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SEISMIC VIBRATION CONTROL OF FLUID STORAGE TANKS USING MAGNETORHEOLOGICAL DAMPERS

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Abstract. *Different energy-dissipating devices have been proposed in literature to attenuate destructive effects of seismic events over fluid storage tanks. These structures have a variety of applications and are the imperative part of critical industries. Failure of these structural systems can result in substantial environmental hazards, huge economic losses, and a heavy toll. Proposed techniques in literature for seismic energy dissipation of fluid contained tanks are mostly centred on passive and active control mechanisms. Passive systems usually add complexities, extra mass, and stiffness to the main system and are not adaptive to changes in future excitations with a stochastic and an uncertain nature. Active control systems on the other hand require a considerable amount of external energy which could render them unreliable in the event of an earthquake. Moreover, robustness and instability are of substantial concerns for systems equipped with these mechanisms. Semi-active control devices using materials with adjustable properties offer advantages of both active and passive systems while removing their drawbacks. In this paper, performance of Magnetorheological (MR) dampers on seismic response reduction of fluid storage tanks has been investigated. Numerical simulations of a circular cylindrical stainless steel liquid storage tank equipped with these dampers and excited at the base have been considered to examine the efficiency of MR dampers in mitigation of seismic effects on these structural systems.*

Keywords: Fluid storage tanks; MR damper; Semi-active vibration control

1 INTRODUCTION

1.1 Fluid Storage Tanks

Fluid storage tanks are widely used in a variety of applications and industries with strategic importance and play a critical role in stability and functionality of the corresponding industries. They are used for storage of various liquids like water, wine, dairy products, oil, etc. in wineries, refineries, nuclear power plants and so on. Ground supported fluid storage tanks which, depending on their applications, may contain various types of fluids, are usually applied for storage, distribution, or processing of the contained liquid. Aspect ratio defined as liquid height to the radius or length of the tank's cross section for circular and rectangular tanks, respectively, plays an important role in their dynamic behaviour under vibrations. Such tanks are susceptible to damage in earthquake-prone regions of the world (Dizhur et al., 2017). Two of the most common types of damage for fluid storage tanks during earthquakes are elephant foot and diamond shape buckling. However, other types of damage including damage to the base connections such as anchor bolt buckling, elongation and/or pull-out, leg buckling, damage to roof due to liquid sloshing, rupture of the shell, etc. can occur depending on the type of support, aspect ratio and size of the tank, and their application, among others. Any damage to these tanks may cause severe environmental hazards and result in disastrous consequences.

Analyzing structural behavior and dynamic modelling of these structural systems has been the focus of attention by different researchers for decades (Haroun, 1983; Veletsos, 1984; Moradi, Behnamfar, & Hashemi, 2018). Different types of damage caused in previous devastating seismic events have been examined and various energy-dissipating devices have been proposed by researchers in literature (Yazdanian et al., 2019). A comprehensive review of different dynamic modelling methodologies, seismic protection devices proposed, and various types of damages observed in previous seismic events have been conducted by authors (Hosseini & Beskhyroun, 2023). Among various modelling techniques proposed so far for these structures, lumped parameter model which is an equivalent mechanical model representing the complex fluid – structure interaction has widely used for analyzing the structural responses of these structures (Haroun, 1983). This technique has formed the basis of structural design of tanks in different seismic codes including Eurocode 8 Part 4 (Standardization, 2006) and NZSEE (NZSEE, 2009). Validation of this technique has been studied by several researchers (Hernandez-Hernandez, Larkin, & Chouw, 2021). While this methodology provides dynamic characteristics of the vibration of the highly complex fluid-structure interaction in a fluid tank with acceptable accuracy they offer efficient and low computational effort compared to more sophisticated Finite Element (FE) based techniques (Hosseini & Beskhyroun, 2023).

