Experimental Validation of a Mass Transit Vehicle Multi-Body System with Integrated Flexible Body

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In recent years, greater emphasis has been placed on the design of high-speed, lightweight precision systems. The design and performance analysis of such systems can be greatly enhanced through transient dynamic simulations, provided that all significant effects are incorporated into the mathematical model. The need for better design, in addition to the fact that many mechanical and structural systems operate in adverse environments, demand the inclusion of many factors that have been ignored in the past.

When these systems are analyzed, neglecting deformation effects can lead to a mathematical model that poorly represents the real system. The interaction of software tools for MBS dynamics, such as SIMPACK, and software tools for finite elements analysis can allow for the optimization of vehicle design. The integration of body flexibility into these models allows for realistic simulation of structures, components and safety systems.

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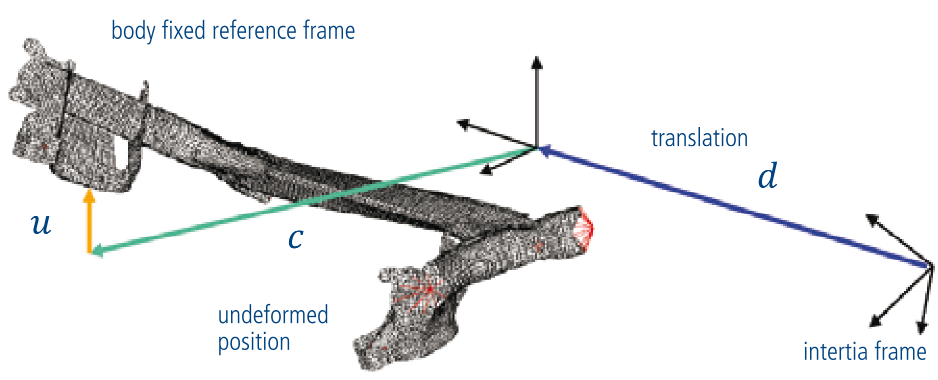

Fig. 1: Deformation description in MBA

This work is concerned with a mass transit vehicle's multi-body dynamics and the reliability of such analysis is proved by experimental validation, i.e., a comparison between vehicle accelerations recorded during running tests and corresponding data computed by simulations.

FLEXIBLE BODY INTEGRATION INTO MBS

In MBS theory, a flexible body motion can be obtained as the superposition of a large reference motion and small displacements with respect to the body reference frame. This gives rise to deformations, as shown in Fig. 1. In order to reduce the system order, this relative displacement field u is described by a linear combination of assumed shape functions ψ_j, with corresponding time dependent weighting factors q_j, similar to an FE approach:

$$(1)\quad u(c,t) = \sum_{j=l}^{n} \psi_j(c) \times q_j(t)$$

This approach, called modal [1], requires the calculation of the shape functions before performing the multi-body analysis, but it is advantageous because it leads to results very close to those obtained by an FE approach, despite using few modes. The mode shapes

Fig. 2: Master DOFs selection for dynamic reduction of carbody (FE Mesh -> nodes of super element)

could be calculated by a structural modal analysis of the single flexible body or static problems. In SIMPACK, all the features of a flexible body, like mass and stiffness matrices and modal content, are defined in a Standard Input Data file (SID), which can be generated by FEMBS, an interface software between SIMPACK and most of the commercial FEA tools.
Often, if FE models have many degrees of freedom (DOF), modal approach is not sufficient to reduce computational burden, and a reduction of dimensions of mass, stiffness and modal matrices is needed. This reduction can be performed by means of techniques like Guyan Reduction (or Static) or Craigh-Bampton Reduction (or Dynamic), each one defining a condensation matrix G_{nm} to transform the matrices of the model, as described below:

$$(2)\quad \begin{cases} u_n = G_{nm} \text{ x } u_m \\ M^{RED} = G^T_{nm} \text{ x } M^{FE} \text{ x } G_{nm} \\ K^{RED} = G^T_{nm} \text{ x } M^{FE} \text{ x } G_{nm} \end{cases}$$

where:

