MATHEMATICAL MODELLING AND SIMULATION OF A TORSIONAL BAR ACTUATED TRUNK LID MECHANISM FOR A PASSENGER CAR

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ABSTRACT

This paper is concerned with the design and analysis of a rear trunk lid mechanism which is used in sedan passenger cars, and actuated by torsional bars. First, the mechanism actually consisting of a four bar linkage is designed regarding certain kinematic aspects. Afterwards, a mathematical model is developed to determine some requirements crucial to design process such as the equilibrium positions and the opening velocity of, and the required hand force to open and close the trunk lid. By dimensioning mechanism and choosing torsional bar stiffness properly, a trunk lid mechanism is obtained, which meets the demands of the higher first equilibrium angle and significantly small hand forces. Hence, this paper presents a practical and easy methodology in the design and analysis of a trunk lid mechanism for design engineers.

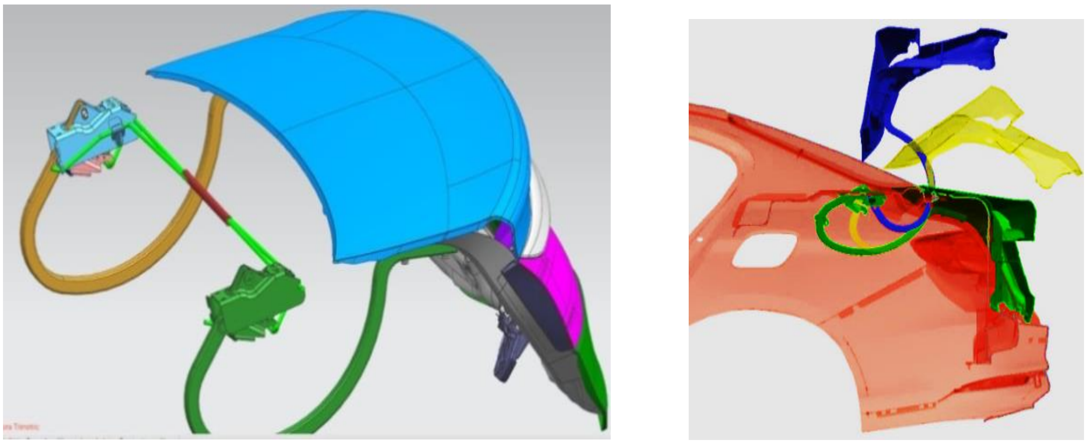
KEYWORDS: *Trunk Lid Mechanism, Torsional Bar Actuated, Four-Bar Linkage*

I. INTRODUCTION

The rear trunk lid mechanisms are used to allow the lid to be lifted up and down. In order to fulfill this simple requirement, there are different types of mechanisms consisted of different structures. Among the various types of these systems, the gas spring and torsional bar actuated mechanisms shown in Figure 1 are commonly used. Both gas spring and torsional bar actuated mechanisms are based on the same kinematic structure, i.e., a four-bar linkage. However, the necessary force to actuate the trunk lid is ensured in different ways. In the gas spring mechanism, there are two pistons to provide the opening force while the torsional bars are used in the other.

These systems have some superiority to each other comparing some properties. For instance, in the torsional bar actuated system, the requested force can be provided exactly in different environmental conditions, whereas the dynamic behavior of gas spring usually deviates from its expected force-displacement characteristic due to the high temperature changing. Another most important difference between these two mechanism is that the torsional bar actuated mechanism has two hinges caused to reduce the effective volume of the baggage while the gas springs mechanism do not use such a structure. The last comparison can be about their cost: The torsional bar actuated system is cheaper than the gas spring mechanism since the last one needs extra devices leading to high costs for manufacture and maintenance.

