J. Marine. Sci. Appl. (2009) 8: 291-297
DOI: 10.1007/s11804-009-8075-3

Research on the dynamic performance of ship isolator systems that use magnetorheological dampers

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Abstract: Isolator systems on ships should ideally be able to simultaneously reduce low frequency vibration response and high frequency shock response. Conventional isolator systems are unable to do so. To solve the problem, a new style isolator system was created. This isolator system consists of a steel coil spring component and a magnetorheological (MR) damper component working in parallel. Experiments on this isolator system were carried out, including tests of vibration reduction and shock resistance. The vibration load frequencies were set from 1-15 Hz, and force amplitudes from 2.94~11.76 kN. The maximum shock input acceleration was 20 g, and impulse width was 10ms. Both the vibration and shock loads were applied using MTS Systems Corporation's hydraulic actuators. The experimental results indicated that the isolator system performs well on system vibration response, with resonance humps of the vibration response obviously reduced after using the MR damper. For the shock experiment, the attenuation of shock response was much faster with increased MR damping. The MR damper's effect on shock moments was very different from its performance in vibration mode. The correlation between MR force and control current was not as evident as it was during vibration loads.

Keywords: magnetorheological fluid damper; vibration reduction; shock resistance; isolator
CLC number: U661.7 **Document code:** A **Article ID:** 1671-9433(2009)04-0291-07

1 Introduction

With the development of modern scientific technology, the level of research on blast and shock resistance performance on ships has been gradually improved. It is well known that shock, secondary bubble pulsing pressure and lagging flow produced by underwater explosion can do a great harm to ship structure, facilities and crew on it; and the vibrations generated by ship equipments are disadvantages for ship working environment and safety too. Both of the shock and vibration mentioned above should be restricted as small as possible at the same time. Currently, isolator system is often used to solve this problem. But the general isolator system on ship is usually a passive spring system whose stiffness is almost steady under all the working condition. It is difficult to satisfy the requirements of reducing vibration and shock at the same time. Hence, seeking for a new type of isolator system which is reasonable, economic and effective becomes a new subject for researchers.

To solve this problem, a new kind of isolator system is presented in this paper. It is composed of steel rope springs and MR (magnetorheological) dampers. The MR dampers and steel rope springs were set in parallel position. This isolator system takes advantages of the high and variable damping property of MR damper to make the ship equipments have a good vibration reduction performance.

Over the years, many research results on MR and its applications were carried out. Jinping Ou did experiment for optimizing configuration of a MR damper, and discussed the influencing factors on the damping force[1]. A non-linear model of MR damper was proposed by Ginder. He determined the static yield stress as the maximum shear stress which was modeled as tensile component in the shear direction of the linear infinite single chains of spherical particles[2]. Carlson proposed that as well as iron-cobalt alloys, iron-nickel alloys with ratio ranging from 90:10 to 99:1 showed a significant increase in the yield stress of MR fluids[3]. Dyke studied the seismic protection of civil structure using MR damper[4]. Spencer reported the phenomenological model for MR damper[5]. A new kind of MR fluid with high yield stress of 100 kPa was reported by Phule[6]. The MR fluid effect is often characterized by Bingham Plastic model and its application on vehicle seat was discussed by Sireteanu[7]. Duan studied the rain-wind-induced cable vibration control over the cable-stayed Dongting Lake Bridge[8]. Several MR fluid devices have been developed for commercial use by the LORD Corporation.

Received date: 2008-10-30.
Foundation item: Supported by the 111 Project. Grant No. B07019.
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2 Experimental model of the isolator system

This experimental model was designed using two main components: MR damper and steel rope spring.

