

Design and Fabrication of Hybrid Operational Modes Magnetorheological Mount

Shuangyi Liang^{1,2}, Chen Chen², Yibu Zhao², Kwanchai Kraitong^{1*}

¹Department of Mechanical Engineering, Faculty of Engineering, Naresuan University, Phitsanulok 65000, Thailand

²Faculty of Mechanical and Automotive Engineering, Guangxi University of Science and Technology, Liuzhou 545006, China

*Corresponding author's email: kwanchaik@nu.ac.th

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ABSTRACT

Magnetorheological (MR) mounts, developed based on the field-dependent damping characteristics of magnetorheological fluids (MRFs), are key vibration isolation components in vehicle powertrain systems and are capable of reducing energy losses induced by structural vibrations. Existing studies have mainly focused on single-mode MR mounts, whose vibration isolation performance is considerably inferior to that of hybrid-mode designs. In this study, a hybrid-mode MR mount integrating both squeeze and flow operational modes is proposed. The proposed design incorporates upper and lower Symmetrical squeeze gaps together with four circumferentially distributed flow channels. The structural configuration of the MR mount is presented in detail, and a static structural finite element analysis is performed to evaluate the maximum principal stresses of the load-bearing components under compressive and tensile loading conditions, which are 368.5 MPa and 230.61 MPa, respectively. The results demonstrate that the designed MR mount exhibits a maximum load-bearing capacity of 7000 N. Subsequently, a comprehensive magnetic circuit design is carried out. The magnetic circuit design indicates that, to ensure effective operation of the MR mount, the number of coil turns must exceed 201 in the squeeze mode and 316 in the flow mode. Finally, the assembly process of the MR mount prototype is described in detail, including the installation sequence of the key structural components, the injection procedure of the MRFs, and the final sealing and fixation steps, thereby demonstrating the feasibility and practical manufacturability of the proposed hybrid-mode MR mount design.

1. Introduction

Magnetorheological fluids (MRFs) are smart materials characterized by variable yield stresses, whose rheological properties can change within milliseconds under the influence of an external magnetic field [1-3]. This remarkable property has inspired numerous applications, including the use of MRFs in automotive mounts for engine vibration isolation. The engine mount ensures that the engine consistently operates within an efficient range and minimizes unnecessary energy losses by optimizing vibration and NVH (noise, vibration, and harshness) performance, thereby contributing to energy saving. Furthermore, by providing effective vibration isolation, the mount establishes a stable and smooth operating environment for various advanced technologies designed to enhance energy efficiency. MR mounts are devices capable of generating controllable vertical forces to attenuate vibration [4,5].

The first critical step in the design of an MR mount is to determine the operational mode of the MRF. The operational modes of MRFs are generally classified into valve mode (i.e., flow mode), squeeze mode, and shear mode [6]. In the flow mode, MRF is driven by the pressure exerted on the fluid within a flow channel. The applied magnetic field acts perpendicular to the flow channel, and changes in the external magnetic field result in corresponding variations in flow resistance, enabling the magnetic field to regulate the vibration damping force [7]. In the squeeze mode, two parallel plates are employed, with one plate fixed and the gap between

the plates filled with MRF. The magnetic field is oriented perpendicular to the plates, and the MRF is forced to flow around the plates when compressed, causing the flow to be blocked along the field lines due to the formation of MRF chains [8]. This results in the piston being constrained, thereby achieving controlled magnetic field damping. In the shear mode, the magnetic field is also perpendicular to two parallel plates, but one plate moves relative to the other, causing the MRF to deform and flow under viscous resistance [9]. Among the three operational modes, the squeeze mode typically produces the highest damping force [3]. Moreover, the squeeze mode requires smaller motion strokes, making it well suited for systems with limited displacement. Engine vibration displacements are generally less than 0.5 mm, classifying the engine as a small-vibration system; therefore, MR mounts for such applications are often designed based on the squeeze mode.

Many researchers have focused on MR mounts operating primarily in a single mode. Y. Shiao and T.-L. Huynh [10] designed a multi-pole squeeze-mode MR mount incorporating six sets of copper coils. Qiang cai et al. [11] addressed the issue of premature magnetic saturation in the damping channels of radial-flow MR mounts with large magnetic core inner diameters, and proposed a flow-mode MR mount featuring a conical damping channel structure. Zhaoxue Deng et al. [12] designed a flow-mode MR mount intended for low-frequency operating conditions, such as automotive start-stop scenarios.

Q. Liu et al. [13] proposed an internal bypass valve-mode (i.e., flow mode) MR mount featuring an extended controllable range of dynamic stiffness and an increased operating stroke.

Conventional MR mounts operate in a single mode, and the range of vibration isolation force provided by the damping and spring stiffness is constrained by the geometric configuration associated with that mode [14]. In contrast, hybrid-mode mounts can operate simultaneously in two modes, whereby the vibration isolation effects of both modes are superimposed, enabling adaptation to a wider range of excitation amplitudes and frequencies [14,15]. Combining different operational modes can further exploit the advantages of each mode [16]. T. M. Nguyen et al. [15] designed a squeeze-flow mixed-mode MR mount in which the individual operational modes function independently of each other. Huang Jincheng et al. [4] proposed a flow-squeeze mixed-mode MR mount suitable for heavy-load operating conditions. Deng, Z [17] designed a squeeze-shear mixed-mode MR mount featuring inclined annular damping channels. In most of the previously developed hybrid-mode MR mounts, the bidirectional vibration excitation of the vehicle engine has not been adequately considered.

Previous studies on MR mounts have demonstrated their adjustable damping characteristics, making them promising for vibration isolation applications. However, these mounts still exhibit several inherent limitations, including restricted damping force output and limited adaptability to complex bidirectional excitations that are typically present in vehicle engine vibrations. In particular, conventional MR mounts often fail to provide sufficient damping in both compression and rebound directions simultaneously, which reduces their effectiveness under the reciprocating and multidirectional excitations generated by vehicle engines. To address these challenges, this study proposes a novel hybrid MR mount that integrates both squeeze and flow operational modes, designed specifically to improve bidirectional vibration isolation performance. The proposed MR mount features symmetric dual-squeeze gaps and four flow channels, which allow it to generate effective damping forces during both compression and rebound phases. This symmetric dual-squeeze configuration ensures that the mount can efficiently respond to reciprocating excitations and maintain stable vibration isolation under the multidirectional load characteristic of vehicle engine operation. The structural design of the hybrid MR mount is described in detail, with a particular focus on its principal load-bearing components, including the upper and lower squeeze plates, the magnetic isolation plates, the piston rod, the upper connecting screw, the upper and lower partitions, all of which contribute to its mechanical robustness.

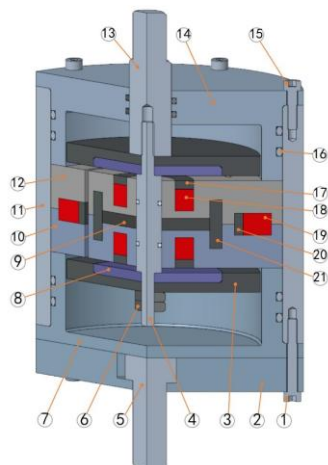


Fig. 1 Section view of the hybrid operational modes MR mount.

