CO-SIMULATION OF A DYNAMIC STIFFNESS TEST OF A SEAT CUSHION USING FINITE ELEMENT AND MULTIBODY DYNAMIC MODELS

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Abstract

Dynamic performance of suspension seats depends on characteristics of both the suspension and the seat cushion. The objective of this study is to develop a hybrid method based on the concept of co-simulation between multibody dynamic and finite element models. The methodology is illustrated via simulating a dynamic stiffness test of a seat cushion. The dynamic stiffness of a cushion was measured on an indenter test rig using broadband random input signals (0.5-25 Hz) of different magnitudes (0.25, 0.5 and 1.0 ms^{-2} r.m.s.) and preloads (400, 600 and 800 N). A finite element model of the seat cushion is built up in MARC and a multibody dynamic model of the test rig is established in ADAMS. During the co-simulation, the multibody model calculates and passes kinematics of the test rig to the finite element model of the cushion. Based on these kinematics the finite element model calculates the force and feeds back to the multibody dynamic model. The hybrid model is calibrated through correlation between measured and computed dynamic stiffness. It is expected that the developed methodology can be extended to modelling of suspension seats where multibody model of the suspension co-simulates with finite element model of the seat cushion.

1. Introduction

Proper design of suspension seat cushion not only helps in supporting the occupant posture but also benefits in reducing seat transmissibility and improving ride quality. Cushion helps to absorb the energy of impact by deforming and spreading the load over a wide area. The isolation of a cushion is determined by the extent to which it attenuates the motion over the complete spectrum of frequency present in the vehicle. In an experimental study it was found that when loaded with a mass of the same weight, a suspension seat (with cushion) showed a lower primary peak frequency in the vertical transmissibility of acceleration from the seat base to the seat surface compared with the seat suspension without cushion when exposed to broadband random vibration (Qiu and Zheng, 2010). Another study on commercially available seat cushions of different densities, thicknesses and compositions showed that equivalent damping coefficient of the cushion material decreased with increase in frequency and peak-to-peak amplitude of vibration, and the decrease was very sharp in lower frequency range up to 3 Hz (Mehta and Tewari, 2010). Responses of seat occupants to vibration were found out as a function of excitation source, type of the vibration and mechanical parameters

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(mass, stiffness and damping) of the cushion (Qassem, 1996). Performance of a seat cushion depends on its static and dynamic characteristics.

Use of Computer Aided Engineering (CAE) methods is increasing in design of suspension seats. The scope of modelling of suspension seats depends on potential applications of the model. To predict transmissibility of a suspension seat, mathematical modelling of the seat-occupant system needs to involve modelling of suspension, cushion and human body. In the suspension seat modelling, cushion is traditionally defined through the use of mass, spring and dampers (Fairley, 1990; Rakheja, Afework and Sankar, 1994; Lewis, 1994; Wu and Griffin, 1995; Tewari and Parasad, 1999; Qiu and Griffin, 2011; Shahzad and Qiu, 2013). A lumped parameter model of a seat suspension and cushion system can be useful in predicting seat transmissibility but it is not convenient or straightforward to model the nonlinear behaviour of cushion and dynamic interaction between the seat and occupant.

In recent years, it becomes a trend that the complex suspension mechanism with nonlinear (stiffness/damping) characteristics and high friction is modelled using multibody dynamics (MBD) approach. This requires looking for an alternative method for cushion modelling in the MBD environment if dynamic interaction of occupant with the suspension seat cushion is of primary interest. One such option could be incorporation of finite element (FE) based model of cushion with MBD models. Finite element methods have been used in modelling cushions (e.g., car seats) for computing contact pressures, contact shear stresses and in-body stresses (Siefert *et. al.*, 2008; Zhang *et. al.*, 2015; Gunter *et*. *al*., 2013; Liu *et. al.*, 2015). However, use of a detailed FE cushion model together with a nonlinear suspension model to predict the suspension seat transmissibility has not been reported. How to effectively integrate the two sub-models that are developed in two different platforms or environments and exchange efficiently and accurately the data between the two sub-models so as to better reflect the dynamics of the suspension seat remains to be a challenging task.