2.1 A summary of MR damper and steel rope spring

Magnetorheological fluids comprise of a carrier fluid, magneticresponsive particles and surfactants or suspension agents. The particles are polarized in the presence of an applied magnetic field, and organized into chains of particles within the fluid. The particle chains apparently increase the viscosity of the fluid. The particles will return to an unorganized state when the magnetic field is removed, which lowers the viscosity of the fluid. MR damper is designed using MR technique. Fig. 1 shows its typical structure sketch. By changing the electricity in the loop, there will be a magnetic field in the MR damper, thus MR fluid will have different mechanical characteristics. This makes the MR damper to be controllable. MR damper has become a new kind of vibration reduction equipment in the domain of vehicles, machines, bridges, and architectures, etc. with its good characteristics of high damping, great adaptation to temperature, fast response, low energy dissipation, simple structure, and continuously controllable damping. Some MR damper equipment has been used in practical projects. Currently, research results indicate that the mechanical characteristics of MR damper are related to adscititious magnetic field, displacement amplitude, and excited frequency. Dynamic constitutive relation of MR fluid is very complicated after rheological change. The dynamic damping of MR damper shows non-linear characteristic.

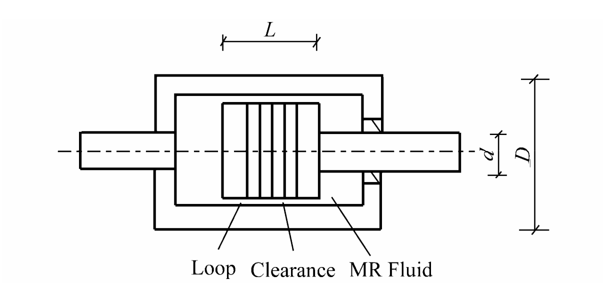


Fig.1 Sketch of MR damper

The MR damper used in this experiment is shown in Fig.2.

Fig.3 illustrates the MR damper's typical characteristic curve. These curves show the force-displacement relations of MR damper with different controllable electricity under the displacement input signal of a sinusoid with amplitude of 20 millimeter and period of 2 seconds.


Fig.2 MR damper in calibrating test

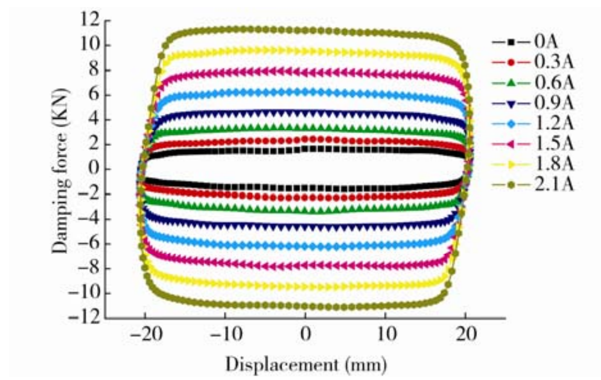


Fig.3 Characteristic curves of MR damper

Steel rope spring is made from steel wire. It has good deformation characteristics, and can make the vibration systems have a lower natural frequency[9,10]. Fig.4 shows the steel rope spring used in this experiment.

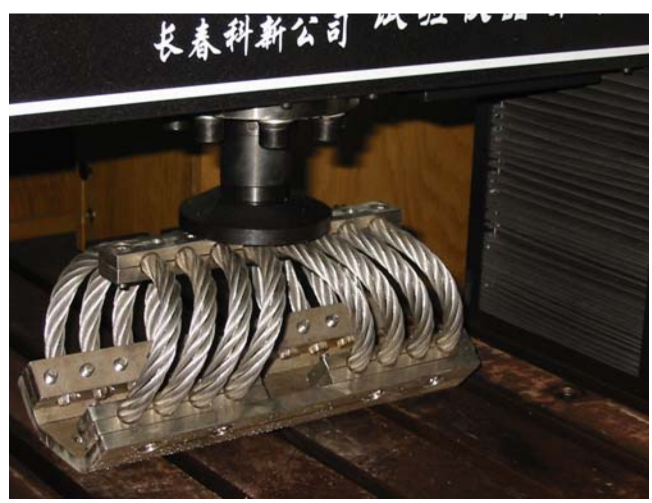


Fig.4 Steel rope spring in calibrating test

Fig.5 shows the typical rigidity characteristic curve of the steel rope spring.

