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Assessment of the impact of sub-components on the dynamic response of a coupled human body/automotive seat system

Romain Barbeau^{a,b}, Thomas Weisser^{a,*}, Raphaël Dupuis^a, Évelyne Aubry^a, Samuel Baudu^b

^aIRIMAS - EA7499, Université de Haute-Alsace, 12 rue des Frères Lumière, 68093 Mulhouse Cedex, France

^bFAURECIA Automotive Seating, ZI Brières les Scellés BP89, 91152 Étampes Cedex, France

Abstract

Automotive seats are complex structures composed of several sub-components and designed to provide the best driving comfort to the occupant. From a mechanical point of view, this amounts to ensuring both optimal body support and vibration isolation. However, considering the multiple interactions between these sub-components and the seated human body, it still remains difficult to identify design trends to *a priori* improve the seat dynamic performances. To handle these issues, a test campaign is first conducted, according to current industrial protocols, to characterize the impact of each seat sub-component on the cushion vertical transmissibility. Then, a multi-body human/seat system is implemented and validated, to assess the influence of all the parameters related to the seat but also to the human body, using global sensitivity analysis methods. It is shown that the foam and cover composing the cushion essentially control the dynamic performances of the seat. However, the variability associated to the internal dynamics of the human body must be taken into account, as it strongly influences the maximum amplitude of the transmissibility of the human/seat system and therefore the vibrational comfort.

Keywords: automotive seat, vibrational comfort, transmissibility measurement, multibody model, sensitivity analysis

1. Introduction

Automotive seats are designed to provide the best driving comfort to the occupant while ensuring high safety levels. Although this complex notion of comfort has been consistently tackled since the late 1980s, it still remains difficult to quantify and to evaluate as it highly relies on subjective assessments of multi-sensory perceptions. Hence, it is usually inversely handled by considering discomfort sources and distinguishing different types of solicitations (e.g. physical, visual, olfactory), as summarized by Bubb *et al.* [1] in the so-called “pyramid of discomfort”. In accordance with this classification, in the case of automotive seats, the baseline anthropometry-related stage proposed by Bubb can be assimilated to postural comfort; the following climate-related one to thermal comfort; and the vibration-related one to vibrational comfort. From a mechanical point of view, postural comfort aims to provide an optimal body

*thomas.weisser@uha.fr

support to lower numbness, fatigue or pain issues [2], while vibrational comfort tends to strengthen these features and to provide an optimal dynamic environment, considering time varying inputs under riding conditions [3].

All the deriving requirements are considered by seat manufacturers through their design process. In order to provide a “globally suitable” seat, the seat occupant is classically represented by a mean or averaged human subject, whose bio-mechanical characteristics and properties are statistically corresponding to a 50th percentile male subject ones [4]. One fundamental rule is to design a seat around the hip-point (H-point) of this seated occupant, representing the theoretical position of the pivot point between the torso and the upper leg part of the body [5]. A proper driving position is ensured by setting the location of this H-point and verifying critical posture angles, and it is typically assessed using pressure distribution measurements [6]. Then, vibrational comfort is usually adjusted by taking into account the vibrations at the seat base. These vibrations result from the tire/road interaction and the thermal engine [7] and are mainly concentrated on a frequency range of 0.5-20 Hz [8], with higher levels between 4 and 8 Hz where the human body is particularly sensitive to vibrations [9]. Once a prototype seat has been built, the most common way to characterize its dynamic performances is to measure its transmissibility, which is defined as the transfer function between the vibrations perceived on the seat surface with regard to the vibrations coming from the base of the seat [2]. Such transmissibility allows quantifying the ability of a system to filter vibrations and its amplitude can therefore be used as a global indicator to evaluate the seat dynamic performances.

However, regarding the technological complexity of current automotive seats, it remains difficult for seat manufacturers to identify *a priori* potential design trends to tailor vibrational comfort. Indeed, while lots of studies have been performed to statically or dynamically characterize some of these sub-components, the derived results are hardly related to these dynamic performances. Moreover, in order to provide value-added products, the automotive industry tends to put particular focus on “personalized comfort”, requiring to take into account the occupant variability through the design process. It is thus essential to assess the impact of some key sub-components on the dynamic response of a seat, when coupled to and/or interacting with a human body exhibiting variable bio-mechanical properties. Regarding the large number of potentially influencing factors and the difficulty to define and to control their levels, an all-experimental approach based on design of experiments remains unattractive. Conversely, a computationally intensive approach based on finite-element analysis would assume that detailed models (*i.e.* geometry, properties, interface management,...) are available for each component of the system, including the human body. Therefore, hybrid approaches using equivalent models identified from experimental measurements seem more adapted to pre-design stages.

The aim of this study is to develop an equivalent model of a coupled human/seat system, identified from experimental characterizations of the main seat sub-components and allowing to determine the sensitivity of its vibrational comfort feature (*i.e.* vertical transmissibility) with regard to its parameter variability. The main contributions of this work are (1) Experimental investigation of the impact of the main seat sub-components on its vertical transmissibility, (2) Implementation of a “lean” multi-body model from both experimental measurements and literature data, and (3) Assessment of the sub-component impact of the vertical transmissibility using global sensitivity analysis methods.

The remainder of the paper is organized as follows. Section 2 summarizes both experimental and modeling works performed since the 2000's to study the mechanical behavior of automotive seats and their sub-components. Section 3 presents an industrial-standard test protocol to investigate the impact of each sub-component of the seat on its dynamic transmissibility. In section 4, a two-dimensional multi-body model of a coupled human body/seat system is developed based on data extracted from both experimental measurements and literature, and a discussion on the accuracy of the initially estimated dynamic transmissibility is proposed. Thereafter, global sensitivity analysis methods are set up in section 5 to precisely characterize the variability of this response with regard to the input parameters of the model, allowing to identify the most impacting ones and their respective effects. Finally, concluding remarks are given in section 6.

2. Review on the dynamic behavior of automotive seats

2.1. Characterization of seat sub-components

Automotive seats have become very complex systems in recent years, as they gather lots of technical solutions aiming at improving the interface between the occupant and its environment, especially during riding. Focusing on its mechanical behavior, the seat can be simplified from more than eighty components to four main sub-components *i.e.* frame, suspension mat, foam pad and cover, as illustrated in figure 1. Such decomposition is also frequently used in the literature [10, 11], as detailed from the bottom up in the following paragraphs.

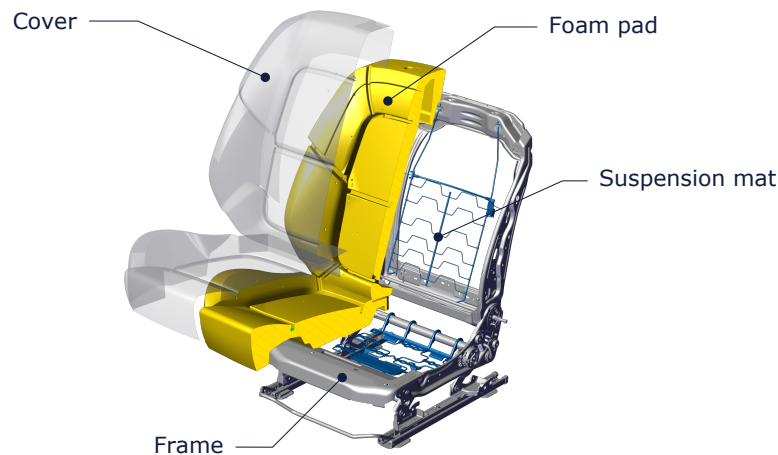


Figure 1: Main sub-components of an automotive seat

The backbone of the seat is composed of a metallic frame structure holding the occupant in a comfortable sitting position, absorbing energy in crash cases and carrying all the components attached to the seat. Some researchers studied the frame behavior under dynamic conditions in order to evaluate its impact on the vibration transfer to the human body. Lo *et al.* [12] studied the modal behaviors of the seat frame alone, then of the complete seat and finally of the complete seat coupled to the occupant. The objective was to predict the vibrational normal modes of the

coupled human/seat system based on the behavior of the raw seat frame. A test protocol was developed to analyze three different seats with six human subjects exhibiting mean human body properties. The authors showed that no structural mode had been measured under 20 Hz for the three different seat configurations. Such results were also derived by Zhang [11], confirming that the frame exhibits a rigid behavior in the vertical direction up to 20 Hz. It may therefore be considered as a negligible sub-component with regard to the dynamic responses of a seat along the vertical direction.

