Transactions of the Canadian Society for Mechanical Engineering

Magnetorheological Elastomer-Based Variable Stiffness Flexible Coupling for Vibration Isolation.

Journal:	<i>Transactions of the Canadian Society for Mechanical Engineering</i>
Manuscript ID	TCSME-2021-0007.R1
Manuscript Type:	Article
Date Submitted by the Author:	03-May-2021
Complete List of Authors:	Syam, Thaer; Qatar University College of Engineering, Mechanical and Industrial Engineering Department Hegazi , Ahmed; Qatar University College of Engineering, Mechanical and Industrial Engineering Department Muthalif, Asan; Qatar University College of Engineering, Mechanical and Industrial Engineering Department Badri, Yousif; Qatar University College of Engineering, Mechanical and Industrial Engineering Department
Keywords:	Magnetorheological Elastomer, Variable Stiffness Coupling, Vibration Isolation
Is the invited manuscript for consideration in a Special Issue? :	Not applicable (regular submission)

SCHOLARONE™
Manuscripts

Magnetorheological Elastomer-Based Variable Stiffness Flexible Coupling for Vibration Isolation

Thaer M. I. Syam¹, Ahmed A. A. Hegazi², Asan G. A. Muthalif³, Yousif Badri⁴

Dept. of Mechanical and Industrial Engineering, College of Engineering, Qatar University, Doha, Qatar

³Corresponding author

E-mail: ¹*thaer@qu.edu.qa*, ²*ah1602411@qu.edu.qa*, ³*drasan@qu.edu.qa*, ⁴*yb1903174@qu.edu.qa*

Draft

Abstract. Rotating mechanical equipment such as pumps and motors could suffer from unattenuated torsional vibration. The causes of such vibration could range from imbalance within the rotating system to misalignment or even operating at frequencies close to the system's natural frequency. Smart composite materials are materials by which the exposure of external stimuli changes their properties. Magnetorheological elastomers (MRE) are smart composite materials by which their mechanical properties, such as stiffness, are changed under a magnetic field. In this article, the introduction of a variable stiffness coupling (VSC) fitted within a shaft for torsional vibration isolation that would adapt and change its attenuation frequency range is presented. The VSC concept on torsional vibration isolation is tested experimentally. MRE samples with 40% volume fraction are fabricated and manufactured using a 3D mold design and fixed within a coupling in a shaft to investigate the magnetic field effect on the torsional rigidity. Impact hammer test is conducted along with an accelerometer to obtain the transmissibility factor analysis. Results show that the vibration level decreases when the magnetic field increases. The 1st natural frequency of the system happened at 26 Hz and moved to 28 Hz when the applied current increases from 0 mT to 12.38 mT. MRE torsional stiffness increased from 37.4 N.m/rad to 61.6 N.m/rad when the current increased from 0 mT to 12.38 mT. The torsional damping coefficient showed a fluctuation in its variation as the damping effect of MR elastomer is ignored.

Keywords: Magnetorheological Elastomer, Variable Stiffness Coupling, Vibration Isolation.

1. Introduction

Magnetorheological elastomer (MRE) is a class of smart materials that can change its rheological properties reversibly due to an applied external magnetic field (Deepalekshmi *et al.*, 2013). Traditionally, MR materials are composed of nano or micro-sized magnetic particles that are placed in a non-magnetic matrix. MRE composites have a solid or gel-like matrix that is non-magnetic and contains suspended magnetic particles distributed within (Samal *et al.*, 2020). MREs' modulus of elasticity could be controlled using the applied magnetic field. Hence, MREs are used in making adaptively tuned vibration absorbers, isolators and sensors for mechanical and magnetic applications [3, 4].