1.2 Semi-Active Control Devices for Fluid Tanks

Vibration control systems and techniques have been targeted for decades to enhance safety, reliability, and resiliency of engineering structures against undesirable vibrations, especially those caused by sever earthquakes. Passive control mechanisms offer reliability whereas they are not adaptable or versatile. On the other hand, active control systems, while removing drawbacks of passive systems, rely on considerable external power for their operation which could be a source of substantial concern during a disaster. Moreover, reliability and robustness are always matters of concern for active systems. Semi-active systems aim to combine advantages of both passive and active systems while eliminating weak points of them.

Passive control devices for energy-dissipation and seismic protection of fluid storage tanks and industrial plants have been developed and applied to these structures in different forms now for decades (Paolacci, Giannini, & De Angelis, 2013). Base isolators and their variants (Shrimali & Jangid, 2002), Horizontal and vertical baffles (Hasheminejad, Mohammadi, & Jarrahi, 2014), are amongst the most developed passive systems for fluid tanks. Active baffles (Hernández, & Santamarina, 2012) and piezoelectric patches (Mehrvarz et al., 2019) can be named as instances of active devices proposed in this field. Semi-active control devices for vibration control and seismic protection of fluid contained tanks have not been explored to the extent as passive and active systems. Examples of such devices proposed for fluid tanks and vessels include variable dampers for making smart base isolation (Iemura, Igarashi, & Kalantari, 2004), the semi-active mechanism proposed by Kobayashi and Koyama (Kobayashi & Koyama, 2010) based on the air spring effect of a gas chamber on the surface of the liquid and operation of a valve, and application of Magnetorheological (MR) dampers (Shrimali, & Kasar, 2012).

1.3 Magnetorheological (MR) Dampers

Magnetorheological (MR) dampers which are in fact the magnetic analogs of Electrorheological (ER) dampers are based on smart MR fluid with adaptable properties. Physical properties such as viscosity of such a fluid can be changed in a few milliseconds (Dyek et al., 1996) upon the application of a magnetic field. Thus, the interactive force produced by the damper can be rapidly controlled and adapted according to the force required at each time step to counteract the applied external force on the system. These dampers are fail-safe as in case of any malfunction in the control system they are turned into passive devices (passive-off mode) and still can continue damping the vibrations. Special features of these dampers such as a wide operational temperature range, and high achievable yield stresses, to name but a few, have made this possible to apply them in a variety of interior and exterior applications for seismic and vibration mitigation of civil structures and infrastructures as well as other industrial applications (Ahamed, Choi, & Ferdaus, 2018).

In this paper, application of these dampers for vibration mitigation of legged ground supported fluid storage tanks against seismic actions is investigated. Numerical simulations of a sample legged fluid storage tank equipped with a MR damper are presented and efficiency of the damper in reducing seismic responses of the tank under the applied base excitations has been studied.

2 DYNAMIC MODELLING

2.1 Dynamic Modelling of Coupled Fluid-Tank-MR Damper System

Figure 1 shows the considered legged fluid storage tank with a MR damper connected rigidly between the base and the ground through one of its legs.

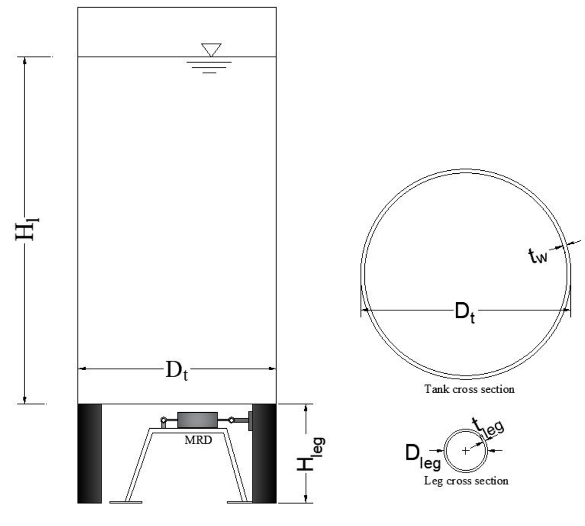

Figure 1 Model of fluid storage tank equipped with a MR damper

The mechanical model of the tank-fluid-damper system is shown in Figure 2. Dynamic equation of motion for the coupled system shown in Figure 2 is written as below,

