On the Contact Interfaces between the Driver and the Vehicle Seat

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ABSTRACT

Automotive seating is characterized by boundary conditions which contain nonlinear contact interfaces. These interfaces experience the vibro-impacts (slaps) and frictional slips. The slaps take place in contact interfaces subjected to high amplitude vibrations, being characterized by very brief duration, rapid dissipation of energy and large accelerations and decelerations, respectively. Including of friction into the contact interface modeling improves the interaction between the driver and the vehicle seat by making the simulation more realistic. The vibro-impacts and the frictional slips can be developed simultaneously in a contact surface. As a result, the friction-induced vibrations interact with the vibrations and waves at the interface, generating sounds and instabilities. Identification of boundary conditions in the driver-seat system is performed by using the concept of the interference distance or penetration. The interference distance is introduced as an optimization problem. It is shown that the optimization problem provides robust solutions to minimum distance and interference problems.

INTRODUCTION

The evaluation of comfort and discomfort of the driver depends on the interaction between the seat properties, driver characteristics, environmental conditions and the driver-seat system, respectively. When the slip and vibro-impact mechanisms are simultaneously developed, the picture of the actual normal forces of the contact interface is changed as shown by Jalali et al. [1] when investigating the micro-vibro-impacts at boundary condition of a clamped Euler-Bernoulli beam. The impact mechanism affects the frictional slip mechanism when they simultaneously develop in a contact interface. Karnopp [2] describes the friction as a function of the relative velocity, while Menq et al. [3] introduced a modified micro-slip model in terms of an elasto-plastic shear layer. Comprehensive investigations of contact friction and the vibro-impact mechanism into the contact interfaces subjected to high amplitude vibrations have been performed by Ferri [4], Berger [5], and Gilardi and Sharf [6], respectively.
Returning to the automotive seating subject, we know that the road and vehicle engines conditions are transferred into the body through the seat. Most car seats have fore- and aft adjustment as well as adjustable seat back inclination for obtaining an acceptable occupant position and also to fitting as large a range of the population as possible (Jonsson [7]). An optimal seatback with seat pan inclination according to (Harrison et al. [8]) is shown in ***Figure.1***. The position of the driver in the car is important for the driving performance. However, the neck injuries in rear-end collisions are still an important problem for the society. Much research effort has been expended for understanding the impacts developing in the contact interfaces (Emaci et al. [9], Moon. and Shawand [10] and Yang and Begeman [11].

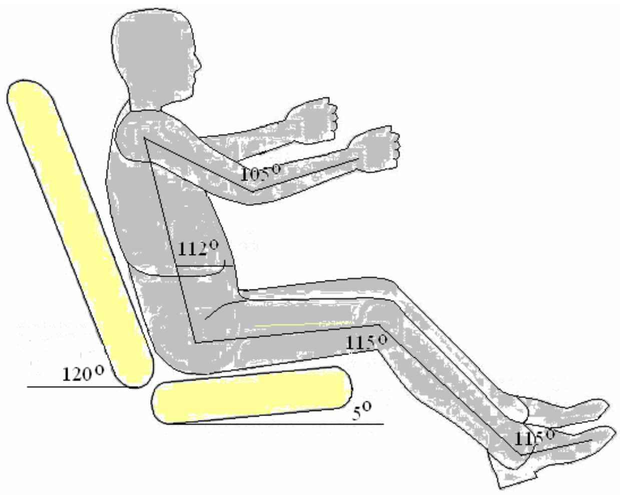

Figure 1. A generic seatback.

A continuous approach of the vibro-contact dynamics is developed in this paper by admitting an explicit relationship between contact force and deformation. In the continuous approach, there is no difference between the impact and the contact, and as a consequence, the methods of the non-impact dynamics can be used to solve the problem. Unlike the discrete formulation, the continuous approach of the vibro-contact dynamics allows for use of any friction model. The friction force can be considered as a combination of forces that resist motion during different energies conversion processes (elastic/plastic contacts viscous dissipation, fracture, and adhesion). In addition, the identification of nonlinear boundary conditions is performed by using the concept of the interference distance or penetration. This concept defines the contact force which in turn depends on the deformation at the contact interface. Generally, the impacts between bodies are defined by the condition of impenetrability (Kim [12]), because the contact can be found by checking the minimum distance between bodies. The contact point is the point where the minimum distance is reached [6] as shown in:

$$\min\left(\frac{1}{2}(p_1-p_2)^T(p_1-p_2)\right),\quad g_1(p_1)\le 0, g_2(p_2)\le 0\,, \qquad (1)$$

where p_1 and p_2 are the position vectors of two points on the two bodies and g_1 and g_2 are the bounding surface constraints. To define the contact force, the interference distance can be computed as

$$\min(-d),\quad g_1(p_1)\le -\frac{d}{2}e_1,\ g_2(p_2)\le -\frac{d}{2}e_2\,, \qquad (2)$$

where d is the interference distance and e_1 and e_2 are the unit vectors.

