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MODELING ISSUES FOR REAL-TIME PERFORMANCE IN FLEXIBLE MULTIBODY SYSTEMS

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Abstract. *In recent years, our group has worked on real-time methods for the dynamics of flexible and rigid multi-body systems. In order to develop such methods, attention must be paid, in an integrated way, to modeling issues, formulation of the equations of motion, numerical integration procedures, and implementation techniques. Achievement of real-time performance in flexible multi-body systems on conventional PC platforms is commonly more challenging than in the rigid case. It can be stated that most methods developed for rigid multi-body systems can be extended to consider flexible members. Consequently, conclusions and recommendations drawn when studying rigid mechanisms in order to achieve real-time performance can be extrapolated to the flexible case. However, some additional "degrees-of-freedom" arise when simulating the motion of flexible multi-body systems, which must be managed by the analyst. They are the choice, for each of the flexible bodies, of the accuracy level to be applied for the description of the corresponding flexible component. This is something affecting the modeling stage, and occurs no matter the selected modeling technique is. In this paper, the described topic is addressed, in the context of a floating reference frame formulation with modal superposition of static and dynamic modes with fixed boundaries, previously developed by the authors. The modal selection process of a complex multi-body system is studied in detail in order to show the effect that the modeling of flexible bodies has on the computational efficiency of the method, emphasizing the need of reaching a compromise between accuracy and efficiency at the time of selecting both the static and dynamic modes. A good selection is considered to be a critical factor when pursuing real-time performance.*

1 INTRODUCTION

During the last years, the research team of the mechanical engineering laboratory has developed a method for the real-time analysis of rigid-flexible Multi-Body Systems (MBS) [1, 2]. Kinematics of the flexible bodies is introduced through the moving frame approach, along with modal superposition to describe the corresponding local deformations [3], carried out by means of both static and dynamic modes defined with respect to a tangent frame [4]. The analyst must decide the number of static and dynamic modes that are relevant in the analysis.

Frequently, a key factor in the diffusion and use of a MBS methodology is the easiness to work with it. When a certain multi-body system, containing flexible bodies, is to be studied through the authors' method, four kinds of parameters are left to the analyst decision. Two of them are related to flexible components modeling: the number of dynamic modes and the finite element density mesh. The other two are related to numerical issues: the time step size and the penalty value to amplify the internal forces caused by constraint violations. A previous work [5] points out that the four mentioned parameters are not independent. For a certain analysis, an optimum number of dynamic modes exists which leads to the corresponding optimum values of the other three parameters. So, at this moment it is interesting to make a study of the modeling parameters which help the analyst to do a modal selection process of the flexible components, in order to obtain optimum results in terms of accuracy and efficiency when employs the mentioned method to perform a MBS simulation.

This method employs natural coordinates [6], points and unit vectors, which connect the body with other mechanism components, called the boundary points and vectors. Moreover, a local reference frame is defined at each body by means of a point and three orthogonal unit vectors. A static mode is defined as the deformed body configuration resulting when a unit displacement of a boundary point (or a unit rotation of a boundary vector) is imposed, while maintaining the rest of the boundaries, points and vectors, fixed. Then, the number of the static modes of a flexible body is determined by the number of points and unit vectors used to model the body. If desired, the analyst could neglect some of these deformation modes by imposing the corresponding constraint equation of null amplitude.

Dynamic modes are defined as the body natural vibration modes with fixed boundaries. These internal deformation modes have the role of improving the representation of the deformation field given by the static modes. It is the analyst work to decide the number of dynamic modes and to select the most important ones. A detailed description of the kinematics of this method can be found in [7].

2 ANALYSIS OF THE FLEXIBLE MULTIBODY MODELING PROCESS

The modal selection is a crucial process in the analysis of a flexible MBS, because it influences both the simulation accuracy and efficiency. The authors' method employs two kinds of deformation modes: static and dynamic.

As mentioned above, the number of static modes is automatically determined by the body modeling. However, the static modes which have no influence in the body deformation could be neglected. For example, it is a common practice to a priori neglect the axial deformation modes of a slender beam. For a simple mechanism, all the static modes could be included and the efficiency would not be influenced. On the contrary, for a complex mechanism a selection of static modes would be essential because the efficiency could be seriously affected.