In most automotive seats, suspension mats are designed to adjust the postural comfort, to ensure that the driver feels well supported, has free access to the steering wheel and the pedals, and has good road vision. Ould Eyih [13] studied the seat behavior under high deformations using a hybrid test/model approach, focusing on the mechanical behaviors of the foam pad and the suspension mat, respectively. An equivalent beam model was notably proposed to determine the suspension mat deflection when submitted to the load corresponding to a seated mean human body. The derived results demonstrated the role of this sub-component on the static adjustment of the occupant. More recently, Choi *et al.* [14] numerically investigated the modal behavior of a backrest suspension mat. A first natural frequency was computed at 34 Hz, corresponding to a bending mode in the fore-aft direction. Reminding that the frequency range considered for vibrational comfort purpose ranges from 0.5 to 20 Hz, these suspension mats do not seem to have any impact on the targeted dynamic behavior. Some experimental validations would however still be required.

Among the four main seat sub-components, a strong focus has been put on foam pads since the early 1960's and more specifically on the mechanical behavior of polyurethane (PU) foams. Lots of investigations have been performed to relate their chemical formulation and manufacturing process to their physical properties and thus to their ability to filter vibrations, in order to meet *a priori* targeted dynamic performances. Ebe and Griffin [15] provided reference works by systematically investigating the impact of different foam properties on the seat cushion transmissibility. They proposed an experimental protocol with different foam blocks placed on the surface of a rigid seat. Firstly, the impact of the foam thickness was investigated, highlighting that the gain at the resonance of the vertical transmissibility increases and the associated frequency decreases as the foam thickness increases. However, as mentioned by the authors, this conclusion would need to be confirmed regarding the wide stiffness variation exhibited by the tested foam blocks. Secondly, the impact of the foam chemical composition was investigated considering four different formulations. These were compared to foam blocks with the same density, and further with foam blocks having identical hardness¹. Some small differences in both resonance gain and frequency were observed and the authors emphasized that formulations referred to as "high-durability" foams (commonly used by seat manufacturers) exhibit a higher resonance gain but the best filtration properties at higher frequencies (*i.e.* after the transmissibility cut-off frequency). These researches were carried on in 2004 by Ebe and Yamasaki [16] who experimentally studied the effect of the density of two different backrest foam pads on the backrest dynamic transmissibility, in the vertical direction. It was clearly observed that a high density foam pad has better filtration performances than a low density one. Although

¹The foam hardness is defined as the stiffness associated to the load required to indent a foam sample at a 25 % compression rate.

these studies were conducted on both cushion and backrest, it can be concluded that changing the foam thickness has more effect than changing its chemical composition (while still ensuring the same density or hardness). Moreover, the density seems to have a significant effect on the filtration performances of the foam pad but it remains hard to further rank these parameters (*i.e.* thickness, composition and density) by importance as the transmissibilities were not measured on the same components (*i.e.* backrest or cushion) for each considered case.

Concerning the impact of the cover on the seat dynamic behavior, some researches were also conducted by Corbridge *et al.* [17], followed by Ebe and Griffin [15]. The former study, concerning train seats, showed that removing the cover seems to have no significant influence on the cushion transmissibility. The latter, concerning automotive seats, showed small differences on the cushion transmissibility due to the air permeability of the cover. Ebe and Griffin [15] argued that the permeability of the different cover types could affect the pneumatic damping of the contained polyurethane foam. However, such conclusions must be critically considered as the different actual trimming processes commonly used by seat manufacturers (e.g. hog ring, clip, hook and loop, . . .) were not considered in these studies.

Although these various studies do provide a meaningful insight on the mechanical behavior of each sub-component of the seat, the fact that they focused on different types of features (e.g. normal modes, dynamic stiffness or transmissibility) makes it difficult to compare and/or combine these results to predict or to characterize their potential influence on vibrational comfort. Moreover, most of these studies were performed on dedicated test benches, under standardized and controlled solicitations, which may not be representative of *in-situ* driving conditions, nor reflect any coupling with the seated occupant. To overcome these issues, modeling approaches are commonly implemented, as exposed in the next sub-section.

2.2. Modeling of a human/seat coupled system

Lots of mathematical models have been developed in the literature to predict the dynamic responses of a coupled human/seat system. These models can be broadly divided into three groups: lumped parameter models, multi-body models and finite element models. Several critical reviews have already been published to detail and to compare each model within or between each group. Hence the reader may refer to the significant contributions by Liang and Chiang [18] and more recently by Mohajer *et al.* [19], concerning lumped parameters models and multi-body ones, respectively, for exhaustive syntheses. Focusing on the approach proposed in the present paper, the authors previously highlighted strong prediction limitations of lumped parameter models [20], inducing the need to develop more complex models with additional directions, to take into account the backrest contribution and internal dynamic interactions of the human/seat coupled system. The following paragraphs are therefore dedicated to multi-body models.

Multi-body models have been significantly considered since the 2000s in the field of bio-mechanics to provide a better description of the human body motions and more accurate dynamic responses of coupled human/seat systems. Such models are classically composed of several rigid bodies, representing different parts of the human anatomy, connected by rotational and/or longitudinal springs and/or dampers. Classically, each body has three Degrees Of

Freedom (DOF) in the sagittal plane: vertical, fore-aft and rotational directions. The first multi-body model describing the human/seat coupled system was introduced by Cho and Yoon [21] to evaluate the ride quality inside a vehicle. Four different bio-mechanical models were studied based on nine DOFs in order to evaluate the transmissibilities to the head, to the back and to the hip, resulting from vertical vibrations. To validate their model, the authors measured acceleration responses on ten subjects under vertical vibration excitations, obtaining a good matching of the transmissibilities at the resonance frequency.

A fourteen DOFs multi-body dynamic model was further proposed by Liang and Chiang [18] based on the models developed by Cho *et al.* [21] and Kim *et al.* [22], to evaluate the impact of the backrest inclination on both apparent mass and seat-to-head transmissibility. This model is composed of five rigid bodies representing thighs, pelvis, upper-body, head and viscera in the sagittal plane. Three different cases were considered (*i.e.* without backrest, with backrest at 12 ° and 21 °), and were validated by comparison to various experimental data. The results showed that fewer vibrations were transmitted with a 21 ° backrest angle. This model was later extended by Kumbhar [23] who added six DOFs to take into account the dynamic behavior of the seat and its sub-components. Three seat configurations were studied (rigid, traditional and suspension seats) to compute transmissibilities and absorbed powers in order to rank the filtration performance of each one. However, it can be noted that this model was not validated with regard to any experimental measurement.

Furthermore, a study was conducted by Kim *et al.* [24] to develop a tool for car seat designers. The occupant and the seat were modeled by three bodies (torso, buttocks and legs) and two bodies (backrest and cushion), respectively. Based on natural frequencies and mode shapes the authors performed a parametric study, highlighting the influence of particular mechanical parameters on the dynamic behavior. A good correlation between simulated frequency responses and experimental ones was also achieved, in terms of locations of resonances and deflection shapes.

Finally, while all these studies essentially focus on the human body aspects, little works investigated the coupling with the seat model and its associated parameters. However, as classically observed [10, 25, 26, 19], when submitted to the occupants weight, the seat foam pad shows a very complex mechanical behavior, essentially deriving from the nonlinear elastic, viscoelastic and multiple dissipation properties of PU foams. As an example, Mohajer *et al.* recently coupled a multi-body human model to a nonlinear seat foam model to estimate directional and sectional ride comfort features. This complete model allowed them to compare the impact of human weight, road class and vehicle speed on standard ride comfort indices.

2.3. Discussion

In view of this synthetic review, it is observed that the purpose of the developed models is often different. Some authors attempted to evaluate the ride quality computing transmissibilities [21]; other focused on the mode shapes of the human body [24]; or investigated the impact of seat properties (e.g. foam properties, backrest inclination, additional suspension) on some vibration responses (Liang and Chiang, 2008; Kumbhar, 2013). Moreover, while some authors considered a linear behavior of the human/seat coupled system [21, 18, 23], others developed nonlinear

model of foam [24]. Finally, it is noticed that almost all the studied vibration responses are deterministic (*i.e.* only based on initial “best-estimate” values of the input parameter). It can be noted that Maciejewski and Krzyzynski recently proposed to apply a global sensitivity analysis approach to determine the isolation properties of a simplified seat suspension system under specific operating conditions [27]. Furthermore human body variability and its impact on these responses has already been observed experimentally [28]. Therefore, the aim of this study is to propose a modeling approach taking into account this human body variability in order to describe the human/seat system behavior under dynamic conditions. The implemented model is based on Kumbhar’s multi-body model [23] and supplemented with experimentally identified values of the equivalent parameters associated to the seat main sub-components. The following sensitivity analyses would finally allow to clearly identify the impacting factors and their influence on the vertical transmissibility of the system.