To verify the feasibility of the design, structural finite element analysis is conducted to evaluate stress distribution and load-bearing capacity under operational conditions. Additionally, a comprehensive magnetic circuit design is presented to optimize the MR fluid response within both the squeeze and flow modes. Based on these analyses, a prototype of the proposed hybrid MR mount is successfully fabricated, demonstrating the practicality of the design. The integration of symmetric dual-squeeze gaps and multiple flow channels enables the mount to provide enhanced vibration isolation for vehicle engines subjected to complex bidirectional vibration excitations, addressing a key limitation of existing MR mount designs and laying a solid foundation for subsequent experimental validation and performance optimization.

2. Design of the proposed hybrid operational modes MR mount

2.1 Structure and working principle

A MR mount with a dual-squeeze gaps and four-channels was designed based on the vibration isolation requirements of vehicle engines, featuring hybrid operation modes of squeeze and flow modes. Fig.1 shows the cut-out view of the hybrid-mode MR mount in this study, which consists of (1) lower screw and (2) screw platen, (3) magnetic isolation plate of lower squeeze plate, (4) piston rod and (5) lower connecting screw and (6) screw nut, (7) lower cover, (8) lower squeeze plate, (9) intermediate magnetic isolation disk and (10) lower partition, (11) shell, (12) upper partition and (13) upper connecting screw, (14) upper cover and (15) upper screw, (16) sealing ring, (17) non-magnetic ring of squeeze mode coil and (18) squeeze mode coil, (19) flow mode coil and (20) non-magnetic ring of flow mode coil, (21) intermediate magnetic isolation cylinder. The screw platen (2) and lower connecting screw (5) serve as the connecting elements for securing the prototype to the experimental apparatus.

The upper connecting screw (13) is machined with external threads at one end and internal threads at the other. The externally threaded end is fastened to the engine, whereas the internally threaded end is coupled to the piston rod (4). Components (3), (4), (6), (8), and (13) are capable of executing small-amplitude reciprocating vertical motions in unison. The lower squeeze gap is formed between the lower squeeze plate (8) and the lower partition (10) as well as magnetic isolation plate of lower squeeze plate (3). The lower squeeze gap and the upper squeeze gap are symmetrically arranged along the interface between the upper and lower partitions. The upper partition (12) and lower partition (10) are configured to cooperate in forming four arc-shaped flow channels, which connect the upper and lower liquid chambers, as illustrated in Fig. 2 and 3.

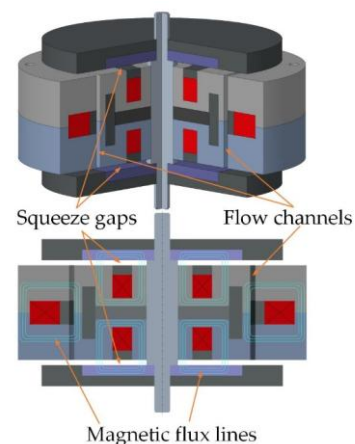


Fig. 2 Schematic diagram of the squeeze gaps and flow channels.

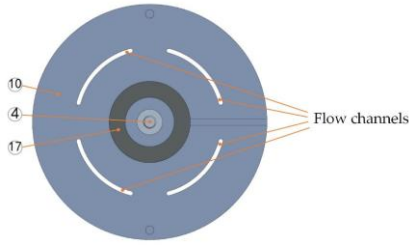


Fig. 3 Distribution of the flow channels on the lower partition.

The vibration of the vehicle engine is transmitted to the MR mount via the upper connecting screw (13), compelling the moving components—such as the piston rod (4) and the upper and lower squeeze plates (8)—to vibrate synchronously. The MR fluid within the squeeze gaps is compressed by the squeeze plates and subsequently displaced toward the surrounding regions. Under the magnetization effect of the annular magnetic field generated by the squeeze mode coil (18)—that is, the magnetic field present in the squeeze gaps as depicted in Fig. 2—the MR fluid undergoes a particle chain formation, transitioning into a solid-like state. This transformation results in an increase in damping force, thereby attenuating engine vibrations. In this stage, the MR fluid operates in the squeeze mode. Following compression by the squeeze plates, the MR fluid in the upper and lower liquid chambers is forced to pass through the flow channels. Within these channels, the magnetic field generated by the flow mode coil (19)—namely, the magnetic field in the flow channels shown in Fig. 2 and Fig. 3—induces further magnetization of the MR fluid, again initiating particle chain formation and producing a solid-like rheological state. This process similarly enhances damping force, dissipating vibrational energy, and here the MR fluid operates in the flow mode.

The combined damping effect generated simultaneously in the symmetrical dual-squeeze gaps and the four flow channels significantly improves the vibration isolation performance of the MR mount.

2.2 Structural design of the principal load-bearing components

Within this structural configuration, the squeeze plate, the magnetic isolation plate associated with the squeeze plate, the partition, the piston rod, and the upper connecting screw together form the primary load-bearing components of the MR mount. These components are critical in ensuring the mechanical integrity and overall stability of the mount under operational loads. In the following section, a detailed description of the structural design of each of these components is provided, including the rationale behind their geometrical configurations, material selections, and mechanical interconnections, accompanied by the corresponding calculations, dimensional specifications, and considerations for stress distribution and load-bearing performance.

2.2.1 Design of the piston rod

The piston rod is centrally located within the squeeze mode coil. To achieve magnetic field isolation and improve magnetic field utilization, the piston rod is fabricated from SUS304, a non-magnetic material. Threads are provided at both ends of the piston rod to facilitate the connection with the upper connecting screw and the fixation of the lower squeeze plate. Since squeeze plates are mounted on both ends of the piston rod, shoulders are machined on the rod to achieve precise positioning of the upper and lower squeeze plates. The central section of the piston rod is sealed to the upper and lower partitions by means of two O-ring seals. The diameter of the piston rod, d , is determined based on the strength requirement given in Eq. (1):

$$d \geq \sqrt{\frac{4F}{\pi[\sigma]}} \quad (1)$$

where, d represents the diameter of the piston rod; F denotes the force applied to the piston rod. The isolation target in this study is a four-stroke, four-cylinder gasoline engine with a weight of approximately 162 kg (excluding vehicle engine oil) [18]. Taking into account the weight of the vehicle engine oil and allowing for an appropriate design margin, the force applied to the piston rod is considered to be 4000 N; and $[\sigma]$ is the allowable stress of the material. Since SUS304 is a brittle material with a tensile strength limit $[\sigma_b] = 520 \text{ MPa}$, and a safety factor $n = 2$ is adopted, the allowable stress $[\sigma]$ is calculated as:

$$[\sigma] = \frac{[\sigma_b]}{n} = 260 \text{ MPa} \quad (2)$$

Substituting the aforementioned parameters into Eq. (1) allows for the calculation of the piston rod diameter, which is determined to be 4.33 mm.