About the trunk lid mechanisms actuated both gas spring and torsional bars, there are very few researches considering the design parameters and procedures in the relevant literature whereas many patents are exist. Baykus and etc. in [1] have designed and analyzed a luggage opening/closing mechanism for the commercial vehicles such as minibuses and buses. Lee and etc. [2] have proposed an innovative design of an automatic opening door mechanism to improve the existing mechanism for opening the door of a mobile shop vehicle. Zhang and Chen [3] have designed a compliant bistable mechanism to keep the lid's two states and compensate the lid weight, which leads to a system that requires only a small input force to switch between the two states. Duran in [4] has designed a new mechanism for opening hatchback-car baggage door. The new design of author allows the door to open vertically and thus it occupies less space behind the car during the opening. Besides that, about the four-bar linkage there is an intensive literature especially on the kinetic analysis and synthesis for different purposes [5-11].



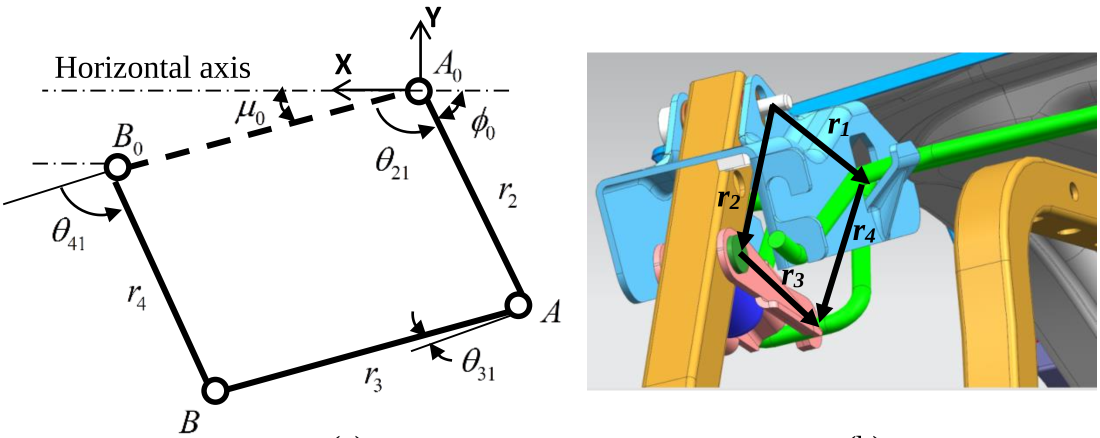
(a) (b)

Figure 1. A torsional bar actuated trunk lid mechanism for sedan cars. (a) Overview, (b) Details.

In this paper, a torsional bar actuated mechanism for trunk lid of a sedan passenger car is designed and analyzed to fulfill the certain requirements: equilibrium positions of the trunk lid and the opening velocity and the necessary hand force to open and close the lid. The kinematics of the four bar linkage is analyzed, and then the quasi-static analysis is carried out analytically. Thus, the equilibrium points of the trunk lid and the necessary hand force are determined as a function of opening angle of the lid. Besides, the dynamic behavior of trunk lid is examined by solving the equation of motion. The simulation results of the mathematical model for a certain trunk lid are illustrated as figures. The authors believe that the design process explained in this paper and the obtained results would be very useful for design engineers in automotive industry.

II. KINEMATIC ANALYSIS OF THE MECHANISM

The trunk lid mechanism the design of which is dealt with in this paper principally consists of a four bar linkage. As seen from Figure 2; the car body, the hook (lid), the hinge and the torsional bar are represented by link 1, link 2, link 3 and link 4, respectively. The schematic of the four-bar linkage is shown in Fig. 2, where the positions of links are described by r_i and θ_I, because the complex numbers method is used in the analysis. The mass moments of inertia of the bars are neglected. In this design, A_0B_0 is selected the reference linkage, φ_0 and ϕ_0 are the initial angles of the A_0B_0 and A_0A linkages respectively.



(a) (b)

Figure 2. The trunk-lid mechanism (a) and its components (b).