PROBLEM FORMULATION AND SOLUTION

The back of the driver surface is modeled as a double curved thin shell of thickness h and the principal radii of curvature R_1 and R_2 as shown in ***Figure 2***.

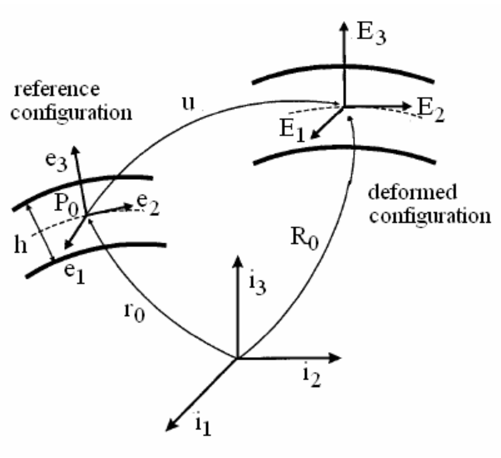

Figure 2. Reference and deformed configurations of the back of the driver shell.

The reference surface area is denoted by Ω . An inertial frame of reference is defined by three mutually orthogonal unit vectors i_1, i_2, i_3 . Position vector of a point of the shell can be written as

$$R(\xi_1,\xi_2,\zeta)=r_0(\xi_1,\xi_2)+u(\xi_1,\xi_2)+\zeta E_3(\xi_1,\xi_2)\,, \qquad (3)$$

where ξ_1,ξ_2 are the material coordinates (the lines of curvatures of the shell reference surface), ζ is the material coordinate along the normal to the reference surface $n=e_3$, r_0 is the position vector of the point P_0 on the reference surface of the shell where the shell has a contact boundary condition, and u is the reference surface displacement vector. E_3 is a unit vector, and E_1, E_2 are the axial and transverse shearing strains develop during deformation. The unknown driver-seat contact domain D_c can be described by the Lamé curve

$$\left(\frac{x}{a}\right)^n+\left(\frac{y}{b}\right)^n=1,\ n>0\,, \qquad (4)$$

where a and b are the radii of the oval shape. The parametric representation of (4) is

$$x(\theta)=a\cos^{2/n}\theta\,,\ y(\theta)=b\sin^{2/n}\theta\,. \qquad (5)$$

The integral motion equation is obtained by applying the Hamilton principle (Bauchau and Choi [13])

$$\int_{t_1}^{t_2}\int_{\Omega}\left\{\delta u(\dot{g}_1-N_{1,1}-N_{2,2})+\delta E(\dot{g}_2-M_{1,1}-M_{2,2}+F_c)\right\}\mathrm{d}\Omega\mathrm{d}t=0\,, \qquad (6)$$

where g_1, g_2 are the linear and angular momentum vectors of the shell, N_1, N_2 are spatial in-plane forces, F_c is the contact force, M_1, M_2 are the bending moments. A constant normal force P is applied to the shell at the contact point P_0 . The shell is excited by forced vibrations $F(t)=f\sin(\omega t)$ having an arbitrary point excitation. The excitation frequency is chosen to be closer to the resonant frequencies of the spine. The nonlinear behavior can be spanned using its first n nonlinear normal modes. We suppose that each contact point permits small deflections. For large amplitude of vibrations the micro-slip and micro-vibro-impact develop in the contact interface of the shell. The associated contact force constrain F_c and the sticking constraint, respectively, are given by

$$F_c=\varphi_c(\delta,\dot{\delta})\,,\quad \varphi_{st}(\xi_{1c},\xi_{2c},\zeta_c,F_t)=0\,, \qquad (7)$$

where δ is the local indentation function

$$\delta(\xi_{1c},\xi_{2c},\zeta_c,\dot{\xi}_{1c},\dot{\xi}_{2c},\dot{\zeta}_c)\,,$$

