal axis). The line 1P representing equal frequency and rotor speed is also plotted. Critical speed values are where the latter line intersects any of the natural frequency curves.

For the example shown in Figure 1, mode 1 (coincident with the horizontal axis) represents a rigid body motion, and the horizontal mode 2 represents a linear motion. Modes 3 and 4 are a mode pair with a negative and positive slope, respectively.

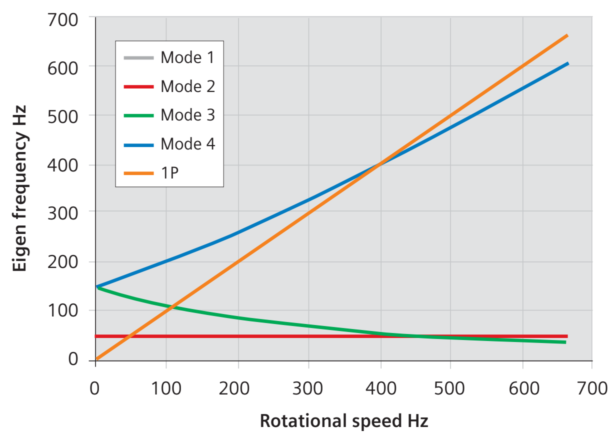

Figure 1.

For unsymmetric rotors there are two critical speeds in the diagram. Between these values the rotor is unstable. For symmetric rotors there is one critical speed at which the rotor is unstable. Based on this, a stability diagram with real eigenvalues or damping as function of rotor speed may be established.

The whirl motion type is also identified from the Campbell diagram as follows: a positive slope of a mode represents a forward, a negative slope a backward whirl motion. The forward (direct) whirl motion is in the direction of the rotation, while the backward (reverse) whirl motion is in the opposite direction from the rotation.

The governing equation of frequency response in modal space with rotor dynamics terms, in a fixed coordinate system, and considering the load to be independent of the speed of the rotation is:

9. $$-\omega_j^2 m + i\omega_j (b + \Omega c) + (k + \Omega h))u(\omega_j) = p(\omega_j); j = l,\ldots m$$

The lower case omega is the circular frequency and this is called an asynchronous solution. The load in this case could still have frequency dependence, as shown by the *m* discrete excitation frequencies defined above.

In the case the load is dependent on the speed of rotation (called synchronous analysis) the governing equation, still in a fixed reference system, is as follows:

10. $$-\Omega_j^2 (m - ic) + i\Omega_j (b - ih) + k)u(\Omega_j) = p(\Omega_j); j = l,\ldots n$$

Here *n* denotes the number of synchronous rotation speeds at which the analysis is executed.

The synchronous analysis case is applicable to various centrifugal loads based on mass or force imbalances of the rotor. The asynchronous analysis is applicable to cases like gravity loads.

The rotating system analysis can also be done in a rotating reference system, and the engineer must specify the system that is more advantageous to a particular problem. In a rotating reference system the rotor dynamic analysis involves additional terms, like the geometric or differential stiffness matrix due to centrifugal stress, and a centrifugal softening matrix. The gyroscopic matrix also contains Coriolis terms in the rotational reference system.

Enforced motion of structures

Enforced motion of structures is another important component of lifecycle simulations in the automobile industry. Analyses with various road excitations are possible with this technology component. The conventional enforced motion formulation in modal space is:

11. $$m_{hh}\,\ddot{\xi}_h + b_{hh}\,\dot{\xi}_h + k_{hh}\,\xi_h = \Phi_{fh}^T\,(P_f - (M_{fs}\ddot{u}_s + B_{fs}\dot{u}_s + K_{fs}u_s))$$

The s-set partition contains the degrees of freedom associated with external excitations in the form of enforced displacements, velocities, or accelerations. The lack of definition of the damping coupling term in the case of modal damping results in accuracy problems in some cases when the excitation amplitude is large in relation to the structure's response.

An alternative formulation presents a solution by applying a so-called constraint mode to represent the base motion.

12. $$\Psi_{fs} = -K_{ff}^{-1}K_{fs}$$

The role of the constraint mode is depicted graphically in Figure 2 in terms of a single degree of freedom system with frequency dependent ground excitation.

The physical displacement u of the single mass is computed as the sum of the constraint mode motion and the alternative physical displacement *v*.

13. $$u_f = v_f + \Psi_{fs}u_s$$

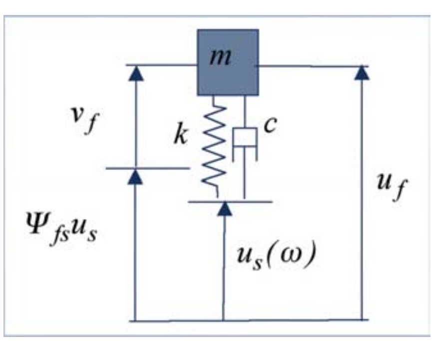

Figure 2.