- u_n: vector of displacements of complete FE-model DOF
- u_m: vector of displacements of reduced FE-model DOF, named "Master Degrees"
- M^{RED}: mass matrix of the reduced FE-model
- K^{RED}: stiffness matrix of the reduced FE-model
- M^{FE}: mass matrix of the complete FE-model
- K^{FE}: stiffness matrix of the complete FE-model

The aim of these techniques is to use as small a number of DOF as possible, translating to a minimum error in the description of the dynamic behavior of the model in the frequency range of interest (see Fig. 2).

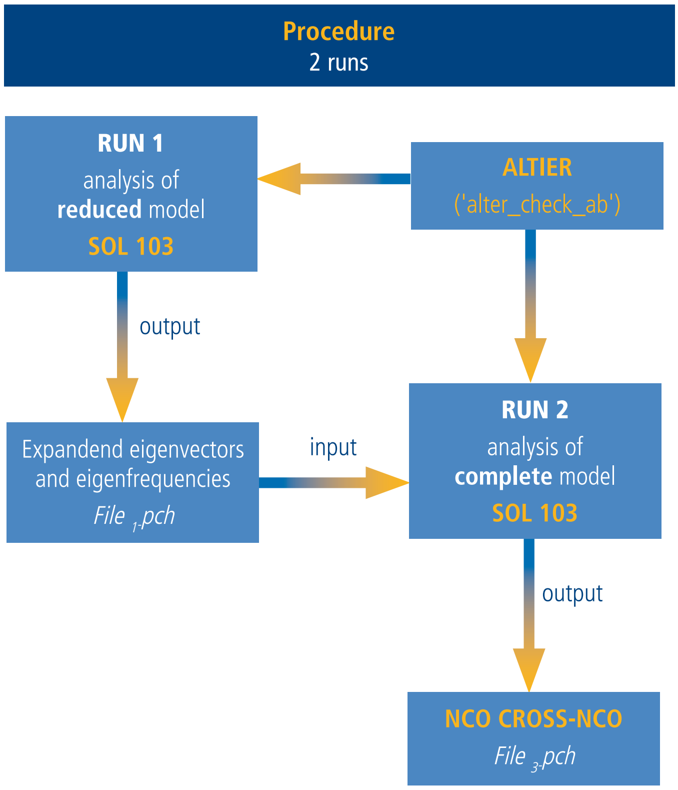

Fig. 3: Procedure for performing and validating FE model reduction

To ensure a proper representation of the full FE-model into the MBS, an analytical verification of reduction process has been conceived, developed and implemented. The procedure developed can be used to optimize the condensation of the full FE-model and consists of three steps:

1. Evaluation of the mode shape correlation between complete and reduced models using a vector criterion, the "Normalized Check Orthogonality", which defines a NCO index as follows:

   $$(3)\quad NCO_{ij} = \left| \Phi_i \text{ x } M^{FE} \text{ x } \Phi_j^{red} \right|$$

   Where M^{FE} is the complete mass matrix, Φ_i are the eigenvectors of the complete FE-model and Φ_i^{red}, the expanded eigenvectors of the reduced one. This index is defined to be in the numerical range between 0 and 1.
2. Evaluation of the dynamic behavior of the mode shapes that are similar, in terms of eigenfrequency difference;
3. Synthesis of a global index of correlation between reduced FE-model and complete FE-model as the ratio between the number of eigenmodes of the reduced FE-model characterized by a low error of NCO and eigenfrequency values, n_{red}, and the total number of eigenmodes of the complete FE-model in the frequency range of interest, n_{eg}.