Normally, the MRE is constituted from non-magnetic elastic matrices, magnetic particles and additives (Bastola and Hossain, 2020). The elastic matrix could be from elastomers such as natural rubber, silicone rubber or polydimethylsiloxane (PDMS) [2, 5, 6]. One example of the magnetic particles that is widely used is the micron-sized carbonyl-iron (CI) powder (Perales-Martínez *et al.*, 2017). The magnetic particles should have high permeability, high saturation magnetization, and low remanent magnetization to maintain a more sensitive response to an external magnetic field applied (Danas, Kankanala and Triantafyllidis, 2012). The distribution of the magnetic particles could be homogenous or into groups (particles aligning into parallel lines). The factor determining the particles' distribution is the magnetic field applied during the crosslinking of the elastomer polymer. If a magnetic field is applied during crosslinking, the resultant elastomer would possess magnetic particles that are aligned into parallel lines (Lee, Kim and Jung, 2018). The main aspect of interest in MRE is the MR effect, where an external magnetic field could influence the elastomer's magnetic particles to move from their original positions to achieve minimum energy states, which leads to a deformation of the elastomer matrix (Boczkowska and Awietj, 2012). Figure . 1 shows the fabrication process of MREs.

MREs have been widely used in vibration isolation applications. (Deng, Gong and Wang, 2006) have investigated the ability to use MREs as tuned vibration absorbers. They have installed a simple setup that included an MRE that acted as the median between an oscillator and a steel beam. By changing the current applied to the MR elastomer coils, the magnetic field acting on the MRE changed, and its stiffness changed. Results revealed that the elastomer's changing stiffness had changed the system's natural (resonance) frequency and the transfer function of the oscillator vibration to the beam. Similar findings were documented by (Tao *et al.*, 2018), where the properties of controllable stiffness and damping are investigated for a novel isolator based on magnetorheological elastomers.

According to the experimental results, the magnetorheological elastomer isolator's initial vertical stiffness and damping coefficient are 1.14×10^6 N/m and 495.8 N s/m, respectively. The relative increase in stiffness and damping is 66.57% and 45.55%, respectively. The isolation transmissibility and root mean square of acceleration response is reduced by 41.2% and 65.3%, respectively, which indicates that the natural frequency is changing.

MRE was used for base isolation of a civil structure; four MRE samples were trapezoidal in geometry (Behrooz, Wang and Gordaninejad, 2014b). Force-displacement results were plotted, and a percentage of 57% rise within the loops is found for different current applied. The work of (Alias *et al.*, 2018) shows an experimental investigation on different MRE samples with various ratios of magnetic particles. Both tension and compression modes were used with other applied current and force. Results depicted that the resisting force of MRE is directly proportional to the applied current, displacement and velocity. The highest force achieved was on the 30% magnetic particles MRE sample with a velocity of 4 mm/s and applied current of 2 A. (Liao *et al.*, 2012) achieved tunable damping and stiffness using four layers of MRE and three coils for magnetic field induction. A novel MRE isolator to control vibration in vehicles, stiffness and damping is increased with magnetic field in both shear and compression modes were proposed by (Behrooz, Wang and Gordaninejad, 2014a). The work of (Lee, Park and Kim, 2019) consisted of isolating the torsional vibration in a shaft using a cylindrical MRE specimen. The magnetic field was generated around the elastomer using magnets. The stiffness values ranged from 23.34 N.m/rad to 36.38 N.m/rad under a magnetic field variation from 0 mT to 150 mT. At this magnetic field range, the natural frequency of the system shifted from 16.8 to 23.5 Hz. Another work by (Hoang *et al.*, 2013) which developed a torsional adaptive tunable vibration absorber using a magnetorheological elastomer for vibration reduction of a powertrain test rig. Their results showed that the system's natural frequency shifted from 10.75 Hz to 16.5 Hz under a magnetic field (0 T – 0.21 T).

Also, the damping ratio of the system was slightly decreased with the magnetic field. (Praveen Shenoy *et al.*, 2018) studied the magnetic field effect on the system parameters comprehended under impulse loading conditions using a custom-built in-house system. Series arrangement of accelerometers was used to differentiate between the torsional and the bending modes of vibration of the system. The tests revealed a shift in torsional frequency in the presence of a magnetic field which elucidates the ability of MRE to work as a potential vibration isolator for torsional systems. Some analytical models of MRE have been studied by (Syam and Muthalif, 2020) to observe their hysteresis behaviour and to plot the transmissibility curves for base isolation. Linear and torsional stiffness and transmissibility analysis

were obtained using FEA for a micro-scale silicon rubber embedded with 12 magnetic particles ME elastomer (Syam *et al.*, 2020). Results showed that the stiffness in both modes increases as the magnetic field increase.