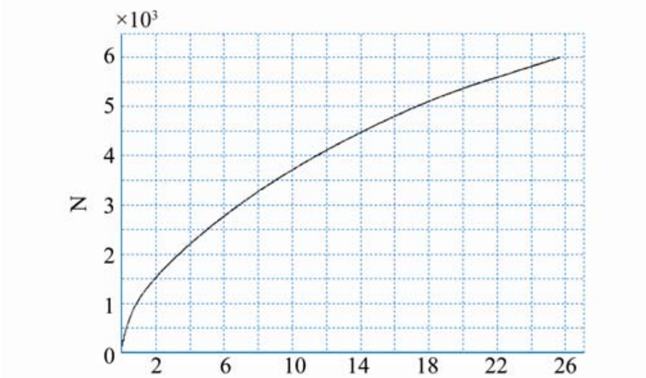
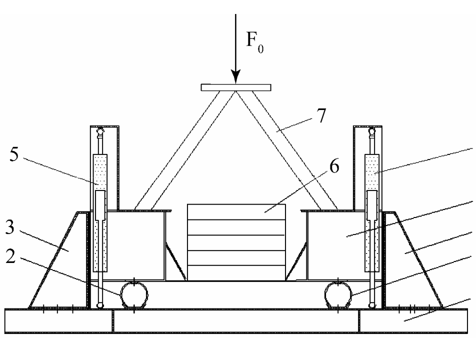


Fig.5 Characteristic curve of steel rope spring

2.2 Experimental model

The isolator vibration system was simplified to a single degree of freedom system. The dimension parameters of model were determined according to the actual smaller engine. The model scale is 1:1. The maximum model mass was confirmed to be 2 000 kg. According to the natural frequency of steel rope spring system about 6Hz, the steel rope spring's equivalent rigidity was determined to be 6.06×10^6 N/m. Fig.6 is the sketch of the experimental model system.



1-lower structure, 2-steel rope spring, 3-travel-limiting structure, 4-uper structure, 5-MRdamper, 6-mass, 7-loading stand.
Fig.6 Sketch of the experimental system

The lower structure was used to simulate the base of the equipment, and the upper structure and mass were for the equipment. Lower and upper structures were connected by MR dampers and steel rope springs. There were 4 MR dampers and 6 steel rope springs being used for this model. The travel-limiting structures were used to restrict the horizontal displacement of the upper structure during impact experiment. The loading stand was used to apply the exciting load on the model. Fig.7 shows the vibration experiment photo.

Fig.7 Vibration experiment photo

3 Vibration experiment

3.1 Boundary condition of the experiment

The exciting force was applied to the experiment system through actuator with the rated load of 10 tons. The actuator was linked to the model through a loading stand.

The main purpose of the vibration experiment is to measure the vibration response under a sinusoidal exciting force. Under the boundary condition of the experiment, many load cases were applied in this experiment. These load cases were varied in different exciting force amplitudes and frequencies, different controllable masses, and different damping forces of MR dampers. The information of load cases is listed below in details, with its equidifferent range for each item.

Exciting force amplitude: 2 000~12 000 N (6 cases).
Exciting force frequency: 1~15 Hz (15 cases).
Controllable mass: 1 000~2 000 kg (6 cases).
Controllable electricity: 0~2 A (8 cases).

3.2 Analysis of experimental results

Fig.8 and Fig.9 show the time-displacement history curves of the experiment with controllable mass of 1 ton, exciting force amplitude of 10 kN, and its frequency of 1Hz. The displacement sensors were located on the upper structure of the experiment model. The curves show that the system displacement response can be well reduced by using this isolator damper.

