Research Article
Journal of Smart Systems Research (JOINSSR) 1(1), 1-10, 2020
Recieved: 10-Oct-2020 Accepted: 19-Oct-2020

SAKARYA UNIVERSITY
OF APPLIED SCIENCES

Fuzzy Logic Control of Vibrations due to Interaction One DOF Vehicle Suspension and Flexible Structure with Tuned Mass Damper

Mehmet Akif KOÇ[1*]

[1]Sakarya Applied Sciences University, Technology Faculty, Mechatronics Engineering

ABSTRACT

In this study, the one DOF quarter car suspension system moving on flexible structure likewise Euler-Bernoulli bridge beam with the simple supported boundary condition is studied. The vibrations due to the interaction between moving car and bridge beam has been reduced with the tuned mass damper (TMD) attached to beam and active control of linear actuator placed to suspension system. Then, a fuzzy logic control algorithm is designed for inspection forces transmitted to vehicle body. For the numerically analysis, three different models have been presented to compare performance's fuzzy logic controller designed in this study. Consequently, it is understood that most effective technique to suppress car body and bridge vibration is method in which TMD and fuzzy logic controller are used together.

Keywords: fuzzy logic control, one DOF car, active control, TMD

Tek Serbestlik Dereceli Süspansiyon ile Ayarlı Kütle Damperli Esnek Yapı Etkileşiminden Kaynaklı Titreşimlerin Bulanık Mantık Kontrolü

ÖZ

Bu çalışmada basit mesnetli Euler-Bernoulli kirişi gibi esnek bir yapı üzerinde hareket eden tek serbestlik dereceli çeyrek araç süspansiyon sistemi çalışılmıştır. Hareketli araç ve köprü kirişi etkileşiminden kaynaklı titreşimler kirişe takılan ayarlı kütle damperi (AKD) ve süspansiyon sistemine takılan lineer aktuatörün aktif kontrolü ile azaltılmıştır. Daha sonra araç gövdesine iletilen kuvvetleri kontrol etmek için bir bulanık mantık kontrolcüsü tasarlanmıştır. Çalışmada kullanılan bulanık mantık kontrolcüsünün performansını test etmek için üç farklı model sunulmuştur. Sonuç olarak araç gövdesi ve köprü titreşimlerini sönümlemede en etkini yöntemin bulanık mantık kontrolcüsü ve AKD'nin beraber kullanıldığı model olduğu görülmüştür.

Anahtar Kelimeler: bulanık mantık kontrol, tek serbestlik dereceli araç, aktif kontrol, AKD.

1 Introduction

The improvement and dissemination of transportation causes a leading role in promoting economic progress, prosperity, and cultural contact in countries. But, as the operating speeds of the transportation increase, the vibrations in the vehicle components, which negatively affect passenger comfort and transportation safety, increase considerably as well. Therefore, reducing and controlling the vibrations in the train components is very significant for passenger comfort, rail holding and riding safety [1], [2]. The moving load problems are commonly used in the field of engineering. One of the most important

* Sorumlu Yazar e-mail: makoc@subu.edu.tr

area moving load used is the military applications and the studies about this area generally focused interaction between moving projectile and barrel. The studies in which moving projectile modelled as moving mass considering Coriolis, centripetal and inertial effects are given by [3]–[6].

The interaction between moving cars and flexible structures is very important problem in engineering because of a heavy vehicle such as high-speed train (HST) moving on the flexible structures like bridge beam causes excessive vibrations due to train bridge interaction (TBI). Because in such a system, the train and the bridge are designed as two different subsystems, and the dynamic response of these two different subsystems is affected by each other through the movement of the train on the bridge [7], [8]. With the developments in computer capacity and appearance of high-capacity computers, studies were launched on complicated simulations which take into consideration the bridge and train dynamics together [9]–[11].

Vibration suppression on TBI models can be analyzed by new engineering techniques with much little cost or results. For example, a train system exposed rail irregularity has primary and secondary suspension system with self-control ability to develop performance of the passenger comfort. Besides, the same technique would be used to approach to a train model with minimum impacts on track and bridges [12]. Besides, A moving train interact with the flexible structure like bridge beam, a passive and active vibration control algorithm has been used to develop its creditableness and strength[13], [14]. The passive suspension systems used on the train have some advantage because of the it is not costly set up process and time consuming. But, increasing of the speed of trains, passive suspension systems are not adequate to suppress vibrations to transmitted to vehicle body. Consequently semi-active suspension systems have been used damping the vibrations transmitted to vehicle body[15]–[18]. A feedback signal is generated from train components to control algorithm. Therefore, effective designed a semi-active suspension system has very impact upon to reduce vibrations.