$$\mathrm{M}\ddot{\mathrm{U}} + \mathrm{C}\dot{\mathrm{U}} + \mathrm{KU} = \xi \mathrm{F}_{\mathrm{MR}} - \mathrm{M}\lambda \ddot{u}_{\mathrm{g}} \tag{1}$$

where, M, C, and K are the mass, damping and stiffness matrices, F_{MR} and $\ddot{u}_g$ are the MR damper force and ground acceleration, ξ and λ are position vectors, andm vectors $\ddot{U}, \dot{U},$ and U

are acceleration, velocity, and displacement vectors of lumped masses shown in Figure 2, respectively. These matrices and vectors are defined as follows,

$$M = \begin{bmatrix} m_r & 0 & 0 \\ 0 & m_i & 0 \\ 0 & 0 & m_c \end{bmatrix}, C = \begin{bmatrix} c_r + c_i + c_c & -c_i & -c_c \\ -c_i & c_i & 0 \\ -c_c & 0 & c_c \end{bmatrix}, K = \begin{bmatrix} k_r + k_i + k_c & -k_i & -k_c \\ -k_i & k_i & 0 \\ -k_c & 0 & k_c \end{bmatrix} \quad (2)$$

$$\ddot{U} = \begin{bmatrix} \ddot{u}_r \\ \ddot{u}_i \\ \ddot{u}_c \end{bmatrix}, \dot{U} = \begin{bmatrix} \dot{u}_r \\ \dot{u}_i \\ \dot{u}_c \end{bmatrix}, U = \begin{bmatrix} u_r \\ u_i \\ u_c \end{bmatrix}, \xi = \begin{bmatrix} -1 \\ 0 \\ 0 \end{bmatrix}, \lambda = \begin{bmatrix} 1 \\ 1 \\ 1 \end{bmatrix} \quad (3)$$

Parameters of the equivalent mechanical model considered in this study for the fluid-tank system can be found in other studies for every specific aspect ratio (Haroun, 1983). In this representation the tank shell has been considered flexible, and only the first impulsive and the fist convective modes are considered. Mass, damping, and stiffness of these modes are shown as m_i, m_c, c_i, c_c, k_i, k_c, while m_r is the fluid mass that moves rigidly with the tank shell, and

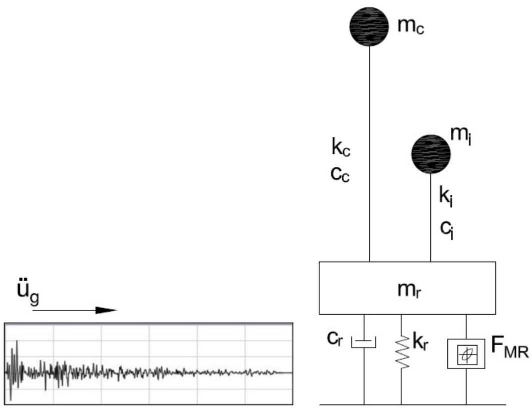

Figure 2 Mechanical equivalent of fluid-tank-damper system

u_r, u_i, and u_c are displacements of the rigid mass, impulsive mass, and convective mass, respectively. Moreover, c_r and k_r represent the damping and stiffness associated with the tank legs.