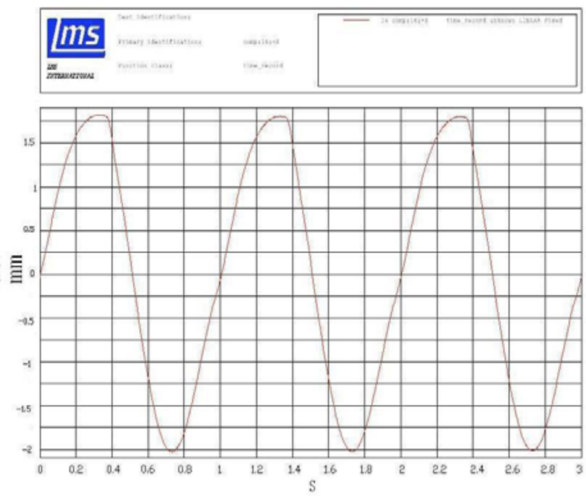


Fig.8 Time-displacement curve when controllable electricity is 0

Fig.9 Time-displacement curve when controllable electricity is 2A

For the convenience of describing the vibration experiment results, the non-dimensional parameters η, β and λ are used in this paper. The definitions of these parameters are shown in Eqs.(1)~(3).

Frequency ratio λ is defined as

$$\lambda = \frac{f}{f_n} \tag{1}$$

where f is the exciting frequency and f_n is the natural frequency of the experiment model system. The parameter λ describes the relationship between these two frequencies.

Force transfer ratio η is defined as

$$\eta = \frac{F_{kc}}{F_0} \tag{2}$$

where F_{kc} is the resultant force transferred from the MR dampers and the steel wire springs, and F_0 is the corresponding exciting force amplitude.

Non-dimensional displacement amplitude ratio β is defined as

$$\beta = \frac{D}{D_0} \tag{3}$$

where D is the vibration displacement amplitude under exciting force amplitude F_0, and D_0 is the static displacement under force F_0.

The parameters η and β describe the vibration reduction effect on force and displacement respectively.

Fig.10 and Fig.11 are the typical curves of this experiment. It shows that the system vibration response can be reduced to different lower levels by changing controllable electricity of MR dampers. The maximum reduction point of vibration response is at the natural frequency of the vibration system, i.e. frequency ratio at 1. The decreasing of the amplitude ratio of force transferred and non-dimensional displacement amplitude ratio is 75% and 71% respectively.

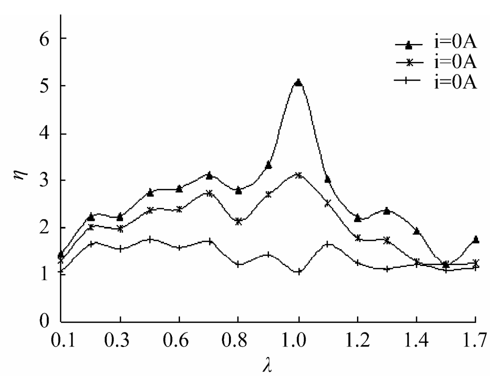


Fig.10 η - λ curves with different controllable electricity of MR

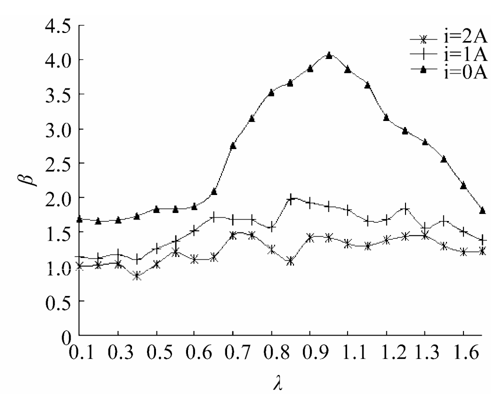


Fig.11 β - λ curves with different controllable electricity of MR

4 Shock experiment

4.1 The boundary conditions

The shock experiment aimed to measure the isolator system's response under the vertical shock load. The shock load was applied by a hydraulic actuator through shock loading stand on the lower structure. The rated output force of the actuator is 50 t. The shock experiment photo is shown in Fig.12.