Artificially intelligent techniques (AIT) are widely used in the engineering applications. AIT has advantage for the reason of has learning and decision ability. Therefore, many papers in literature works has been employed AIT [19], [20]. To reduce vibration due to vehicle road or train rail interaction fuzzy logic control algorithms are used in many studies. The comparation fuzzy logic controller with the PID one in term of quarter car vibration reducing, some paper has been studied in the literature [21], [22]. The results show that the performance of the fuzzy logic controller better and PID controller. Acceleration of the vehicle body has been reduced with the using of thee fuzzy logic controller.

In this study, an active suspension system with fuzzy logic controller has been modelled to reduce excessive vibrations occurred on the moving car body and flexible bridge beam. Firstly, the mathematical model of the moving car and Euler-Bernoulli bridge beam is presented. Then, the fist two natural frequency and mode shapes of the bridge beam are calculated. The critical velocity of the moving car equal to bridge natural frequency are determined with formulation presented in this study. The effect of the TMD and active suspension control upon excessive vibration which occur on the bridge beam is studied in detail. With the proposed the method given in this study, the interaction between flexible structures and moving vehicles is analyzed easily without the need for costly and time-consuming experimental studies.

2 Theoretical Analysis

Figure 1 shows a vehicle bridge interaction model with TMD. The vehicle moves from the left side of the bridge beam to right side with constant velocity v. The vehicle model given by Figure1 has been modelled one DOF and consist of vehicle body, suspension spring and damping elements. On the other hand, the suspension system has a linear actuator which controlled with a fuzzy logic controller designed in this study. The parameters m, k and c represent car body, suspension spring and damping coefficients respectively. Besides, the parameter y(t) is the time dependent vertical displacement of vehicle body. The vehicle location on the beam according to left reference point of the beam is defined by time dependent parameter $x=vt$. The actuator time dependent forces transmitted to vehicle body and flexible bridge beam is determined by parameter $U(t)$.

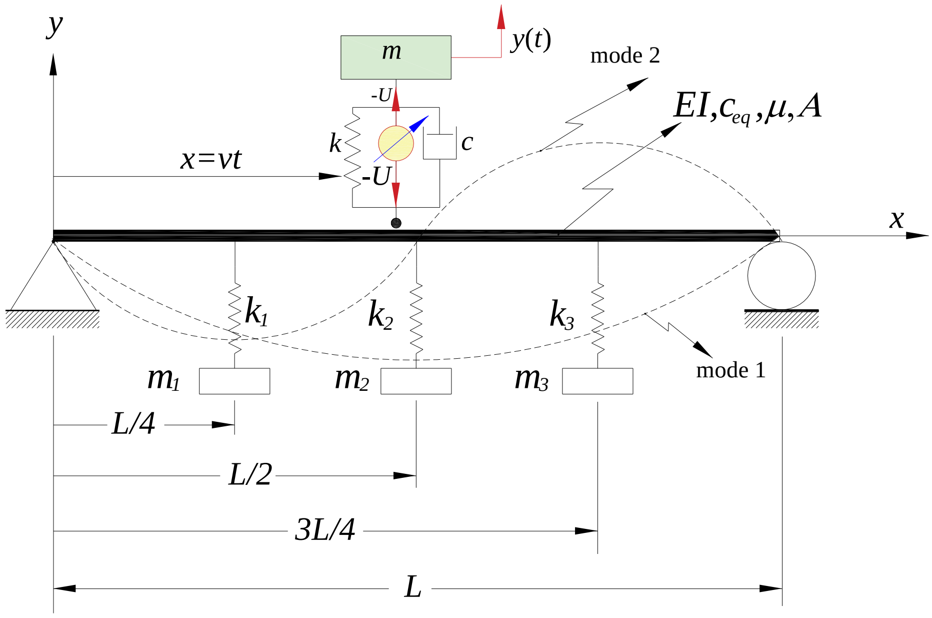

Figure 1: Physical model of the single DOF quarter car suspension, bridge and TMD used in this study.

The flexible bridge beam has been modelled as according to Euler-Bernoulli beam with simple supported boundary conditions as shown in Figure 1. The parameters EI, c_{eq}, μ, A represent bridge rigidity, equivalent damping, per unit length mass and bridge beam cross-section area respectively. The length of the bridge beam is defined by parameter L. As shown in Figure 1, three different vibration absorbers called in this study "Tuned Mass Damper" attached to Euler-Bernoulli bridge beam. The parameter $w_b(x,t)$ is the vertical deflection of the bridge beam at any position x and at any time t. The parameter m_i (i=1,2,3) represents TMD's mass and k_i(i=1,2,3) is the TMD's spring coefficient. The TMDs are placed on quarter, half and $3L/4$ bridge length because of the peak points of the first and second modes equal to certain location at quarter, half and $3L/4$.