In this paper, MRE and coupler design is tested in practice, per the previous studies in the field. The investigation is mainly about the relationship between the electric current applied and the intensity of the magnetic field and the effect on the MRE stiffness in torsional semi-active vibration isolation application. Also, through the mounted accelerometer on the experimental setup, the transmissibility factor's change will be determined and reported. The novelty of this article is the contribution to making a new design for flexible coupling using MRE for torsional vibration isolation. Also, this work contributes to minimizing the effect of the torsional vibration in rotary machines, which can be extended to rotating shafts applications; the vibration analysis in this work depends on the concept of tangential acceleration, which is relevant to the circular motion.

2. Mathematical modelling and equations used

The mathematical model of the system including the MRE coupling starts with a fundamental equation of motion of single degree of freedom with input angular displacement $\theta(t)_{in}$, and output angular displacement $\theta(t)_{out}$. This can be shown in Figure . 2. Exact mathematical modeling of any physical system leads to nonlinear differential equations which complicate the modeling. Since exact solution of linear differential equations can usually be determined, assumptions are often made to linearize the problem. Linearization of the problem often leads to the elimination of nonlinear terms in governing equations or the approximation of nonlinear terms by linear terms (Elliott, Tehrani and Langley, 2015). Therefore, the modeling of the system is assumed to be linear. The equation of motion that shows a set of differential equations obtains the system performance is as follows

$$\sum T = J\ddot{\theta} \quad (1)$$

$$J(\ddot{\theta}_{out}) + c_{mre}(\dot{\theta}_{out} - \dot{\theta}_{in}) + K_{mre}(\theta_{out} - \theta_{in}) = 0 \quad (2)$$

where J is the mass moment of inertia, $\dot{\theta}_{out}$ is the output angular velocity, $\dot{\theta}_{in}$ is the input angular velocity, θ_{out} is the angular displacement, θ_{in} is the input angular displacement. K_{mre} and c_{mre} are the stiffness and damping coefficient, respectively; of MRE which can be expressed in this form:

$$J(\ddot{\theta}_{out}) + c_{mre}\dot{\theta}_{out} + K_{mre}\theta_{out} = c_{mre}\dot{\theta}_{in} + K_{mre}\theta_{in} \quad (3)$$

The transfer function of this system can be obtained by applying the Laplace transform to differential equation Eq.

(3). Applying of $\theta(t) = \vartheta(s)$

$$Js^2\vartheta(s)_{out} + c_{mre}s\vartheta(s)_{out} + K_{mre}\vartheta(s)_{out} = c_{mre}s\vartheta(s)_{in} + K_{mre}\vartheta(s)_{in} \quad (4)$$

By taking the common factors, Eq. (4) can be expressed in this form

$$[Js^2 + c_{mre} + K_{mre}]\vartheta(s)_{out} = [c_{mre} + K_{mre}]\vartheta(s)_{in} \quad (5)$$

The transfer function can be obtained by taking the ratio of the output and the input as follows

$$\frac{\vartheta(s)_{out}}{\vartheta(s)_{in}} = \frac{c_{mre}s + K_{mre}}{Js^2 + c_{mre}s + K_{mre}} \quad (6)$$

The angular displacement transmissibility factor equation can be obtained firstly by converting Eq. (6) from the (s-domain) to the frequency domain (ω -domain) in terms of amplitude and phase. By replacing $s = j\omega$

$$\frac{\vartheta(j\omega)_{out}}{\vartheta(j\omega)_{in}} = \frac{c_{mre}(j\omega) + K_{mre}}{J(j\omega)^2 + c_{mre}(j\omega) + K_{mre}} = \frac{c_{mre}j\omega + K_{mre}}{-J\omega^2 + c_{mre}j\omega + K_{mre}} \quad (7)$$

The angular displacement (torsional) transmissibility factor (TT) can be obtained by taking the magnitude of Eq. (7)

$$TT = \left| \frac{\vartheta(\omega)_{out}}{\vartheta(\omega)_{in}} \right| = \left| \frac{\theta_{out}}{\theta_{in}} \right| = \sqrt{\frac{(c_{mre}\omega)^2 + (K_{mre})^2}{[c_{mre}\omega]^2 + (K_{mre} - J\omega^2)^2}} \quad (8)$$

where TT is the torsional transmissibility factor, and ω is the natural frequency in (rad/s).