3. Experimental characterization of seat sub-components

3.1. Test bench setup

To objectively evaluate the dynamic behavior of a seat and to avoid any repeatability issue, a standard test protocol using a lead buttock [29] is commonly performed in the automotive industry. The test bench, illustrated in figure 2, is mainly composed of a 54.8 kg mass² with a buttock shape (*i.e.* lead buttock), placed on a seat cushion of a premium car. The mass was oriented at an 18 ° angle to the horizontal. A 3 ° tilt was also considered to compensate for the thighs flexibility with regard to a human subject (the femur angle is equal to 18 ° for a typical automotive posture). The H-point of the whole buttock component was aligned with the H-point of the seat to guarantee a realistic pressure distribution on the cushion. The seat was fixed on a jig considered as rigid on the studied frequency range. The track angle was set to 5 ° to the horizontal, to reproduce vehicle conditions. The seat and the jig were rigidly mounted on the shaker and excited by a 1 m.s⁻² RMS white noise signal on 2 - 20 Hz for eight minutes. The vertical transmissibility between the seat base and the cushion surface was measured on the last six minutes, using two accelerometers.

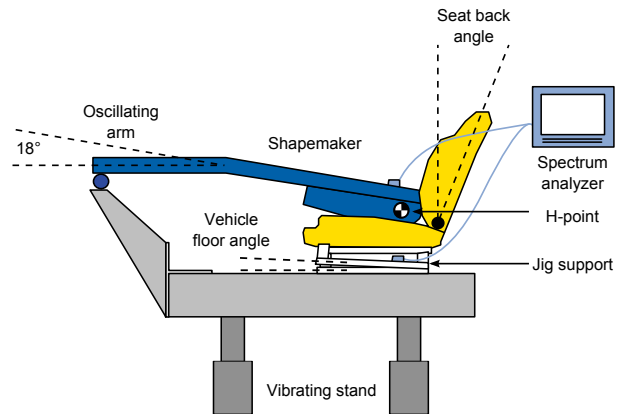
Classically, the relevant properties that can be derived from seat transmissibilities are: the resonance frequency f_r and gain g_r , depicting the vibration amplification; the cut-off frequency f_c , corresponding to the beginning of the filtration zone (*i.e.* where the transmissibility is lower than 1); and the area A under the transmissibility curve, reflecting the total energy in the system. Additional post-processed features could also be determined from the associated raw data, e.g. wavelet transfer functions [31], to further investigate the dynamic behavior of such complex coupled systems. However these are still rarely used in the automotive industry. Figure 3 illustrates the transmissibility in the vertical direction of the seat cushion with the lead buttock, highlighting a first resonance at 6.9 Hz, a gain at resonance equal to 4.8, and a filtration zone starting at 10.0 Hz.

Based on this test protocol, the proposed approach in this study is to separate each sub-component (*i.e.* cover, suspension mat and foam pad) to quantify its impact on the cushion vertical transmissibility. As discussed in the

²Mass representing the weight applied to the cushion by a seated mean human, according to the ISO5982 standard [30]



(a)



(b)

Figure 2: Lead buttock measurement facilities: (a) test-bench; (b) schematic

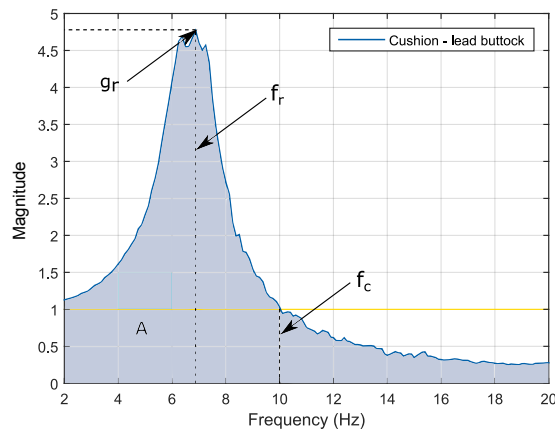


Figure 3: Typical measurement of the vertical cushion transmissibility and derived properties

introduction, the frame is not considered in the test campaign as its rigid behavior has already been highlighted on the studied frequency range along this direction .

3.2. Sub-component impact on the transmissibility

3.2.1. Cover

In a first set of experiments, the impact of the cover was investigated considering four configurations of the seat, as illustrated in figure 4:

- figure 4 (a): full trimming, *i.e.* the complete seat cushion;
- figure 4 (b): without trimming, *i.e.* the seat cushion is composed of the frame, the suspension mat and the foam pad;

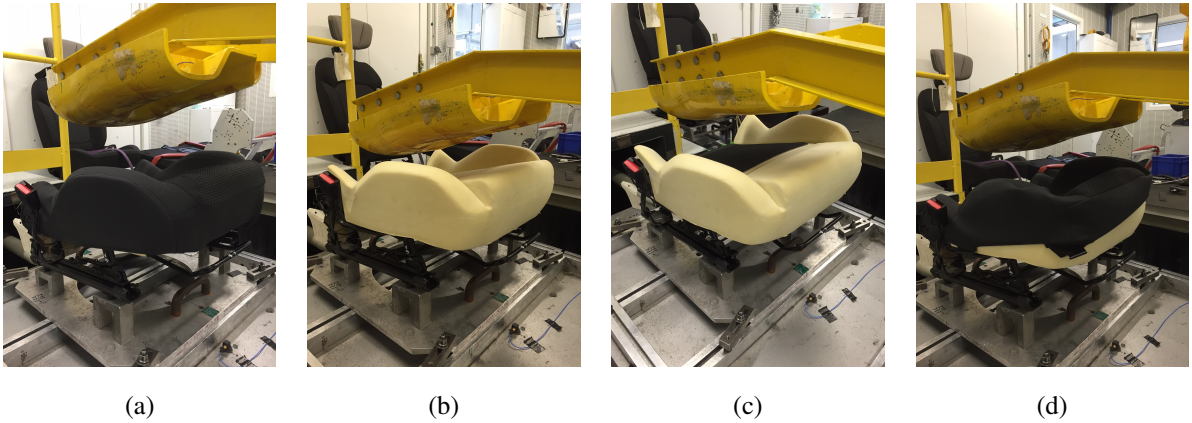


Figure 4: Seat configurations to investigate the impact of the trim cover

- figure 4 (c): without trimming, with cover material, *i.e.* the seat cushion is composed of the frame, the suspension mat, the foam pad and a fabric cover sample placed on the insert of the cushion;
- figure 4 (d): without frame trimming, with cover trimming, *i.e.* the seat cushion is complete but the trimming between the frame and the cover is removed; Only the trimming between the cover and the foam pad is kept.

The obtained transmissibilities are depicted in figure 5 (a). The reference configuration (*i.e.* full trimming) shows a resonance at 7.2 Hz with a 5.6 gain. When the trimming is removed, the transmissibility resonance frequency decreases (6.6 Hz), reflecting a 9 % softening, and its gain increases (6.7). By comparison between the curves with and without cover material, it can be observed that the cover material has no significant impact on the response. Moreover, the measured transmissibilities for the reference configuration and the configuration without the trimming between the frame and the cover show similar behaviors. This suggests that the trimming tension between the frame and the cover has no impact on the seat transmissibility and that only the trimming tension between the foam pad and the cover impacts the seat transmissibility.

A second set of tests was performed to investigate the impact of the material constituting the cover. The seat with fabric cover was compared to the same seat but the cover was replaced by a leather one. The obtained cushion transmissibilities for both configurations are illustrated in figure 5 (b). It is observed that the material of the cover and the tension induced by the material itself have no significant impact on the vertical transmissibility of the cushion.