And $\xi_{1c}, \xi_{2c}, \zeta_c$ are the components of the position vector of a contact point. So, the contact force F_c can be explicitly written as

$$F_c = A\delta^n + B\delta^p\dot{\delta}^q , \tag{8}$$

where n, p, q and constants, the coefficient A depends on the material and the geometric properties of the bodies in contact, and B is defined with respect to the coefficient of restitution. Both coefficients also depend on the flexural and normal stiffnesses of the boundary condition. In the continuous model the damping depends on the indentation since the contact area increases with deformation. The first term in (8) represents the contact constraint expressed as a functional between the contact force F_c at the contact point coordinates $\xi_{1c}, \xi_{2c}, \zeta_c$ and their time derivatives. Therefore, the normal contact force F_c given by $(7)_1$ depends on the local indentation δ (Hunt and Crossley [14]). Next, the friction F_t at the contact point during sticking can be defined as (Stronge [15])

$$F_t = C\delta_t + D\dot{\delta}_t , \tag{9}$$

where δ_t is the tangential component of displacement at the contact point, C is the tangential stiffness and D is defined with respect to the coefficient of restitution $0 \le e \le 1$. The energy released during restitution W_r at the dynamical contact (impact) can be calculated as the negative work done by F_c during the collision

$$W_r = -\int F_c \dot{\delta} \mathrm{d}t . \tag{10}$$

The coefficient of restitution can be calculated from

$$e = \sqrt{\frac{W_r}{-W_c}} , \tag{11}$$

where W_r is the energy released during restitution and W_c is the energy absorbed during compression. The unknowns of the problem are the contact vectors, the contact forces and the friction forces, respectively. These unknowns are determined by solving (6) by using the cnoidal method (Munteanu and Donescu [16]). The general solutions of (6), denoted by S, can be written in the form

$$S = S_{\mathrm{lin}} + S_{\mathrm{nonlin}} , \tag{12}$$

where the linear and nonlinear terms express the linear and nonlinear superposition, respectively, of cnoidal functions

$$S_{\mathrm{lin}} = 2\sum_{k=0}^{n} \alpha_k \mathrm{cn}^2(\omega_k\zeta; m_k) ,$$

$$S_{\mathrm{nonlin}} = \frac{\sum_{k=0}^{n} \beta_k \mathrm{cn}^2(\omega_k\zeta; m_k)}{1 + \sum_{k=0}^{n} \gamma_k \mathrm{cn}^2(\omega_k\zeta; m_k)} . \tag{13}$$

where $0 \le m_k \le 1$ are cnoidal moduli, and the quantities ω_k, α_k, β_k and γ_k are depending on φ_c and φ_{st} (see Appendix). The presence of the contact with no material overlapping condition is detected by checking the minimum distance between bodies. A genetic algorithm is used to solve the constrained optimization problem (1) and (2). Figure 3 displays the contact domain D_c containing several contact points detected on the back seat at the beginning of an impact, for high amplitude vibrations. In this figure the "contact point" 1 was chosen for the subsequent analysis. The size of the "contact point" depends on the mesh of the genetic algorithm.

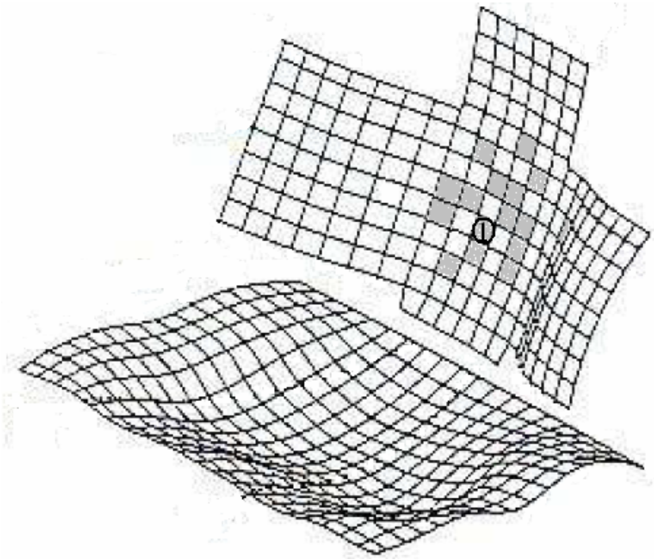

Figure 3. The contact domain D_c.

Figure 4 shows the identified contact force F_c with respect to δ for the contact point 1, for four resonant frequencies of the spine. Certain frequencies in the range of the resonance frequency of the human body and specially the spine have been proved to cause the low back pain. The resonance frequency of the spine is about 4.75-6.25 Hz (Matsumoto and Griffin [17]). ***Figure 5*** shows the variation of the contact force F_c with respect to δ at the contact point 1 for the same resonant frequencies. The variation of the friction F_t with respect to tangential component of displacement δ_t at the contact point 1 is displayed in ***Figure 6*** for the same resonant frequencies.

