Dynamic Stress Analysis of a Bus Systems

***H. S. Kim, #Y. S. Hwang, #H. S. Yoon**

Commercial Vehicle Engineering & Research Center

Hyundai Motor Company

772-1, Changduk, Namyang, Whasung, Kyunggi-Do, Korea

ABSTRACT

This paper presents the effective method for dynamic stress analysis of structural components of bus systems or general mechanical systems. The proposed method is the hybrid superposition method that combined finite element static and eigenvalue analysis with flexible multibody dynamic analysis. In the stress recovery, dynamic stresses are calculated through sum of pseudostatic stresses and modal acceleration stresses, which are obtained by applying the principle of linear superposition to the modal acceleration method. The proposed method is more effective than conventional methods, that is, *the mode displacement method* or *the mode acceleration method*. Numerical example of bus systems estimates the efficiency and accuracy of the proposed method.

* Author : Tel: +82-0339-369-6179, e-mail: coreawill@yahoo.co.kr

[#] Coauthor

INTRODUCTION

In structural component design of bus systems, the appropriate criterion for fatigue failure should be based on consideration of failure modes of the specific component being designed. If the structural component is to withstand millions of cycles of load application, a criterion for fatigue failure must be used. The fatigue damage caused by repeated dynamic loads depends on the number of cycles and the frequency of the occurrence of significant stresses. Therefore, for accurate prediction of fatigue failure in structural components, accurate prediction of dynamic stress time histories is required. Experimental testing method may be the most exact for evaluation of dynamic stresses or strains in components of bus systems. The experimental method, however, requires at least one prototype bus system. The structural components in the prototype are then subjected to laboratory durability tests or real vehicle durability tests on proving grounds or fields in which the loading life cycle is applied at an accelerated rate.

Owing to growth in the power of digital computers, several analytical approaches for dynamic stress calculation in structural components have been developed to speed design cycles. One method is *Quasi-static Method combined with Rigid Multibody Dynamic Simulation*[1]. In this method, a bus system simulated by modeling its components as rigid bodies. From the rigid multibody dynamic simulation of bus systems, loads in joints and D'Alembert acceleration loads on structural components of concern are generated. These loads are applied the finite element models of structural components, so that quasi-static stress time histories that used for dynamic stress time histories may be computed. However, it does not consider the flexibility of structural components and the effects of such flexibility on dynamically induced loads, such as joint reaction loads and inertia loads. Therefore, this method may be useful for very stiff systems but inaccurate for flexible systems

To consider flexibility effects in dynamic stress calculation, another method can be used ; i.e., *Mode Superposition Method combined with Flexible Multibody Dynamic Simulation*[2],[3]. In this method, instead of full nodal coordinates, a few component modes are used to represent the deformation field in each structural component for flexible multibody dynamic analysis. Dynamic stress time histories in the structural component are then calculated by linear superposition of modal stresses, multiplied by corresponding modal coordinate time histories that are generated from flexible multibody dynamic analysis of the bus systems. When the modal stress superposition method are used for stress calculation, accuracy of stress is dependent on the components modes that are used in flexible multibody dynamic analysis. This means that inappropriate modes are selected, the resulting stresses will be inaccurate. Therefore, this method must consider the selection of component modes in order to improve accuracy.

To improve the accuracy of dynamic stress calculation, the other method can be used ; i.e., *Mode Acceleration Method combined with Flexible Multibody Dynamic Simulation*[4]. In this method, dynamic stress time histories in the component are calculated by summing pseudostatic stresses and modal acceleration stresses at each integration time step. Pseudostatic stresses are obtained from static analysis of structural components using dynamic loads at each integration time step and modal acceleration stresses are calculated by linear superposition. This method is more accurate than the previous method because of static correction of deleted vibration normal modes, but less efficient than the previous one because of increased computational cost; i.e. computational time and hardware space. Therefore, new methodology must be considered in order to improve efficiency.

The above three conventional methods are combined to form the hybrid superposition method(4) that improves the accuracy and efficiency of dynamic stress prediction. A brief theory and A conceptual procedure of the hybrid superposition method are presented. Numerical example of bus structures shows the effectiveness of the proposed method.