State-space representation of equation (1) can be formulated according to equation (4) which represents a sixth order dynamical system,

$$X = A\dot{X} + Bu^* \quad (4)$$

where state and control input matrices A, and B are defined as follows,

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{k_r+k_i+k_c}{m_r} & -\frac{c_r+c_i+c_c}{m_r} & \frac{k_i}{m_r} & \frac{c_i}{m_r} & \frac{k_c}{m_r} & \frac{c_c}{m_r} \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_i}{m_i} & \frac{c_i}{m_i} & -\frac{k_i}{m_i} & -\frac{c_i}{m_i} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ \frac{k_c}{m_c} & \frac{c_c}{m_c} & 0 & 0 & -\frac{k_c}{m_c} & -\frac{c_c}{m_c} \end{bmatrix}, B = \begin{bmatrix} 0 & 0 \\ -\frac{1}{m_r} & -1 \\ 0 & 0 \\ 0 & -1 \\ 0 & 0 \\ 0 & -1 \end{bmatrix} \quad (5)$$

while state and control vectors are expressed as $X = [x_1 \quad x_2 \quad x_3 \quad x_4 \quad x_5 \quad x_6]^T = [u_r \quad \dot{u}_r \quad u_i \quad \dot{u}_i \quad u_c \quad \dot{u}_c]^T$ and $u^* = [f_{MR} \quad \ddot{u}_g]^T$, respectively.

It is assumed that absolute accelerations of the system through accelerometers at the locations of rigid mass, impulsive mass, convective mass, as well as the displacement of the rigid mass are measured. Regarding the mentioned measurements, the output equation is formulated according to equation (6),

$$Y = CX + Du^* \tag{6}$$

where the output vector is $Y = [\dot{x}_2 \quad \dot{x}_4 \quad \dot{x}_6]^T$, and measurement and direct transition matrices are represented according to equation (7),

$$C = \begin{bmatrix} -\frac{k_r+k_i+k_c}{m_r} & -\frac{c_r+c_i+c_c}{m_r} & \frac{k_i}{m_r} & \frac{c_i}{m_r} & \frac{k_c}{m_r} & \frac{c_c}{m_r} \\ \frac{k_i}{m_i} & \frac{c_i}{m_i} & -\frac{k_i}{m_i} & -\frac{c_i}{m_i} & 0 & 0 \\ \frac{k_c}{m_c} & \frac{c_c}{m_c} & 0 & 0 & -\frac{k_c}{m_c} & -\frac{c_c}{m_c} \end{bmatrix}, D = \begin{bmatrix} -\frac{1}{m_r} & 0 \\ 0 & 0 \\ 0 & 0 \end{bmatrix} \tag{7}$$

MR damper force is calculated based on the modified Bouc-Wen hysteresis dynamic representation of the behaviour of this damper. According to this dynamic modelling, the force produced by the damper is formulated using equations (8-14) (Spencer et al., 1997),

$$f_{MR} = \alpha z + c_0(\dot{x}_M - \dot{y}) + k_0(x_M - y) + k_1(x_M - x_0) \tag{8}$$

$$\dot{z} = -\gamma|\dot{x}_M - \dot{y}|z|z|^{n-1} - \beta(\dot{x}_M - \dot{y})|z|^n + A(\dot{x}_M - \dot{y}) \tag{9}$$

$$\dot{y} = \frac{1}{(c_0+c_1)}\{\alpha z + c_0\dot{x}_M + k_0(x_M - y)\} \tag{10}$$

$$\alpha(\tau) = \alpha_a + \alpha_b\tau \tag{11}$$

$$c_1(\tau) = c_{1_a} + c_{1_b}\tau \tag{12}$$

$$c_0(\tau) = c_{0_a} + c_{0_b}\tau \tag{13}$$

$$\dot{\tau} = -\mu(\tau - \vartheta) \tag{14}$$

where $\alpha_a, \alpha_b, c_{1_a}, c_{1_b}, c_{0_a}, c_{0_b}, \mu, k_0, k_1, A, \beta, \gamma, x_0$, and n are the 14 parameters associated with the modified Bouc-Wen model, and τ is result of the first-order filter equation (14). Displacement of MR damper is shown by x_M.