3.2.2. Suspension mat

To quantify the impact of the actual suspension mat on the vertical transmissibility, a complementary mat with a stiffer behavior was designed. A quick experimental modal analysis showed a first bending mode at 180 Hz, which can be considered as rigid on the studied frequency range. Figure 6 depicts the transmissibility comparison between the complete seat with the rigid mat and the complete seat with the standard suspension mat. It is observed that the

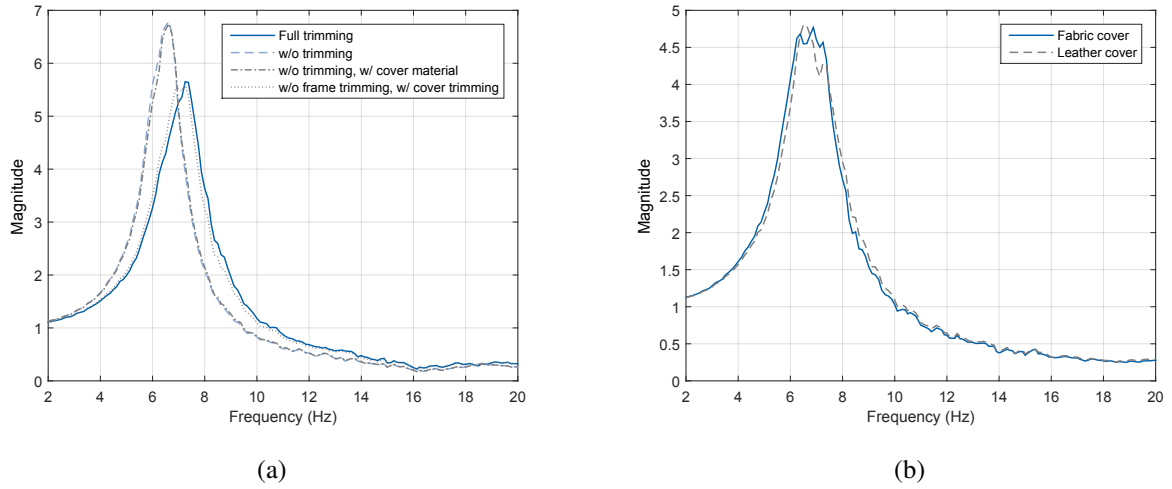


Figure 5: Cover impact on the vertical cushion transmissibility: (a) Impact of the trim; (b) Impact of the cover material

resonance frequency of the transmissibility, initially measured at 7.2 Hz, does not evolve whatever the considered suspension mat; and the cut-off frequency is similar for both mats (10.8 Hz and 10.4 Hz). However, the resonance gain is reduced by 10.5 % using a rigid mat. This confirms that the suspension mat slightly deteriorates the dynamic performances of the cushion, as stated by Ould Eyih [13].

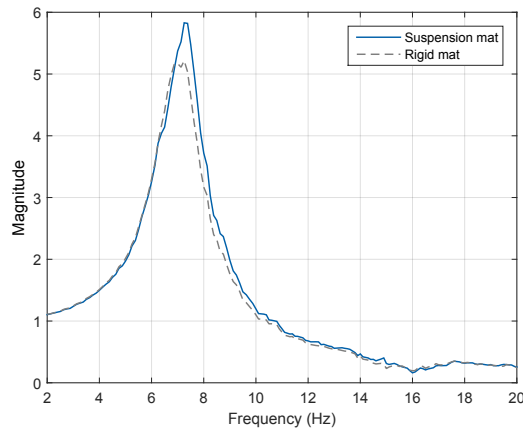


Figure 6: Suspension mat impact on the vertical cushion transmissibility

3.2.3. Foam pad

A final comparison was performed to quantify the impact of the foam pad on the vertical cushion transmissibility. A measurement of a complete seat frame, with a suspension mat and a foam pad on top, was compared to a second one, without any foam pad, as shown in figure 7. To perform this latter test, a new indenter was designed to replace the buttock shape of the lead buttock device, in order to fit with the upper shape of the suspension mat and to avoid

any unwanted friction. This indenter is only in contact with the flexible area of the suspension mat in the vertical direction and it is shaped to avoid any contact with the neighboring components and to respect the H-point location of the system. Three additional masses were mounted on the top of the indenter to reach 7.1 kg, corresponding to the mass of the removed buttock shape, thus ensuring a valid comparison with the previous transmissibility tests (*i.e.* a total mass of the mobile mass equal to 54.8 kg). The obtained comparison is depicted in figure 8. The transmissibility measured without the foam pad shows a 4.3 resonance gain at 6.9 Hz and a cut-off frequency at 11.4 Hz. This result highlights that the foam pad is the main seat sub-component impacting the vertical cushion transmissibility and will require a particular consideration in the modeling process presented in the next section.

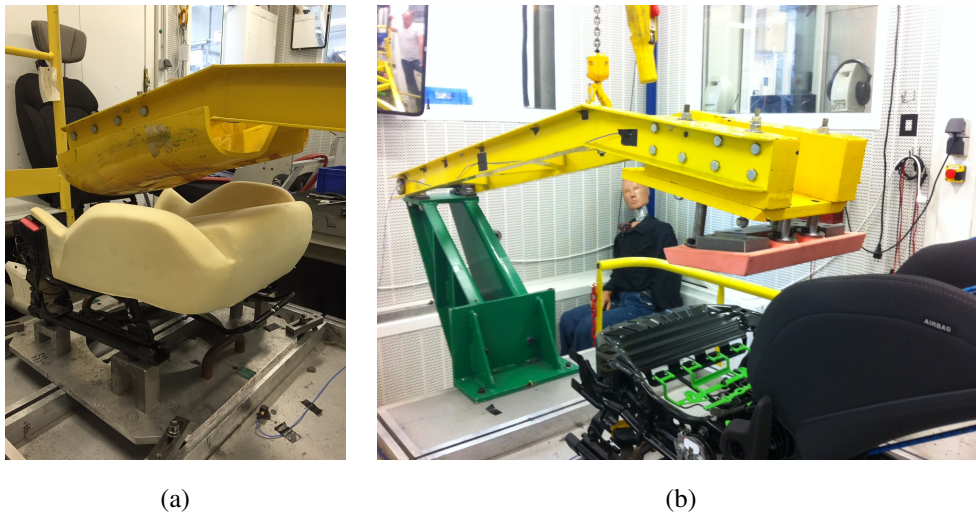


Figure 7: Measurement protocol for foam pad impact investigation: (a) with foam pad; (b) without foam pad

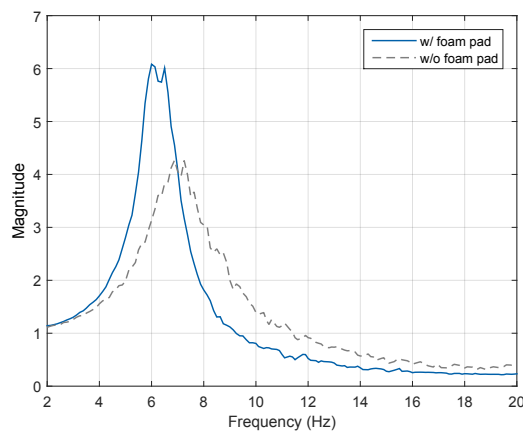


Figure 8: Foam pad impact on the vertical cushion transmissibility

3.2.4. Discussion

This test campaign gives a characterization of the seat dynamic behavior based on an experimental protocol currently available in the automotive industry [29], in order to observe the isolated impact of three sub-components. Their respective impacts in the dynamic transmissibility properties, defined as the relative errors with regard to the initial complete seat configuration, are summarized in table 1.

Configuration	f_r	g_r	f_c
w/o frame trimming	-	-	-2.8 %
w/o cover trimming	-8.3 %	+19.6 %	-11.2 %
w/ rigid mat	-	-10.4 %	-3.7 %
w/o foam pad	+11.3 %	-28.3 %	+22.6 %

Table 1: Impact of the sub-components on the dynamic transmissibility properties

The main conclusion is that the vibrational response of the seat is mainly influenced by two sub-components among four, namely the foam pad and the cover. While the behavior of the foam has already been emphasized in the literature, the impact of the trimming tension has been clearly illustrated. However, more in depth investigations would be needed to describe this complex interaction between the foam and the cover. Therefore, the following modeling step will consider the foam pad and the cover as a unique sub-component with equivalent stiffness and damping properties.