HYBRID SUPERPOSITION METHOD

The hybrid superposition method is defined as a computational dynamic stress analysis that employs hybrid sum of the pseudostatic stress and the modal acceleration stress, which are obtained by the principle of linear superposition in order to improve the efficiency of the modal acceleration method.

The structural component in bus systems can be modeled in equilibrium with body loads including inertia loads, Coriolis loads, centrifugal loads, gravity forces by ignoring corresponding loads resulted from elastic deformation and surface loads including joint reaction loads, externally applied loads, etc. The so called dynamic equilibrium equation for a structural component is then obtained in the form

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{C}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{f}^{surface} + \mathbf{f}^{body}_{rigid} \tag{1}$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are mass matrix, damping matrix, stiffness matrix. $\mathbf{u}$, $\dot{\mathbf{u}}$ and $\ddot{\mathbf{u}}$ are displacement, velocity and acceleration vector as nodal coordinates. $\mathbf{f}^{body}_{rigid}$ and $\mathbf{f}^{surface}$ are body loads and surface loads vector, respectively.

If body loads and surface loads that act on the component at time t are known, stresses in the component at time t can be calculated by the static analysis ; i.e., the dynamic problem is reduced to an equivalent static problem, in the conventional manner, by the application of D'Alembert's principle. Therefore, the static equilibrium equation is converted from the dynamic equilibrium equation of Eq. 1 as

$$\mathbf{K}\mathbf{u} = \mathbf{f}^{surface} + \mathbf{f}^{body}_{rigid} - \mathbf{C}\dot{\mathbf{u}} - \mathbf{M}\ddot{\mathbf{u}} \tag{2}$$

Consider a component that is in static equilibrium under D'Alembert body loads and surface loads at time t. Since the component is in statically equilibrium, stress in the component at time t can be calculated by executing conventional finite element static analysis with D'Alembert body loads and surface loads. If boundary conditions of the component have free conditions, Static analyses with inertia relief scheme are executed. Stress σ_i at node i with viscous damping assumption at time t can be written as

$$\sigma_i(t) = \sum_{j=1}^{Ns} \Sigma_{ij}^{surface} f_j^{surface}(t) + \sum_{k=1}^{Nb} \Sigma_{ik}^{body} f_k^{body}(t) - \sum_{r=1}^{Nr} \left(\frac{2\xi_r}{\omega_r} \right) \Sigma_{ir}^{mode} \dot{q}_r(t) - \sum_{r=1}^{Nr} \left(\frac{1}{\omega_r^2} \right) \Sigma_{ir}^{mode} \ddot{q}_r(t) \quad (3)$$

where *Ns* and *Nb* are the number of surface loads and body loads, respectively. $\Sigma_{ij}^{surface}$ and Σ_{ik}^{body} are static stress coefficients that are the stresses at node i, due to unit load of $f_j^{surface}(t)$ which is j-th component of surface loads and unit load of $f_k^{body}(t)$ which is k-th component of body loads, respectively. Σ_{ir}^{mode} is the modal stress coefficients that are the modal stresses at node i and ξ_r and ω_r are the modal damping factor and the natural frequency of r-th kept normal mode, respectively. $\dot{q}_r(t)$ and $\ddot{q}_r(t)$ are the velocity and acceleration of r-th modal coordinate, respectively.

In Eq. 3, the first and the second terms in the right hand side are pseudo static stresses that are obtained by the principle of linear superposition in order to improve the efficiency of the modal acceleration method. The third and the fourth terms are modal velocity stresses and modal acceleration stresses. In the conventional modal acceleration method, pseudostatic stresses are obtained from static analysis of structural components at each integration time step. In the proposed method, however, the pseudostatic stresses are computed from (*Ns* + *Nb*) static analysis. Since the numbers of static analysis are less than the numbers of the mode acceleration method, Computing time and hardware resources are required less than the mode acceleration method.

Time histories of body loads, surface loads and modal coordinates are accurately predicted by carrying out computer simulation of the bus system, using flexible multibody dynamic analysis codes such as DADS[5]. Static stress coefficients are calculated once through static analysis. Modal stress coefficients are calculated once through eigenvalue analysis, using a finite element analysis code such as MSC.Nastran[6].