Therefore, the minimum diameter of the piston rod is selected as $d_1 = 6 \text{ mm}$, to satisfy the strength requirements. Other dimensional calculations are omitted here for brevity. The engineering schematic and main dimensions of the piston rod are shown in Fig. 4 and Table 1.

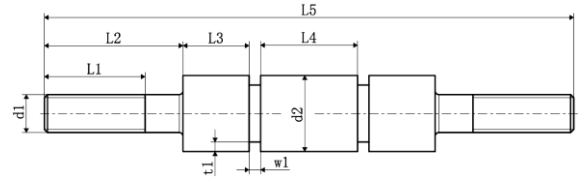


Fig. 4 Diagram of the piston rod.

Table 1 The main dimensions of the piston rod.

Parameter	Numerical Value(mm)
L1	16
L2	22
L3	10.5
d1	6
d2	12

2.2.2 Design of the squeeze plate and the magnetic isolation plate

The squeeze plate is located on one side of the squeeze gap. To ensure that the magnetic flux lines pass perpendicularly through the squeeze gap, the squeeze plate material should be a magnetic conductive material with low magnetic reluctance. Meanwhile, during the operation of the MR mount, the squeeze plate exerts a squeezing force on the MRF and must possess certain impact resistance. Therefore, in this study, the squeeze plate material is selected as 45# steel.

The magnetic isolation plate associated with the squeeze plate prevents the magnetic flux lines generated by the squeeze coil from passing through the squeeze plate into the upper chamber, which would otherwise affect the magnetic field intensity in the squeeze gap. Therefore, the magnetic isolation plate of the squeeze plate must be made of a magnetic shielding material. Common magnetic shielding materials include copper, aluminum alloys, and stainless steel. In this structure, since the isolation plate must withstand pressures caused by engine vibrations and requires adequate strength, stainless steel is selected as the material for the isolation plate. Referring to commonly available stainless-steel types on the market, the austenitic stainless steel SUS304, with grade 06Cr19Ni10, is chosen for its excellent magnetic isolation properties.

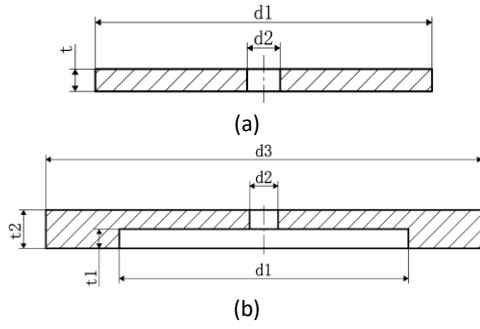


Fig. 5 (a) Squeeze plate diagram and (b) magnetic isolation plate diagram.

Table 2 The main dimensions of the squeeze plate and isolation plate.

Parameter	Numerical Value(mm)
d1	61
d2	6
t	4
t1	4

Considering both the low magnetic reluctance requirement of the squeeze plate and ensuring the magnetic isolation effect of the squeeze plate's magnetic isolation plate, the designed squeeze plate and its magnetic isolation plate are shown in Fig. 5, with main dimensions detailed in Table 2.

2.2.3 Design of the partition

Three sets of excitation coils are wound into coil slots on the upper and lower partitions. The material surrounding the coils should exhibit low magnetic reluctance; therefore, 45# steel is selected. The upper partition includes an additional lead-out hole compared to the lower partition. Fig. 6 shows the structure and size definition of the upper partition. The design rationale for the upper partition is presented below.

In this structure, the thickness of the partition is an important dimension to consider. The length of the flow channel is related to the thickness of the partition. An excessively short damping channel length reduces the number of magnetic flux lines passing vertically through the channel, thereby diminishing the chain formation effect of the MRF within the channel and consequently affecting the damping force output of the entire device. The partition must be machined with coil slots and grooves to accommodate the magnetic isolation components; hence, its thickness cannot be too small to ensure sufficient structural strength. Furthermore, the assembly of the upper and lower partitions, piston rod, and squeeze plates determines the height of the squeeze gap.

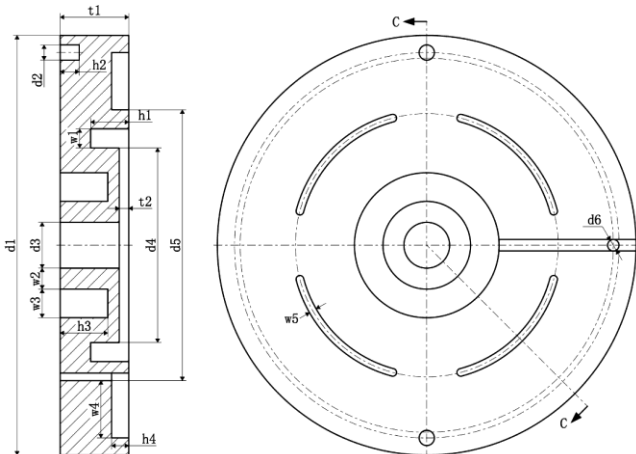


Fig. 6 Diagram of the upper partition.

Considering the engine excitation amplitude range of 0.5 mm to 1.5 mm [19], the gap dimension is increased accordingly; however, an excessively large gap height would increase magnetic reluctance in the squeeze region, adversely affecting the perpendicular passage of magnetic flux lines through the squeeze region [20]. Therefore, the squeeze gap height is set to 2 mm [20]. Based on the piston rod dimensions designed in Section 2.2.1, the partition thickness is set to $t_1 = 18 \text{ mm}$. According to the maximum diameter of the piston rod, the diameter of the through-hole matching the piston rod is determined as $d_3 = 12 \text{ mm}$.

Magnetic flux lines must pass around the coil slots, and the magnetic reluctance at the squeeze coil location impacts the magnetic field intensity in the squeeze gap. The width w_2 is one of the critical parameters influencing this reluctance. Increasing w_2 raises the magnetic reluctance, whereas reducing it compromises the structural strength. Increasing w_2 raises the magnetic reluctance, whereas reducing it compromises the structural strength. Balancing these considerations, w_2 is chosen as 5.5 mm. The squeeze coil is equipped with a magnetic isolation ring, which is interference-fitted with the annular coil slot on the partition to clamp the squeeze coil, preventing direct contact between the MR fluid and the coil. The high magnetic reluctance of the isolation ring ensures that the magnetic flux lines generated by the squeeze coil pass perpendicularly through the squeeze region, allowing the MRF in the squeeze gap to effectively form chain structures.