4. Multi-body model

4.1. Model description

As previously stated in section 2.3, the coupled human/seat system implemented in this paper is based on the twenty DOFs multi-body model initially proposed by Kumbhar [23] and it has been adapted as depicted in figure 9. It is composed of five rigid bodies, with their respective masses and centers of gravity, representing the occupant: the thighs (m_1, g_1), the pelvis (m_2, g_2), the torso (m_3, g_3), the head (m_4, g_4) and the viscera (m_5, g_5). Three additional bodies represent the seat: the seat frame (m_{st}), the cushion (m_c) and the backrest (m_{bc}). Each mass except m_5 , has two translational motions (fore-aft and vertical ones) and one rotational motion. The viscera have only two translational motions. All masses except mass m_5 are connected to each other by joints ($j_{(i=1;\dots;4)}$) formed by translational and torsional sets of springs (K_i, K_{ri}) and dampers (C_i, C_{ri}). The viscera are connected by a fore-aft and vertical set of springs and dampers. The contacts between seat masses and body masses are represented by points $c_{(i=1;2;3;25;35)}$. The fore-aft (K_{hi}, C_{hi}) and vertical (K_{vi}, C_{vi}) springs and dampers are associated to their respective masses.

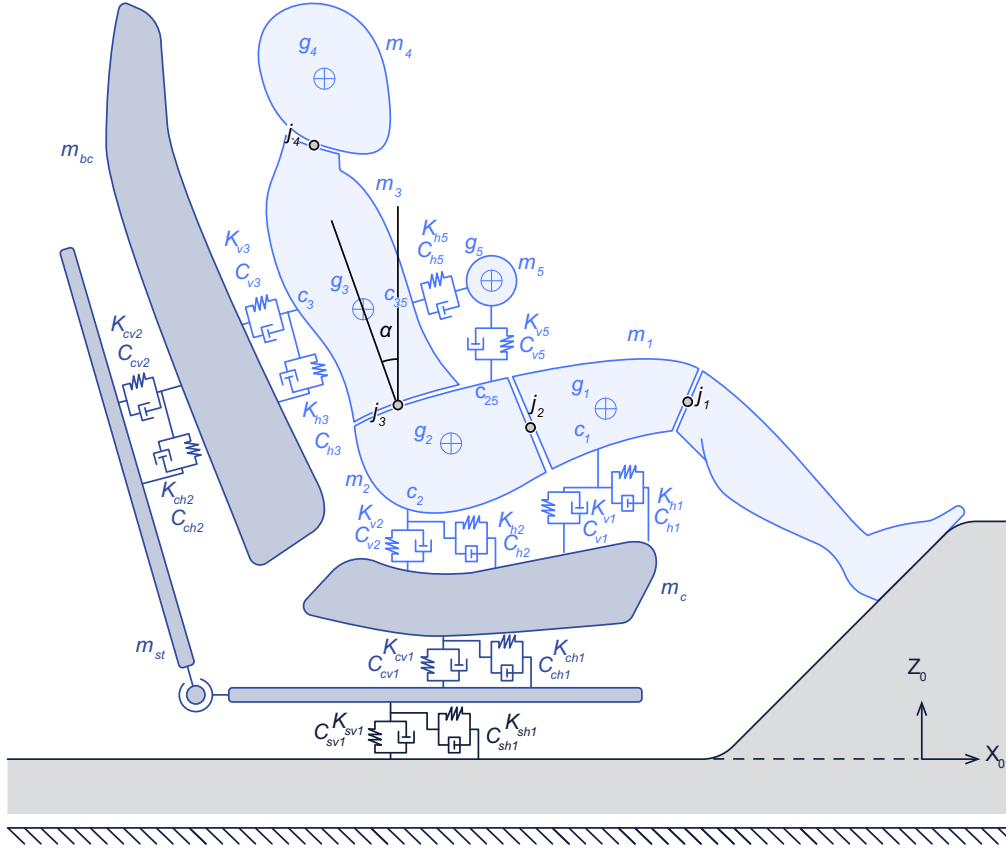


Figure 9: Human/seat model adapted from Kumbhar's study [23]

The twenty equations of motion associated to each degree of freedom of the coupled human/seat model were classically derived using Newton's second law. For example, considering the seat frame sub-component, with mass m_{st} , the equations of motion along the vertical direction are given by:

$$m_{st}\ddot{z}_{st} = f_{0S_z} + f_{SCS_z} + f_{BRCS_z}, \quad (1)$$

where:

- $f_{0S_z} = -K_{sv1}(z_{st} - z_0) - C_{sv1}(\dot{z}_{st} - \dot{z}_0)$, is the vertical component of the force applied by the ground on mass m_{st} ,
- $f_{SCS_z} = -K_{cv1}(z_{st} - z_c) - C_{cv1}(\dot{z}_{st} - \dot{z}_c)$, is the vertical component of the force applied by mass m_c on mass m_{st} ,
- $f_{BRCS_z} = -\sin(\alpha)K_{cv2}(\cos(\alpha)x_{st} + \sin(\alpha)z_{st} - \cos(\alpha)x_{bc} - \sin(\alpha)z_{bc})$
 $-\cos(\alpha)K_{ch2}(-\sin(\alpha)x_{st} + \cos(\alpha)z_{st} + \sin(\alpha)x_{bc} - \cos(\alpha)z_{bc})$
 $-\sin(\alpha)C_{cv2}(\cos(\alpha)\dot{x}_{st} + \sin(\alpha)\dot{z}_{st} - \cos(\alpha)\dot{x}_{bc} - \sin(\alpha)\dot{z}_{bc})$

$-\cos(\alpha)C_{ch2}(-\sin(\alpha)\dot{x}_{st} + \cos(\alpha)\dot{z}_{st} + \sin(\alpha)\dot{x}_{bc} - \cos(\alpha)\dot{z}_{bc})$, is the vertical component of the force applied by mass m_{bc} on mass m_{st} .

The same approach is performed considering each DoF of the coupled human/seat system, leading to a set of equations of motion. Finally, under both steady-state and harmonic excitation assumptions, this can be expressed in the frequency domain under matrix form as:

$$\left[-\omega^2\mathbf{M} + j\omega\mathbf{C} + \mathbf{K}\right]\mathbf{x}(\omega) = \mathbf{f}(\omega), \quad (2)$$

where $\mathbf{M} \in \mathbb{R}^{(20;20)}$, $\mathbf{C} \in \mathbb{R}^{(20;20)}$, $\mathbf{K} \in \mathbb{R}^{(20;20)}$ represent the mass, damping and stiffness matrices, respectively; $\mathbf{x} \in \mathbb{C}^{(20;1)}$ is the displacement vector (ω is the excitation frequency and $j^2 = -1$). Finally, $\mathbf{f} \in \mathbb{R}^{(20;1)}$ represents the input excitation vector corresponding to the kinematic forces generated by the base excitation. Depending on the displacements and/or the velocities prescribed to the seat-base, elastic and/or dissipative forces are generated at the interface between the seat-base and the coupled human-body/seat system. These depend on the interface parameters (i.e. K_{sh1} , K_{sv1} , K_1 and C_{sh1} , C_{sv1} , C_1), the position of sub-component 1 (i.e. the locations of points g_1 and j_1) and the value of the excitation (i.e. x_0 , z_0 and \dot{x}_0 , \dot{z}_0)³.

4.2. Model validation

The coordinates of each body part (depending on the backrest angle α) and their equivalent parameters were directly extracted from Liang and Chiang's study [18], representing a mean human (71.3 kg total mass). They are given in Appendix A. A traditional seat design was considered, as opposed to actively or passively suspended seats, with suspension parameters (K_{sv1} and K_{sh1}) considered as rigid. The cushion parameters, given in table 2, were derived from the lead-buttock measurements presented in a previous study [20]. The nominal values of the seat parameters are given in table 2.

Parameter	Suspension	Cushion
Mass (kg)	$m_{st} = 16.8$	$m_c = 16.8$
Horizontal stiffness (N.m ⁻¹)	$K_{sh1} = 1.10^8$	-
Vertical stiffness (N.m ⁻¹)	$K_{sv1} = 1.10^8$	$K_{cv1} = 85866$
Vertical damping (N.s.m ⁻¹)	-	$C_{cv1} = 448$

Table 2: Nominal seat parameters

To validate the model, the suggested approach consists firstly in studying the biodynamic responses of the seated human body exposed to whole-body vibrations. As widely discussed in the literature [33, 21, 18], several magnitudes

³In order not to overburden the article, the reader is referred to the works of Kumbhar *et al.* [23, 32] to read the complete set of equations associated to the dynamic equilibrium of this 20 DoFs human/seat model and to derive the associated global matrix expressions.

can be considered to assess the dynamic response of a human/body, namely the driving-point mechanical impedance, seat-to-head transmissibility, or the apparent mass. The latter is defined as the ratio between the force and the acceleration at the interface between the surface of a rigid seat and the seated human body [34]. It is therefore only representative of the occupant's morphology and posture. Considering the nominal set of human body parameters, a 12° backrest angle, and assuming a rigid behavior of the seat model (by arbitrarily fixing the stiffness parameters of the seat to 1.10^8 N.m^{-1}) the apparent mass is computed and compared to the experimental and simulated data extracted from Liang and Chiang's study [18], as illustrated in figure 10. The apparent mass computed with the proposed model globally fits to the reference data with a 8 % gap at the resonance frequency and 2 % gap at the maximum amplitude. The main discrepancies are located before the amplitude resonance where the proposed model tends to slightly underestimate the response level. Regarding these results, the human-body part of the model can be considered as validated.