In order to investigate the dynamic behavior of a four-bar mechanism, the necessary kinematic relations are derived from the following loop equation in the complex number notation:

$$r_2 e^{i\theta_{21}} + r_3 e^{i\theta_{31}} = r_1 + r_4 e^{i\theta_{41}} \tag{1}$$

By using the conjugate of Eq. (1) the so-called Freudenstein equation is obtained as follow:

$$K_1 \cos\theta_{41} + K_2 \cos(\theta_{41} - \theta_{21}) + K_3 = 0 \tag{2}$$

where

$$K_1 = 2r_1 r_4$$

$$K_2 = -2r_2 r_4 \tag{3}$$

$$K_3 = r_1^2 + r_2^2 - r_3^2 + r_4^2 - 2r_1 r_2 \cos\theta_{21}$$

After the following definitions are introduced in Equation (2)

$$a = K_1 + K_2 \cos\theta_{21}$$

$$b = K_2 \sin\theta_{21} \tag{4}$$

$$c = K_3$$

and it is rearranged we obtain the following:

$$a\cos\theta_{41} + b\sin\theta_{41} + c = 0 \tag{5}$$

Using half angle tangent formulas for θ_{41} is obtained from Equation 5. The angle θ_{31} is also calculated in the same way.

III. KINETIC ANALYSIS OF THE TRUNK LID MECHANISM

In this section, the kinetic analysis of the trunk-lid is treated to determine the hand forces and the angular speed of the lid. Finding this speed is important to predict whether the trunk lid opens smoothly or not. If the lid opens at high speeds, it may hit and injure the user, thus, a detailed kinetic analysis is necessary before designing such a mechanism.

3.1. Quasi Static Analysis of the Mechanism

The external forces and moments acting on the mechanism as the door is in the closed position are: the weight of the lid, the lock force, and the torsional moment of the bars. Since there are two mechanisms at both sides of the lid, if each mechanism is considered individually, weight force and lock force should be divided into two. If two mechanisms are to be regarded as one mechanism, the torque produced by a single bar must be doubled. Here, two mechanisms will be considered as a single mechanism.

In Figure 3, all forces and moments acting on the mechanism are demonstrated. As shown in the figure, the weight of the lid and the lock force will be shifted to the point A in force-moment pairs. The lock force can be taken as zero since it is disappear as soon as the lid starts to open. Also, the hand force is not necessary at the equilibrium positions of the trunk lid. In order to determine the equilibrium positions of the trunk lid, all of these forces and moments must be defined as functions of the mechanism parameters.

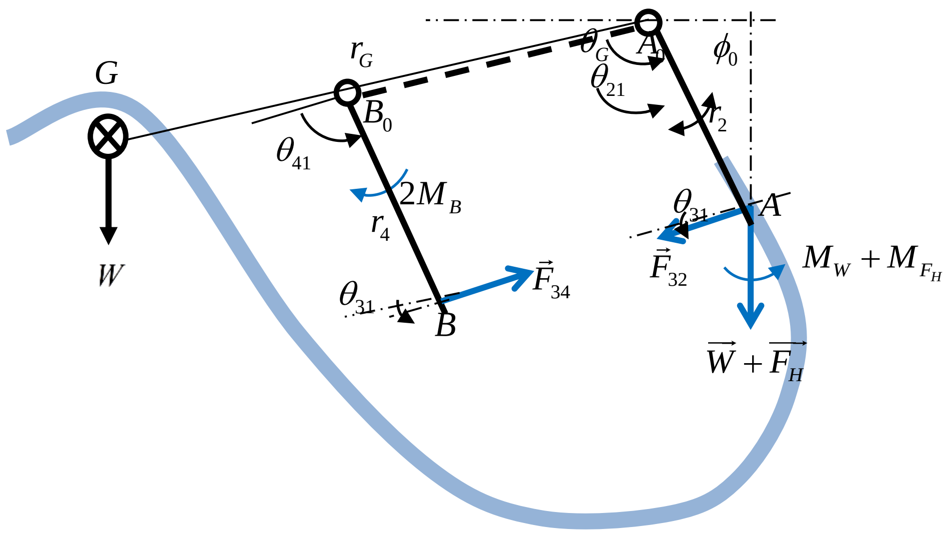


Fig. 3: Forces and moments acting on parts of mechanism

The moment of the weight of trunk lid is:

$$M_W = \left(r_G \cos(\theta_G - \theta_{21}) + r_2 \cos(\phi_0)\right)W \tag{6}$$

where r_G and θ_G are the length and direction angle of the positon vector of center of gravity of the trunk lid, and the ϕ_0 is the initial assembling angle as shown in Figure 2.

M_B is the torsional moment of a single bar and calculated by the formula below:

$$M_B = b_k(\varphi_0 + \beta_0 + \theta_{41}) \tag{7}$$

where φ_0 and β_0 are the initial angles of the mechanism as shown in Figure 2, and b_k is the torsional stiffness of the bar given by

$$b_k = \frac{GI_p}{L} \tag{8}$$

where G is shear modulus of torsional bar's material, L is the operational length of torsion bar and I_p is polar area moment of inertia that equals to $I_p = \pi d^4/32$

The reduced torsional moment is obtained using usual reduction methods in mechanisms:

$$M_{B,red} = \left(\frac{r_2}{r_4}\frac{\sin(\theta_{21} - \theta_{31})}{\sin(\theta_{41} - \theta_{31})}\right)2M_B \tag{9}$$

In the mathematical model the seal forces are also considered since they force the lid to open. These forces apply in five different regions circumferentially along the contour of the lid. The contact of every part of seal with the lid vanishes gradually as the lid is opening. The total moment of the resultants of these distributed seal forces can be expressed as follows:

$$\sum_{i=1}^{5} M_{Fi} = \sum_{i=1}^{5} F_{Fi} x_{Fi} \tag{10}$$

where the F_{Fi} and x_{Fi} are the seal forces and the vertical distances of these forces to the reference rotational point A_0 respectively. In here, this information are not given intentionally.

The total opening moment in the mechanism is obtained as:

$$M_T = M_{B,red} + \sum_{i=1}^{5} M_{Fi} \tag{11}$$

In order to find the necessary hand force, the quasi static condition of every bar is considered as shown in Figure 4. In the mechanism, the total moment about B_0 must equal to zero due to the equilibrium of link 4. Thus, the F_{34} that is also equal to the minus of the internal force F_{23} can be obtained as below:

$$F_{34} = \frac{2M_B}{r_4 \sin(\theta_{41} - \theta_{31})} \tag{12}$$

The moment of the internal force F_{32} can be written as follows:

$$M_{F_{32}} = \left|\overrightarrow{A_0 A} x \overrightarrow{F_{32}}\right| = r_2 \sin(\theta_{21} - \theta_{31}) F_{32} \tag{13}$$

The moment of lock force is:

$$M_{F_H} = \left(r_H \cos(\theta_H - \theta_{21}) + r_2 \cos(\phi_0)\right) F_H \tag{14}$$

Similarly, from the equilibrium of the second bar, the total moment about A_0 must be equal to zero, hence, the hand force F_H is obtained as following equation:

$$F_H = \frac{r_2 \sin(\theta_{21} - \theta_{31})}{r_4 \sin(\theta_{41} - \theta_{31})} \frac{2M_B}{r_H \cos(\theta_H - \theta_{21})} - \frac{r_G \cos(\theta_G - \theta_{21})}{r_H \cos(\theta_H - \theta_{21})} W \tag{15}$$

3.2. Equation of Motion of the Trunk Lid

The equation of motion of trunk-lid is derived to analyze the dynamical behavior of the trunk lid. Equation 10 represents the rotational motion of trunk-lid with respect to y axis. It will be solved by numerically according to the fourth order Runge-Kutta method.