2.2 Vibration Control Scheme

Since in this investigation only the acceleration responses of the system are measured for the control system design of the coupled fluid-tank system, H_2/LQG control scheme (Spencer et al., 1994) is applied as the primary control design. Moreover, due to the application of a semi-active control mechanism, i.e. MR damper, the calculated desired control force cannot be commanded. Therefore, a secondary controller is required to command the control signal to the damper. For this purpose, Clipped-Optimal Control (COC) scheme (Tseng, & Hedrick, 1994) using acceleration feedback is employed to acquire the command signal to the damper. Based on the selected control scheme, the desired optimal control vector would be obtained using equation (15),

$$u = -K\tilde{X} \tag{15}$$

where $\tilde{X}$ is the vector of estimated states using the Kalman-Bucy filter, and K is the full-state feedback gain matrix based on the Linear Quadratic Regulator (LQR) problem. Equation (15) minimizes the performance index stated in equation (16) below,

$$J = \int_0^\infty (Y^TQY + u^TRu)dt \tag{16}$$

In this equation, Q and R are weighting matrices which weight selected measured responses of the system and control force, respectively. Thus, the observer-controller transfer function of the system is represented in the closed form of equation (17),

$$K_{Con}(s) = K(sI - (A - K_EC))^{-1}[K_E\, B_1 - K_ED_1] \tag{17}$$

where K_E is the Kalman filter gain matrix, while B_1 and D_1 are the first columns of matrices B and D, respectively. Finally, the control signal using the secondary controller would be acquired using equation (18),

$$\vartheta = V_{max}H\{(u - f_{MR})f_{MR}\} \tag{18}$$

3 NUMERICAL EVALUATIONS

To investigate the efficacy of MR dampers in vibration mitigation of the model fluid-tank system shown in Figure 1, an example of a stainless-steel circular cylindrical tank with a height of 2m, a radius of 0.850m, and a shell thickness of 0.0005m is considered. Legs are stainless-steel pipes with a thickness of 0.002m, height of 0.5m, and radius of 0.05 m. Other specifications of the tank are modulus of Elasticity, E= 200 GPa, poison's ratio ν=0.3, density of steel, $\rho_s = 7850\ kg/m^3$, and density of water, $\rho_s = 1000\ kg/m^3$. Damping ratios of 2% and 0.5% have been considered for impulsive and convective modes of fluid vibrations. Damping ratio for legs has been regarded similar to that of impulsive mass and shell body, i.e. 2%. This fluid-tank system is modeled using one impulsive mass (first fundamental impulsive mode) and one convective mass (first fundamental convective mode) to represent the fluid-structure interaction between the tank shell and the fluid domain (Haroun, 1983). The fluid-tank-MR system is examined under the 1979 El Centro earthquake with the PGA of 0.449g. The considered MR damper in this study has a capacity of around 2.45 KN, and the input voltage to the damper is regarded in the range of 0 to 5 V. Parameters of modified Bouc-Wen model for the MR damper are considered as below (Gao, 2012),
$\alpha_a = 1921.141\frac{N}{m}, \alpha_b = 5882.51\frac{N}{V.m}, c_{1_a} = 2089.263\frac{N.s}{m}, c_{1_b} = 14384.918\frac{N.s}{V.m}, c_{0_a} = 651.4718\ \frac{N.s}{m},\ c_{0_b} = 1043.7559\ \frac{N.s}{V.m}, \mu = 60\ s^{-1},\ k_0 = 1940.405\frac{N}{m}, k_1 = 1.751268\frac{N}{m}$, $A = 155.32, \beta = 36332.07\ m^{-2}, \gamma = 36332.07\ m^{-2}, x_0 = 0.00\ m$, and $n = 2$.