The partitions, intermediate magnetic isolation cylinder, and intermediate magnetic isolation plate cooperate to separate the two squeeze-region magnetic fields and the flow-region magnetic field into individual magnetic fields, thereby preventing mutual interference and improving magnetic field utilization efficiency. Dimensions d_4 , t_2 , w_1 , and h_1 determine the structure of the intermediate magnetic isolation cylinder and plate. A gap of 6.5 mm is reserved outside the squeeze coil for magnetic flux passage, thus

$$d_4 = 2 \times \left(\frac{d_3}{2} + w_2 + w_3 + 6.5 \right) = 51 \text{ mm} \quad (3)$$

Given that the MRF has higher magnetic reluctance than magnetic conductive materials, the reluctance of the flow channel increases with its width. Therefore, the flow channel width should not be too large. However, a channel that is too narrow may induce oil film effects that slow the MRF flow. Typically, MR damping channels have widths ranging from 0.8 to 2.0 mm; thus, the flow channel width w_5 is chosen as 2 mm. Outside the groove of the intermediate magnetic isolation cylinder, a 3 mm thick magnetic conductive material is reserved to allow magnetic flux lines generated by the flow-mode coil to pass. Consequently,

$$d_5 = d_4 + 2w_1 + 2 \times 3 + 2w_5 = 71 \text{ mm} \quad (4)$$

The flow mode incorporates a magnetic isolation ring, analogous to that of the squeeze mode coil, with a thickness of 3.5 mm. Considering the isolation ring thickness, the flow mode coil slot width w_4 is 15 mm, and slot depth h_4 is 4.5 mm. A 4.5 mm gap is reserved outside the coil slot for magnetic flux passage, resulting in an outer partition diameter

$$d_1 = d_5 + 2w_4 + 2 \times 4.5 = 110 \text{ mm} \quad (5)$$

Since the flow mode coil is clamped between the upper and lower squeeze plates within an enclosed area, a through-hole is machined on the end face of the upper partition to lead out the flow mode coil wiring.

Table 3 Dimensions of the upper partition.

Parameter	Numerical Value(mm)
d1	110
d2	4
d3	12
d4	51
d5	71
d6	3
w1	3.5

To prevent rotation of the partitions during vibration, which could affect wiring, a blind positioning pin hole is machined on the partition end face. This pin hole mates with the end cover to restrict the rotation of the upper and lower partitions. Table 3 presents the definitions and corresponding values of all dimensions of the partition.

2.2.4 Design of the upper connecting screw

The upper connecting screw serves as the component that links the MR mount to the vibration source and has no magnetic isolation requirements; therefore, its material is specified as 45# steel. Since the MR mount prototype is attached to the suspension lugs of the test bench via M16 threads, the nominal diameter of the external thread is set to $d_1 = 16 \text{ mm}$, with a thread depth of $L_1 = 20 \text{ mm}$. A 2 mm shaft shoulder is reserved for lug positioning, yielding $d_2 = d_1 + 2 \times 2 = 20 \text{ mm}$. The bottom threaded hole diameter is determined with reference to the piston rod external thread diameter in Section 2.2.1, and is taken as $d_3 = 6 \text{ mm}$. When determining the thread depth L_2 , the thickness of the magnetic isolation plate associated with the squeeze plate and the distance from the piston rod end face to the shaft shoulder are considered, resulting in $L_2 = 17 \text{ mm}$. The internal thread hole depth L_3 is set to $L_3 = L_2 + 0.5 \times d_3 = 20 \text{ mm}$. A clearance is maintained between the protruding end of the connecting screw and the top surface of the upper end cover, for which $L_4 = 59 \text{ mm}$ is adopted. Fig. 7 illustrates the structural schematic of the upper connecting screw, while Table 4 summarizes the dimensional definitions and values.

The subsequent section will present the simulation analysis, with the materials of the key components summarized in Table 5.

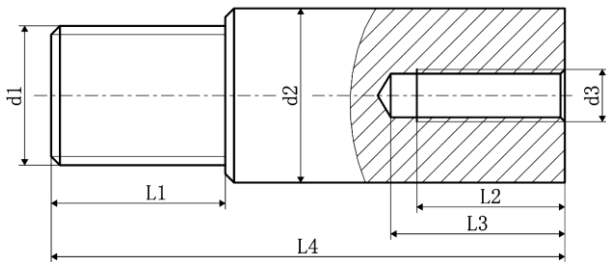


Fig. 7 Diagram of the upper connecting screw.

Table 4 Dimensions of the upper connecting screw.

Parameter	Numerical Value(mm)
d1	16
d2	20
d3	6
L1	20
L2	17
L3	20
L4	59

Table 5 The materials of the key components.

Material	Components NO. (Fig. 1)
Copper	(18) (19)
45# steel	(2) (7) (8) (10) (12) (13) (14)
SUS304	(3) (4) (11) (17) (20) (21)

2.2.5 Structural finite element analysis

To evaluate the load-carrying capacity of the designed structure, a finite element analysis (FEA) of the MR mount was conducted using the Static Structural module in ANSYS. The material properties of the components of the designed MR mount are listed in Table 6.

Static structural analysis models for the upper connecting screw under both compressive and tensile loading conditions were developed in ANSYS/Workbench, with material properties from Table 6 assigned in the Engineering Data module. To reduce computational cost, the analysis was limited to the key load-bearing components. The simplified simulation model of these key components was imported into the geometry module. Use the smart grid division with tetrahedral element type and size of 5 mm. For components subjected to compression and tension, the element size was refined to 2.5 mm. The generated grid is shown in Fig. 8. After meshing, the total number of elements and nodes in the model under the compressive loading condition were 186,383 and 301,842, respectively, whereas under the tensile loading condition, the model comprised 185,866 elements and 305,139 nodes.

Table 6 The material properties of MR mount components.

Project	Copper	45# steel	SUS304
Elastic modulus (N/m^2)	1.17×10^{11}	2.09×10^{11}	1.9×10^{11}
Poisson's ratio	0.3	0.269	0.29
Density (Kg/m^3)	8.94×10^3	7.89×10^3	8×10^3
Shear modulus (N/m^2)	4.68×10^{10}	8.23×10^{10}	7.5×10^{10}
Yield strength (N/m^2)	6.2×10^7	3.55×10^8	2.05×10^8
Tensile strength (N/m^2)	4.16×10^8	6×10^8	5.2×10^8

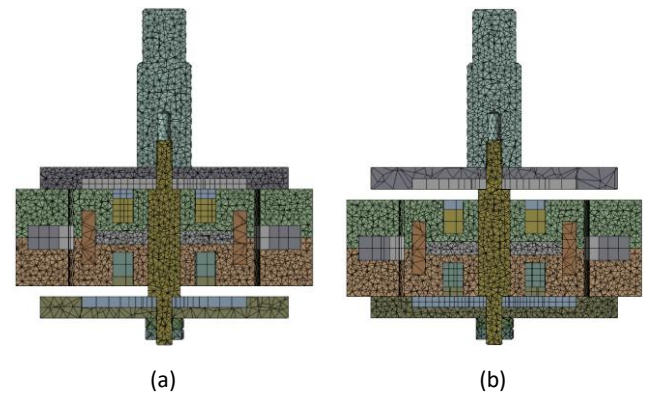


Fig. 8 Meshing diagram: (a) compressive loading conditions; (b) tensile loading conditions.