Development of a combined MBD and FE model requires multidiscipline simulation such that different programmes can effectively communicate with each other during system simulation and produce a coupled and meaningful solution. Co-simulation is a general approach for joint simulation of models developed with different tools where each tool treats one part of the modular coupled problem. Intermediate results are exchanged between these tools during simulation. ADAMS (ADAMS version 2014.1, multibody dynamic software, 2014) can be used to model a nonlinear seat suspension and MARC (MARC 2013.1, nonlinear finite element software, 2013) can be used to model a nonlinear seat cushion. With a coupled model it would be possible to increase model fidelity in predicting suspension seat transmissibility as well as study the seat-occupant interaction using one integrated model thus accelerating the design process.

This paper develops and presents a methodology of connecting the multidiscipline models. The implementation of the proposed method is demonstrated by developing a model of simulating an indenter rig test. It is envisaged that the finite element model of the cushion can be combined with a multibody dynamic model of the indenter test rig through co-simulation and the developed methodology can be readily extended to model a dynamic system involving seat suspension and cushion.

2. Measurement of cushion dynamic stiffness

Dynamic behaviour of a seat cushion can be determined by measuring its dynamic stiffness. Dynamic stiffness is defined as complex ratio of forces transmitted through the cushion to the input displacement in frequency domain. To measure the dynamic stiffness of a suspension seat cushion and its dependency on load, amplitude and frequency of excitation, tests were conducted on the indenter rig at the Human Factors Research Unit, Institute of Sound and Vibration Research, University of Southampton.

2.1 Method

2.1.1 Apparatus

The indenter rig is equipped with a Ling V860 electro-dynamic vibrator. A seat cushion was mounted vertically up on the test rig as shown in Figure 1. Motion at the vibrator platform was measured using an Entran EGCS_D0_10V accelerometer. The accelerometer had an operating range of $\underline{+}$10g and a sensitivity of approximately 10mV/g. Force at the indenter head was measured by Kistler 9321A force transducer which had sensitivity around 3,69 pC/N. All transducers were calibrated before the test. The signals were acquired using *HVLab* data acquisition and analysis system via 50 Hz anti-aliasing filter with a sampling rate of 256 samples per second. Signal processing was conducted with a frequency resolution of 0.25 Hz.