IMPLEMENTATION

As discussed in the previous section, the hybrid superposition method obtains dynamic stress time histories by the principle of linear superposition. To calculate dynamic stress time histories that consist of pseudostatic stresses, modal velocity stresses and modal acceleration stresses as in Eq. 3, time dependent terms; dynamic loads and velocities and accelerations of modal coordinates, and time independent terms; stress coefficients are necessary. Therefore, two basic analyses are required. One is flexible multibody dynamic analysis to calculate dynamic loads and modal coordinates of the structural component. The other is finite element static analysis that is to calculate static stress coefficients and finite element eigenvalue analysis that is to compute modal stress coefficients. Figure 1 shows an outline of the proposed method and its conceptual data flow, from structural finite element analysis to stress history calculation.

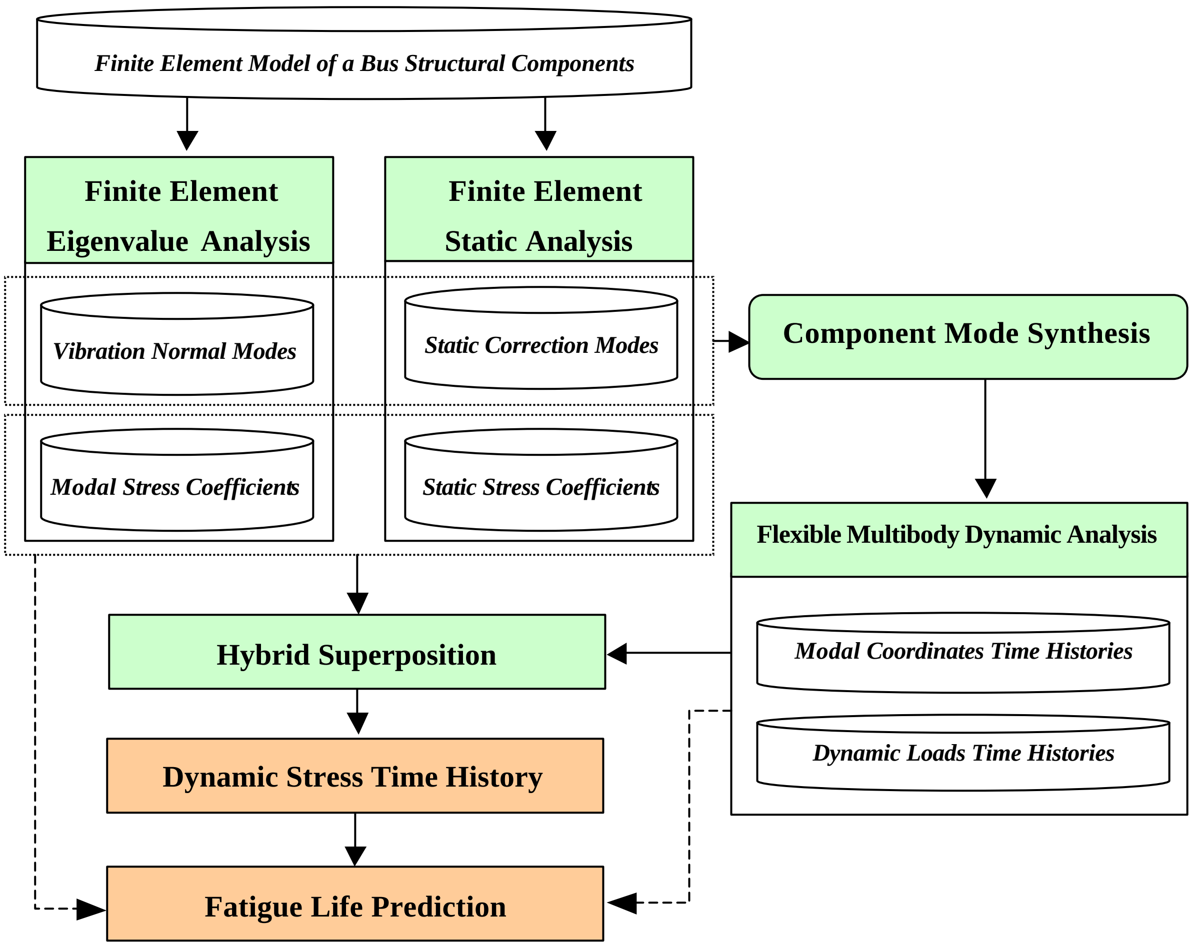


Fig. 1 Conceptual procedure and data flow of the hybrid superposition method

Using existing flexible multibody dynamic and finite element analysis codes such as DADS and MSC.Nastran, the hybrid superposition method can be implemented as follows.