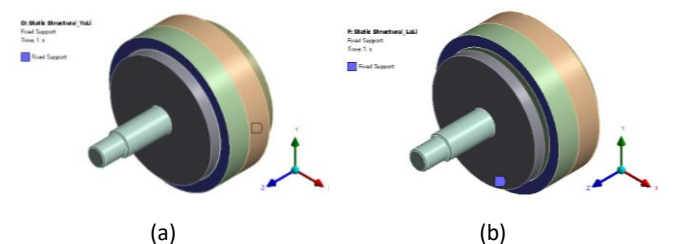


Fig. 9 Diagram of fixed constraint: (a) compressive loading conditions; (b) tensile loading conditions.

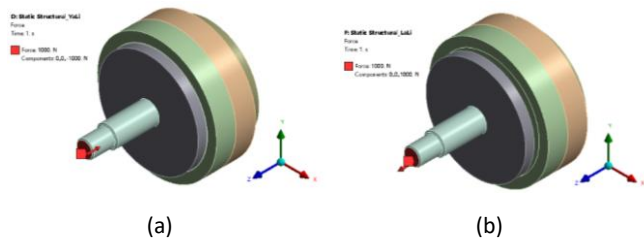


Fig. 10 Diagram of the load application model: (a) compressive loading conditions; (b) tensile loading conditions.

Fixed constraints were applied to the annular contact surfaces between the upper and lower partitions and the covers, as shown in Fig. 9.

A stepwise load, incremented based on a base value of 1000 N, was applied to the model. A time-dependent force was imposed on the end face of the upper connecting screw, wherein the Z-axis component of the applied load was defined as “ $-1000 \times \text{time}$ ” for compression and “ $=1000 \times \text{time}$ ” for tension, as illustrated in Fig. 10.

Fig. 11 presents the maximum principal stress of the entire model under different loading conditions. The maximum principal stress at Step 10, corresponding to a load of 10,000 N, is shown in Fig. 12 and 13.

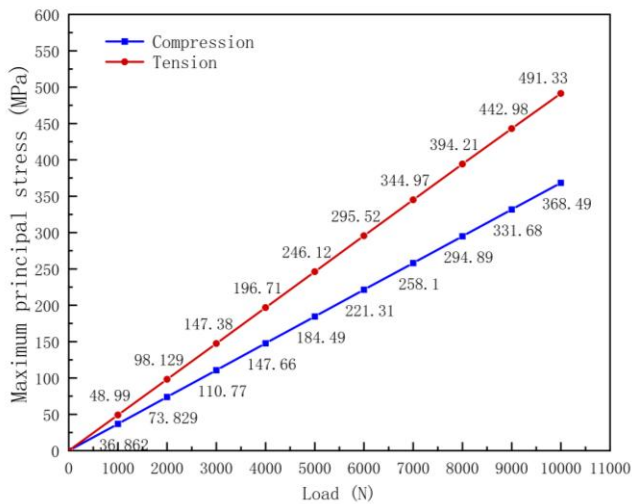


Fig. 11 Maximum principal stress curve.

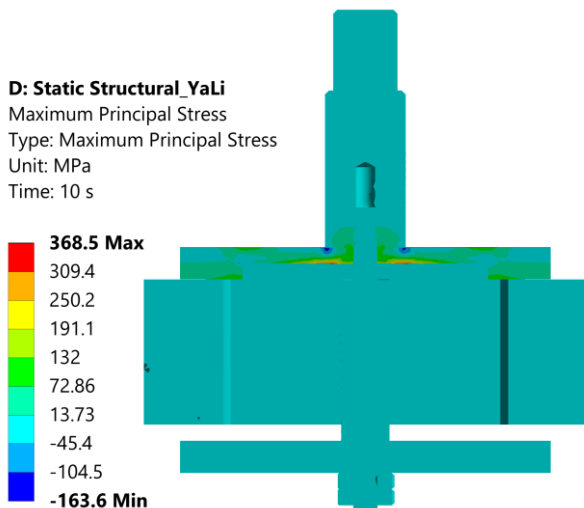


Fig. 12 Diagram of maximum principal stress under compressive loading conditions (10,000 N).

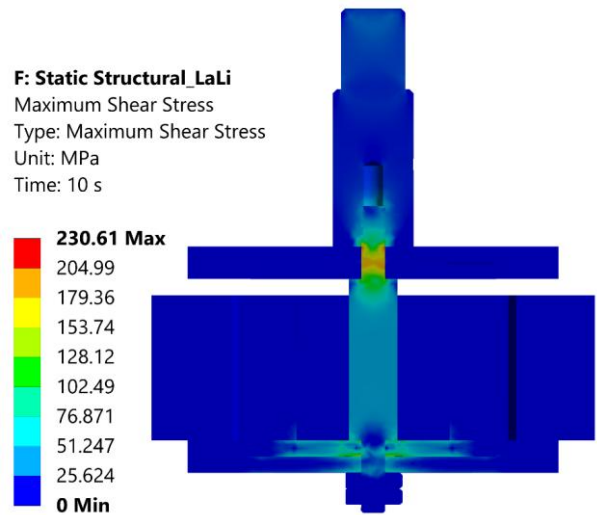


Fig. 13 Diagram of maximum principal stress under tensile loading conditions (10,000 N).

From the maximum principal stress distribution obtained through finite element analysis, it can be observed that under compressive loading, the maximum stress reaches 368.5 MPa, occurring at the lower end surface of the magnetic isolation plate associated with the upper squeeze plate. In contrast, under tensile loading, the maximum stress is measured at 230.61 MPa, located at the interface between the outer cylindrical surface of the piston rod and the magnetic isolation plate of the upper squeeze plate. These stress concentrations indicate the critical regions where the mount components are most susceptible to deformation or potential failure under operational loads. Finite element results indicate that the maximum principal stress under compressive loading is significantly higher than that under tensile loading. This is mainly attributed to the fact that, under compression, the external load is transmitted from the engine to the piston rod through the upper connecting bolts and subsequently transferred to the upper squeeze plate, where it is concentrated at the lower end of the magnetic isolation plate, leading to strong geometric constraints and rigid surface contact that induce pronounced stress concentration and local bending effects. In contrast, under tensile loading, the force is primarily transmitted through the interface between the piston rod and the magnetic isolation plate, allowing partial stress redistribution and resulting in a lower peak principal stress.