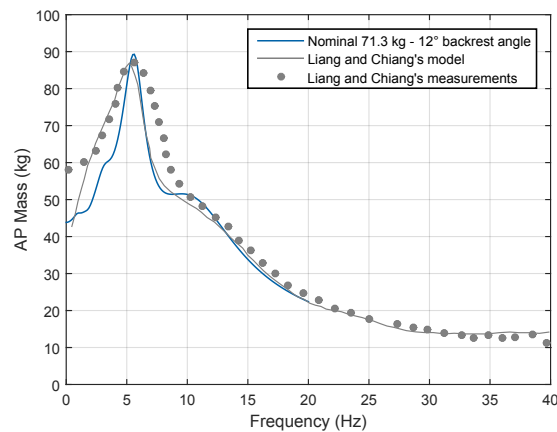


Figure 10: Comparison of computed apparent mass of the human body model

Secondly, a test-model comparison is performed to evaluate the accuracy of the complete coupled human/seat model. Hence, transmissibility measurements were carried out on the same seat, using the same test facility as depicted in figure 2 but replacing the lead-buttock with a human seated human subject. These tests were carried out in accordance with The Code of Ethics of the World Medical Association (Declaration of Helsinki) for experiments involving humans. The posture was adjusted to a standard-driver one with a 21° backrest angle, and harmonic displacements with a unitary displacement amplitude were applied at the seat base. However, it must be emphasized that the considered human subject was a bit heavier than the theoretical mean one, which led to re-calibrate the parameters of the human body reach a 80 kg total mass. The obtained results are illustrated in figure 11. Two experimental measurements were performed to check the repeatability. The first resonance frequency around 4.3 Hz is effectively estimated and the second one is present around 7 Hz. The filtration zone is also globally well predicted. However, the resonance amplitude is still underestimated.

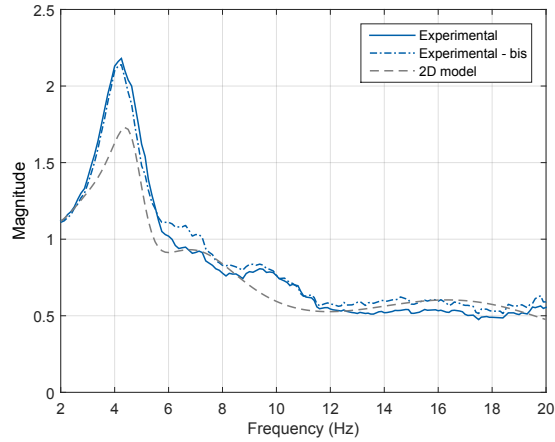


Figure 11: Test-model comparison of the vertical cushion transmissibility

4.3. Discussion

In view of these results, some improvements may seem necessary to improve the model accuracy. Two types of modifications could be considered, namely parametric and non-parametric ones. The latter are related to high modifications of the model, for example to take into account the influence of the friction forces at the feet by adding a dedicated rigid body and computing the associated reaction forces at the interface with the rest of the human body model. The former are related to the experimental characterization of the mass, stiffness and damping parameters of both seat and human body models.

However, despite the tuning of the total mass of the human body model, all the considered parameters have been kept to their nominal “best-estimate” values, coming from the literature or from sub-component tests. Moreover, no model-updating step was initiated. As an example, Desai *et al.* recently proposed a multi-objective-based optimization procedure to determine optimal sets of parameters in order to improve the prediction of concurrent dynamic features (e.g. seat to head transmissibility, apparent mass) [35]. While such approaches could have resulted in better correlations, this would not have given any answer concerning the experimentally observed variability. Indeed, regarding the previous results, it is difficult at this stage, to determine which of the human body or the seat has the most important impact on the observed transmissibilities. Therefore, sensitivity analysis are further proposed to study how the input parameters of the model and their associated uncertainties impact its outputs.

5. Sensitivity analysis

Refining the approach proposed in the previous discussion, the objective of this section is to determine which of the human body or the seat has the greatest impact on the observed transmissibility, using global sensitivity analysis methods. In the last twenty years, lots of work has been published concerning the development and the application of such methods, as tools to increase the understanding on relationships between input and output parameters of

complex models (see the reference book by Saltelli *et al.* [36]). These are usually divided into three categories: screening techniques, methods based on the variance decomposition of the output and sample-based analysis. In the following sub-sections, they are each briefly introduced and adapted to study the behavior of the proposed human/seat model.

5.1. Qualitative approach

Considering the large number of input parameters typically associated to complex models, screening techniques are used to identify the most impacting factors, with the lowest computational cost. These techniques are often based on One-At-a-Time (OAT) experiments: the impact of each input factor is evaluated one after the other, which does not allow investigating possible interactions. The main drawback of such methods is that they only provide qualitative measures of the sensitivity, not quantitative ones, leading to a relative ranking of the input factors.

The proposed human/seat coupled system is composed of forty-one input parameters related to the human body and sixteen to the seat. To investigate their influence on the model outputs, they are further allowed to take different levels of variation between specified bounds around their nominal values. Regarding the masses of the human body, the variation is arbitrarily set to be comprised between 70 % (5th human percentile) and 140 % (95th human percentile) of the nominal values. The backrest angle can vary between 0 ° and 84 °, in order to take into account new postures for future industrial trends. The remaining input factors are arbitrarily bounded between ± 50 % of their nominal values, due to their unknown variability.

The considered outputs are directly derived from the computed cushion transmissibility: the resonance frequency (f_r) and the area under the transmissibility curve (A), cf. figure 3. The underlying idea is that, from a seat manufacturer's point of view, the dynamic behavior of a seat is considered as "better" when the area and the resonance frequency are the lowest possible.

In this study, the method initially proposed by Morris [37, 38] is used for this purpose. The sensitivity indexes provided by this method are: the absolute value of the overall effect μ and its standard deviation σ . The former allows detecting input parameters that exhibit an important influence. The latter gives information on coupling factors and non-linear effects. For each input factor, whose possible values are discretized into p levels, the Morris indexes are calculated on a number of iterations $N = r(k + 1)$ where k is the number of input parameters and r the number of successively considered sampling points, also called "trajectories". The elementary effect of the j^{th} variable, for the i^{th} sample of the input factors \mathbf{x}^i , denoted E_j^i , is defined as⁴:

$$E_j^i = \frac{f(\mathbf{x}^i + \Delta \mathbf{e}_j) - f(\mathbf{x}^i)}{\Delta} \quad , \quad (3)$$

where Δ is a multiple of $\frac{1}{p-1}$ and \mathbf{e}_j is a unitary vector whose j^{th} component is equal to 1 and the others are identically

⁴The reader is referred to the reference book by Saltelli *et al.* [36] for more theoretical and computational details about the Morris screening technique.

null. Morris indexes are further determined according to the following equations:

$$\mu_j = \frac{1}{r} \sum_{i=1}^r |E_j^i| \quad \text{and} \quad \sigma_j = \sqrt{\frac{1}{r} \sum_{i=1}^r \left(E_j^i - \frac{1}{r} \sum_{i=1}^r E_j^i \right)^2} . \quad (4)$$

Figure 12 shows the derived results (for 100 discretization levels and 500 trajectories). This OAT method allows dividing the input factors into three groups depending on their associated effects, *i.e.* negligible (low μ and low σ), linear without interactions (high μ and low σ), non-linear or with interactions (high μ and high σ). Three input parameters for the f_r criterion are highlighted: m_5 , K_{v5} and K_{cv1} . Concerning the A criterion, no parameter appears as impacting. It is observed that for the vertical cushion transmissibility, only one parameter coming from the seat is important: the vertical stiffness K_{cv1} .