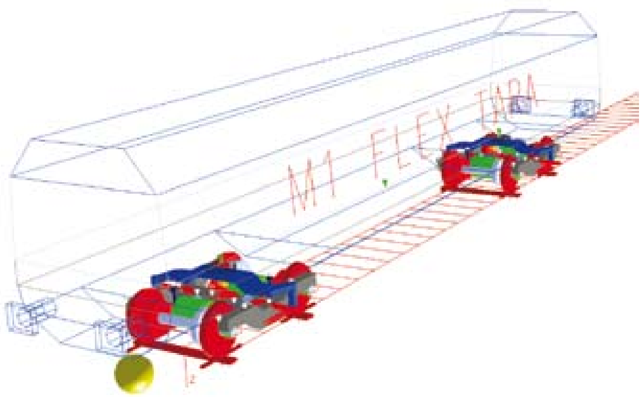

Fig. 4: MBS of Metro Rome Line C

A further check concerning the mode shape is the evaluation of the mutual cross-correlation of the expanded eigenvectors of the reduced FE-model (CROSS-NCO index), in order to assess the validity and consistency of the applied reduction technique. Due to matrix and vector sizes, common software for programming, like MATLAB®, cannot allow one to directly manage the modal analysis results for NCO evaluation due to hardware memory constraints. Therefore, an algorithm was developed directly by means of DMAP, "Direct Matrix Abstraction Pro-

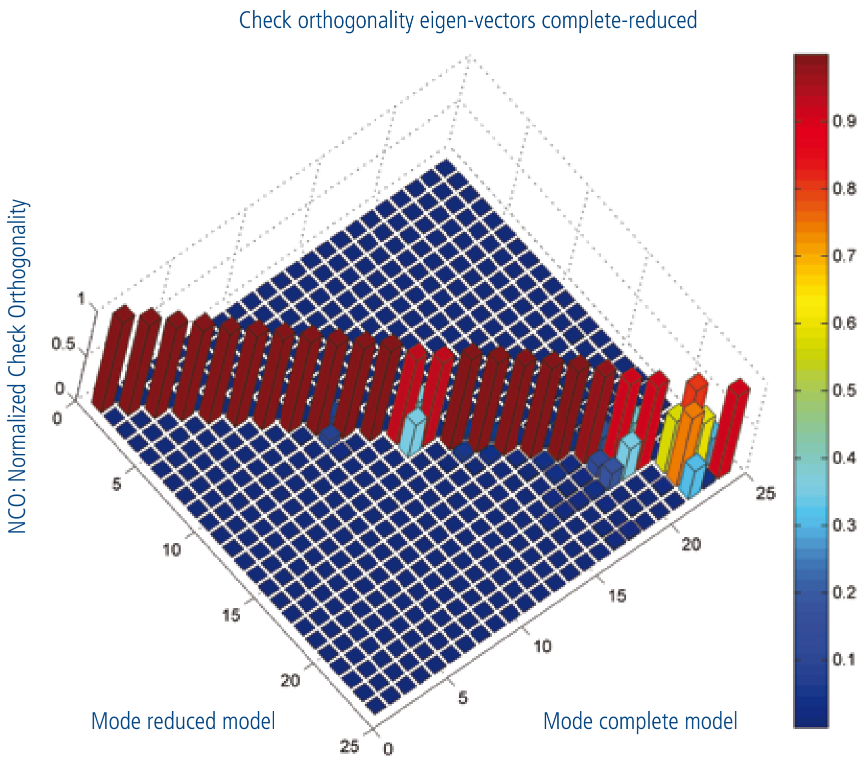

Fig. 5: NCO matrix of carbody reduced FE model

gram” of NASTRAN, which can be included in the Input File Deck for the modal analysis. Hence, a procedure has been implemented that reduces the FE-model and evaluates NCO indices in two runs of NASTRAN (see Fig. 3).