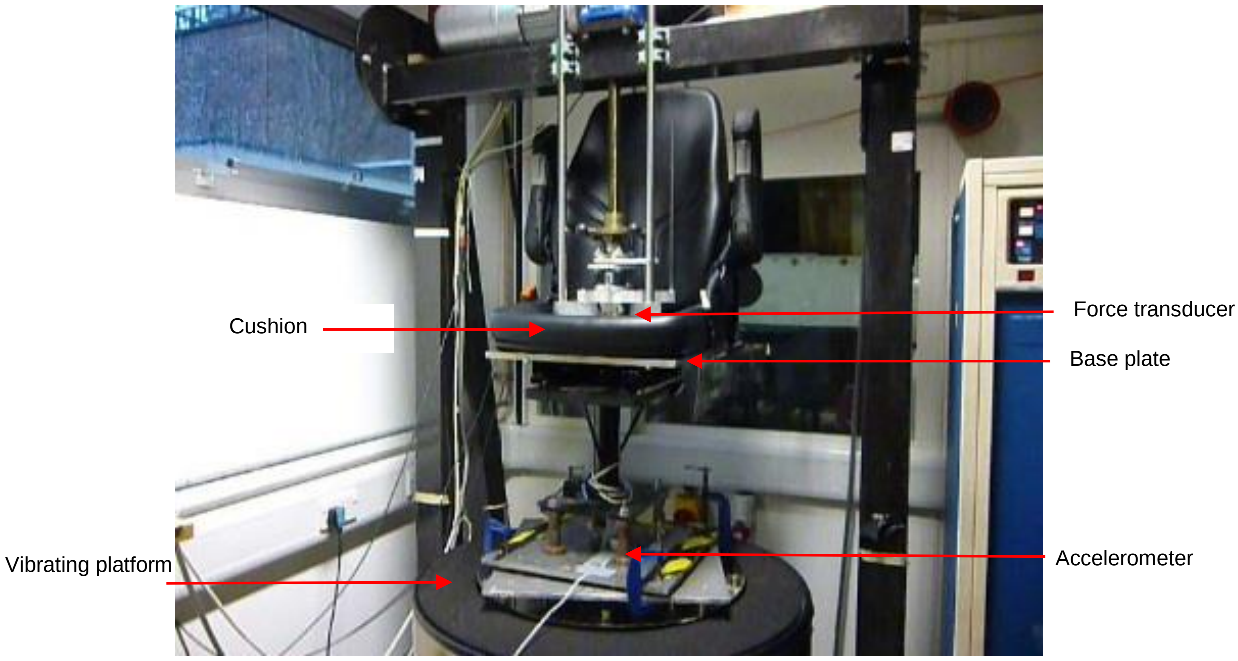


Figure 1 Seat cushion mounted on the indenter rig

2.1.1 Stimuli

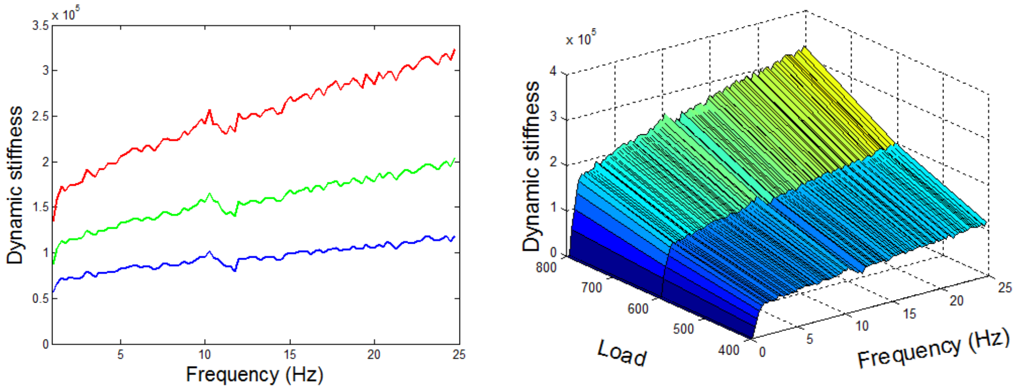
Broadband random input signals of frequency range (0.5 – 25 Hz) and magnitudes 0.25, 0.5 and 1.0 ms^{-2} r.m.s. were used to vibrate the platform. In total, eight test runs with combinations of frequency, amplitude and three preloads (400, 600 and 800 N) were conducted (Table 1).

Table 1 Broadband random stimuli at the vibration platform used in the cushion test

Preload	400 N	600 N	800 N
Vibration magnitude (0.5 – 25 Hz)	0.25 ms^{-2} rms 0.5 ms^{-2} rms 1.0 ms^{-2} rms	0.25 ms^{-2} rms 0.5 ms^{-2} rms 1.0 ms^{-2} rms	0.25 ms^{-2} rms 0.5 ms^{-2} rms

2.2 Measurement results

Cushion was rigidly connected to the base plate and indenter head was lowered down on cushion to get the required preload. Then the vibration platform was excited using the defined input signal. Duration for each test run was 60 sec. The vertical acceleration at the platform and the contact force at the indenter-cushion interface were measured and the dynamic stiffness was calculated after signal normalization. Figure 2 shows the behaviour of measured cushion dynamic stiffness.



(a) (b)

Figure 2 (a) Cushion dynamic stiffness with different preloads at vibration magnitude 0.25 ms^{-2} r.m.s.: — 400 N; — 600 N; — 800 N. (b) Cushion dynamic stiffness with different preloads at vibration magnitude 0.25 ms^{-2} r.m.s. (3D view)

Experimental results showed frequency dependency of dynamic stiffness of the seat cushion. The behaviour of the cushion was also found as a function of preload and vibration manganite. Increase in vibration magnitude resulted in decrease in dynamic stiffness, whereas increase in preload resulted in increase in dynamic stiffness (Figure 3 (a), (b) and (c)).