As shown in Figure 1 , for the simple supported Euler-Bernoulli beam, the natural frequency of the jth vibration mode is given by [23]

$$w_j^2 = \frac{j^4 p^4 EI}{mL^4} \ (rad/s) \tag{1}$$

In Equation (1), the parameter ω_i is the circular natural frequency of the bridge beam (rad/s), parameter j represents number of the mode, the parameter E represents the elastic modulus (N/m^2) of the beam, I represent the moment of inertia (m^4) of the cross-section of the beam. Using Eq. (1) the natural frequency of a simply supported beam in Hz is expressed as follows:

$$f_j = \frac{\omega_j}{2\pi} = \frac{j^2\pi}{2L^2}\left(\frac{EI}{\mu}\right)^{1/2} (Hz.), \tag{2}$$

The ratio of the excitation frequency $\omega=\pi v L^{-1}$ *of* a vehicle on the bridge to the natural frequency ω_i of the bridge (i=1, 2, 3, 4) is called the speed parameter and is expressed as in Eq. (3). When $\omega = \omega_i$, resonance occurs, which can be destructive for the bridge.

$$a = \frac{w}{w_i}, \tag{3}$$

When Eq. (3) and the excitation frequency of a moving vehicle on the bridge are rearranged, the following is obtained:

$$\alpha = \frac{\omega}{\omega_j} = \frac{\omega}{2\pi f_j} = \frac{vL}{j^2\pi}\left(\frac{\mu}{EI}\right)^{1/2} = \frac{v}{v_{cr}}, \tag{4}$$

Using Eq. (21), v_{cr} is expressed as follows:

$$v_{(cr)j} = 2f_jL = \frac{\pi}{L}\left(\frac{\mu}{EI}\right)^{1/2}, \tag{5}$$

Using the parameters given in Table 1 for the Euler-Bernoulli bridge beam for the used in this analyses, mode shapes of the beam have been plotted as shown in Figure 2. Then, Frequency of the first two modes of vibration of the flexible structure are f_1=0.2108 Hz, f_2=0.8432 Hz. Consequently, the critical vehicle velocity corresponding to these frequencies are v_{cr1}=42.15 m/s, v_{cr2}=168.62 m/s.

Table 1. Properties of the vehicle and bridge.

Bridge		**Vehicle parameters**			
L **(m)**	100	***m*** **(kg)**	2172	c_2 **(Ns/m)**	10000
E **(Gpa)**	207	***k*** **(kg)**	85439.6	m_1 **(Ns/m)**	10000
I **(m⁴)**	0.174	***c*** **(kg)**	2219.6	m_2 **(Ns/m)**	10000
μ **(kg/m)**	20000	k_1 **(N/m)**	280401.59	m_3 **(Ns/m)**	10000
c_{eq} **(Ns/m)**	1750	k_2 **(N/m)**	17525.08	k_3 **(N/m)**	280401.59

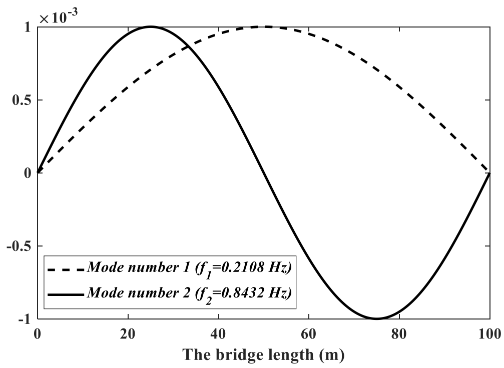

Figure 2: The mode shapes of the flexible bridge beam used in this study.

In this study, only first and second mode shapes have been used for the numerically analysis because of the bridge beam is a large and big structure. If a structure has a large and big body, this structure vibrates effect of the low frequency vibration modes and it is adequate considering only low vibration modes to calculation of the dynamic response of the system. To demonstrate effect of the mode number upon solution accuracy, Figure 3 shows bridge beam mid-point displacement considering three different mode numbers (n=2,3,8). As shown in figure, increasing of the mode number used calculation has no effect upon bridge dynamic by reason of structure size.

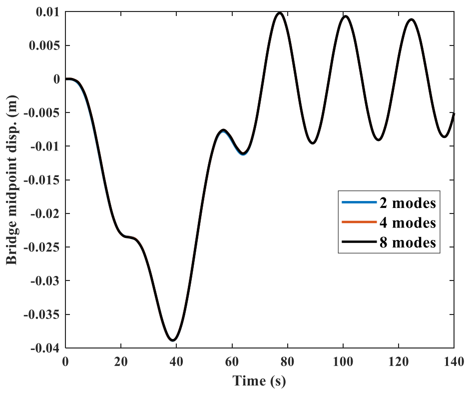

Figure 3: The effect of the mode number upon solution accuracy.

In formulation of the bridge, TMD and single DOF quarter car analyses the following assumptions will be adopted:

- The bridge beam is modelled as according to Euler-Bernoulli beam with simple supported boundary conditions.
- The vehicle has been modelled as single DOF quarter car suspension.
- The velocity of the car is considered as constant and any acceleration and deceleration effect are excluded from the scope of the study.