MULTI-BODY SYSTEM OF VEHICLE

The subject of this work is the mass-transit vehicle “Metro Roma linea C”, manufactured by Ansaldobreda, which will be in service in 2012. This is a bidirectional vehicle composed of six carriages with two running gears for each one; the general layout is symmetric. Hence, a simplified MBS of Metro Roma line C has been created. The model is composed of the head motor carriage only, as shown in Fig. 4, because it is known that each car body can be considered independent of the others in lateral and vertical dynamics, and both are more important than the longitudinal dynamics [4]. The modeled flexible body was the car body. This model was chosen in order to compare comfort indexes calculated with multi-body analysis with those coming from running test results. For the flexible body description, only 250 DOF have been retained using the dynamic reduction, obtaining a correlation global index of 92 % according to the validation procedure mentioned above. The NCO matrix is shown in Fig. 5.

EXPERIMENTAL VALIDATION OF MBS OF METRO ROMA-LINE C

The reliability of multi-body analysis results, in terms of flexible body implementation, has been proved by an experimental validation, i.e., a comparison between vehicle accelerations recorded during running tests and corresponding data computed by simulations.

In order to do this, test conditions were recreated. First of all, inertial properties of vehicle multi-body system were set according to a weighing test. Further, the test track was modeled in terms of geometry and measured irregularities. Accuracy of track description was demonstrated by comparing lateral and vertical accelerations of

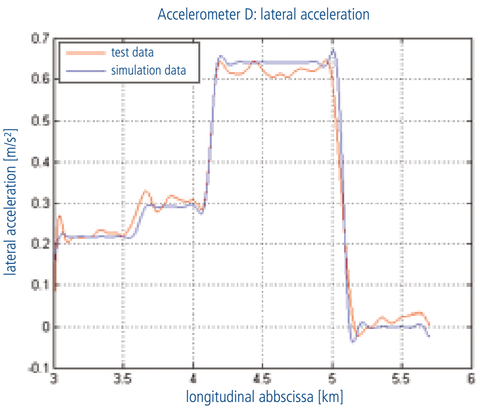

Fig. 6: Quasi-static signal comparison of bogie lateral acceleration in an emergency stop scenario

