

# Improvement of Virtual Vehicle Analysis Efficiency with Optimal Modes Selection in Flexible Multi-Body Dynamics

Geunsoo Ryu and Jaewook Jeon  
 Hyundai Motor Company

## ABSTRACT

In the analysis for durability or R&H performance with the full vehicle multibody models, the need for component flexibility is increasing along with demand for more precise full vehicle system. The component elastic deformations are usually expressed by modal superposition from component normal mode analysis with finite element model for reducing model size and simulation time. Although the simulation results of MBD analysis are more accurate according to increasing the number of flexible body and modes, the increasing of flexible components makes worse simulation time and convergence in MBD analysis. Especially, in the MBD analysis including a flexible upper body, in substitution for large number degree of freedom FE model such as trimmed body, it should take a few times longer than the case of rigid upper body

This paper proposes the methods of reducing computational cost with adequate mode selections without the loss of simulation accuracy in the flexible MBD. An optimal modes selection method employs modal participation factor calculated with only modal deformation shapes at the interface nodes of flexible body. Examples are provided that demonstrate the performance of the method and also how to select essential modes for using only external (interface) modes shape information.

**CITATION:** Ryu, G. and Jeon, J., "Improvement of Virtual Vehicle Analysis Efficiency with Optimal Modes Selection in Flexible Multi-Body Dynamics," *SAE Int. J. Passeng. Cars - Mech. Syst.* 6(1):2013, doi:10.4271/2013-01-1193.

## INTRODUCTION

For dynamic analysis of ride comfort, vehicle handling and durability with Multibody Dynamic scheme(MBD), the component flexibility is important factor for increasing the accuracy of simulation results in many research. Especially, vehicle body global deformations accordance with torsional, bending stiffness effects on vehicle dynamic performance. Many investigations for a flexible multibody system which consists of elastic and rigid components have focused on how the flexible components can be formulated and combined with other component equations. For linear elastic structures, a physical model of a structure with applied loads can be transformed to the modal domain for reducing computational cost. Most flexible multibody dynamic scheme adopted this approach. In the MBD simulation, rigid vehicle upper body is substituted with modal based flexible vehicle upper body including mass, mode shapes and eigenvalues calculated from normal modes analysis of finite element trimmed body model. In practical engineering situations, it is commonly used criterion that all modes within a frequency range that is two or three times the maximum frequency of interest are retained. However, if the flexible part is complex and has a

large number of degree of freedom such as FE based trimmed vehicle body model as depicted in [figure 1](#), it is important to characterize the contribution of the subsystem modes of the flexible body.

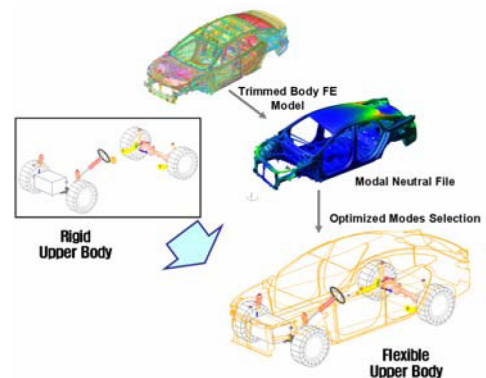


**Figure 1. Mutibody Dynamics Model using Flexible Chassis and Body Components**

In this paper an algorithmic basis for numerically quantifying the participation of modes of flexible body is developed. The algorithm is underpinned by known structural dynamic, and specifically, modal analysis theory. The justification for such a process stems from the need to understand the influence of the original