As listed in Table 5, both the magnetic isolation plate of the squeeze plate and the piston rod of the MR mount are fabricated from SUS304 stainless steel, chosen for its non-magnetic properties, high strength, corrosion resistance, and suitability for repeated cyclic loading conditions. According to the piston rod design specifications described in Section 2.2.1 and the material properties provided in Table 6, and considering a safety factor of 2, the allowable stress for SUS304 is determined to be $[\sigma] = 260 \text{ MPa}$. One fundamental prerequisite for the safe operation of the proposed MR mount is that the maximum stresses obtained from the numerical finite element analysis should remain below this allowable stress limit, ensuring both structural reliability and operational safety. Based on the results from the 10-step FEA simulation presented in Fig. 11, it can be concluded that under compressive loading, the magnetic isolation plate of the upper squeeze plate can safely sustain a load up to 7000 N without exceeding the material's allowable stress. Similarly, under tensile loading, the piston rod demonstrates the capacity to withstand a load exceeding 10,000 N, indicating

a substantial margin of safety in tension. Since the MR mount is designed to operate under both tensile and compressive loading conditions during engine-induced vibrations, the ultimate load capacity of the proposed MR mount is conservatively determined to be 7000 N, which corresponds to the limiting compressive stress condition. This assessment confirms that the structural design and material selection of the MR mount are adequate to ensure safe and reliable operation under the anticipated service loads.

3. Magnetic circuit design

Fig. 14 illustrates the magnetic circuit diagram of the hybrid operational modes MR mount, while Table 7 provides the introduction of definitions and numerical of its structural parameters. The designed magnetic circuit comprises three excitation coils: the upper and lower squeeze mode coils and a flow mode coil.

As depicted in Fig. 14, the magnetic flux lines generated by the flow mode coil traverse vertically through the flow channels within the upper and lower partitions, thereby forming a closed magnetic loop. This loop encompasses six magnetic reluctances, denoted as R11 to R16. The configuration also includes two squeeze mode coils. The magnetic flux lines produced by the upper squeeze mode coil originate in the upper partition, pass vertically through the upper squeeze gap to reach the upper squeeze plate, and subsequently return through the same gap from the upper squeeze plate back to the upper partition, thus establishing a closed magnetic loop. This loop similarly consists of six magnetic reluctances, denoted as R21 to R26. The magnetic flux path of the lower squeeze mode coil exhibits a mirror-symmetric configuration relative to that of the upper squeeze mode coil. Fig. 15 presents the equivalent magnetic circuit diagram of the hybrid operational modes MR mount.

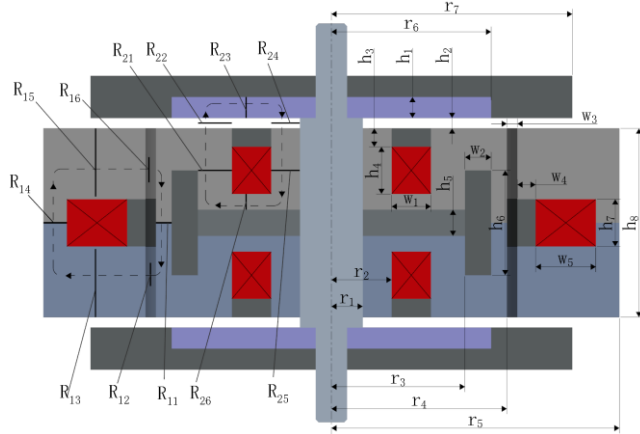


Fig. 14 Magnetic circuit of hybrid operational mode MR mount.

Table 7 Dimension parameters of hybrid operational mode MR mount

Parameter	Numerical Value (mm)	Parameter	Numerical Value(mm)
Partition radius (r_5)	55	Magnetic isolation cylinder height (h_6)	20
Squeeze plate radius (r_6)	30.5	Height of flow mode coil (h_7)	9
Magnetic isolation plate radius(r_7)	46	Height of upper and lower partitions(h_8)	36
Squeeze plate height (h_1)	4	Width of squeeze mode coil (w_1)	7.5
Squeeze gap height (h_2)	2	Magnetic isolation cylinder thickness (w_2)	5
No-magnetic ring height of squeeze mode coil (h_3)	3.5	Width of flow channel(w_3)	2
Height of squeeze mode coil (h_4)	9	No-magnetic ring width of flow mode coil (w_4)	3.5
Intermediate magnetic isolation disk height (h_5)	5	Width of flow mode coil (w_5)	11.5

Table 8 The main characteristic parameters of the MRF-J25T magnetorheological fluid[22].

Project	Parameter
Density	2.56 g/cm ³
Zero magnetic field viscosity	0.8 Pa·s
Shear stress (5000 Gs)	50 KPa
Temperature range	-40~130 °C
Magnetization performance (Ms)	365.29 KA/m

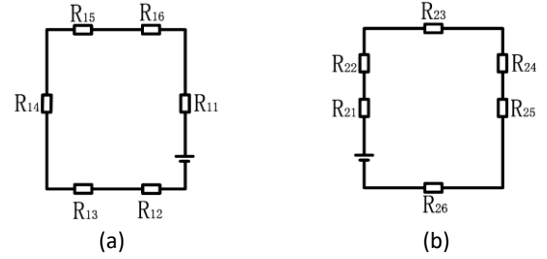


Fig. 15 Equivalent magnetic circuit: (a) Flow mode; (b) Squeeze mode

The magnetic reluctance of each component is determined according to the following expression:

$$R_i = \frac{l_i}{\mu_0 \mu_i S_i} \quad (6)$$

where l_i denotes the length of the magnetic flux path within the circuit that passes vertically through the magnetically conductive material, $\mu_0 = 4\pi \times 10^{-7}$ H/m is the absolute permeability, μ_i is the relative permeability of the magnetic material, and S_i is the cross-sectional area through which the magnetic flux traverses the material vertically. For more complex cases of magnetic reluctance calculation, S_i may alternatively be defined as the ratio of the volume of the cylindrical region penetrated by the magnetic flux to the corresponding flux path length [21].

In the prototype part of this study, the MRF-J25T magnetorheological fluid developed by the Chongqing Materials Research Institute was employed, and its principal characteristic parameters are listed in Table 8. The magnetic circuit utilized 45# steel with uniform material properties as the magnetic conductive medium.

Given the following parameters:

The relative permeability of MRF is $\mu_{MRF}=3$, and the relative permeability of 45# steel is $\mu_{45}=2000$. For the flow mode section, the magnetic reluctances R_{11} to R_{16} are determined according to Eq. (7)-(10).

$$R_{11} = \frac{h_8}{\mu_0 \mu_{45} \pi [r_4^2 - (r_3 + w_2)^2]} \quad (7)$$

$$R_{12}=R_{16}=\frac{w_3^2}{\mu_0\mu_{MRF}\pi[(r_4+w_3)^2-r_4^2]\frac{h_8-h_7}{2}} \quad (8)$$

$$R_{13}=R_{15}=\frac{[r_5-(r_4+w_3)]^2}{\mu_0\mu_{45}\pi[r_5^2-(r_4+w_3)^2]\frac{h_8-h_7}{2}} \quad (9)$$

$$R_{14}=\frac{h_7}{\mu_0\mu_{45}\pi[r_5^2-(r_4+w_4+w_5)^2]} \quad (10)$$

For the squeeze mode section, the magnetic reluctances R_{21} to R_{26} are determined according to Eq. (11)-(16).