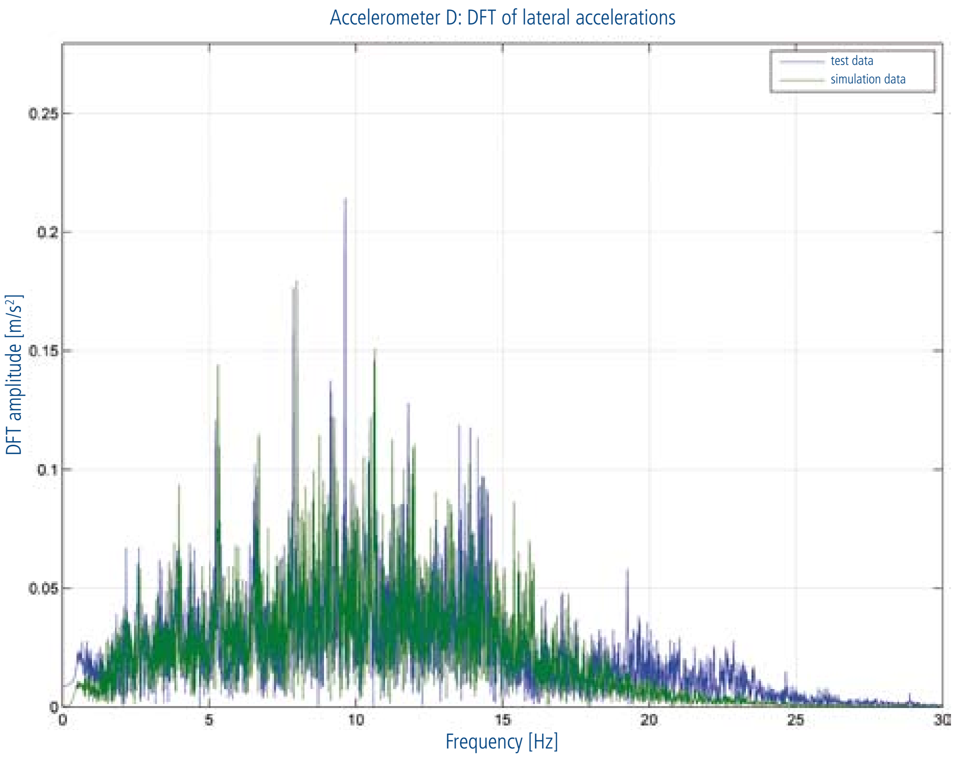

Fig. 7: Frequency content comparison of bogie lateral acceleration

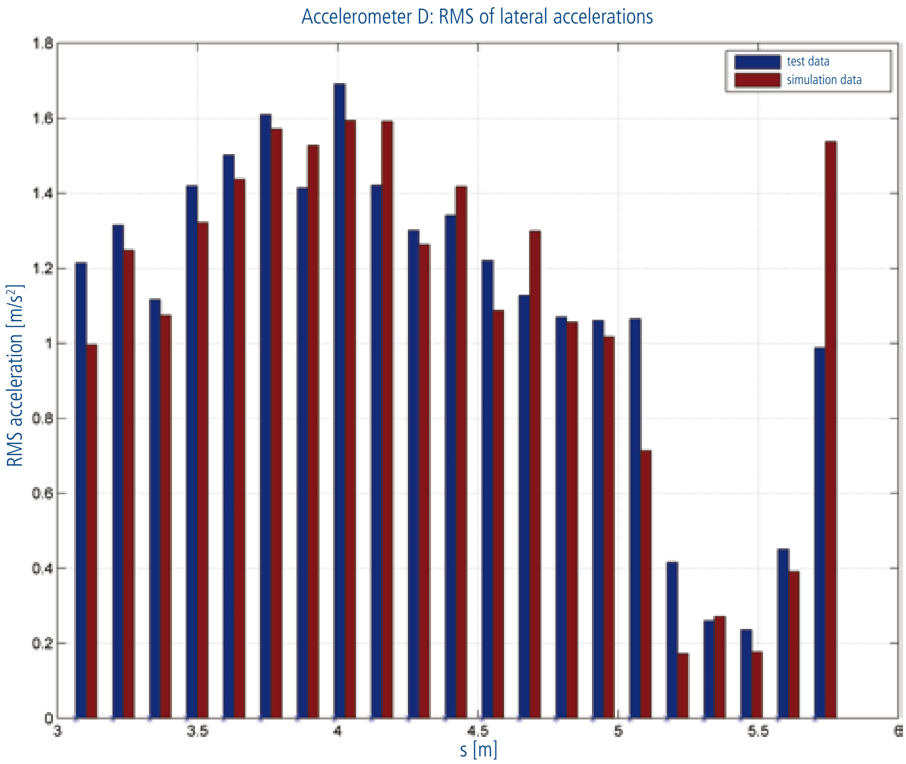

Fig. 8: RMS value comparison of bogie lateral acceleration

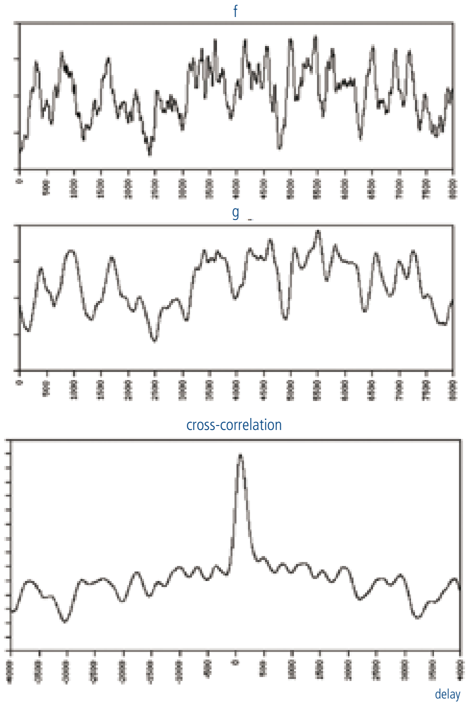

Fig. 9: Cross-correlation function for two signals.