## FLEXIBLE MULTIBODY DYNAMIC

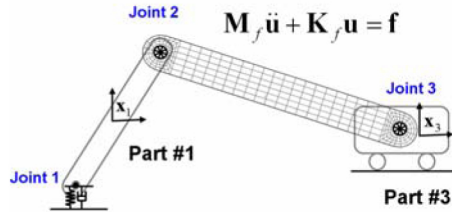


Figure 2. Four bars crank-slider system model with a flexible connecting rod

### Rigid multibody dynamics

In the simulation software for multibody dynamics, mechanical systems, consisting of a finite number of bodies, are interconnected in a way that allows for a large relative motion between the bodies. These interconnections consist of force elements, such as spring, dampers and joints. Joints constrain the relative motion between the interconnected bodies and as a result they are cause of constraining forces. If connecting rod part (2<sup>nd</sup> part) in figure 2 is rigid, general purpose multibody programs like ADAMS, which are widely used in industry, generate both the kinematics and dynamic equations formed with DAE(Differential Algebraic Equation) as shown in equation (1).

$$\begin{bmatrix} \mathbf{M}_T & \mathbf{M}_{TR} \\ \mathbf{M}_{TR}^T & \mathbf{M}_R \end{bmatrix}^i \begin{Bmatrix} \ddot{\mathbf{x}} \\ \dot{\omega} \end{Bmatrix}^i + \begin{bmatrix} \mathbf{0} \\ \omega \times \mathbf{M}_R \omega \end{bmatrix}^i = \begin{Bmatrix} \mathbf{F} \\ \mathbf{T} \end{Bmatrix}^i \quad (i=1,3)$$

$$\Phi = \Phi(\mathbf{x}, \theta, t) = 0 \quad (1)$$

where  $\ddot{\mathbf{x}}$  and  $\dot{\omega}$  are respectively translational acceleration and angular acceleration of  $i^{\text{th}}$  part. The matrix  $\mathbf{M}_T$  is associated with the translation of body reference coordinates and a constant diagonal matrix composed of total mass of body. The matrix  $\mathbf{M}_{TR}$  represents the inertia coupling between the translation and rotation of the body reference. We may also write the matrix  $\mathbf{M}_R$  associated with the rotational coordinates of the body reference. Kinematic constraints between different components in the multibody system can be written in a vector form as  $\Phi$  linearly independent constraint equation. The kinematic constraints can be described as the general system differential equations of motion with Lagrange's multipliers

### Mode-based flexible multibody dynamics

In the four bars crank-slider system model in figure 2, the connecting rod part (2<sup>nd</sup> part) is deformable body in space.

This deformable body can be expressed with finite element model having mass, stiffness matrix and nodal displacement vector  $\mathbf{u}$  as depicted in equation (2)

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{F} \quad (2)$$

The commercial multibody dynamics code such as ADAMS employs modes-based reduced order modeling for flexible components. The deformable body in equation (2) can not directly be used in multibody dynamic code because finite element parts have too large number of degree of freedom to solve with other rigid body components in general DAEs solver. The modes-based reduced order models are embedded in DAEs and solved with other rigid body components for computation efficiency. the nodal deformation of deformable body in equation (2) can be written as

$$\mathbf{u}(t) = \sum_{p=1}^N \Psi_p \mathbf{q}_p(t) \approx \sum_{p=1}^M \Psi_p \mathbf{q}_p(t) \quad (3)$$

where  $N$  is the number of degree of freedom in finite element model,  $\Psi_p$ ,  $\mathbf{q}_p$  are each  $p^{\text{th}}$  mode shape vector and modal coordinate as a function of time. In equation (3), we can make reduced order model approximated with adequate  $M$  number of modes. It is not necessary to use all mode shapes for most practical problems. It is general idea that good approximate solutions can be obtained via superposition with only first few mode shapes. The matrix  $\Psi$  can be normalized with mass matrix  $\mathbf{M}$  and then one can transform general nodal coordinate in equation (2) to modes based coordinate  $\mathbf{q}$  as equation (6)

$$\Psi^T \mathbf{M} \Psi \ddot{\mathbf{q}} + \Psi^T \mathbf{K} \Psi \mathbf{q} = \Psi^T \mathbf{F} \quad (4)$$