To design the adopted control scheme, the weighting matrix Q is selected to weight accelerations of lumped masses with a factor of 1 and $R = 10^{-16}$. The vibrational responses of the system under the considered ground motion are calculated and compared in different scenarios to evaluate the effectiveness of added dampers to the fluid-tank system. These scenarios include, the uncontrolled case, controlled system with COC controller, as well as two other scenarios in which MR dampers are regarded as merely passive devices that require no control signal command at each time step. These passive modes are passive-off and passive-on in which the voltage to the damper is held at 0 V (no voltage applied at all), while in the other case it is held at the maximum level of 5 V through the whole control process. Table 1 shows peak structural response of the system under these scenarios. In this table $\ddot{u}$ is the absolute acceleration of each mass, u is its displacement relative to the ground, and f_{MR} is the generated force by the MR damper corresponding to each control strategy. Efficacy of MR damper in seismic response attenuation of the system has been examined for three different aspect ratios as the operational condition which governs the behaviour of the system under base excitations.

Table 1 Peak responses of tank-liquid-MR damper system under El Centro 1979 earthquake

Control scenario	u_r (cm)	$\ddot{u}_r$ (cm^2/s)	u_i (cm)	$\ddot{u}_i$ (cm^2/s)	u_c (cm)	$\ddot{u}_c$ (cm^2/s)	f_{MR} (N)
S=1.00 (Broad tank)							
Uncontrolled	6.67e-3	695.15	7.47e-3	749.74	16.08	649.02	-
COC	5.67e-3	556.45	5.82e-3	538.78	16.08	649.00	118
Passive-off	6.29e-3	661.86	7.04e-3	713.06	16.06	648.34	14
Passive-on	5.63e-3	445.72	5.76e-3	475.41	16.06	648.49	147

S = 2.00 (Slender tank)							
Uncontrolled	1.80e-2	640.90	2.10e-2	737.91	15.95	676.30	-
COC	1.50e-2	629.75	1.76e-2	675.66	15.95	676.29	343
Passive-off	1.71e-2	616.13	2.01e-2	709.62	15.92	675.14	35
Passive-on	1.15e-2	476.03	1.36e-2	570.10	15.93	675.57	502
S = 3.00 (Slender tank)							
Uncontrolled	3.23e-2	662.26	4.12e-2	790.77	15.93	676.12	-
COC	2.27e-2	499.65	3.02e-2	632.28	15.93	676.11	664
Passive-off	3.34e-2	659.36	4.30e-2	805.24	15.93	675.99	48
Passive-on	1.87e-2	459.97	2.46e-2	520.84	15.93	676.19	698

It can be seen from Table 1 that MR damper has been able to reduce vibrational response of the fluid tank system. Reductions in responses of the convective mass are not considerable which could be justified by very different vibrational period and frequency of this mode from impulsive and rigid modes. Reductions in structural seismic responses increases by the increase of the aspect ratio and the force produced by the damper. Depending on the aspect ratio and control scheme, MR damper was able to reduce peak relative displacements of rigid and impulsive masses between 15% to 40%. Corresponding reduction percentage points for absolute accelerations of these masses are between 20% to 37%.

4 CONCLUSIONS

Various seismic energy-dissipating devices have been introduced in literature for seismic retrofitting and vibration mitigation of fluid tanks. Most of these devices are passive systems which their characteristics cannot be adapted during future unknown excitations. In this research, application of MR dampers as smart semi-active mechanisms that combine features of active and passive systems while removing drawbacks of each has been examined. Results of numerical simulations show that efficiency of the damper in vibration attenuation of fluid tanks depends highly on the aspect ratio. For the sample tank model considered in this study, installation of one MR damper at the base decreased relative displacements and absolute accelerations of base as well as the impulsive mass. Responses of the convective mass did not change significantly. This can be justified as convective mass vibrates with a period and frequency considerably far away from the impulsive and rigid masses. Further investigations are required to examine influence of different factors on the extent of attenuations in seismic vibrational responses of fluid tanks by controlling them via MR dampers.

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