With these assumptions the kinetic energy of the system given by Figure 1 has been derived by Equation (1):

$$E_k = \frac{1}{2}\left\{\int_0^L \mu\left[\dot{w}_b^2(x,t)\right]dx + m\dot{y}^2(t) + m_1\dot{y}_1^2(t) + m_2\dot{y}_2^2(t) + m_3\dot{y}_3^2(t)\right\}, \qquad (6)$$

On the other hand, the potential energy of the system given by Figure 1 has been stated as follows:

$$E_p = \frac{1}{2}\left\{\begin{array}{l}\int_0^L EIw_b''^2(x,t)dx + k_1\left[y_1(t) - w_b(L/4,t)\right]^2 \\ +k_2\left[y_2(t) - w_b(L/2,t)\right]^2 + k_3\left[y_3(t) - w_b(3L/4,t)\right]^2 + k\left[y(t) - w_b(\xi(t),t)\right]^2 H(x,\xi(t))\end{array}\right\} \qquad (7)$$

The parameter *H(x)* is the Heaviside function. For any point *x* on the bridge beam and any time t, transverse deformation is written by Galerkin function as follows:

$$w_b(x,t) = \sum_{i=1}^{n}\varphi_i(x)\eta_{bi}(t), \qquad (8)$$

The parameter η_{bi} and φ_i represent i^{th} generalized modal coordinate and mode shape function respectively. The orthogonality of the modes can be expressed as:

$$\begin{array}{l}\int_0^L \mu\varphi_i(x)\varphi_j(x)dx = N_i\delta_{ij}, \\ \int_0^L EI\varphi_i''(x)\varphi_j''(x)dx = \Pi_i\delta_{ij},\end{array} \qquad (9)$$

In Eq. (4) δ_{ij} (i, j=1,2,…,n) are the Kronecker delta function. For the vehicle- bridge system the Rayleigh's dissipation function can be expressed as below:

$$D = \frac{1}{2}\left\{\int_0^L \left\{c_{eq}\dot{w}_b^2(x,t)dx + c[\dot{y}(t) - \dot{w}_b(\xi(t),t)]^2 H(x - \xi(t))\right\}\right. \qquad (10)$$

In Eq. (5) c_{eq} is the equivalent damping coefficient of bridge beam. By using the Galerkin function given by equation (8), the equations of motion of the system are obtained by Lagrange equation g,iven by Equation (10).

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial L}{\partial\dot{y}(t)}\right) - \frac{\partial L}{\partial y(t)} + \frac{\partial D}{\partial\dot{y}(t)} = U \ , \qquad (11)$$

Accordingly, the equation of motion for car body vertical vibrations can be obtained by:

$$\ddot{y} = \left\{-c[\dot{y}(t) - \dot{w}_b(x,t)] - k[y(t) - w_b(x,t) + U]\right\}m^{-1} \qquad (12)$$

Then, Lagrange equation for the TMDs is written as follows:

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial L}{\partial \dot{y}_i(t)}\right)-\frac{\partial L}{\partial y_i(t)}+\frac{\partial D}{\partial \dot{y}_i(t)}=U \;,\left(i=1,2,3\right) \tag{13}$$

The equation of the motion of the first, second and third vibration absorbers have been developed as follows:

$$\ddot{y}_1=\left\{-k_1[y(t)-w_b(L/4,t)]\right\}m_1^{-1} \tag{14a}$$

$$\ddot{y}_2=\left\{-k_2[y(t)-w_b(L/2,t)]\right\}m_2^{-1} \tag{14b}$$

$$\ddot{y}_3=\left\{-k_3[y(t)-w_b(3L/4,t)]\right\}m_3^{-1} \tag{14c}$$

the equation of motion representing the bridge dynamics is written as follows:

$$\begin{aligned} &N_i\ddot{\eta}_i(t)+c_{eq}\varphi_i^2(x)\dot{\eta}_i(t)+\Pi_i\eta_i(t)+\Lambda\varphi_i\left(\xi_1(t)\right)\left\{f_c+U+c\left[\dot{w}_b\left(\xi(t),t\right)\Lambda_1-\dot{y}(t)\right]+k\left[w\left(\xi(t),t\right)\Lambda-y(t)\right]\right\}\\ &+k_1\left[w_b(L/4,t)-y_1(t)\right]+k_2\left[w_b(L/2,t)-y_2(t)\right]+k_3\left[w_b(3L/4,t)-y_3(t)\right]=0 \quad i=1,2,3,4. \end{aligned} \tag{15}$$

where Λ is:

$$\Lambda=\begin{cases}1, & for\ 0\le t<t_1\\ 0 & elsewhere,\end{cases} \tag{16}$$

3 Designing of the Fuzzy Logic Controller

In this study, to control of the linear actuator given by Figure 1, a fuzzy logic controller has been designed as shown in Figure. In this context, the output of the controller is defined as $r=y(t)-w_b(x,t)$ which equals to difference between car body vertical displacement and bridge beam vertical deflection at the contact point of the car upon beam. On the other hand, as the reference input is defined the parameter R and which equals to zero. The error value of the controller is calculated by the parameter e with calculation of the formulation $e=R-r$. Figure 4 shows fuzzy logic controller used in this study for the vibration control of moving car and flexible beam. As shown in figure, the input parameter is defined by symbol R, on the other hand, the output parameter is obtained by r. Then, the error value is calculated by difference with the input and output. After derivation of the error, this parameter entered the fuzzy controller.