When dealing with modeling issues, a common idea is that the more dynamic modes are introduced, better accuracy of the results is obtained. However, this in not always true, because the introduction of higher frequency modes in the flexible MBS simulation could hinder the integration process, thus increasing the simulation error. This problem could be avoided by reducing the time-step size, but then the efficiency would be decreased too. Moreover, the efficiency is always decreased by the introduction of a larger number of modes than needed. There are two reasons: the time-step reduction and the size of the finite element mesh needed to fit correctly the high frequency dynamic modes.

Mode selection is not a simple task, since the interesting modes are not always those under a certain maximum frequency, as established in previous works [4, 8, 9]. When simple systems are analyzed, a large number of dynamic modes can be tried, along with a trial-error methodology to achieve an accurate and efficient model of the flexible multi-body system. However, with complex systems, the simulation process is not so fast and a modal selection process would be convenient.

Recently, the Modal Participation Factors (MPF) method has been successfully employed [8] to estimate the dynamic modes which must be included in the model of a complex flexible multi-body system. Such method has been adapted to the example we are dealing with in this paper, in order to quantitatively estimate the relevance of the modal selection process. To this end, rigid-body simulation of the system has been carried out, and the most critical position has been identified. The complete process is described in the next section.

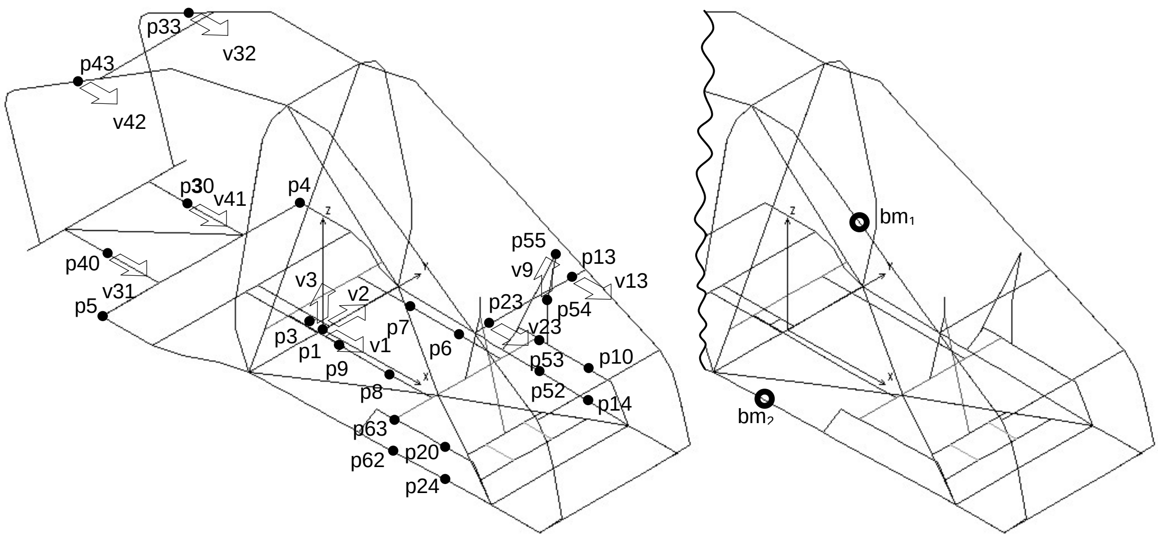

Figure 1: a) Boundary points and vectors of a flexible chassis model, b) Points where results are obtained.

3 MODAL SELECTION IN A CAR SIMULATION

In an earlier work [2], a complex multi-body system was analyzed. It consists of a car, where all the components were considered rigid except the chassis, a structure in steel tubing shown in Figure 1, which was considered a flexible body. The whole model consists of 58 points, 16 unit vectors, 5 distances, 1 angle, 60 static modes and 5 dynamic modes, which gives a total number of variables of 293 for the whole multi-body system.

A maneuver simulation lasting 15 seconds was performed. Starting from rest, the car accelerates and travels direct until reaching an approximate speed of 5 m/s, then drops down a kerb of 8 cm height, and finally brakes until complete stop is attained. In this way, the bending modes of the chassis are excited.