Equation of motion is found as follows:

$$M_T - M_W - k_1 \dot{\theta}_{21} - k_2 \dot{\theta}_{21}^2 - I_K \ddot{\theta}_{21} = 0 \tag{16}$$

where k_1 and k_2 are the equivalent viscous damping coefficient and the drag force coefficients, respectively. I_K is the mass moment of inertia of the lid about rotation axis of the trunk-lid. Some design parameters used in the simulations are given in Table 1.

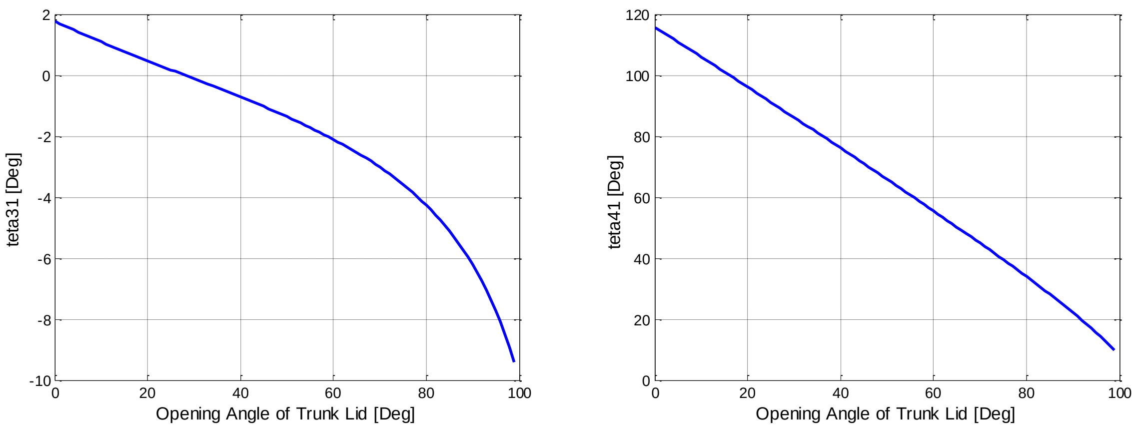
Table 1. Designed parameters and the other necessary quantities for a sample trunk lid for sedan passenger vehicle

Dimensions of the designed four-bar mechanism		The initial conditions of mechanism and values the trunk lid			
r_1	80 mm	r_G	420,8 mm	β_0	10,8°
r_2	75 mm	r_H	377,9 mm	W	131,5 N
r_3	85 mm	θ_G	23,7°	I_K	2,1355 kgm^2
r_4	75 mm	θ_H	37,6°	G	75000 N/mm^2
d	6 mm	φ_0	20°	k_1	12,9
L	932 mm	ϕ_0	40°	k_2	10,5

IV. SIMULATION RESULTS

In this part, the numerical results associated with the equilibrium positions of trunk lid, the necessary hand forces for opening/closing the door and the angular velocity of the lid obtained by computer simulations will be given.

In Figure 4, the kinematic quantities are demonstrated versus opening angle of trunk lid. Figure 4a and 4b show the variation of θ_{31} and θ_{41} for each point of opening angle.



(a) (b)

Figure 4: Variation θ_{31} (a) and θ_{41} (b) versus the opening angle of the lid.

Figure 5 helps us find the equilibrium positions of the trunk lid. The first equilibrium position is 9.2° while the second one is 83.25°. These results are obtained from the intersection points of the two curves. The real opening angle of trunk lid about 92°. It can be concluded from this results that the trunk lid can open up to 9.2° without any hand force. This angle can be bigger if the parameters of mechanism and torsional bar diameter adjust differently.