Step 1. For flexible multibody dynamic analysis, component deformation modes must be selected for flexible bodies in the bus system that may require finite element vibration and static analysis using finite element analysis codes. In flexible multibody dynamic analysis, the sub-structuring technique may be used, if critical region can be modeled as a separated body from the remaining structure.

Step 2. Using flexible multibody dynamic analysis code, the bus system of interest is simulated under the specified scenario to obtain dynamic load time histories applying on each structural component and modal coordinate time histories of each component.

Step 3. To calculated static stress coefficients in Eq. 3, finite element static analysis is carried out through applying each unit load at the finite element node points where corresponding dynamic load is applied. To calculate modal stress coefficients in Eq. 3, finite element eigenvalue analysis is carried out. For a finite element model that is high dimensional, the conventional sub-structuring technique may be used to calculate stress coefficients in local area.

Step 5. As in Eq. 3, static stress influence coefficients that are obtained in step 3 are multiplied by corresponding to magnitudes of dynamic load time histories from step 2. Modal stress influence coefficients that are obtained in step 4 are multiplied by corresponding to magnitudes of modal coordinates time histories from step 2. Performing the multiplication and summation at each integration time step yields dynamic stress time histories at every time step.

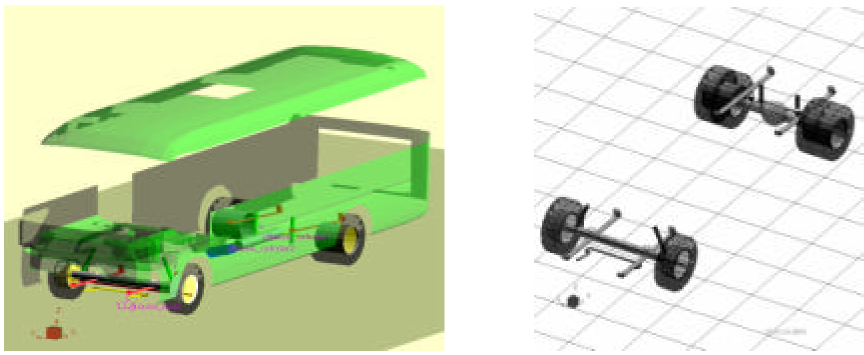
Step 6. Finally, fatigue lives of structural components are predicted by use of dynamic stress time histories from step 2 or hybrid superposition of results from step 2 and step 3.

NUMERICAL EXAMPLE

To demonstrate the proposed method, a real bus system in Fig. 2 is chosen. Dynamic stress time histories in the local area of the B-pillar joint structure are computed as the bus traverses a bump.

Figure 2. Mini Bus

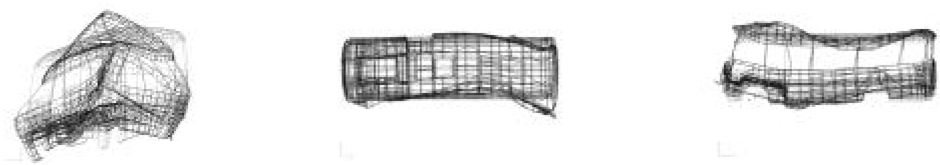
For dynamic simulations, The multibody model of the bus systems is modeled as in Fig. 3. To include nonlinear effects of leaf springs and shock absorbers, bump stoppers, stabilizer bars and bushes in the suspension system, nonlinear properties measured from component experiments are used. Mechanical parts, such as engine, air conditioner and other nonstructural mass elements are lumped to nodes of the body structure with spring and damper elements.



(a) Total system (b) Chassis

Figure 3. Multibody model of the bus systems

To consider the flexibility effect of the body structure, it is modeled as a flexible body. The MSC.Nastran code is employed for eigenvalue and static analyses with free boundary conditions to compute component modes. Component modes for the body structure are chosen as combination of four vibration normal modes in Fig. 4 and eight static correction modes; i.e. residual inertia relief attachment modes[(3),(7)]. Static correction modes are defined at the joints between body structure and leaf springs.