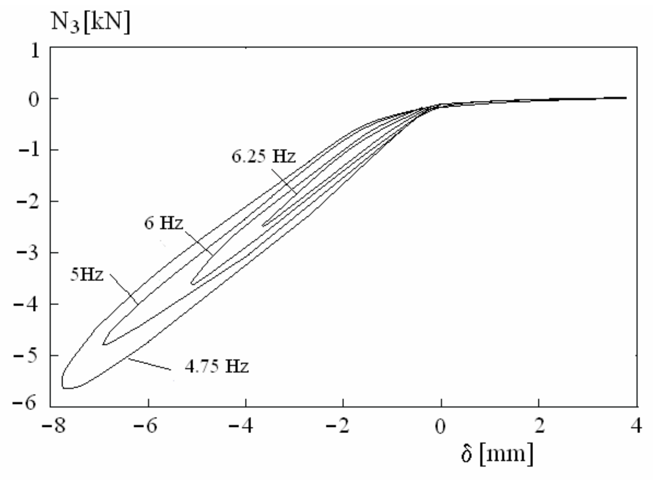

Figure 4. Contact force F_c ***with respect to*** δ ***in the contact point 1***.

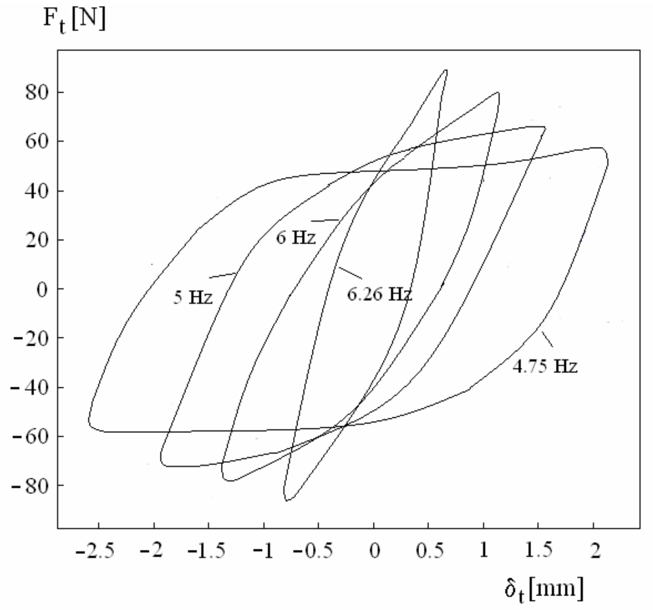

Figure 5. Friction force F_t ***with respect to*** δ_t ***in the contact point 1***.

The optimization problem provides robust solutions to minimum distance and interference problems. The time variation of the ratio between the area of the contact domain D_c and a chosen reference contact area, during the impact is presented in ***Figure 7*** for the same resonant frequencies as before. When the slip and vibro-impact mechanisms are simultaneously developed, the vibro-impact modifies the size of the contact domain. As a consequence the driver-seat system is subject to elastic and/or plastic deformation with dissipation of energy. The collision between bodies is central with the mass centers on the line of impact.

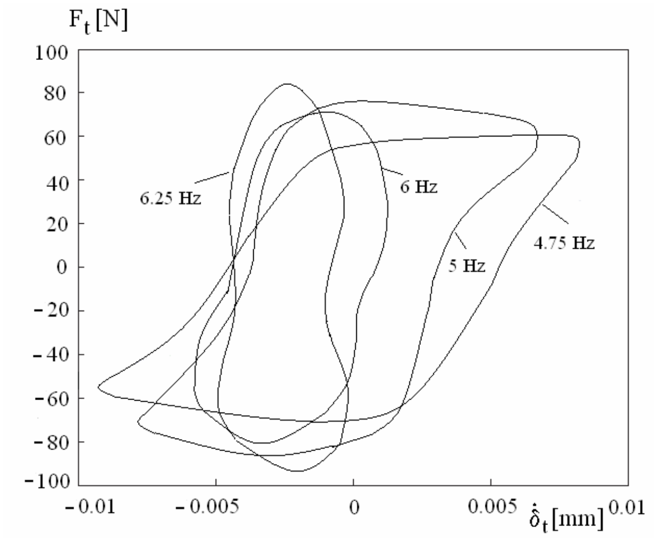

Figure 6. Friction force F_t ***with respect to*** $\dot{\delta}_t$ ***in the contact point 1***.