The limitations of assuming a linear model are that the narrower frequency band of vibration attenuation (Sun *et al.*, 2018) and lower robustness (Li *et al.*, 2015) compared to the non-linear system. Since the mass. Non-linear isolators have a wider frequency band of vibration attenuation and can effectively reduce vibration with lighter auxiliary mass, since the mass of the system is not the main objective of this study, the linear model assumption showed satisfying results in terms of shifting the natural frequency away from the fundamental frequency. This is to avoid the non-linear modeling complexity and errors (Jiffri *et al.*, 2014).

3. Experimental Methodology and Setup

3.1 Fabrication of MRE based coupling

A silicone-based rubber which is molded to fit within a fabricated coupling was used. RTV136-60 silicone was chosen due to its high shear strength (higher than 29 N/mm) and flexibility (elongates more than 450%). Its high viscosity ensured that the particles would not precipitate at the bottom of the newly made elastomer on adding magnetic particles during curing. The volume fraction (VF) of the magnetic particles to the polymer is 40%. This

value is chosen based on the work of (Dargahi, Sedaghati and Rakheja, 2019). The particular MRE sample with a 40% volume fraction revealed a high increase in the MRE storage modulus under magnetic flux. The properties of the used MRE are given in Table 1.

The RTV136-60 constituted from two parts (part A: base and part B: catalyst) mixed manually at a ratio of 1000:35 for an estimated period of 15 minutes. The fabrication of a mold to cure the RTV136-60 was based on the design of an existent rubber model that was widely used within couplers available in the market. The shape of the rubber part of the coupler was across. The proposed MREs were designed to extended heights of 30 mm and 50 mm to inspect their performance change further. The two parts of the RTV136-60 were utilized to make a quantity of 70 mL of the elastomer. The mixing was done manually but thoroughly, and the mixture was poured as slowly as possible into the mold to release as much air bubbles as possible from the mixture. Some remaining air bubbles would not affect the required portion of the mold. Figure . 3 (a) shows the mold used to manufacture the MRE, and Figure . 3 (b) shows the MRE used for this work. Figure . 3 (c) shows the aluminium coupler parts.

3.2 Experimental Methodology

A modal analysis is done in this experiment done using an impact hammer test. This test is done to get the transmitted load of the shaft before and after the presence of MRE. The response needed in this test is the frequency response function (FRF) which represents the transfer function of the system. The transfer function is the ratio between the output response and the input excitation. The response obtained from this test is the frequency response in terms of the ratio of the excitation force (F) to the output acceleration (a_t). The impact was given on the right plate, as shown in Figure .4 (c) and the response is obtained using the accelerometer fitted on the left plate as in Figure .4 (b).

The FRF needed in this experiment must be in an angular domain because the application is to obtain the angular transmissibility factor. The FRF obtained from the data acquisition system is as follows (Al-Khazali, 2011)

$$FRF = \frac{a_t}{F} = \frac{\alpha r}{T} = \frac{\alpha}{T} r^2 \quad (9)$$

where a_t is the tangential acceleration and F is the excitation force input, r is the distance from the shaft centre to the accelerometer axis, α is the angular acceleration, and T is the torque. From the FRF, half bandwidth theory is applied to obtain the damping ratio and the torsional stiffness of the MRE as the magnetic field increases. The damping ratio ζ can be expressed by (Papagiannopoulos and Hatzigeorgiou, 2011)

$$\zeta = \frac{f_2 - f_1}{2f_n} \quad (10)$$

where f_1 and f_2 are the frequencies in Hz measured at $(\frac{1}{\sqrt{2}}) TF_{maximum}$, where TF is the transfer function factor. f_n is the natural frequency. The frequency f (Hz) can be represented in ω (rad/s) using $f = 2\pi/\omega$.