The bending moments were obtained at two points, bm_1 and bm_2, which are located in the chassis bars indicated in Figure 1.

For this simulation, a preliminary modal selection process was performed. All the static and dynamic modes which could reasonably influence the system behavior were included.

The local reference frame is defined by point p1 and unit vectors v1, v2, v3 (see Figure 1), so that chassis boundaries are 24 points and 10 vectors. In theory, we must include three modes for each boundary point and two modes for each boundary unit vector. Then, the total number of possible static modes is 92. However, not all these modes were taken into account since the contribution of some of them to the chassis deformation can be neglected, as in the case of static modes producing axial deformation of the steel tubes. So, only 60 static modes were considered. Besides, 5 dynamic modes were included, covering the natural frequencies up to 100 Hz. The MBS car model was validated by comparing the resulting stresses at the mentioned points of the chassis coming from both measurement and simulation. So, it can be said that the employed modal basis allowed the correct modeling of the system.

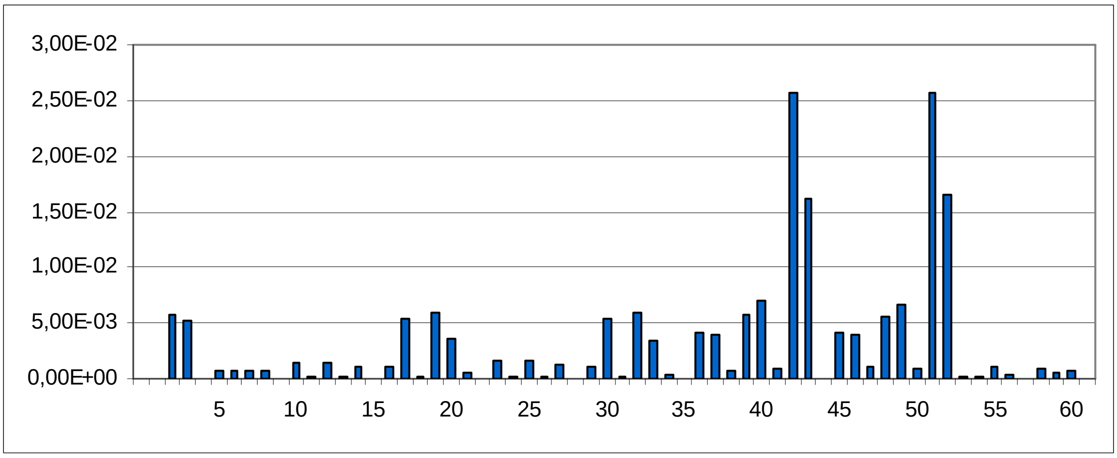

Figure 2: Maximum amplitude of each static deformation mode.

M	**MPF (%)**	**M**	**MPF (%)**	**M**	**MPF (%)**	**M**	**MPF (%)**	**M**	**MPF (%)**	**M**	**MPF (%)**
1	0	11	1	21	2	31	0	41	4	51	100
2	22	12	6	22	0	32	23	42	100	52	64
3	20	13	1	23	6	33	14	43	63	53	1
4	0	14	4	24	1	34	2	44	0	54	0
5	3	15	0	25	6	35	0	45	16	55	4
6	2	16	4	26	1	36	16	46	16	56	1
7	3	17	21	27	5	37	15	47	4	57	0
8	2	18	0	28	0	38	3	48	22	58	3
9	0	19	23	29	4	39	22	49	26	59	2
10	6	20	14	30	21	40	27	50	4	60	3

Table 1: Modal Participation Factors (MPF).

In order to carry out the modal selection process in complex systems, the MPF method will be applied to the flexible chassis. The objective is to obtain a new chassis model, where the non relevant modes are neglected. This new flexible body model will improve the efficiency of the car simulation while maintaining the accuracy.