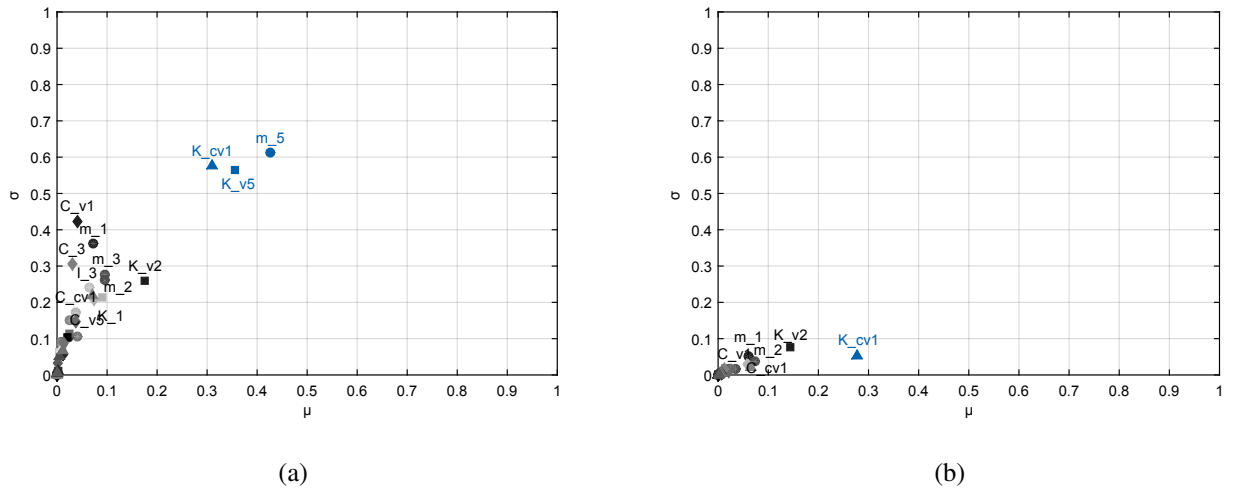


Figure 12: Results of the Morris method on: (a) f_r criterion; (b) A criterion

While the Morris analysis allows identifying the most impacting parameters among the fifty-seven ones of the model, it does not give any quantitative information. The next step thus consists in performing a variance-based global sensitivity analysis called FAST analysis, to quantify the importance of these identified parameters and also, to highlight potential coupling effects between these parameters.

5.2. Quantitative approach

Variance-based methods aim to decompose the variance of the model output y into fractions which can be attributed to inputs $\mathbf{x} = [x_1, \dots, x_{np}]^T$. The advantage of these methods is their ability to cover the whole design space and to quantify the effect of the interactions between input factors, even when facing non-linear output responses. Most of these methods are based on decomposing the variance of the output $V(y)$ as:

$$V(y) = V(E(y|x_i)) + E(V(y|x_i)) , \quad (5)$$

where $E(y|x_i)$ and $V(y|x_i)$ are the expected value (or expectation, long run average value) and the variance of y when the input factor x_i is fixed to an *a priori* chosen value, respectively. The variance of the conditional expected value $V_i = V(E(y|x_i))$, also called the “main effect” of x_i on y , is usually considered as a good measure of the sensitivity of y with regard to x_i . In order to derive higher order effects representing the interactions between the input parameters on the output, the previous decomposition is classically extended, as proposed by Sobol [39]:

$$V(y) = \sum_{i=1}^{n_p} V_i + \sum_{1 \leq i < j \leq n_p} V_{ij} + \sum_{1 \leq i < j < k \leq n_p} V_{ijk} + \dots + V_{1\dots n_p}, \quad (6)$$

where:

$$V_i = V(E(y|x_i)), \quad (7)$$

$$V_{ij} = V(E(y|x_i, x_j)) - V_i - V_j, \quad (8)$$

$$V_{ijk} = V(E(y|x_i, x_j, x_k)) - V_{ij} - V_{ik} - V_{jk} - V_i - V_j - V_k, \quad (9)$$

and so on. It can be noticed that in the case of a model without interactions, only the V_i coefficients are different from 0. Similarly, higher order sensitivity indexes are obtained by normalizing the previous coefficients by $V(y)$. Furthermore, these coefficients also often normalized, leading to $S_i = \frac{V_i}{V(y)}$, which are called the “first order effects” sensitivity index and conveniently lie between [0, 1].

While it is theoretically possible to derive all these higher order coefficients ($2^{n_p} - 1$ coefficients), computations are usually limited to the second order ones as the associated numerical costs quickly become prohibitive. However, the sensitivity analyses are often complemented with the so-called “total effect” coefficients, corresponding to the sum of all terms of any order that include the x_i factor. For example, if $\mathbf{x} = [x_1, x_2, x_3, x_4]^T$, then the total effect associated to the x_2 factor is defined as:

$$S_2^T = S_2 + S_{12} + S_{23} + S_{24} + S_{123} + S_{124} + S_{234} + S_{1234}. \quad (10)$$

It can be noticed that, for example, if only the first and second order indices are initially computed, the interactions of higher order terms can be estimated by subtracting these values to the total effect coefficients⁵.

In the case of engineering applications, the complexity of the model does not allow to analytically derive all these coefficients which are usually estimated numerically using either the method of Sobol or FAST (Fourier Analysis Sensitivity Test), or extensions. The former, based on random samplings, allows all the coefficients to be derived but is computationally cost effective [36]. To overcome this issue, the FAST method has been proposed to estimate the first order term, based on a Fourier transform of the output variance. An extended FAST method has been later proposed by Saltelli *et al.* allowing to also derive the total effect terms⁶.

⁵As the numerical implementation of this methods goes out of the topic of this article, the reader is referred to the works of Saltelli *et al.* [36, 40] for more theoretical and computational details about this method.

⁶*ibid.*

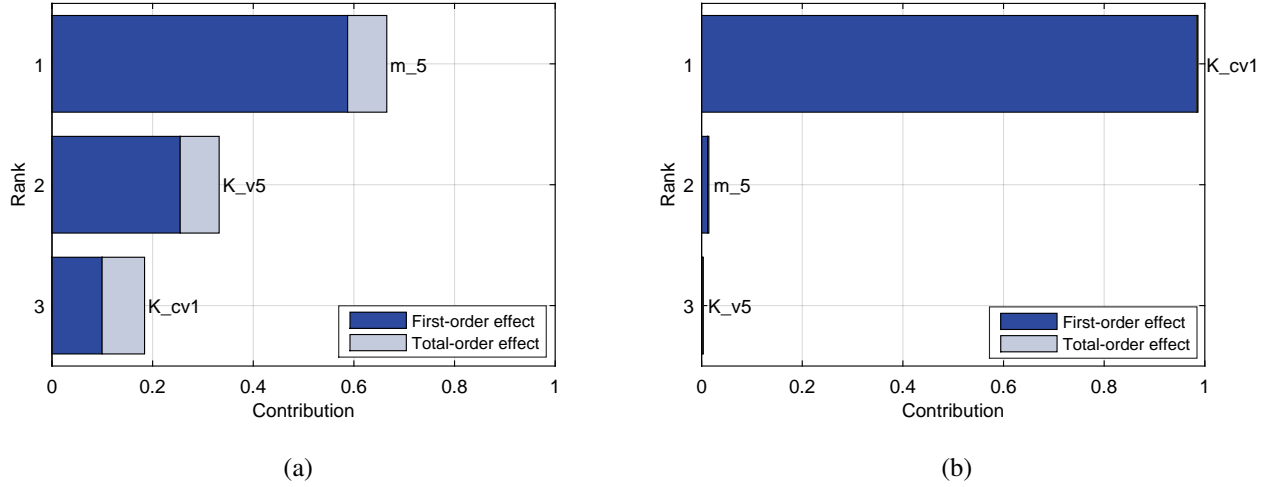


Figure 13: Results of the Extended-FAST method on: (a) f_r criterion; (b) A criterion

Figure 13 illustrates the obtained results: dark blue bars represent the first order coefficients and light blue ones represent the total effects, reflecting the weight of all the interactions involving the considered parameters. It is observed in figure 13 (a) that parameters m_5 and K_{v5} , related to the human body, exhibit the main contributions (0.58 and 0.25 for the f_r criterion, respectively). This confirms the previous parameter ranking derived from the Morris analysis. Low interactions are also noticed between the input parameters (around 0.08). Concerning the A criterion (figure 13 (b)), it is shown that the vertical stiffness K_{cv1} has the most influence on the transmissibility area variation (contribution of 0.98) but has no interaction with the other selected input parameters.