The torsional stiffness k_t of the MRE is obtained from the frequency domain representation of the signal (corresponding to the maximum value of the FFT). For the torsional natural frequency (ω_n), k_t for the given rotational inertia, J is estimated from the relation (Papagiannopoulos and Hatzigeorgiou, 2011)

$$\omega_n = \sqrt{\frac{k_t}{J}} \quad (11)$$

where ω_n is the natural frequency of the system in rad/s. k_t is the torsional stiffness of MRE, and J is the mass moment of inertia of the disc attaching the end of the shaft. The damping coefficient c , of MRE, can be obtained in terms of the torsional stiffness k_t , a torsional moment of inertia J , and the damping ratio ζ by using the critical damping c_c as follows

$$c_c = 2\sqrt{Jk_t} \quad (12)$$

$$c = \zeta c_c \quad (13)$$

3.3 Experimental Setup

A test rig is shown in Figure . 5 was designed and fabricated to validate the results. It mainly consists of two shafts, which were coupled together with the MRE-fitted coupler. The MRE-fitted coupler was designed to be put within the solenoid's inner vicinity to achieve maximum benefit from the generated magnetic field. A small plate was welded on the shaft to fit an accelerometer. A plate was also welded on the other shaft for the impact hammer. The two shafts were fixed in their positions using ball bearings. Also, a disc was fitted on the shaft holding the accelerometer to provide an inertial motion to the shaft. A DC regulated power supply was utilized to control the current intensity within the solenoid, and the FW Bell Gauss/Tesla meter was used to measure the magnetic field intensity within the solenoid. Both the impact hammer and the accelerometer were connected to a data acquisition system (DAQ) using NI 9234, which is then connected to the dynamic signal analyzer (DSA) (CF-H2

TOUGHBOOK). DSA calculated the torsional transfer function in real-time. The MRE fitted along with the coupling is shown in Figure . 6.

Figure . 7 shows a simple schematic of the experimental setup used. The MRE utilized was 13% iron particles by mass. As soon as the impact hammer touched the impact plate (channel 1), the data acquisition system (DAQ) captured the system's response (channel 2) via the accelerometer attached on the other side of the MRE-fitted coupler. At every 1 A (3.16 mT) increment of current, the magnetic field at the solenoid was first measured, then the impact hammer was utilized five times to do the impact test. The five responses achieved at each current increment were averaged to ensure the consistency of the results reported and eliminate any anomaly. The current applied was varied from 0 A (0 mT) to 3 A (12.38 mT).

4. Experimental Results and Discussion

Figure . 8 shows the relationship between the applied current and the magnetic field distributed throughout the coil. It shows that there is a direct relationship between the current and the magnetic field. The test was carried out for the different applied current i.e., different magnetic field.

The torsional transfer function factor was evident through the experiment as in Figure . 9 that it varies on changing the current applied. The variation occurs on the whole excitation frequency range as it shows some fluctuations and different peaks. The spectrum range of 15 Hz - 40 Hz is where the transfer function factor reaches its maximum values, indicating resonance. That is why the low-frequency range of 0 Hz - 150 Hz is the most critical range for the experimental setup used, which is similar to most rotatory equipment that suffers high vibration amplitude at low excitation frequencies (Wen *et al.*, 2018). In higher frequencies, the value of the transmissibility factor starts varying more randomly. Further analysis of 0 Hz - 150 Hz is done as shown in the zoomed figure in Figure . 10.

Figure . 10 provides a clearer demonstration of the MRE effect. As the current applied is increased, the torsional transfer factor diminishes in value and the natural (resonance) frequency shifts to the right (increases). Without using current (no magnetic field), the system's natural frequency is at nearly 26.5 Hz, and the torsional transfer function factor is at 0.62. This is under the passive stiffness of the MRE. At the same frequency with an applied current of 1 A (3.16 mT), the torsional transfer function factor drops to 0.56. As the current is applied and increased from 1 A (3.16 mT) to 3 A (12.38 mT), the natural frequency starts increasing to 28 Hz, 30.5 Hz, and 34

Hz accordingly as in Figure . 11 (c) also, and the factor starts decreasing to 0.6, 0.59, and 0.55 respectively. Therefore, a minimum torsional transfer function factor can be achieved.