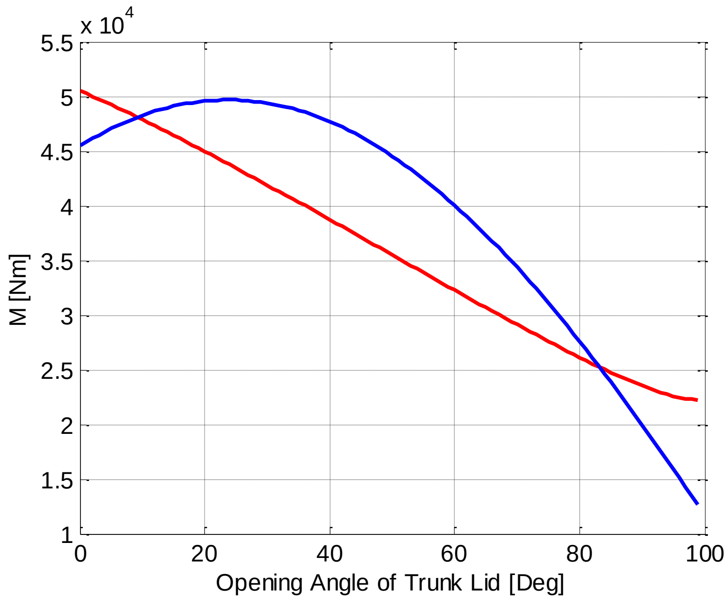


Figure 5: Variation of the moments of hand force (red line) and torsion moment (blue line) with opening angle of the lid. Intersection points correspond the equilibrium positions.

In Figure 6, the necessary force to actuate the trunk lid versus opening position is illustrated. It can be seen from this figure that the hand force is zero in the equilibrium positions since the torsional moment is balance the weight moment. In positions where the hand force is positive, the weight moment of trunk lid cannot be reversed and therefore the lid must be opened with the help of hand force and so, the maximum opening force is about 15 N. For the closing the trunk lid 12,6 N force in the negative direction is necessary. These hand forces are quite small comparing the maximum allowed hand force that is about 40 N.

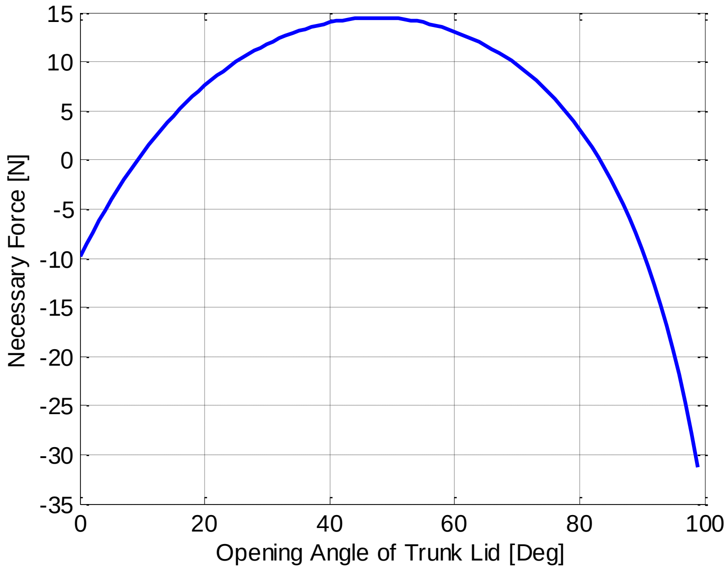


Figure 6: Necessary force versus opening angle of the trunk lid.

Figure 7 shows the dynamic response of the trunk lid after opening immediately. The first equilibrium state occurs at about 9.2 as expected. The total time spent to be stagnant is almost two second in the specified conditions. It can be adjust by changing the design parameters.