$$R_{21}=\frac{h_3+h_4}{\mu_0\mu_{45}\pi[r_3^2-(r_2+w_1)^2]} \quad (11)$$

$$R_{22}=\frac{h_2}{\mu_0\mu_{MRF}\pi[r_3^2-(r_2+w_1)^2]} \quad (12)$$

$$R_{23}=\frac{(r_6-r_1)^2}{\mu_0\mu_{45}\pi(r_6^2-r_1^2)h_1} \quad (13)$$

$$R_{24}=\frac{h_2}{\mu_0\mu_{MRF}\pi(r_2^2-r_1^2)} \quad (14)$$

$$R_{25}=\frac{h_3+h_4}{\mu_0\mu_{45}\pi(r_2^2-r_1^2)} \quad (15)$$

$$R_{26}=\frac{(r_3-r_1)^2}{\mu_0\mu_{45}\pi(r_3^2-r_1^2)\left(\frac{h_8}{2}-\frac{h_5}{2}-h_3-h_4\right)} \quad (16)$$

The definitions and corresponding values of the parameters used in the calculation of magnetic reluctances are provided in Table 7 and 8. By substituting these values into the Eq. (7)-(16), the magnetic reluctance of each component can be determined, as summarized in Table 9.

The total magnetic reluctance of the flow pattern is defined as the sum of components R_{11} through R_{16} , yielding a value of 392,256 m/H. Conversely, the total reluctance for the extrusion pattern is calculated as the sum of R_{21} to R_{26} , resulting in a value of 2,408,826 m/H.

The total magnetic flux Φ can be determined using the magnetic circuit analogue of Ohm's law:

$$\Phi = \frac{NI}{R} = B_i S_i \quad (17)$$

where N represents the number of coils to be round, I is inputted the current, R is magnetic reluctance, B_i is the magnetic flux density in the magnetorheological region, and S_i denotes the flux area of flow channel and squeeze gap.

Table 9 Magnetic reluctances value of each part of hybrid operational mode MR mount.

Magnetic reluctances	Value(m/H)	Magnetic reluctances	Value(m/H)
R_{11}	23759	R_{21}	5476
R_{12}	181379	R_{22}	584112
R_{13}	2022	R_{23}	21264
R_{14}	1695	R_{24}	1755369
R_{15}	2022	R_{25}	16457
R_{16}	181379	R_{26}	26148

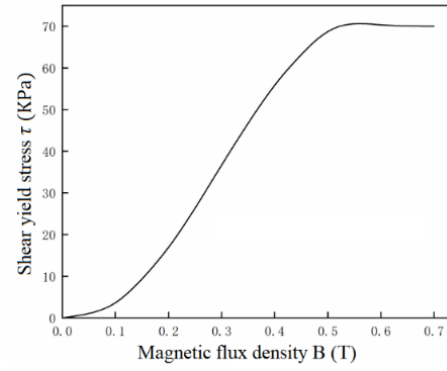


Fig. 16 B- τ curve of MRF-J25T.

The relationship between the shear yield stress τ of the MRF-J25T magnetorheological fluid and the magnetic flux density B was experimentally determined, as illustrated in Fig. 16. The MRF attains its maximum shear yield stress when the magnetic flux density reaches approximately 0.55 T. In this study, it is anticipated that an excitation current of $I = 2$ A will produce a magnetic flux density of about 0.55 T within the MR region.

By substituting the parameter values into Eq. 18, the required number of coils turns for both the flow mode and the squeeze mode can be determined. Specifically, for the flow mode section, the number of coil turns is given as:

$$N_f = \frac{BS_{12}R_f}{I} = 316 \text{ turns} \quad (18)$$

Similarly, for the squeeze mode section, the number of coils turns can be expressed as:

$$N_s = \frac{BS_{24}R_s}{I} = 201 \text{ turns} \quad (19)$$

4. Assembly of the prototype

The assembly of the proposed MR mount prototype was carried out based on the designed structural configuration and material specifications. The magnetorheological fluid used in this study is MRF-J25T, developed by Chongqing Materials Research Institute, China, and its principal properties are summarized in Table 8.

The excitation coil was fabricated by winding enamel-coated copper wire with a diameter of 0.6 mm. This configuration ensures a stable and safe current-carrying capacity in the range of approximately 1.4 A to 2.2 A, which is sufficient to generate the required magnetic field for the operation of the MR mount.

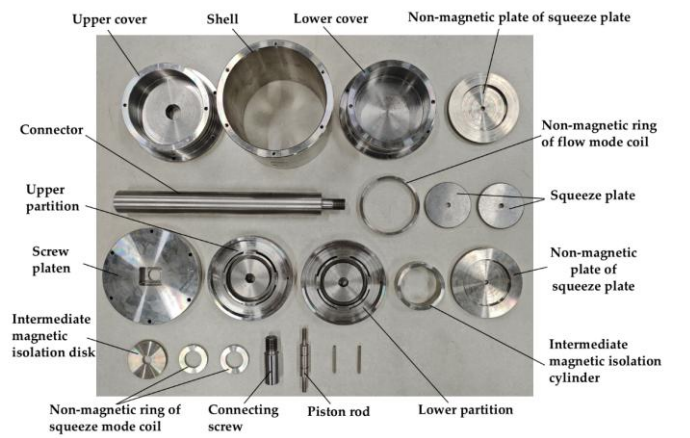


Fig. 17 The components of MR mount prototype.

All components were then manufactured and assembled according to the design requirements. The assembled prototype of the MR mount, including its key components, is illustrated in Fig. 17.

To ensure proper assembly and functional performance, the prototype was assembled following a carefully controlled procedure. The lower partition was first combined with the non-magnetic ring of the flow-mode coil to form the winding frame. Enamel-coated copper wire with a diameter of 0.6 mm was then wound around the outer surface of the non-magnetic ring to fabricate the flow-mode coil. Since the energizing direction of the coil directly affects the internal magnetic field distribution, the coil leads were clearly marked to prevent errors during subsequent current loading. According to the magnetic circuit design presented in Section 3, the number of coil turns was designed to exceed 316 for the flow mode and 201 for the squeeze mode.

After completing the winding of the flow-mode coil, the intermediate magnetic isolation disk and cylinder were mounted onto the lower partition. The lead wire of the flow-mode coil was routed through the wire outlet hole of the upper partition, followed by compression assembly of the upper partition with the lower structure. The resulting subassembly is shown in Fig. 18.

Subsequently, the squeeze-mode coils were fabricated by winding more than 201 turns of copper wire into the coil slots of both the upper and lower partitions. The coil leads were similarly marked to ensure correct electrical connections. The non-magnetic rings of the squeeze-mode coils were then installed onto the partitions using an interference fit to secure the coils in position, as illustrated in Fig. 19.

To protect the electrical connections, all coil leads were enclosed in heat-shrink tubing. The assembled substructure was then inserted into the cylindrical shell, with the coil leads guided through the outlet hole, as shown in Fig. 20. Coil pressure strips were subsequently installed onto the upper and lower partitions using an interference fit to further secure the leads. Electrical continuity was verified using a multimeter to ensure proper circuit connections.