The variation of torsional stiffness and damping coefficient with current applied was investigated and calculated using Eqs. (10, 11). The findings as indicated in Figure . 11 (a) illustrate that the torsional stiffness of the system increases on increasing the magnetic field generated. However, that is the expected behaviour from the MRE-fitted coupler, which proves its functionality. The torsional stiffness has increased from 37.4 Nm/rad to 61.1 Nm/rad by the magnetic field's influence from 0 A (0 mT) to 3 A (12.38 mT). Eq. (11) was used to calculate the torsional stiffness using the moment of inertia of the disc ($J = 0.00135 \text{ kg.m}^2$); and using the natural frequency of the shaft at a given magnetic field value. Figure . 11 (b) shows that the damping coefficient increases on increasing the magnetic field generated and then decreases and increases after that, which shows no specific variation as expected. The change in the damping coefficient (Eqs. (12,13)) is minimal as it increased from $0.043 \text{ kg.m}^2/\text{rad.s}$ to $0.071 \text{ kg.m}^2/\text{rad.s}$. In essence, on increasing the current applied to the electromagnet surrounding the MRE-fitted coupler, the magnetic field within increases, the torsional stiffness and damping coefficient fluctuate. Hence, the system's natural frequency increases and the torsional transfer function decreases within the same frequency. Table 2 summarizes the findings of this work.

5. Conclusion

In this article, 40% volume fraction MRE was fabricated and fitted into a VSC for semi-active torsional vibration isolation application. The mathematical model was derived and obtained to get the angular transmissibility factor formula. MRE was tested using an impact hammer test as a way of conducting a modal analysis. Torque was applied on one side of the shaft, and using the concept of angular acceleration, the FRF was obtained and plotted against the excitation frequencies. It was found that the resonance frequency happens at a low range of frequencies. The system's natural frequency is shifted from 26.5 Hz to 28 Hz when 3.16 mT field is applied on the system. This means that vibration isolation has occurred so that the vibration levels are reduced. The torsional stiffness had increased from 37.4 Nm/rad to 61.6 Nm/rad when the applied magnetic field increased from 0 mT to 12.38 mT. Results revealed that MREs' damping characteristics do not change significantly under a magnetic field, proving the literature. This work showed that semi-active torsional vibration isolation of a rotating shaft using VSC is achieved. This work can be enhanced and improved by using an inertial shaker to generate an input sinusoidal wave instead of

an impact hammer. Besides, further investigations can be made on the properties of the MRE is such applications by studying the variation of the volume fraction and the composition of the polymer matrix.

Funding Acknowledgement

This publication was partially supported by Qatar University International Research Collaboration Grant No. IRCC-2020-017. The findings achieved herein are solely the responsibility of the authors

Conflict of Interest Statement

The authors declare no conflict of interest in preparing this article.