(a) Torsion (b) Lateral Bending (c) Vertical Bending

Figure 4. Vibration normal modes

The flexible multibody dynamic simulations are carried out by using DADS code with the scenario which the bus traverse a sinusoidal bump in Fig. 5 with a constant forward speed of 15km/hour.

The A-point in the B-pillar joint connected the waist rail in Fig. 6 is the critical area where a few durability problems have occurred. When the bus traverses over the bump, large dynamic stresses are anticipated in the joint.

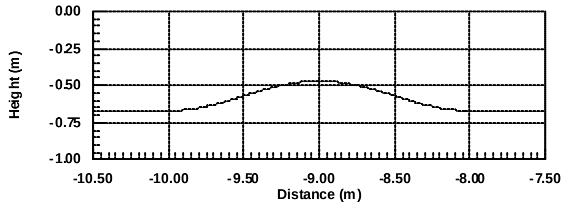


Figure 5. Bump profile

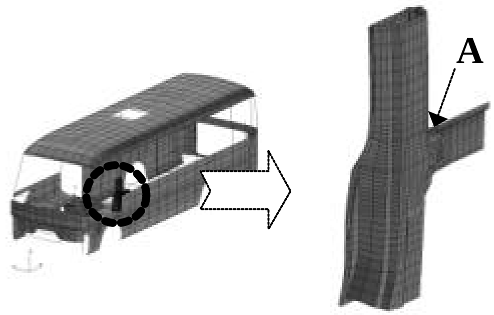
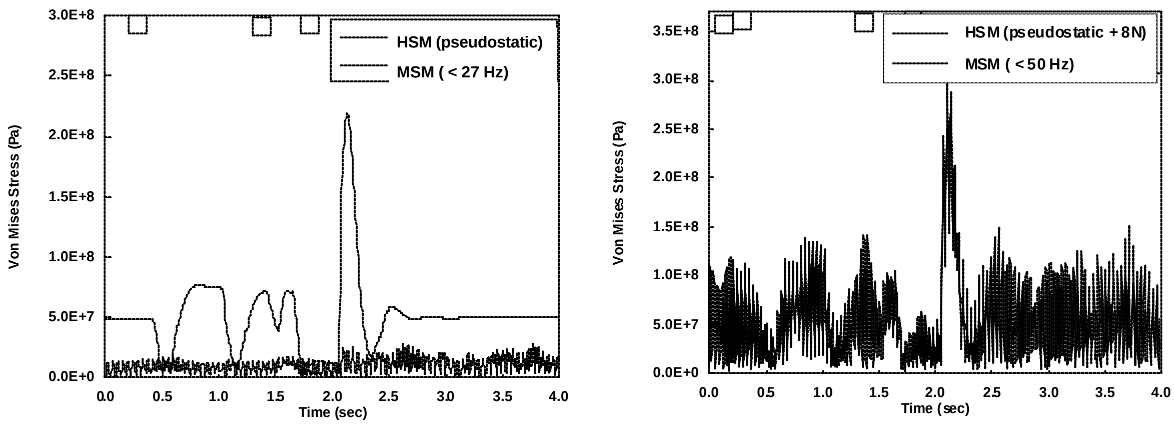


Figure 6. Critical area of B-pillar sub-structure

Dynamic load and modal coordinate time histories are calculated from flexible multibody dynamic simulation and stress coefficients are obtained from finite element static analyses.

Figure 7 shows the dynamic stress time histories at the A-point, which are obtained using the mode superposition method with vibration normal modes to 27Hz ; (MSM<27Hz) in (a) and 50Hz ; (MSM<50Hz) in (b) and the hybrid superposition method with only pseudostatic stresses ; (HSM pseudostatic) in (a) and pseudostatic stresses and eight fundamental vibration normal modes under 50Hz ; (HSM : pseudostatic + 8N) in (b). The figure shows that the modal superposition method will be improved by addition of more vibration modes in the dynamic simulation. On the other hand, the hybrid superposition method shows almost the same trend regardless of the number of vibration modes and will converge to the exact solution by addition of more vibration modes.