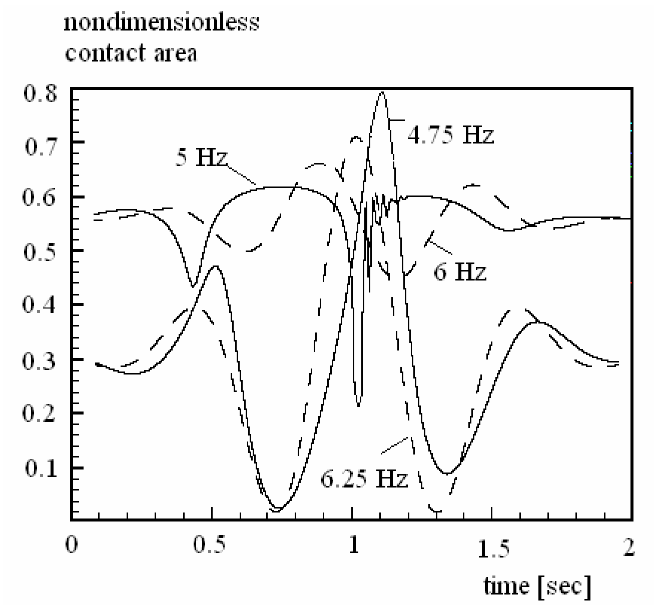

Figure 7. Time variation of the contact area D_c

CONCLUSIONS

The paper is concerned on the modeling of coupled vibro-impacts and frictional slips at the boundary condition in the driver-seat system. Driver position had a double risk compared with other passenger seat positions. The contacting bodies experience contacts and vibro-impacts at multiple points simultaneously. Each contact point allows small deformations in normal and tangential directions at the contact interface. The interaction between the human and the seat occurs in a short time and exhibits energy transfer and dissipation.

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APPENDIX

Cnoidal method (Munteanu and Donescu [16])

The general solution of the nonlinear ODE or PDE can be written in the terms of the *theta function* representation

$$\theta(x,t) = 2\frac{\mathrm{d}^2}{\mathrm{d}x^2}\log\Theta_n(\eta_1,\eta_2,...,\eta_n), \tag{A1}$$

where Θ is the *theta function* defined as

$$\Theta_n(\eta_1,\eta_2,...,\eta_n) = \sum_{M\in(-\infty,\infty)}\exp(\mathrm{i}\sum_{i=1}^{n}M_i\eta_i + \frac{1}{2}\sum_{i,j=1}^{n}M_iB_{ij}M_j), \tag{A2}$$

with n the number of degrees of freedom for a particular solution of the equation, and

$$\eta_j = k_jx - \omega_jt + \phi_j,\ 1\le j\le N. \tag{A3}$$

In (A3), k_j are the wave numbers, the ω_j are the frequencies and the ϕ_j are the phases. Also we have

$$M\eta = Kx - \Omega t + \Phi,$$

$$M = [M_1, M_2, ..., M_n],$$

$$K = Mk,\ \Omega = M\omega, \Phi = M\phi.$$

The integer components in M are the integer indices in (A2). The matrix B can be decomposed in a diagonal matrix D and an off-diagonal matrix O, that is $B = D + O$. Further, we write the solution (A1) under the form

$$\theta(\eta) = 2\frac{\partial^2}{\partial t^2}\log\Theta_n(\eta) = \theta_{lin}(\eta) + \theta_{int}(\eta). \tag{A4}$$

The first term θ_{lin} represents a linear superposition of cnoidal functions

$$\theta_{lin} = \sum_{l=1}^{n}\alpha_l\mathrm{cn}^2[\omega_lt; m_l], \tag{A5}$$

with

$$q = \exp(-\pi\frac{K'}{K}),\ K = K(m) + \int_0^{\pi/2}\frac{\mathrm{d}u}{\sqrt{1\text{-}m\sin^2u}},\ K'(m_1) = K(m),\quad m + m_1 = 1.$$

The second term θ_{int} represents a nonlinear superposition or interaction among cnoidal functions

$$2\frac{\mathrm{d}^2}{\mathrm{d}t^2}\log[1 + \frac{F(t)}{G(t)}] \approx \frac{\beta_k\mathrm{cn}^2(\omega t, m_k)}{1 + \gamma_k\mathrm{cn}^2(\omega t, m_k)}. \tag{A6}$$