References

- Al-Khazali, H. A. (2011) *Calculations of Frequency Response Functions(FRF) Using Computer Smart Office Software and Nyquist Plot under Gyroscopic Effect Rotation*, *IRACST-International Journal of Computer Science and Information Technology & Security (IJCSITS)*.
 ALCHEMIX® PU 7231 | Enhanced Reader (no date). Available at: moz-extension://63792d48-c9fb-4e84-82c8-fc58689b85ea/enhanced-reader.html?openApp&pdf=https%3A%2F%2Fwww.alchemie.com%2Fmedia%2Fcustom%2Fupload%2FTDS-RTV-136.pdf (Accessed: 12 April 2021).
- Alias, N. F. *et al.* (2018) 'Experimental investigation of static properties of magnetorheological elastomer', *Iranian Journal of Science and Technology - Transactions of Mechanical Engineering*. Springer International Publishing, 42(2), pp. 185–197. doi: 10.1007/s40997-017-0081-5.
- Ausanio, G. *et al.* (2014) 'Magneto-piezoresistance in Magnetorheological elastomers for magnetic induction gradient or position sensors', *Sensors and Actuators, A: Physical*. Elsevier, 205, pp. 235–239. doi: 10.1016/j.sna.2013.10.009.
- Bastola, A. K. and Hossain, M. (2020) 'A review on magneto-mechanical characterizations of magnetorheological elastomers', *Composites Part B: Engineering*. Elsevier Ltd, p. 108348. doi: 10.1016/j.compositesb.2020.108348.
- Behrooz, M., Wang, X. and Gordaninejad, F. (2014a) 'Modeling of a new semi-active/passive magnetorheological elastomer isolator', *Smart Materials and Structures*. Institute of Physics Publishing, 23(4). doi: 10.1088/0964-1726/23/4/045013.
- Behrooz, M., Wang, X. and Gordaninejad, F. (2014b) 'Performance of a new magnetorheological elastomer isolation system', *Smart Materials and Structures*, 23(4). doi: 10.1088/0964-1726/23/4/045014.
- Boczkowska, A. and Awietj, S. (2012) 'Microstructure and Properties of Magnetorheological Elastomers', in *Advanced Elastomers - Technology, Properties and Applications*. InTech. doi: 10.5772/50430.
- Danas, K., Kankanala, S. V. and Triantafyllidis, N. (2012) 'Experiments and modeling of iron-particle-filled magnetorheological elastomers', *Journal of the Mechanics and Physics of Solids*, 60(1), pp. 120–138. doi: 10.1016/j.jmps.2011.09.006.
- Dargahi, A., Sedaghati, R. and Rakheja, S. (2019) 'On the properties of magnetorheological elastomers in shear mode: Design, fabrication and characterization', *Composites Part B: Engineering*. Elsevier Ltd, 159, pp. 269–283. doi: 10.1016/j.compositesb.2018.09.080.
- Deepalekshmi, P. *et al.* (2013) 'Advances in Elastomers: Their Blends and Interpenetrating Networks-State of Art, New Challenges and Opportunities', in, pp. 1–9. doi: 10.1007/978-3-642-20925-3_1.
- Deng, H. X., Gong, X. L. and Wang, L. H. (2006) 'Development of an adaptive tuned vibration absorber with magnetorheological elastomer', *Smart Materials and Structures*. IOP Publishing, 15(5), p. N111. doi: 10.1088/0964-1726/15/5/N02.
- Gavrilovici, A. M. *et al.* (2019) 'Magnetodielectric effects in magnetorheological elastomers based on polymer fabric, silicone rubber, and magnetorheological suspension', *Advances in Polymer Technology*. John Wiley and Sons Inc., 2019. doi: 10.1155/2019/1983547.
- Hapipi, N. *et al.* (2019) 'The field-dependent rheological properties of plate-like carbonyl iron particle-based magnetorheological elastomers', *Results in Physics*. Elsevier B.V., 12, pp. 2146–2154. doi: 10.1016/j.rinp.2019.02.045.
- Hoang, N. *et al.* (2013) 'Development of a torsional dynamic absorber using a magnetorheological elastomer for vibration reduction of a powertrain test rig', *Journal of Intelligent Material Systems and Structures*, 24(16), pp. 2036–2044. doi: 10.1177/1045389X13489361.
- Jiffri, S. *et al.* (2014) 'Feedback Linearisation for Nonlinear Vibration Problems'. doi: 10.1155/2014/106531.
- Lee, C. W., Kim, I. H. and Jung, H. J. (2018) 'Fabrication and Characterization of Natural Rubber-Based Magnetorheological Elastomers at Large Strain for Base Isolators', *Shock and Vibration*. Hindawi Limited, 2018. doi: 10.1155/2018/7434536.
- Lee, K.-H., Park, J.-E. and Kim, Y.-K. (2019) 'Design of a stiffness variable flexible coupling using magnetorheological elastomer for torsional vibration reduction', *Original Article Journal of Intelligent Material Systems and Structures*, 30(15), pp. 2212–2221. doi: 10.1177/1045389X19862378.