(a) Case 1 (b) Case 2

Figure 7. Dynamic stresses time histories

Fig. 8 shows FFT(Fast Fourier Transformation) of dynamic stress time histories in terms of frequencies, which are obtained using the mode acceleration method with vibration normal modes to 50Hz ; (MAM<50Hz) and the hybrid superposition method with pseudostatic stresses and eight fundamental vibration normal modes under 50Hz ; (HSM : pseudostatic + 8N). The figure shows that FFT of the hybrid superposition method are similar to that of the modal acceleration method. Using SGI Indigo R10000 engineering workstation, total solving time and hardware space of the hybrid superposition method, however, required 970.7 sec and 7.2 GB, respectively. On the other hand, them of the modal acceleration method required 2223.6 sec and 12.5 GB.

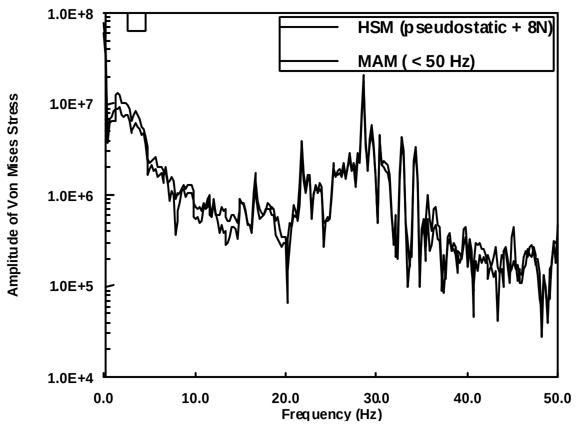


Figure 8. Fast Fourier Transform of dynamic stress time histories

These imply that the hybrid superposition method is more accurate than the mode superposition method and more efficient than the mode acceleration method. This figure clearly shows improvements of the proposed method over the conventional methods in the computation of the dynamic stress time histories in structural components of bus systems and the other mechanical systems.

CONCLUSION

In this paper, the hybrid superposition method is compared with the conventional methods; i.e. the mode superposition method and the mode acceleration method. In the mode superposition method, stress can be computed only when vibration normal modes are defined for the component. Thus, if the component is very stiff and is modeled as a rigid body, zero stresses will be obtained. Even for flexible bodies, the accuracy of stress that obtained will depend on the selection of vibration normal modes. In the hybrid superposition method, however, dynamic loads that are accurately predicted in flexible multibody dynamic simulation are used in stress calculation, hence avoiding the dependence on vibration normal modes. In the mode acceleration method, pseudostatic stresses are computed from a great number of static analyses, where the number of static analyses is equal to the number of total integration time steps. In the hybrid superposition method, however, the pseudostatic stresses are computed from a small number of static analyses, where the number of static analyses is equal to that of total dynamic loads, so that computing time and hardware resources are required less than the mode acceleration method. Therefore the hybrid superposition method is more accurate than the mode superposition method and more efficient than the mode acceleration method.

REFERENCE

1. Winfrey, R. C., "Elastic Link Mechanism Dynamics," Transaction of ASME, Journal of Engineering for industry, Vol. 93, N0. 1, pp. 268-272, Feb. 1971.
2. Liu, T. S., "Computational Methods for Life Prediction of Mechanical Components of Dynamic Systems," Ph. D. Thesis, The Univ. of Iowa, 1987.
3. Yoo. W. S., and Haug, E. J., "Dynamics of Articulated Structures. Part I : Theory," Journal of Structural Mechanics, Vol. 14, No. 1, pp. 105-126, 1986.
4. Kim, H. S., "Dynamic Stress Analysis of a Flexible Body in Multibody System for Fatigue Life Prediction," Ph.D. Thesis, Inha University, Incheon, Korea, 1999.
5. *DADS User's Manual*, Rev. 8.0, Computer Aided Design Software, Inc., IA 5221, 1995.
6. *MSC/NASTRAN User's Manual*, Ver. 70, The MacNeal-Schwendler Corporation, June 1997.
7. Craig. R. R., and Chang, C. J., "On the Use of Attachment Modes in Substructure Coupling for Dynamic Analysis," Dynamic & Structural Dynamics, AIAA/ASME 18^{th} Structures, Structural Dynamics & Material Conference.