In this case, we have an important advantage: a correct and validated modal basis is available and employed as reference. The MPF can be obtained directly from the amplitude of the deformation modes at the most critical instants of the simulation. Looking at the stress peaks on plots presented in [2], it is difficult to select only one critical point of time. Therefore, three points of time have been selected: 5.67 s, 6.81 s and 8.04 s. For each static mode, the maximum amplitude reached among the three critical instants has been selected and represented in Figure 2, so that the most relevant modes can be detected.

The MPF are obtained employing these maximum amplitudes, as a percentage of the largest modal amplitude. They are shown in Table 1, which includes the same information offered in Figure 2 but in a numerical way.

The modal selection process is established considering the MPF values. Table 2 shows four different models of the chassis. Each of them includes a different number of deformation modes. The modal selection criterion is displayed in the second column. The number of selected modes appears in the third column.

Model	Selection criterion for modal inclusion	Number of included modes	CPU-Time (s)	Error bm_1 (%)	Error bm_2 (%)
Reference	All modes included (No modal selection)	5 dynamic and 60 static modes	55	0.00	0.00
1	Only static modes	0 dynamic and 60 static modes	44	0.31	0.04
2	Modes with MPF > 1%	0 dynamic and 43 static modes	30	0.43	1.49
3	Modes with MPF > 10%	0 dynamic and 20 static modes	15	0.61	22.85
4	Modes with MPF > 20%	0 dynamic and 6 static modes	10	12.30	53.57

Table 2: Modal selection criterion and simulation results.

These four models have been employed to simulate the car maneuver mentioned above. The corresponding CPU-times have been registered in the fourth column of Table 2. The CPU-times have been obtained on an AMD Athlon XP 2600 @ 1.92 GHz computer processor. A bending moment error value, at points called bm_1 and bm_2, has been obtained for each simulation and reflected in the fifth and sixth columns of Table 2, respectively.

The bending moment error has been established for each simulation by comparing the actual simulation results with the reference simulation results. It has been calculated following Equation (1). First, the history of the bending moment has been recorded at every time-step for both the simulation of reference and the simulation being evaluated. Then, the error is obtained as,

$$error = \left(\frac{1}{1501} \frac{1}{\left| m \right|_{\max}} \sum_{i=0}^{1500} \left| m_i - m_i^* \right| \right) x\,100 \tag{1}$$

where 1501 is the number of values considered (steps of 1 ms during 15 s of simulation), m_i represents the history of the bending moment for the current simulation, m_i^* is the same mag