The piston rod assembly was then prepared by installing an O-ring seal onto the piston rod, followed by mounting the squeeze plate and magnetic isolation plate at one end and fastening them with two M6 nuts. This assembly was integrated into the previously prepared substructure. The squeeze plate, magnetic isolation plate, and connecting screw were then installed at the opposite end of the piston rod.



Fig. 18 Flow mode coils winding and compaction: (a) the flow mode coils and (b) install the partition.

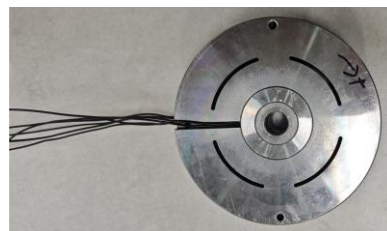


Fig. 19 Upper partition after installation of the squeeze-mode coil and the magnetic isolation ring.

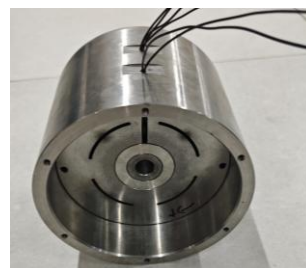


Fig. 20 The partition assembly is assembled into the shell.



Fig. 21 (a) MR mount prototype after MRF injection (b) MR mount prototype.

Following this, an O-ring seal was mounted onto the lower cover, which was then assembled into the structure. The screw and screw platen were installed sequentially, and the entire assembly was securely fastened to the cylindrical shell.

The injection of MR fluid constituted a critical step in the assembly process. The MR fluid was introduced into the cylindrical housing while the piston rod was intermittently moved up and down to facilitate the filling of the lower chamber. The filling process was carefully monitored, and the absence of air bubbles during overflow indicated that the lower chamber had been completely filled. The MR fluid was then continuously injected into the upper chamber until it reached approximately two-thirds of the chamber volume, as shown in Fig. 21a. The upper cover, pre-installed with an O-ring seal, was subsequently mounted and secured with screws. Finally, adhesive was applied to seal the lead wires and the outlet hole, as illustrated in Fig. 21b, completing the assembly of the hybrid-mode MR mount prototype.

For experimental testing, the assembled MR mount prototype was mounted onto the electro-hydraulic servo damping suspension test rig. A connecting component and lifting lugs compatible with the test rig were attached to the threaded end of the prototype, as shown in Fig. 22.

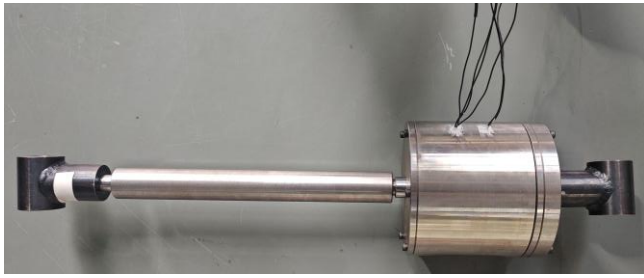


Fig. 22 MR mount prototype after installation of the connecting component and lifting lugs.

5. Conclusion

In this study, a novel MR mount integrating squeeze and flow operational modes has been proposed by exploiting the field-dependent variable viscosity characteristics of MRF. The proposed MR mount is specifically designed to provide effective vibration isolation under bidirectional excitation conditions commonly generated by vehicle engines. By integrating the advantages of both operational modes within a single structural configuration, the proposed design aims to achieve enhanced controllable damping performance and improved adaptability to complex dynamic loading environments.

The squeeze mode of the proposed MR mount is realized through the arrangement of symmetrically configured upper and lower squeeze gaps, enabling the MR fluid to generate controllable resistance under bidirectional vibration excitation. This configuration allows the MR mount to respond effectively to vehicle engine-induced vibrations acting in both upward and downward directions. In parallel, the flow mode is implemented by incorporating four flow channels, allowing the MR fluid to flow under magnetic-field-controlled shear conditions. The combined squeeze-flow hybrid configuration significantly broadens the adjustable damping range and enhances the vibration isolation performance of the MR mount under bidirectional and broadband excitation conditions, compared with conventional single-mode MR mount designs.

The structural design of the MR mount was described in detail, with particular attention paid to the primary load-bearing components. A static finite element analysis was conducted to assess the mechanical performance and structural safety of the proposed design. Under an applied axial load of 10,000 N, the simulation results indicate that the maximum principal stress reaches 368.5 MPa in compression and 230.61 MPa in tension. Based on the material properties and allowable stress limits of the load-bearing components, the ultimate load capacity of the MR mount was conservatively determined to be 7,000 N, which satisfies the load requirements for typical engine mounting applications.

Furthermore, a comprehensive magnetic circuit design was carried out, as the magnetic circuit represents the core functional subsystem governing the controllability of the MR mount. The excitation requirements for both operational modes were analyzed, and the results demonstrate that the number of coil turns must exceed 201 in the squeeze mode and 316 in the flow mode to ensure effective magnetic field strength in the active regions of the MR fluid. These results provide important guidance for the electromagnetic design and optimization of hybrid-mode MR mounts.

Based on the proposed design and analysis, a prototype of the hybrid-mode MR mount was successfully fabricated and assembled, confirming the feasibility of the integrated squeeze-flow concept. The proposed MR mount is intended to provide controllable damping

for vibration reduction in mechanical systems, highlighting its potential application in vibration isolation.

Future work will focus on the development of a damping force prediction model, as well as the characterization of the relationship between the damping force and the applied current used to generate the magnetic field. In addition, detailed numerical analyses of the magnetic circuit and flow field will be conducted. These efforts will be complemented by experimental investigations, including mount damping force tests and principal stress tests under compressive and tensile loading, to systematically evaluate the vibration isolation performance of the proposed magnetorheological mount under various operating conditions.

List of abbreviations

MR	=	Magnetorheological
MRFs	=	Magnetorheological fluids
FEA	=	Finite Element Analysis

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Author Contributions:

Shuangyi Liang: Conceptualization, Methodology, Software, Formal analysis, Investigation, Writing – original draft. **Chen Chen:** Methodology. **Yibu Zhao:** Software. **Kwanchai Kraitong:** Supervision, Resources, Writing – review & editing.

Declaration of Interest Statement

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

Shuangyi Liang and Kwanchai Kraitong, hereby declare that an Invention Patent Application, Number: 2501007349, for Magnetorheological Mount was filed with the Department of Intellectual Property (DIP), Thailand on 27 December 2025, and that said application is currently pending, making this product 'Patent Pending'.

Shuangyi Liang, Chen Chen and Yibu Zhao, hereby declare that an Invention Patent Application, Number: 2025115811040, for Magnetorheological Mount was filed with China National Intellectual Property Administration (CNIPA) on 31 October 2025, and that said application is currently pending, making this product 'Patent Pending'.

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