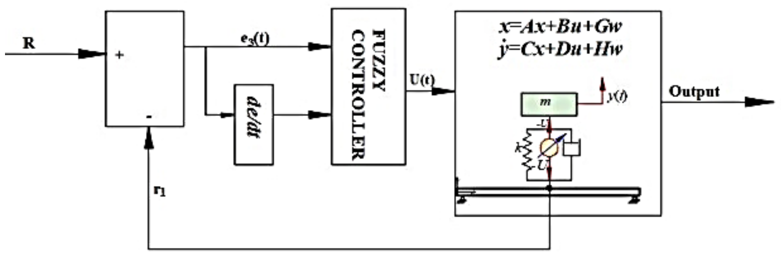

Figure 4: Fuzzy logic controller used in this study.

Figure 5 shows fuzzy controller schemas for moving suspension car. In the Figures 5, input and output parameters of the fuzzy logic controller used in this paper are seen clearly. The parameters error and deviation of the error are defined as inputs for controller.

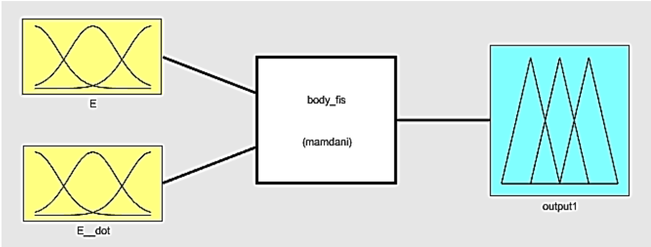

Figure 5: The input and output parameters for fuzzy logic controller used in this study.

Figure 6 show s membership functions of the fuzzy logic controller used in this study. As shown in figures, the input parameter error has been represented by five numbers membership functions. On the other hand, the input parameter deviation of the error has been presented by three membership functions. The output parameter actuator force is represented by nine number member ship functions. The all membership functions are selected as triangular geometry.

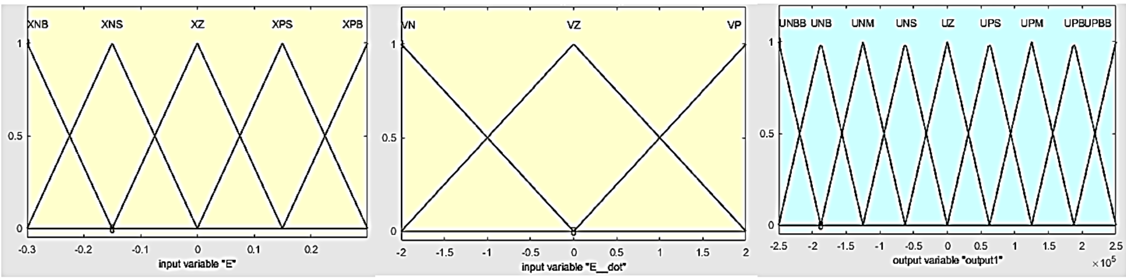

Figure 6: The membership functions of input and output parameters.

Before starting analysis, it must be defined fuzzy rules which indicate relationships between input and output parameters. As mentioned earlier, the input parameters are defined by error and derivatives of error. The error parameter has been converted to fuzzy by the five membership functions. Likewise, the parameter derivation of the error converted to fuzzy with the three membership functions. The fuzzy rule between these input and output parameter given by Table 2. Figure 7 shows output surface according to input parameters.

Table 2. Fuzzy laws used in controller.

U_i		$\frac{de_i(t)}{dt}$		
		VN	**VZ**	**VP**
$e_i(t)$	**XNB**	UNB	UNM	UNS
	XNS	UNM	UNS	UZ
	XZ	UNS	UZ	UPS
	XPS	UZ	UPS	UPM
	XPB	UPS	UPM	UPB

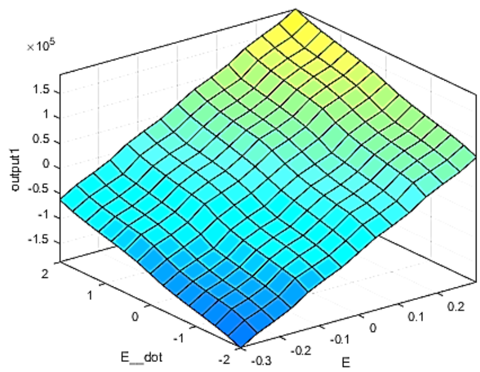

Figure 7: The output surface according to input parameters.