$$\Psi^T \mathbf{M} \Psi = \mathbf{M}_g = \begin{bmatrix} 1 & 0 & 0 \\ 0 & \ddots & 0 \\ 0 & 0 & 1 \end{bmatrix}$$

$$\Psi^T \mathbf{K} \Psi = \mathbf{K}_g = \begin{bmatrix} \omega_1^2 & 0 & 0 \\ 0 & \ddots & 0 \\ 0 & 0 & \omega_M^2 \end{bmatrix} \quad (5)$$

$$\mathbf{M}_g \ddot{\mathbf{q}} + \mathbf{K}_g \mathbf{q} = \Psi^T \mathbf{F} = \mathbf{F}_g \quad (6)$$

This type of modal data has mode shapes and natural frequencies including constraint modes and can be exported by commercial finite element analysis software such as NASTRAN, and/or ABAQUS with modes based modal stress vector. The configuration of a deformable body is identified by using two sets of coordinates: reference and elastic coordinates. Reference coordinates define the location and orientation of a selected body reference. Elastic coordinates

on the other hand, describe the body deformation with respect to the body reference. The motion of the body is then defined as the motion of its reference plus the motion of the material points on the body with respect to its reference as follow [equation \(7\)](#)

$$\begin{bmatrix} \mathbf{M}_T & \mathbf{M}_{TR} + \mathbf{M}_{TRq}(\mathbf{q}) & \mathbf{M}_{Tq} \\ \mathbf{M}_{TR}^T + \mathbf{M}_{TRq}^T(\mathbf{q}) & \mathbf{M}_R + \mathbf{M}_{RRq}(\mathbf{q}, \dot{\mathbf{q}}) & \mathbf{M}_{Rq}(\mathbf{q}, \dot{\mathbf{q}}) \\ \mathbf{M}_{Tq}^T & \mathbf{M}_{Rq}^T(\mathbf{q}, \dot{\mathbf{q}}) & \mathbf{M}_g \end{bmatrix} \begin{Bmatrix} \ddot{\mathbf{x}} \\ \dot{\mathbf{q}} \\ \ddot{\mathbf{q}} \end{Bmatrix} + \mathbf{Q}(\mathbf{q}, \dot{\mathbf{q}}, \mathbf{x}, \boldsymbol{\omega}) + \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & \mathbf{K}_g \end{bmatrix} \begin{Bmatrix} \mathbf{x} \\ \boldsymbol{\omega} \\ \mathbf{q} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F} \\ \mathbf{T} \\ \mathbf{F}_g \end{Bmatrix} \quad (7)$$

In [equation \(7\)](#), the mass matrix of the flexible body includes coupling terms of body deformation and translation and rotational rigid body motion written as  $\mathbf{M}_{Tq}$ ,  $\mathbf{M}_{TRq}(\mathbf{q})$ ,  $\mathbf{M}_{Rq}(\mathbf{q}, \dot{\mathbf{q}})$ ,  $\mathbf{M}_{RRq}(\mathbf{q}, \dot{\mathbf{q}})$ . These mass matrix terms are not constant matrix changing value of the magnitude and the rate of body deformation.  $\mathbf{Q}(\mathbf{q}, \dot{\mathbf{q}}, \mathbf{x}, \boldsymbol{\omega})$  is the quadratic velocity that includes the gyroscopic and coriolis force components

## OPTIMAL MODES SELECTION (OMS)

For linear elastic structures, a physical model of a structure with applied loads can be transformed to the modal domain and expressed for the reduction of analysis time and efforts. In practical engineering situations, it is commonly used criterion that all modes within a frequency range that is two or three times the maximum frequency of interest are retained. However, if the flexible part is complex and a large number of degree of freedom such as FE based trimmed vehicle body model, it is important to characterize the contribution of the subsystem modes of the flexible body to the multibody dynamic system.