- Li, H. *et al.* (2015) 'Robustness study of the pounding tuned mass damper for vibration control of subsea jumpers', *Smart Materials and Structures*. Institute of Physics Publishing, 24(9), p. 095001. doi: 10.1088/0964-1726/24/9/095001.
- Liao, G. J. *et al.* (2012) 'Development of a real-time tunable stiffness and damping vibration isolator based on magnetorheological elastomer', *Journal of Intelligent Material Systems and Structures*, 23(1), pp. 25–33. doi: 10.1177/1045389X11429853.
- Papagiannopoulos, G. A. and Hatzigeorgiou, G. D. (2011) 'On the use of the half-power bandwidth method to estimate damping in building structures', *Soil Dynamics and Earthquake Engineering*, 31(7), pp. 1075–1079. doi: 10.1016/j.soildyn.2011.02.007.
- Perales-Martinez, I. A. *et al.* (2017) 'Enhancement of a magnetorheological PDMS elastomer with carbonyl iron particles', *Polymer Testing*. Elsevier Ltd, 57, pp. 78–86. doi: 10.1016/j.polymertesting.2016.10.029.
- Praveen Shenoy, K. *et al.* (2018) 'Experimental investigation of torsional vibration isolation using Magneto Rheological Elastomer', *MATEC Web of Conferences*, 144, pp. 1–11. doi: 10.1051/mateconf/201714401007.
- Samal, S. *et al.* (2020) 'Magneto-rheological elastomer composites. A review', *Applied Sciences (Switzerland)*. MDPI AG. doi: 10.3390/app10144899.
- Sun, S. *et al.* (2018) *Development of a nonlinear adaptive absorber based on magnetorheological elastomer*, *Journal of Intelligent Material Systems and Structures*. doi: 10.1177/1045389X17733053.
- Syam, T. *et al.* (2020) '3D numerical modelling and analysis of a magnetorheological elastomer (MRE)', *Journal of Vibroengineering*, pp. 1–15. doi: 10.21595/jve.2020.21377.
- Syam, T. M. I. and Muthalif, A. G. A. (2020) 'Hysteresis behaviour of different magnetorheological elastomer models: Modelling and simulation', *Vibroengineering Procedia*, 31, pp. 7–14. doi: 10.21595/vp.2020.21408.
- Tao, Y. *et al.* (2018) 'Design and experimental research of a magnetorheological elastomer isolator working in squeeze/elongation–shear mode', *Journal of Intelligent Material Systems and Structures*. SAGE Publications Ltd, 29(7), pp. 1418–1429. doi: 10.1177/1045389X17740436.
- Wang, Q. *et al.* (2017) 'Study on an improved variable stiffness tuned mass damper based on conical magnetorheological elastomer isolators', *Smart Materials and Structures*. Institute of Physics Publishing, 26(10), p. 105028. doi: 10.1088/1361-665X/aa81e8.
- Wen, H. *et al.* (2018) 'The transmissibility of a vibration isolation system with ball-screw inerter based on complex mass', *Journal of Low Frequency Noise, Vibration and Active Control*, 37(4), pp. 1097–1108. doi: 10.1177/1461348418769650.

Table 1. MRE properties used (ALCHEMIX® PU 7231 | Enhanced Reader, 2010)

Physical Properties @ 25 °C	RTV136-60
Mixing Ratio (by weight)	100:2~5
Mixing Viscosity (cps)	35000-40000
Cure Time (Hours)	8-10
Tensile Strength (MPa)	≥ 4.3
Elongation (%)	≥ 450
Shrinkage (%)	≤ 0.2

Table 2. Summary of the results obtained.

Applied current (A)	Magnetic Field (mT)	f_n (Hz)	Torsional stiffness (Nm/rad)	Torsional damping coefficient (kg.m ² /rad.s)
0	0	26.5	37.42702	0.042755
1	3.16	28	41.78396	0.071936
2	8.32	30.52	49.64352	0.061316
3	12.38	34	61.61002	0.071289

Figure . 1. Simplified fabrication schematic of isotropic and anisotropic MREs (Hapipi *et al.*, 2019).

Figure . 2. Representation of the input (force) and the output (acceleration) responses for the mathematical model derivation.

Figure . 3. (a) 3D printed mold used to manufacture the MRE, and (b) fabricated MRE sample and (c) aluminum coupler details.

Figure .4. (a) Top view of the experimental setup, (b) accelerometer orientation showing the concept of the tangential acceleration, and (c) location of the impact and orientation showing the force-torque relation.

Figure . 5. Experimental Setup.

Figure . 6. MRE fitted within the coupling.

Figure . 7. Schematic representation of the experimental setup.

Figure . 8. Magnetic field vs applied current.

Figure . 9. Transfer function vs excitation frequency.

Figure . 10. Transfer function vs excitation frequency in the low-frequency range.

Figure . 11. (a) Torsional stiffness vs magnetic field, (b) Torsional damping coefficient vs magnetic field, and (c) Natural frequency vs magnetic field.