3. Development of the co-simulation model

3.1 Basic concept of co-simulation method

The co-simulation between ADAMS and MARC was manipulated through an independent interface - ADAMS Co-simulation Interface System (ACIS) and was based on the concept of glue code (Elliot, 2002) which implemented a simple control algorithm allowing asynchronous communication of variables between the two software. Co-simulation between the codes worked such that kinematics evaluated in ADAMS were imposed on MARC while the forces calculated in MARC were applied to ADAMS. Figure 4 shows the basic working principle.

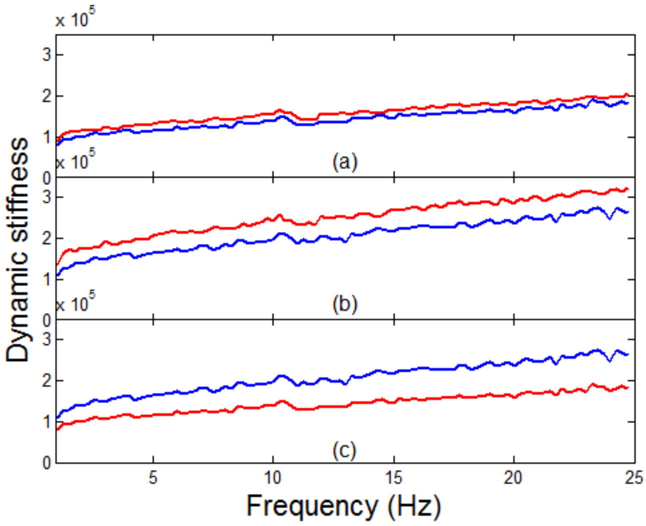


Figure 3 (a) Effect of input vibration magnitude on cushion dynamic stiffness under 600 N preload: — 0.25 ms^{-2}; — 0.5 ms^{-2}. (b) Effect of input vibration magnitude on cushion dynamic stiffness under 800 N preload: — 0.25 ms^{-2}; — 0.5 ms^{-2}. (c) Effect of preload on cushion dynamic stiffness under input vibration magnitude 0.5 ms^{-2}: — 600 N; — 800 N.

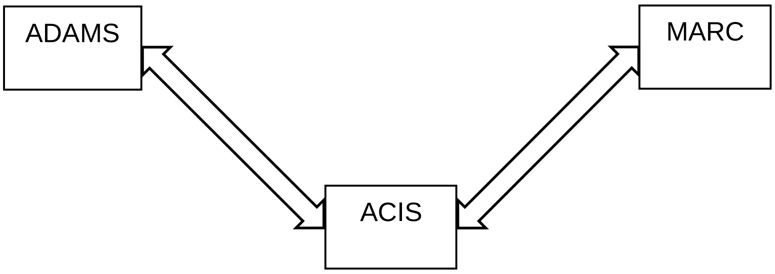


Figure 4 Co-simulation method

3.2 Cushion model

The seat cushion tested in this study was made of polyurethane foam. The behaviour of foam material can in general be described as nonlinear and strain rate dependent with high energy dissipation characteristics and hysteresis in cyclic loading. It is a hyperelastic cellular elastomer that presents a significant viscoelastic behaviour (Haan, 2002). A CAD model of the cushion was meshed in MARC using 4-node linear tetrahedron element (tet4) and the total number of elements was 37152 (Figure 5 (a) and (b)).

Cushion material was represented using a hyperplastic material model. This material model was characterized by means of strain energy density function, W (MARC theory and user information, 2013).

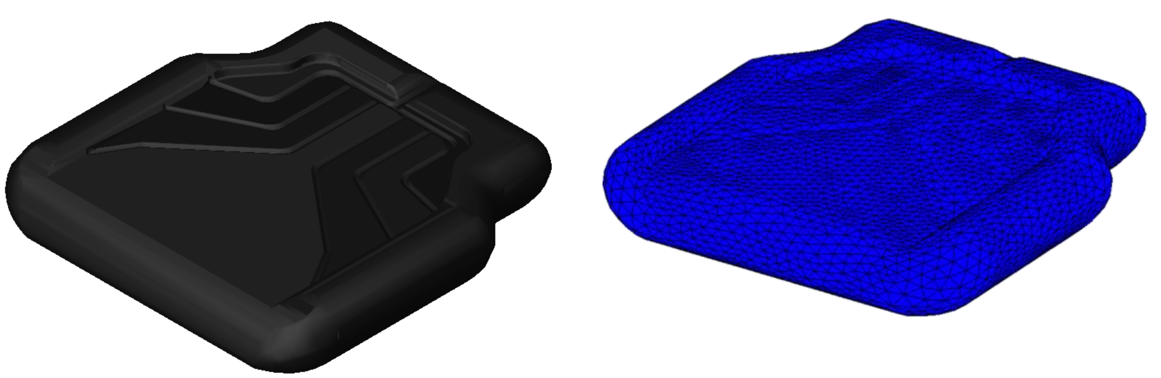
$$W = \sum_{n=1}^{N} \frac{\mu_n}{\alpha_n}\left(\lambda_1^{\alpha_n} + \lambda_2^{\alpha_n} + \lambda_3^{\alpha_n} - 3\right) + \sum_{n=1}^{N} \frac{\mu_n}{\alpha_n}\left(1 - J^{\beta_n}\right) \tag{1}$$