Fig.12 The photo of shock experiment

The shock input was determined according to the corresponding shock specification. Fig.13 shows the acceleration, velocity, and displacement history curves of the shock input. The maximum shock input acceleration was 23 g and the corresponding pulse width was 10 ms.

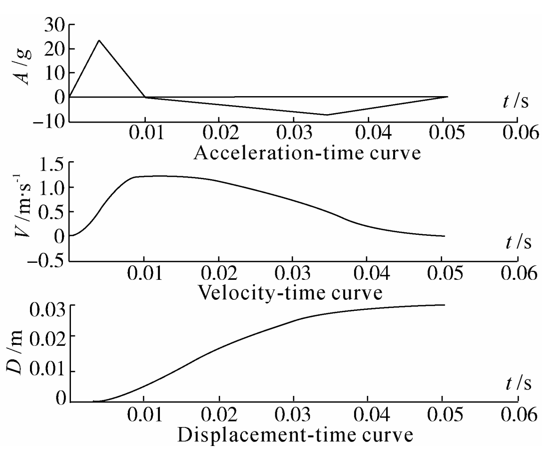


Fig.13 Shock input curves

4.2 Analysis of experimental results

There were 11 electricity cases in the shock experiment, and they were equidifferented between 0~2 A. The model controllable mass was 2 tons. Fig.14 and Fig.15 show the time domain acceleration input and response respectively.

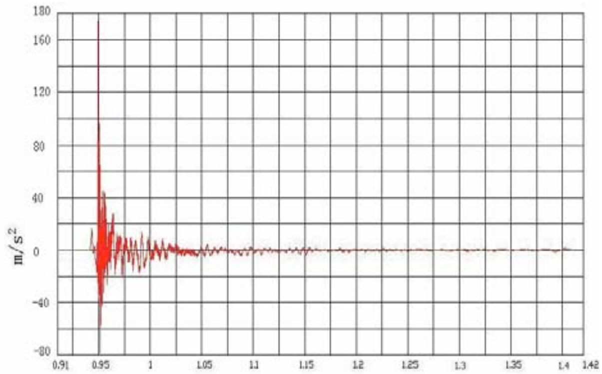


Fig.14 The shock acceleration input

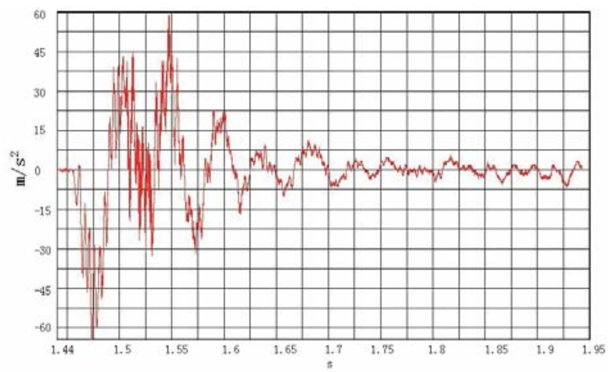


Fig.15 The shock acceleration response

For the convenience of describing the results of the shock experiment, the non-dimensional parameters P_a and η_a are used in this paper. The definitions of these parameters are shown in Eqs.(4) and (5).

Acceleration transmissibility P_a is defined as

$$P_a = \frac{a_o}{a_i} \times 100\% \tag{4}$$

Where a_0 is the acceleration peak value of shock response of upper structure, and a_{i} is the acceleration peak value of shock input on the lower structure. This parameter describes the impact resistance effect on acceleration at the impact moment.

Acceleration attenuation factor η_a is defined as

$$\eta_a = \frac{a_a}{a_f} \times 100\% \tag{5}$$

where a_a is the second peak value of the two adjacent wave crests in the acceleration response history, and a_f is the corresponding first peak value. This parameter describes the vibration acceleration attenuation after the impact moment.