bogies computed by a running simulation to the corresponding ones recorded during the tests. More in depth, quasi-static signals, frequency content and root mean square values above 5-seconds period (RMS) of bogie accelerations were analyzed. The quasi-static signal of bogie lateral acceleration depends on track geometry features, i.e., curve radius and rail superelevation, and vehicle run speed; whereas, signal frequency content analysis underlines track irregularity effect. Finally, root mean square values synthesize both aspects and provide global information about track features.

During this activity, the need for an objective criterion for setting out signals similitude has risen; hence, a signal comparison index (SCI) was defined, which was cross correlated and concept-based. The cross-correlation function $C(t)$ can be computed for a pair of functions $f(t)$ and $g(t)$ in the following way:

$$C(t) = \int_{-\infty}^{+\infty} f^*(\tau) \times g(t+\tau)\, d\tau \tag{4}$$

and usually has a peak for $t = 0$ whether functions are similar or correlated (see Fig. 9). The idea was to compare the cross-correlation peak computed simulation acceleration signals C_0 with the auto-correlation peak of test signal A_0; SCI has been defined in a way to result in a range between 0 and 1:

$$SCI = 1 - \left| \frac{C_0 - A_0}{A_0} \right| \tag{5}$$

This comparison has led to positive results and demonstrated that interaction between vehicle and environment was correctly described in MBS. Figs. 6, 7 and 8 show quasi-static signal, frequency content, and RMS values comparison of bogie lateral acceleration, respectively.

Lastly, comfort tests were simulated and car body acceleration signals compared. During running tests, three accelerometers were placed above the car body floor, at the two bogie pivot locations, and the middle car body. Thus, correspondent signals were calculated by multi-body analysis. These were

		SCI	N_{MV} simulation	N_{MV} test	Diff %
Pivot 1	X	0.74	2.47	2.22	+11%
	Y	0.95			
	Z	0.97			
Mid	X	0.74	2.85	2.75	+4%
	Y	0.94			
	Z	0.99			
Pivot 2	X	0.74	2.10	2.30	-6%
	Y	0.94			
	Z	0.85			