where N is polynomial order, μ_n is coefficient of initial shear modulus, α_n is a material constant, and β_n is coefficient of degree of compressibility, $\lambda_1^{\alpha_n}$, $\lambda_2^{\alpha_n}$ and $\lambda_3^{\alpha_n}$ are stretch ratios and J is the elastic volume ratio. Foam model parameters μ_n , α_n and β_n were obtained by performing a non-linear least square fit with the cushion test data. The coefficient β_n is related to the Poisson's ratio v_n. In order to account for the thin cell-wall structure of the foam which allows wall buckling under pressure without lateral resistance, principal strains are assumed to be fully de-coupled which means Poisson effect was neglected (Grujicic, *et. al.*, 2009).

It was assumed that the cushion behaviour was isotropic and it can be described by a time dependent shear and bulk modulus (MARC theory and user information, 2013). However, time dependency of the bulk modulus is generally quite weak in this type of material, thus viscoelastic portion of material model was restricted to the shear modulus which is defined in equation 2 (Grujicic *et. al.*, 2009)..

$$G(t) = G_0 - \sum_{n=1}^{N} G_n \left(1 - e^{-\frac{t}{\tau_n}} \right) \qquad (2)$$

where G_0 is shear modulus independent from relaxation data, τ_n is relaxation time and G_n is relaxation magnitude.



(a) (b)

Figure 5 (a) A CAD model of the cushion (b) A FE model of the cushion in Marc

3.3 <u>Test rig model</u>

Test rig consisting of indenter head and vibrating platform was modelled in ADAMS. All parts of the test rig were modelled as rigid bodies and were interconnected through kinematic joints (Figure 6). Indenter head was constrained with rig frame through translation joint and could move in the vertical direction. The base plate was rigidly connected with vibrating platform which was connected with the frame through translation joint and could move vertically up and down.