With applying the modal reduction to this alternative physical displacement, the alternative formulation of the enforced motion in modal space is as follows:

14. $$m_{hh}\,\ddot{\zeta}_h + b_{hh}\,\dot{\zeta}_h + k_{hh}\,\zeta_h = \Phi_{fh}^T\,(P_f - (M_{fs} + M_{ff}\Psi_{fs})\ddot{u}_s - (B_{fs} + B_{ff}\Psi_{fs})\dot{u}_s)$$

Note that the alternative modal solution vector is different than from the conventional solution and they are related via the not trivially computable equation,

15. $$\xi_h = \zeta_h + \Phi_{fh}^T M_{ff}\Psi_{fs}u_s$$

Recovering the conventional modal solution components may be desirable for modal solution metrics computations, since modal energies or contributions are defined in terms of the conventional solutions. Due to the complexity of the above relation, however, if these metrics are required, the engineer is better off using the conventional method.

The alternative method's apparent accuracy advantage is in the computation of damping forces in this scenario. The alternative formulation leads to the damping force computed from the "relative" (base excitation removed) physical displacement between the free and enforced set. This difference is manifested in better force responses in the alternative method, especially in the low frequency range of the excitation.

Interfacing to multi-body analysis

Interfacing to multi-body analyses is also an important component of automotive lifecycle simulations. The wheel control mechanisms, for example, as shown in Figure 3, are often analyzed in separate processes with the dynamic behavior of the flexible car body and engine models captured in modal and inertia matrices and attached to the rigid structure.

For interfacing to multi-body analysis, we define an a-set of degrees of freedom that consists of the physical boundary nodes of the flexible model related to the multi-body model. The flexible body model first is statically condensed onto this set of degrees of freedom by a reduction of the stiffness, mass and load matrices.[6]

Then an o-set eigenvalue problem, describing the interior, "omitted" set of degrees of freedom of the flexible model, is solved. This produces a matrix of mode shapes $[\phi om]$, where the m is the number of eigenvalues computed. The modal stiffness and modal mass matrices are computed as shown in equation (3) with these mode shapes. These computations are similar to the earlier discussed modal space computations; however, the selection of the a-set degrees of freedom and the extent of the modal reduction still require considerable engineering intuition.[7] The modal matrices along with the corresponding mode shapes, as well as the o-set node locations, are then exported to the multi-body solver tool.

In order to represent the kinetic energy of the dynamically reduced flexible body with respect to a global or ground reference frame, a set of inertia invariants are also computed. They are based on the movement of any node point of the flexible structure with respect to the ground reference frame. Assume a rotation vector representing the motion of the local coordinate system with respect to the global coordinate systems:

16. $$\omega = \omega_x i + \omega_y j + \omega_z k$$

Figure 3.

Since in the multi-body simulation the reduced flexible body is moving as a rigid body, this vector is considered to be the same for all nodes. Let us also introduce a skew-symmetric rotation matrix (to facilitate vector products) as

17. $$\tilde{\omega} = \begin{bmatrix} 0 & -\omega_z & \omega_y \\ \omega_z & 0 & -\omega_x \\ -\omega_y & \omega_x & 0 \end{bmatrix}$$

Based on this, the velocity of a node point in the global reference frame is computed as

18. $$v_i = v_0 + \tilde{\omega}A(r_i + u_i) + A\dot{u}_i$$

Here the r denotes the location vector of a particular node in the local reference frame attached to the flexible body. The u vector contains the translational displacement components of a particular node point i and the dot indicates the time derivative, i.e. local velocities. The A matrix contains the direction cosines describing the relation between the ground and the local coordinate systems. The global velocities are denoted by v, i indicating node i and 0 indicating the local coordinate system's origin's velocities.

The kinetic energy of the flexible body with respect to above rotation may be computed as

19. $$T = \frac{1}{2}\sum_{i=l}^{n} (m(i)v_i^T v_i + \omega^T I_i \omega)$$

In the above the n denotes the number of nodes in the flexible body finite element model and $m(i)$ is the modal mass matrix component of the particular node, already computed in the reduction phase. The inertia invariant matrices are of the form

20. $$I_i = m(i)\tilde{\Phi}_{om}(i,j)\tilde{\Phi}_{om}(i,k);\ j,k = 1,\ldots m$$

where the tilde indicates the skew-symmetric reordering of any vector into the 3 by 3 matrix form as shown in equation (17). This in essence produces vector products between the i-th node's components of any (j,k) pairs of the mode shapes vectors spanning the modal space. This is a rather time consuming operation, but necessary to model the physical behavior of the flexible body in the rigid body motion space.

The application of the technology is ideally executed in an integrated process with the following steps:

a. Flexible model pre-analysis
   - calculate mode shapes and transformation matrices
   - compute modal mass, stiffness and inertia invariants

b. Multi-body system analysis based on the rigid model
   - evaluate dynamic equilibrium in time
   - calculate dynamic load for each time step

c. Flexible model post-analysis
   - based on dynamic loads compute physical displacements
   - based on the displacements, evaluate stresses and strains

Conclusion

We believe that the lifecycle simulation initiative of Siemens is a visionary step towards the future. It is an important goal of Siemens PLM Software to supply tools for the engineers to further automate the product development process. NX™ NASTRAN® software is an industry leading component of the product development tool set of the NX suite that includes design, manufacturing, and product data management offerings.

The recent Version 5 of NX Nastran incorporates the methods described above, and enables companies to increase the efficiency and value of their virtual product development process. NX Nastran is available on today's hardware, ranging from a simple cluster built from the multitude of CAD workstations, idle during the night, to dedicated high performance compute servers.

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