Table 1: Comfort index comparison for Metro Rome line C

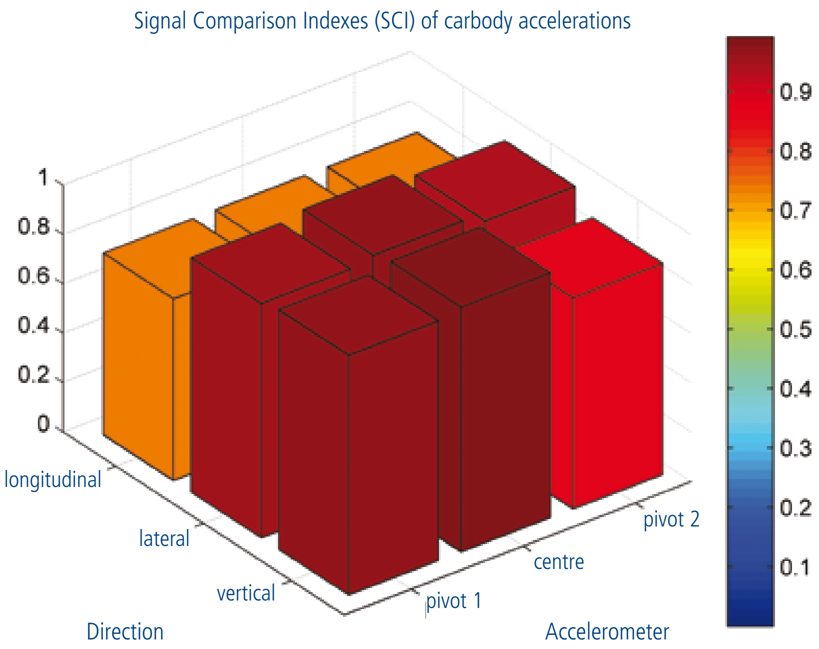

Fig. 10: SCI histogram for carbody accelerations

first filtered as prescribed in UNI ENV 12299 comfort norm [3], then frequency content and 5-second RMS values were computed in order to obtain N_{MV} mean comfort indexes [3]. A sensitivity study about flexible car body description was done, analyzing SCIs and comparing test/simulation mean comfort indexes. In Fig. 10, car body acceleration SCIs for all motion directions were plotted.

The result is that an accurate modeling process, i.e., consistent car body deformation behavior and correct description of system-environment interaction leads to very good results for computed accelerations, as shown in Table 1.

THE NEW IMPROVED SINGLE PROCESS FOR VEHICLE DESIGN

Incorporating multi-body analysis capability and reliability, an improved single process for vehicle design is proposed which integrates both structural and dynamic requirements (see Fig. 11):

1. Firstly, a simplified multi-body simulation (MBS) model of vehicle has to be defined by means of vehicle typology and applicable standards.
2. Then, flexible bodies obtained from reduced FE-models of the components are implemented in the MBS model after the reduction procedure is validated by the analytical method.
3. Such an MBS model is used for performing running simulations in several operative conditions.
4. Results obtained from the multi-body analysis are directly used for verifying running dynamics requirements. Concerning structural requirements, computed load conditions for the bogie-frame are used for stress analyses.

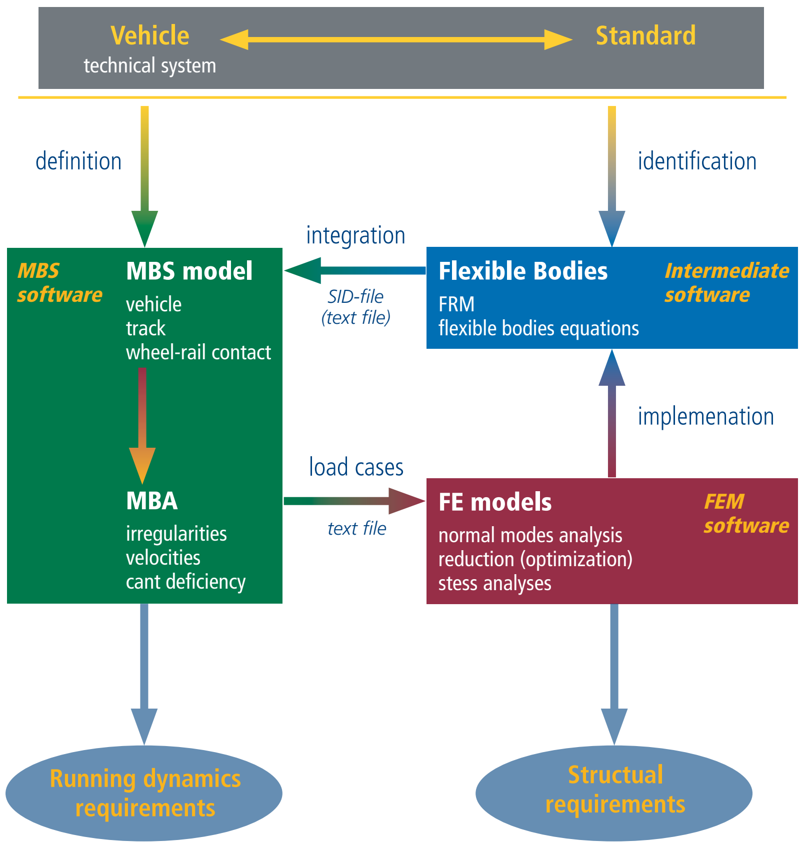

Fig. 11: Integrated vehicle design process

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