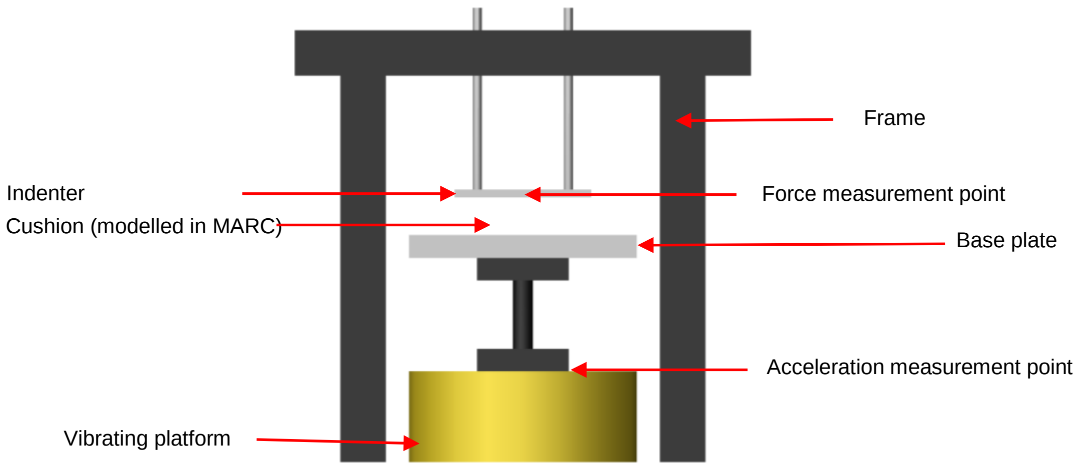


Figure 6 The MBD model of the indenter test rig in ADAMS

4. Correlation of cushion dynamic stiffness

Dynamic stiffness of the suspension seat cushion was calculated by co-simulating the models of the cushion and the test rig described in the previous section using the method outlined in section 3.1. The coupled model of the cushion and the test rig was simulated by running dynamic motion analysis in ADAMS and nonlinear finite element quasi static analysis in MARC through ACIS. Dynamic stiffness was then calculated using cushion force, calculated by MARC, and platform displacement, calculated by ADAMS. Figure 7 (a), (b) and (c) shows the correlation of the measured and predicted cushion dynamic stiffness under different preloads and with varying input vibration magnitudes.

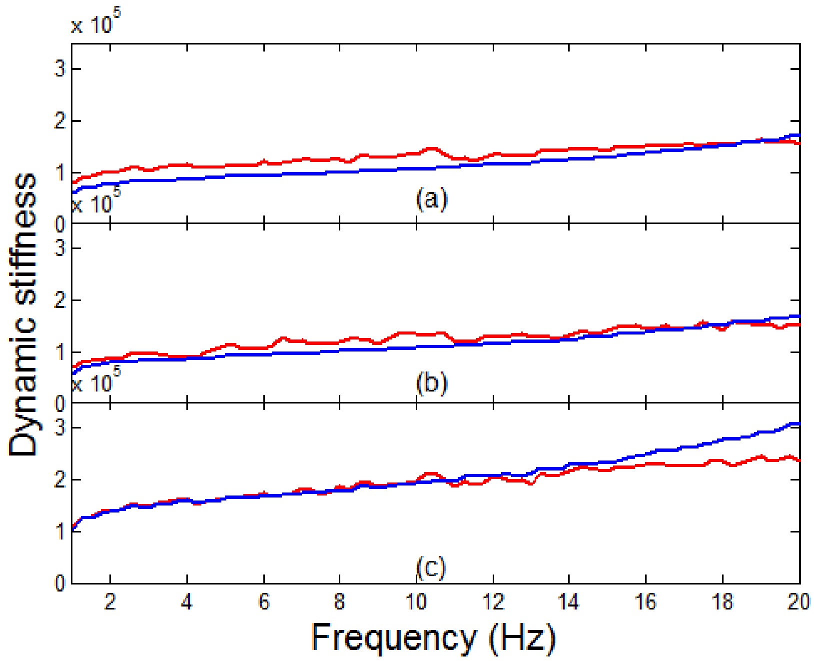


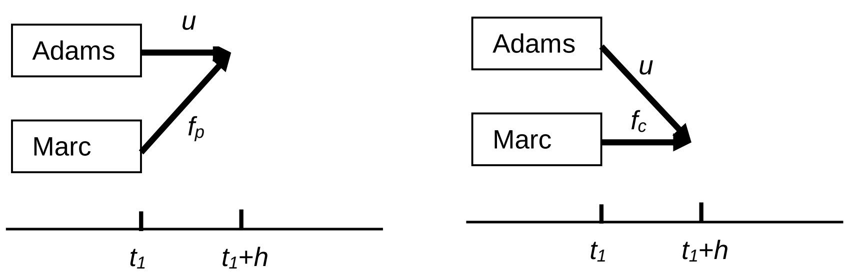
Figure 7 (a) Comparison of measured and predicted dynamic stiffness with preload 400 N and vibration magnitude 0.5 ms^{-2} r.m.s. (b) Comparison of measured and predicted dynamic stiffness with preload 400 N and vibration magnitude 1.0 ms^{-2} r.m.s. (c) Comparison of measured and predicted dynamic stiffness with preload 800 N and vibration magnitude 0.5 ms^{-2} r.m.s.. — Measured; — Predicted.