4 Results

For the numerically analysis moving suspended car and flexible structure's parameters given by Table 1 are used. The moving car constant velocity is determined v=90 m/s. Figure 8 shows different cases for the numerically analysis. For example, Figure 8a presents a moving car body with the simple supported Euler-Bernoulli beam without TMD and fuzzy logic controller. This model is the simplest model used in the study and is used to compare the performance of other models. In the Figure 8b, a linear actuator has been attached to moving car's suspension system and introduced a simple supported Euler-Bernoulli beam with three TMDs. First TMD has been placed on bridge at location L/4 and it is called in this study "*Left L/4*". The second one is placed to L/2 bridge beam and it is defined as "*Mid L/2*". The last one is the mounted on the bridge 3L/4 location and it is mentioned that as "*Right 3L/4*". Otherwise, Figure 8c shows that a simple supported Euler-Bernoulli beam with three TMDs similar to Figure 8b and moving suspended car without linear actuator and fuzzy logic controller.

Figure 9a-b show displacement and acceleration of the car body for the different cases given by Figure 8a-c respectively. As shown in Figure 9a, the maximum vibration of the car body is occurred in the without TMD given by Figure 8a. Besides, minimum vibration of the car body has been determined with TMD and fuzzy controller case as shown in Figure 8b. The vertical displacement values of the car body are determined 1.02, 2,73 and 4.45 mm respectively for the cases with TMD and fuzzy controller, with TMD and without TMD. On the other hand, the maximum vertical accelerations of the car body are 0.01378, 0.075, 0.1595 m/s^2 for the cases with TMD and fuzzy controller, with TMD and without TMD respectively. According to these results it seems clearly that best effective method to reduce vibration due to the fact that moving car and flexible structure is the with TMD and fuzzy controller combined algorithm because of the TMD and fuzzy controller case has two vibration damping system. In the TMD and fuzzy controller case, there is external force transmitted to moving car body to active control vibration in addition to TMD.

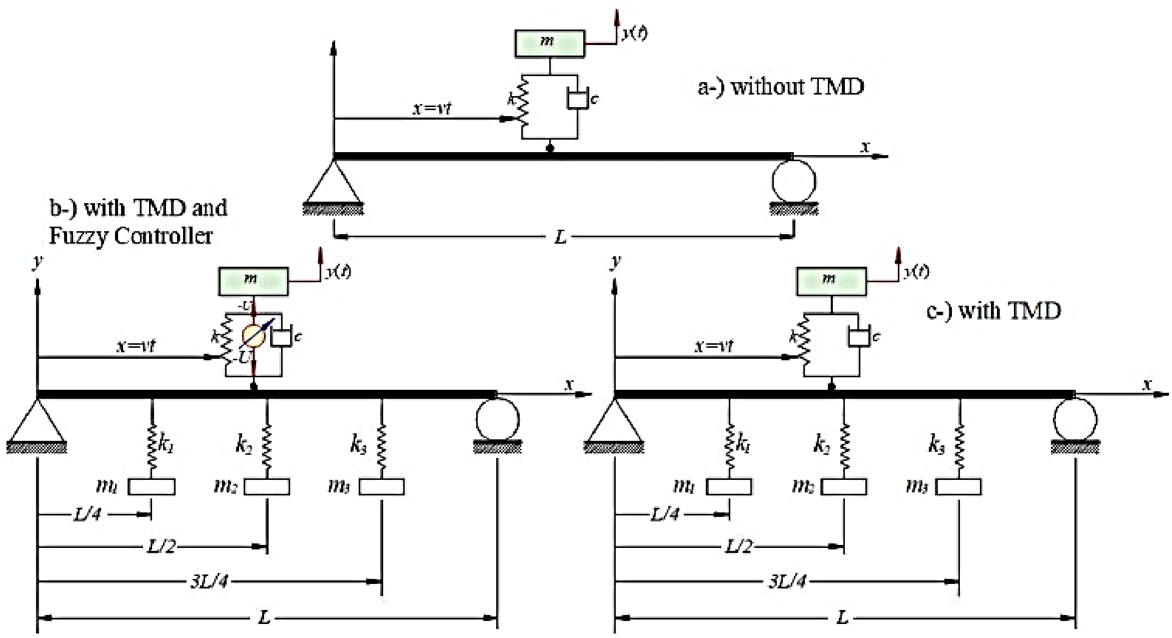

Figure 8: Different cases used in this study for the numerical analysis.

Figure 10a-b shows bridge midpoint transverse displacement and vertical displacements of the vibration absorbers attached to Euler-Bernoulli beam as shown in Figure 8b-c. As shown in Figure 10a, minimum bridge mid-point vertical displacement is determined in TMD and fuzzy controller case. The midpoint vertical displacements are defined by 5.7, 6.18 and 10.4 mm for the TMD and fuzzy controller, with TMD and without TMD respectively. Moreover, Figure 10b shows vertical displacements of the TMDs given by Figure 8b-c for the case TMD and fuzzy controller as shown in Figure 8b. Due to the TMDs used in this study has only mass and spring element, the vibration amplitude of the waves is not damped.