## Multibody Dynamic Analysis Including Flexible Body Components

In previous chapter, the equation of motion having rigid components and flexible body component is mentioned. The connecting rod in crank -slider model as depicted in [figure 2](#) can be written as [equation \(7\)](#), other rigid parts can be written as [equation \(1\)](#). In the multibody systems, the system coordinates are not independent because of the specified motion trajectories as well as mechanical joints such as universal, prismatic and revolute joints. The kinematic constraints can be introduced to the dynamic formulation by using a set of nonlinear algebraic constraint equation. At the flexible body part, the constraint equation  $\Phi(\mathbf{x}, \boldsymbol{\theta}, \mathbf{u}, t) = 0$  is the function of interface point nodes as shown in [figure 3](#)

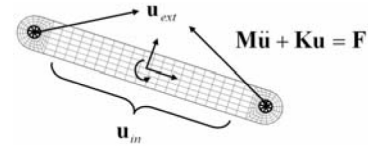


Figure 3. External node definition of flexible connecting rod in [figure 1](#)

The displacement field vector of flexible body can be expressed with  $\mathbf{u}$ , in the body fixed reference coordinates system, can be written in terms of linear combination of finite series of mode shape vectors in [equation \(3\)](#). This displacement field can be approximated with M number of modes including normal modes(component free vibration modes) and constraint modes and also divided by external nodes(interface nodes) and internal nodes as follow

$$\mathbf{u} = \begin{Bmatrix} \mathbf{u}_{in} \\ \mathbf{u}_{ext} \end{Bmatrix} = \begin{bmatrix} \Psi_{nor}^{in} & \Psi_{con}^{in} \\ \Psi_{nor}^{ext} & \Psi_{con}^{ext} \end{bmatrix} \begin{Bmatrix} \mathbf{q}_{nor} \\ \mathbf{q}_{con} \end{Bmatrix} \quad (8)$$

where mode shape matrix  $\Psi$  can be partitioned with 4 sub-matrix forms  $\Psi_{nor}^{in}$ ,  $\Psi_{con}^{in}$ ,  $\Psi_{nor}^{ext}$ ,  $\Psi_{con}^{in}$  and also modal coordinates  $\mathbf{q}$  can be divided by two types of motions. Then the integrator of analysis solver finds the solution each part of flexible multibody dynamics satisfying constraint equation  $\Phi(\mathbf{x}, \boldsymbol{\theta}, \mathbf{u}, t) = 0$  between rigid and flexible parts in whole time steps. One can know that the constraint equation related to flexible body has a set of only  $\mathbf{u}_{ext}$  as follow

$$\begin{aligned} \mathbf{u}_{ext} &= \Psi_{nor}^{ext} \mathbf{q}_{nor} + \Psi_{con}^{ext} \mathbf{q}_{con} \\ &= \Psi_{ext}^{ext} \mathbf{q} \end{aligned} \quad (9)$$

In [equation \(9\)](#), the displacement vector of external nodes  $\mathbf{u}_{ext}$  is superposition of only the mode shapes of external nodes and the solution of modal coordinates in [equation \(7\)](#) as follow

$$\ddot{\mathbf{q}}_{nor} + \mathbf{K}_g^{nor} \mathbf{q}_{nor} = \Psi_{nor}^{ext T} \mathbf{F}_{ext} + \mathbf{R}_{nor} \quad (10)$$

$$\ddot{\mathbf{q}}_{con} + \mathbf{K}_g^{con} \mathbf{q}_{con} = \Psi_{con}^{ext T} \mathbf{F}_{ext} + \mathbf{R}_{con} \quad (11)$$

[Equation \(10\)](#), [\(11\)](#) are general forms of modal transient equation including the coupling effects  $\mathbf{R}$  of rigid and deformable body motion. As a results of [equation \(10\)](#), [\(11\)](#) in the multibody dynamic analysis including flexible parts, flexible part can has only the information of mode shapes of external nodes and eigenvalues of component.