5. Discussion

A multibody dynamic model generally consists of rigid bodies interconnected through kinematic and compliant connections and if required a model can be built fully or partially using flexible bodies. The flexible parts in the MBD environment are generally based on the modal flexibility approach using orthogonalized Craig Bampton modes (Ottarsson, 2000) and their behaviour is considered as linear having small deformation. However, experimental study has shown that seat cushion exhibits nonlinear behaviour which makes the above mentioned modelling method of flexibility inadequate for this kind of applications.

To address this limitation, this paper has investigated the possibility of integrating a MBD and an FE model to produce a coupled solution of a nonlinear system with a view to applying the similar techniques to modelling of a suspension seat-cushion-occupant system in the next step. For this purpose, simulation of a dynamic stiffness test of a seat cushion was taken as an exemplary case. Interaction between the cushion and test rig was modelled through two surfaces, one at the top and the other at the bottom of the cushion. These surfaces were glued to FE model of the cushion and were attached to MBD test rig model through force element in ADAMS. During the co-simulation ADAMS calculated cushion deformation due to preload and input vibration, whereas MARC computed resultant force due to deformation.

The concept of co-simulation used in this study was based on the data interpolation and extrapolation. Assume that both ADAMS and MARC solvers are at time t_1 and would like to proceed with the next time step t_1+h, where h is the step size (Figure 8 (a) and (b)).



(a) (b)

Figure 8 (a) Co-simulation - ADAMS advancing for next step (b) Co-simulation - MARC advancing for next step

ADAMS takes the next time step and computes the displacement using the predicted force (f_p) value. This predicted force value is obtained by extrapolation of the computed force values of MARC up to time t_1. Next MARC takes its simulation step using interpolated displacement of u up to time t_1+h from ADAMS and computes the force f_c at the interaction point. The difference between predicted force f_p and computed force f_c is a measure of the error in co-simulation (ADAMS co-simulation interface, 2014). This interpolation-extrapolation strategy could improve accuracy of the solution over the conventional co-simulation technique where data exchanged between the software remains constant between communication intervals. In ACIS, exchanged data can be interpolated either linearly or quadratically. Moreover, ACIS also allows variable asynchronous communication meaning two solvers can be run at different step size. The provisions of asynchronous communication and interpolation options may help in optimizing the simulation run time as mostly FE analyses are computationally expensive as well as help in increasing the fidelity of coupled solutions.

To use the Foam material model in FE calculation, it was required to define the order *N* and unknown parameters μ_n, α_n , $\lambda_1^{\alpha_n}$, $\lambda_2^{\alpha_n}$, $\lambda_3^{\alpha_n}$, G_n and τ_n. Values of these unknown parameters are generally identified from the experimental results obtained from uniaxial compression test and shear test as well as from normalized shear modulus vs. time relaxation data. However, as the main objective of this study was to develop a co-simulation between MBD and FE models, so the values of unknown parameters were initially taken from literature (Grujicic *et. al.*, 2009) and an order of *N*=2 was used to build the FE cushion model. These parameters were further adjusted in the process of matching the predicted dynamic stiffness with the measured one. Comparison between the measured and predicted dynamic stiffness of the cushion (Figure 7) showed promising results indicating that MBD and FE models developed in different platforms can be combined and simulated together. However, there is a need to further improve the cushion model and investigate the effect of different co-simulation options such as selection of step size and interpolation/extrapolation of exchanged data as discussed earlier.

In design of suspension seats, it is a common practise to develop a FE model of seat cushion to study behaviour of the cushion and its interaction with seat occupant. This study has provided a good starting point to further expand this approach and use the FE cushion model to develop complete suspension seat model and study its response to vibration. In a similar way, a MBD model of a seat suspension that defines its nonlinear behaviour can be developed through detailed modelling of spring, damper, and consideration of mechanism friction and structural flexibility. Using the approach developed in this study, this nonlinear suspension model can be combined with a FE model of the cushion to form a coupled model for predicting seat transmissibility.

6. Conclusion

Suspension seat cushion exhibits a nonlinear behaviour which is a function of load, input vibration magnitude and excitation frequency. This study has demonstrated a proof of concept of developing a co-simulation method between MBD and FE models. The developed method can be extended to modelling and analysis of a coupled nonlinear suspension and cushion system for predicting transmissibility of suspension seat with occupant exposed to vibration of varying magnitudes and frequencies.

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