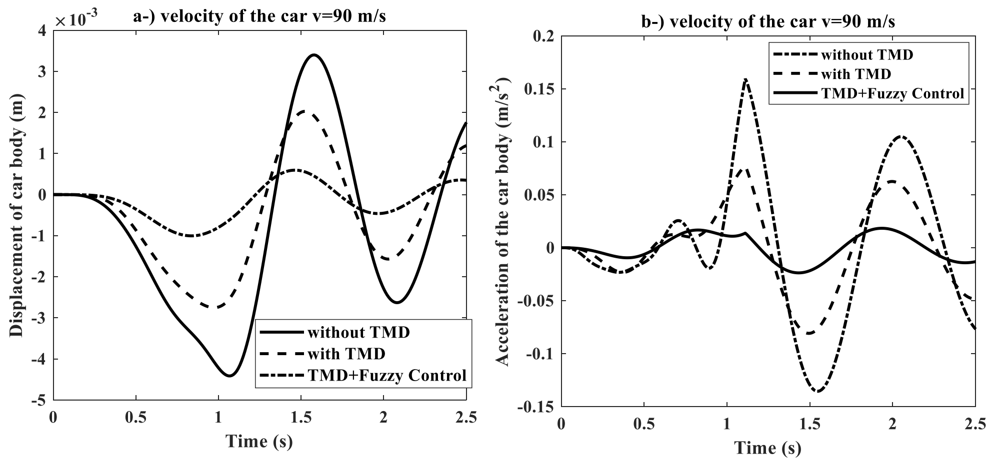

Figure 9: The vertical displacement and acceleration of the moving car.

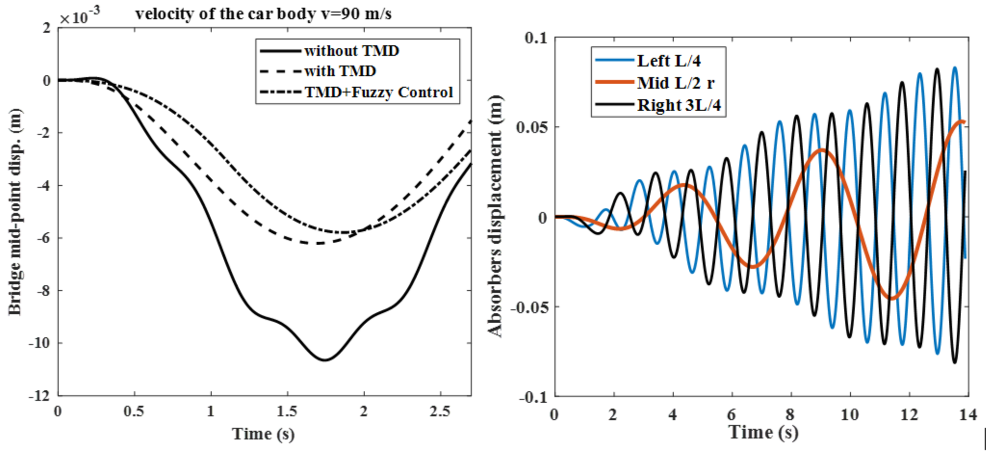

Figure 10: The bridge midpoint transverse displacement and TMD's vertical displacement.

5 Conclusions

In this study, it is demonstrated that fuzzy logic controller is verry effective control algorithm to suppress vibration caused by interaction between moving car and simple supported boundary condition Euler-Bernoulli beam. Moreover, the idea that placed vibration absorbers TMD to first and second mode's peak points is important technique in terms of reducing vibration due to moving car and flexible structures. Using mathematical model proposed in this study, someone can develop easily more complicated model considering multiple vehicle axle, vehicle body, passenger seats, road roughness. So, more detail analysis can be obtained without the need for time consuming and costly experimental works.

References

[1] J. Han, Y. Hayashi, P. Jia, and Q. Yuan, "Economic Effect of High-Speed Rail: Empirical Analysis of Shinkansen's Impact on Industrial Location," *J. Transp. Eng.*, vol. 138, no. 12, pp. 1551–1557, Dec. 2012.

[2] X. Jin, "A measurement and evaluation method for wheel-rail contact forces and axle stresses of high-speed train," *Measurement*, vol. 149, p. 106983, Jan. 2020.

[3] İ. Esen and M. A. Koç, "Optimization of a passive vibration absorber for a barrel using the genetic algorithm," *Expert Syst. Appl.*, vol. 42, no. 2, pp. 894–905, 2015.

[4] I. Esen and M. A. Koç, "Dynamic response of a 120 mm smoothbore tank barrel during horizontal and inclined firing positions," *Lat. Am. J. Solids Struct.*, vol. 12, no. 8, 2015.