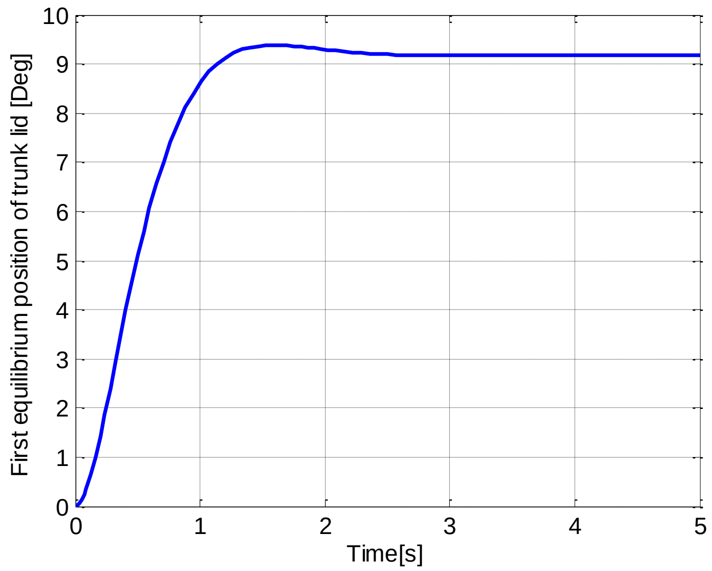


Figure 7: Dynamic response of the lid immediately after opening. First equilibrium state occurs at about 9.2°

In the Figure 8, the time history of the angular velocity of the trunk lid about the first equilibrium position is demonstrated. It should be noted that this speed must be check if it is under the maximum allowed opening velocity of trunk lid or not. The maximum angular velocity of trunk lid is about 0.22 rad/sn and this value is under the requested velocity which is 0.5 rad/s.

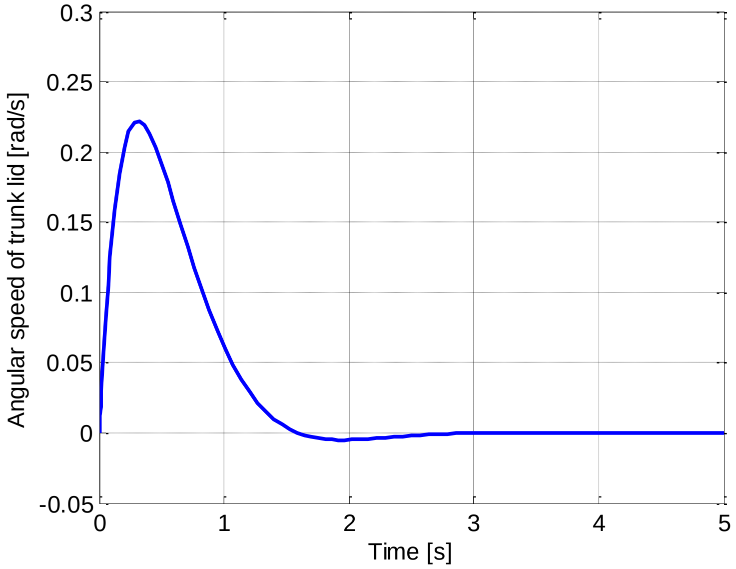


Figure 8: Time history of the angular velocity of trunk lid until it reaches the first equilibrium position.

V. Conclusion

In this study, a rear trunk lid mechanism actuated by torsional bars is mathematically modeled and simulated. A four-bar mechanism is designed regarding certain kinematic aspects providing some requirements crucial to design process such as the equilibrium positions and the opening velocity of, and the required hand force to open and close the trunk lid. By dimensioning mechanism and choosing torsional bar stiffness properly, a trunk lid mechanism allowing the lid to open automatically up to a certain angle is obtained, which fulfills the demands of the higher first equilibrium angle and significantly small hand forces. Thus, the results of this paper are so important for the design process of a trunk lid mechanism for sedan passenger cars.

VI. Future work

More future works can be done about design of the mechanism for providing higher first equilibrium positions of trunk lid by optimising the dimensions of linkages and other parameters. Also, it is possible to allow the trunk lid to open itself by using bigger torsional stiffness of bars. However, in this time a damper should be added for decelerating the angular velocity of trunk lid; otherwise it causes the driver to be injured and to rear glass damage.

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