These analyses allowed identifying which input parameters have the most impact on the output transmissibility of the model and quantifying their respective contributions. However, in terms of seat design, it seems interesting to visualize the evolution of the transmissibility involved by the variation of these identified parameters using sample-based analysis.

5.3. Evolution of the cushion transmissibility

Sample-based analysis use several evaluations of the model from random inputs in order to determine the uncertainty on the model prediction and the contribution of each input parameter [41]. Monte Carlo simulations are usually used: in a first step, probability distributions are specified (*i.e.* type and parameters) for each input parameter; then, a high number of samples are generated accordingly; finally, the model is evaluated for each set of input parameters. Various algorithms are available to generate these samples, so as to cover most of the design space: Latin Hypercube sampling is subsequently used. Considering the lack of knowledge on the probability distribution associated to each parameter, these have been arbitrarily defined as uniform, in agreement with the previously defined bounds.

Figure 14 (a) depicts 1000 computed cushion transmissibilities where only the previously identified important

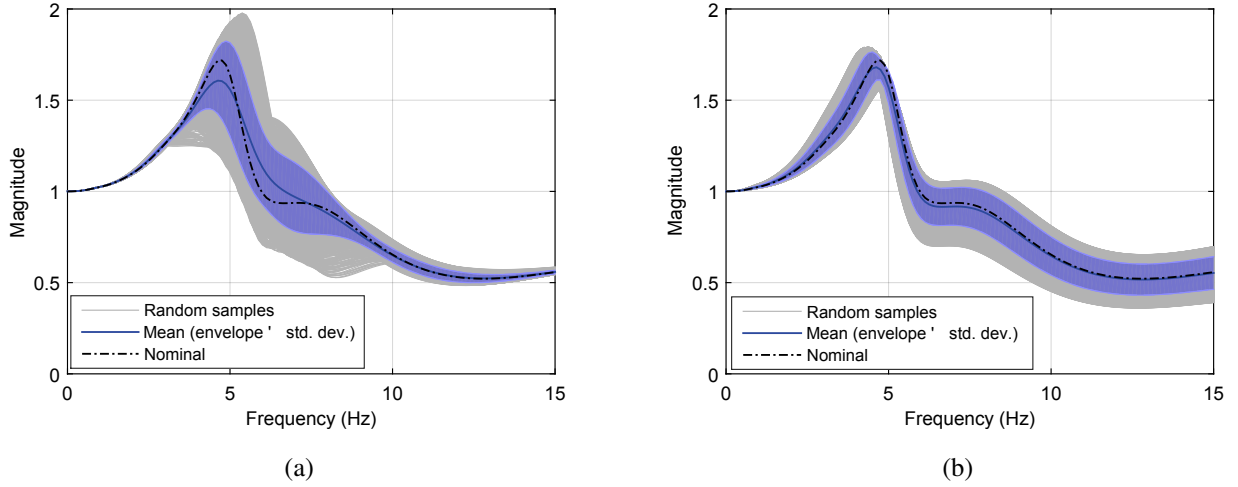


Figure 14: Evolution of the cushion transmissibility: (a) human body variation; (b) seat variation

input parameters from the human body (m_5 and K_{v5}) are varying⁷. The blue shaded area represents the standard deviation of these transmissibilities, quantifying the dispersion of the response at each frequency. A low standard deviation indicates that the transmissibilities tend to be close to the mean (blue curve) of the set, while a high standard deviation indicates that the output responses are spread out over a wider range of values. The black curve represents the nominal transmissibility.

Then, the input parameters associated to the human body are blocked to their nominal values, and only the most impacting seat parameter (K_{cv1}) is varying. A new sampling is performed to illustrate its impact on the vertical cushion transmissibility, as depicted in figure 14 (b). It is observed that while the evolution range induced by the variation of K_{cv1} is lower between 3 and 9 Hz, it is higher from 9 Hz than the evolution induced by the variation of human body parameters (which is also confirmed by the narrower corridor defined by the standard deviation).

Therefore, it can be concluded that the input parameters of the human body have more impact than the seat ones on the transmissibility around the resonance frequency and that only the seat parameter has some impact on the filtration area.

6. Conclusion

A hybrid test/model approach was proposed to assess the impact of the various sub-components of an automotive seat and of the occupant morphology, on the dynamic response of a coupled human body/seat system. The seat cushion transmissibility was chosen as a unique output feature since it is commonly used in the automotive industry as a vibrational comfort indicator. The test campaign, performed on an industrial standard test bench, allowed to

⁷To ensure the representativeness of the output variability, the size of the random samples can be determined by verifying the convergence of *a posteriori* indicators, such as cumulative sums [42]

strengthen some of the trends already mentioned in the literature, especially the impact of the foam pad and its complex interactions with the trimmed cover. A two-dimensional multi-body model was implemented to simulate the response of coupled human/seat system. Based on nominal parameter values extracted from the previous experimental characterization (for the seat) and from the literature (for the human body), a deterministic solution was derived, showing an acceptable correlation with regard to measurements performed with a real human subject. This model was further used to evaluate, both qualitatively and quantitatively, the influence of each model parameter on the cushion transmissibility, using global sensitivity analysis methods. The results showed that the variability of the human body, especially the parameters associated to the internal dynamics of the body, is the most impacting factor. It essentially influences the area around the transmissibility resonance, where the seat amplifies the incoming vibrations. Furthermore, the dynamic behavior of the seat is mainly controlled by the stiffness parameter of the cushion, which has a high impact on the filtration area. Consequently, from a seat manufacturer point of view, it is concluded that the human body variability has to be considered to provide “personalized” comfort for the occupant and that the design space to improve the vibrational comfort is strongly depending on the dynamic behavior of the cushion. Hence, complementary investigations are needed to better describe the interaction between the foam, the cover and the seated occupant.

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Appendix A. Parameters of the human body model

The values of the human model parameters are directly extracted from Liang and Chiang’s study [18]:

Points	c_1	c_2	c_{25}	c_3	c_{35}	j_1	j_2	j_3	j_4	g_1	g_2	g_3	g_4	g_5
X(mm)	133	-19	-26.9	-57	-56.3	329	-17	-84	-8	133	-23	-26.8	18.1	-27
Z (mm)	0	0	137	386	237.4	52	84	139	626	70	97	474	726	237

Table A.3: Coordinates of the human body in erect posture

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Body parameters	Body 1	Body 2	Body 3	Body 4	Body 5
Masses (kg)	$m_1 = 20.3$	$m_2 = 11.0$	$m_3 = 19.87$	$m_4 = 7.25$	$m_5 = 12.9$
Inertia (kg.m ²)	$I_1 = 1.16$	$I_2 = 0.68$	$I_3 = 1.53$	$I_4 = 0.402$	-
K_h (N.m ⁻¹)	$K_{h1} = 614$	$K_{h2} = 905$	$K_{h3} = 17200$	-	$K_{h5} = 1930$
K_v (N.m ⁻¹)	$K_{v1} = 16710$	$K_{v2} = 121300$	$K_{v3} = 2300$	-	$K_{v5} = 18370$
C_h (N.s.m ⁻¹)	$C_{h1} = 14$	$C_{h2} = 15$	$C_{h3} = 334.5$	-	$C_{h5} = 79$
C_v (N.s.m ⁻¹)	$C_{v1} = 8010$	$C_{v2} = 47$	$C_{v3} = 154$	-	$C_{v5} = 197$

Table A.4: Human body parameters

Joint parameters	Joint 1	Joint 2	Joint 3	Joint 4
Translational stiffness (N.m ⁻¹)	$K_1 = 23550$	$K_2 = 6400$	$K_3 = 299$	$K_4 = 113700$
Rotational stiffness (N.m.rad ⁻¹)	$K_{r1} = 220$	$K_{r2} = 162$	$K_{r3} = 328$	$K_{r4} = 915$
Translational damping (N.s.m ⁻¹)	$C_1 = 154$	$C_2 = 61$	$C_3 = 1790$	$C_4 = 66$
Rotational damping (N.m.s.rad ⁻¹)	$C_{r1} = 104$	$C_{r2} = 30$	$C_{r3} = 724$	$C_{r4} = 340$

Table A.5: Human joint parameters

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