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The Powerful Linear Subsystem Solver for Flexible Bodies

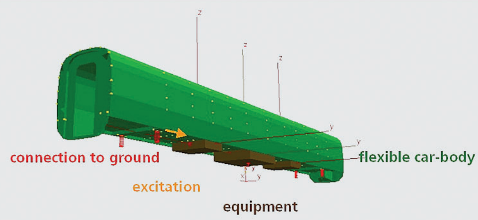

Fig. 1 A Flexible Car-Body of a Railway Vehicle with Attached Devices

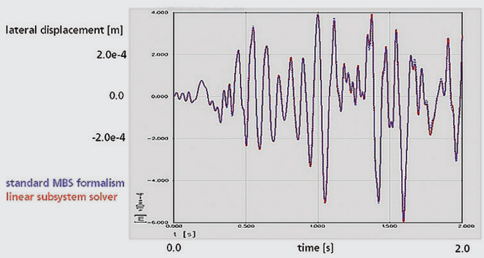

Fig. 2 The Lateral Displacement of the Mid-Point of the Car-Body's Floor Calculated by the Standard MBS Formalism and the New Linear Subsystem Solver

The modal approach is a powerful method to represent flexible bodies in multi-body systems. In many cases only a small number of modal degrees of freedom are required to correctly represent the flexible body deformation in SIMPACK, meaning the simulation tasks can be performed within seconds. However, for a large number of interface loads or moved loads on flexible bodies the number of modal degrees of freedom can drastically increase. The new solver enables the user to considerably increase the number of modal degrees of freedom without a significant increase in the required computational time. This enables improved accuracy and an easier handling of the FEMBS interface.

STATE OF THE ART

The integration of flexible bodies in multi-body systems is an advanced technology, but can still be performed without difficulty. A detailed finite element representation of a vehicle component, needs to be reduced to a modal representation to be incorporated into a multi-body model. Limiting the number of modes enables small and fast simulation models, however this raises the question of whether enough modes are selected for the correct representation of the elastic body.

In current multi-body system codes, the equations of motion consist of a combination of the linear modal representation of flexible bodies and the non-linear equations that describe the surrounding multi-body system, which includes the non-linear rigid body motion of the flexible bodies. The system of linear and non-linear equations is then solved using time integration procedures that are designed for non-linear mathematical problems.

WHY MORE POWER?

The increasing demands regarding the level of detail in the flexible body representation in multi-body systems leads to both an increased number of modes and linear equations. The representation of local deformations near the attachment points of a flexible body, for example, requires a large number of high-frequency modes that reduce the step sizes during the time integration and therefore lengthen the computational times. When moving loads on flexible bodies the problem becomes apparent; for each nodal position of the moved load, a frequency response mode has to be calculated in order to represent the local deformation under this load. This, therefore, drastically increases the number of required modes.

With this new solver technique the complete system of equations is subdivided into the linear modal equations representing the flexible body and the non-linear equations that represent the surrounding multi-body system. A powerful semi-analytic solver is applied to the linear modal equations and the non-linear part is still solved by the standard integration procedures for non-linear multi-body systems. These two methods are then coupled together as a co-simulation. An equidistant time grid is used to exchange data between both subsystems. Meanwhile, within the macro steps, the linear subsystem and the non-linear equations are solved independently of each other, which is clearly advantageous for the solution of the linear modal equations. Using orthogonal mode shapes means a single modal equation for each mode is achieved, which is then used to represent the flexible body deformation. The linear problem, with n mode shapes, is therefore solved with n equations, instead of having to solve n x n equations. In addition, each modal equation can be analytically solved, if the generalised modal loads are assumed to be polynomial functions and are updated within each new macro step. Finally, the linear subsystem solver evaluates n analytical functions, similar to the so-called Duhamel's integration. The result is, with the linear subsystem solver, that the calculation time of the flexible body in the multi-body system depends almost linearly on the number of selected modes.

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The co-simulation scheme of a non-linear multi-body system and a linear flexible subsystem enables the user to use a large number of modes and set-up very detailed flexible body representations for multi-body system analyses, whilst keeping the computational time down.

EXAMPLES

Figure 1 shows the car-body of a railway vehicle, which has various different devices attached - represented by rigid body masses. Local deformations of the car-body are caused by the connections between the car-body and the devices and the secondary suspension springs, which were modelled as linear spring damper elements connected to ground. These forces in the multi-body system can influence the car-body deformations and for each of the interfacing degree of freedoms a frequency response mode is required. Additionally, normal modes from 9.5 Hz up to 40.5 Hz were considered, and altogether a total of 81 modes were required to calculate the flexible body deformation precisely within SIMPACK. The excitation was applied in the longitudinal direction, at excitation frequencies up to 30 Hz, at an attachment point of a yaw damper on the car-body.

Figure 2 shows a comparison of the standard MBS formalism and SIMPACK's new MBS formalism with the built-in linear subsystem solver. The results for the lateral movement at the mid-point of the floor of the car-body are almost coincident, whilst the SIMPACK's linear subsystem solver, for this application, is 10 times faster than the standard formalism. The calculation time for the new solver is approximately linear with the number of selected modes, whereas the standard MBS formalism shows a tremendous increase in the calculation time. This is caused mainly by the high frequencies in the frequency response modes and the large number of modes, see Table 1.

CONCLUSION

The linear subsystem solver enables the user to select a large number of modes to represent flexible bodies in SIMPACK without significantly increasing computational time. The advantages are both increased accuracy and a simplified mode selection. INTEC is currently testing the new solver for both stability and accuracy.

number of modes	computation time standard MBS formalism [s]	computation time linear subsystem solver [s]
29	11.4	8.9
51	29.6	16.8
81	269.1	26.9

number of modes	frequency range of eigenmodes [Hz]	frequency range of frequency resp. modes [Hz]
29	9.56 - 40.55	-
51	9.56 - 40.55	55.24 - 198.87
81	9.56 - 40.55	55.24 - 1151.50

Tab. 1 Comparison of the Calculation Times Obtained by the Standard MBS Formalism and the New Linear Subsystem Solver