[5] M. A. Koç, İ. Esen, and Y. Çay, "Tip deflection determination of a barrel for the effect of an accelerating projectile before firing using finite element and artificial neural network combined algorithm," *Lat. Am. J. Solids Struct.*, vol. 13, no. 10, pp. 1968–1995, 2016.

[6] İ. Esen and M. A. Koç, "35 mm Uçaksavar Topu Namlusu için Titreşim Absorberi Tasarımı ve Genetik Algoritma ile Optimizasyonu," in *Otomatik Kontrol Ulusal Toplantısı-TOK 2013*, 2013, pp. 513–518.

[7] M.-K. Song, H.-C. Noh, and C.-K. Choi, "A new three-dimensional finite element analysis model of high-speed train–bridge interactions," *Eng. Struct.*, vol. 25, no. 13, pp. 1611–1626, 2003.

[8] J. D. Yau, M. D. Martínez-Rodrigo, and A. Doménech, "An equivalent additional damping approach to assess vehicle-bridge interaction for train-induced vibration of short-span railway bridges," *Eng. Struct.*, vol. 188, no. January, pp. 469–479, 2019.

[9] K. Youcef, T. Sabiha, D. El Mostafa, D. Ali, and M. Bachir, "Dynamic analysis of train-bridge system and riding comfort of trains," *J. Mech. Sci. Technol.*, vol. 27, no. 4, pp. 951–962, 2013.

[10] B. Biondi, G. Muscolino, and A. Sofi, "A substructure approach for the dynamic analysis of train-track-bridge system," *Comput. Struct.*, vol. 83, no. 28-30 SPEC. ISS., pp. 2271–2281, 2005.

[11] M. Majka and M. Hartnett, "Effects of speed , load and damping on the dynamic response of railway bridges and vehicles," *Comput. Struct.*, vol. 86, pp. 556–572, 2008.

[12] M. Valášek, W. Kortüm, Z. Šika, L. Magdolen, and O. Vaculín, "Development of semi-active road-friendly truck suspensions," *Control Eng. Pract.*, vol. 6, no. 6, pp. 735–744, 1998.

[13] J. F. Wang, C. C. Lin, and B. L. Chen, "Vibration suppression for high-speed railway bridges using tuned mass dampers," *Int. J. Solids Struct.*, vol. 40, no. 2, pp. 465–491, 2003.

[14] Z. Y. Gao, B. Tian, D. P. Wu, and Y. S. Chang, "Study on semi-active control of running stability in the high-speed train under unsteady aerodynamic loads and track excitation," *Veh. Syst. Dyn.*, pp. 1–14, Sep. 2019.

[15] M. J. Thoresson, P. E. Uys, P. S. Els, and J. A. Snyman, "Efficient optimisation of a vehicle suspension system, using a gradient-based approximation method, Part 1: Mathematical modelling," *Math. Comput. Model.*, vol. 50, no. 9–10, pp. 1421–1436, 2009.

[16] P. E. Uys, P. S. Els, and M. Thoresson, "Suspension settings for optimal ride comfort of off-road vehicles travelling on roads with different roughness and speeds," *J. Terramechanics*, vol. 44, pp. 163–175, 2007.

[17] S. K. Sharma and A. Kumar, "Ride comfort of a higher speed rail vehicle using a magnetorheological suspension system," *Proc. Inst. Mech. Eng. Part K J. Multi-body Dyn.*, vol. 232, no. 1, pp. 32–48, 2018.

[18] Q. Zhu, L. Li, C. J. Chen, C. Z. Liu, and G. Di Hu, "A Low-Cost Lateral Active Suspension System of the High-Speed Train for Ride Quality Based on the Resonant Control Method," *IEEE Trans. Ind. Electron.*, vol. 65, no. 5, pp. 4187–4196, 2018.

[19] M. Düğenci, A. Aydemir, İ. Esen, and M. E. Aydın, "Creep modelling of polypropylenes using artificial neural networks trained with Bee algorithms," *Eng. Appl. Artif. Intell.*, vol. 45, pp. 71–79, Oct. 2015.

[20] H. Altınkaya, İ. M. Orak, and İ. Esen, "Artificial neural network application for modeling the rail rolling process," *Expert Syst. Appl.*, vol. 41, no. 16, pp. 7135–7146, 2014.

[21] M. Metin and R. Guclu, "Active vibration control with comparative algorithms of half rail vehicle model under various track irregularities," *JVC/Journal Vib. Control*, vol. 17, no. 10, pp. 1525–1539, 2011.

[22] Z. Yang, J. Zhang, Z. Chen, and B. Zhang, "Semi-active Control of High-speed Trains Based on Fuzzy PID Control," *Procedia Eng.*, vol. 15, pp. 521–525, Jan. 2011.

[23] W. K. Gawronski, *Advanced Dynamics and Active Control of Structures*. 2004.