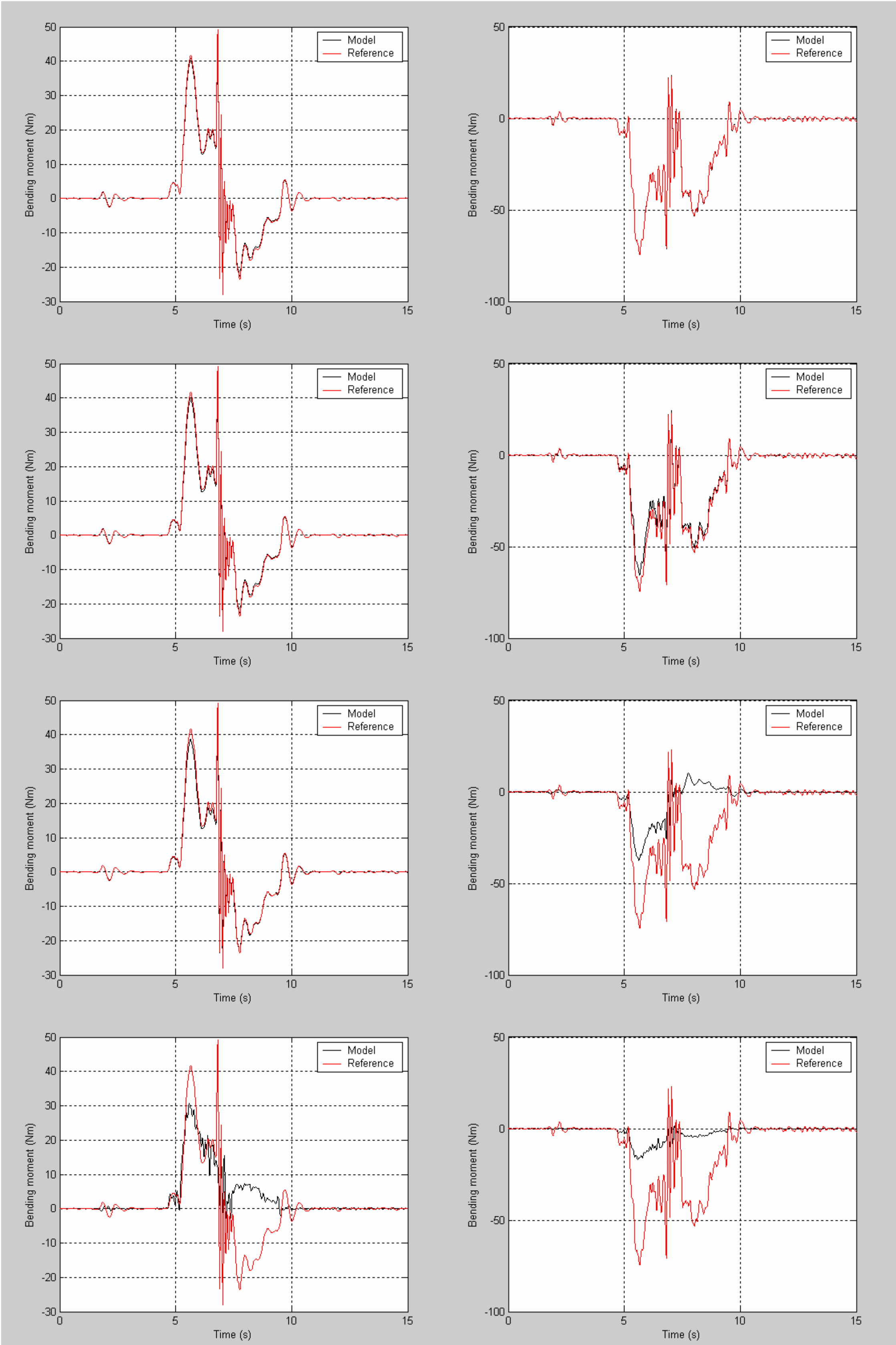

Figure 3: Comparison of the results: models 1, 2, 3 and 4 versus reference model.

nitude for the reference simulation, and $|m|_{max}$ is the maximum absolute value of the magnitude during the simulation. The resulting errors have the form of percentages.

The results of the four simulations proposed are shown in Figure 3. The plots which represent the history of the bending moment at point bm_1 are located on the left. Those which correspond to point bm_2 are located on the right. All plots compare the bending moment results of a model simulation with the reference model simulation.

In the first row of plots, the analysis of the results obtained through model 1 allows to establish that the dynamic modes have no influence. In the second row, the plots indicate that the use of 43 static modes instead of 60 lead to results which are similar in accuracy, but the efficiency is notably improved. A 45% CPU-time reduction is achieved by model 2 with respect to the reference model. In third row, the plots corresponding to model 3, which includes 20 static modes, indicate that the model is accurate enough to determine the bending moment at the top bar (point bm_1). Conversely, important errors have been obtained in the bending moment representation at the bottom bar (point bm_2). Real-time performance is achieved in this simulation. Finally, the plots on the fourth row show the comparison between model 4 and the reference model, indicating that this modeling cannot represent the bending moments undergone by the chassis in the maneuver.

4 CONCLUSIONS

- The authors have recently proposed a method for real-time dynamic analysis of rigid-flexible multibody systems, based on the floating frame of reference formulation, with static and dynamic modes. The analyst must decide the number of deformation modes and which of them are relevant in the simulation.
- The static and dynamic modal selection process influences the simulation accuracy and efficiency. For a specific problem, a larger number of modes always lead to a more inefficient simulation but not necessarily to a more accurate simulation. Moreover, the relevant modes do not have to be those of lower frequencies.
- The importance of the modal selection process in the achievement of real-time in complex rigid-flexible multi-body simulations has been quantitatively studied through an illustrative example.
- The authors’ method can benefit from the modal participation factors method to have an initial estimation of which are the most relevant modes. However, the usual criterion of selecting those modes which have amplitude greater than 1% of the maximum modal amplitude can result a little severe.

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