Fig.16 illustrates the relation between acceleration transmissibility and controllable electricity. It shows that the peak values of acceleration don't have obvious difference in changing the controllable electricity. Tracking the MR damper's force in the experiment indicated that the reason of this phenomenon was that the peak value of the MR dampers' force was almost not changed by the controllable electricity during the shock experiment.

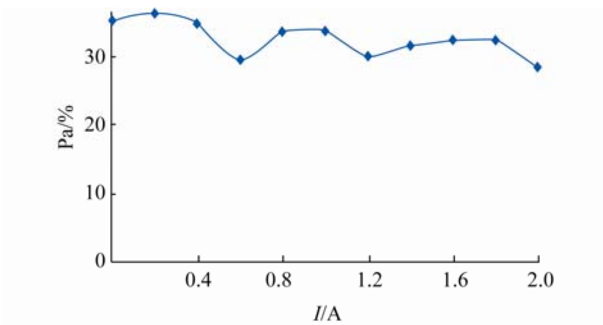


Fig.16 The acceleration transmissibility

The relation between the peak value of MR damper's force and controllable electricity is shown in Fig.17.

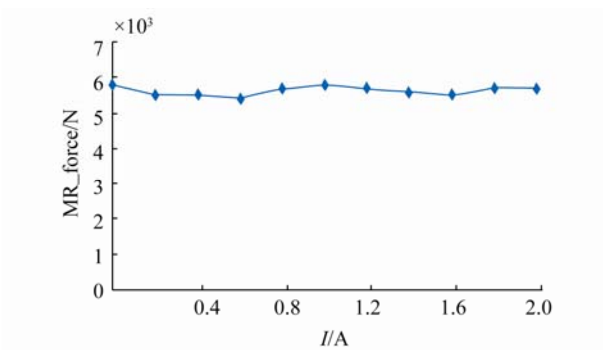


Fig.17 Peak value of MR force with control current

Fig.17 shows that in the shock experiment, the MR damper showed a big different characteristic compared to its performance in the vibration experiment. The MR damper's force was almost not changed by changing controllable electricity. MR damper's impact mathematical model should be different from the vibration model.

Fig.18 illustrates the relationship between acceleration attenuation factor and controllable electricity.

It shows that the acceleration attenuation factor became smaller with the increase of controllable electricity. According to the definition of the acceleration attenuation factor, it's clear that the bigger the system damping, the smaller the attenuation factor. So this experimental phenomenon shows that the MR damper has a good effect on the vibration after impact like that in the vibration experiment.

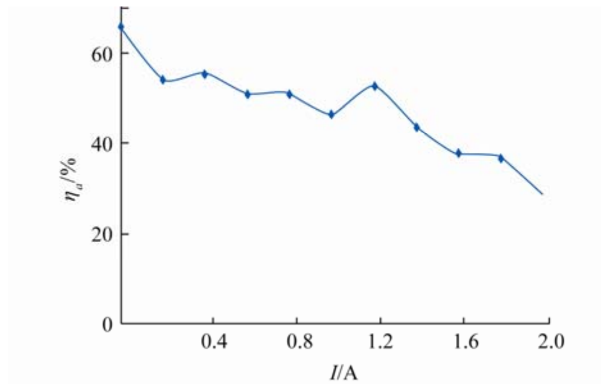


Fig.18 Velocity damping factor

Conclusions

In the scope of this study, the experiments of isolator system based on MR have been carried out. It shows that: The isolator system has a good effect on vibration reduction in low frequency range. The force transmissibility is about 1 with the exciting frequency varying from 1 Hz to 15 Hz. The vibration response has been reduced evidently.

The MR damper has a little effect on the shock response peak value with changing of the controllable electricity. Shock response is not evident and disciplinarian. At the shock moment, the MR damper's force is approximately constant, and is not changed with changing of the controllable electricity. The mechanical characteristic of impact is different from that of vibration. The MR damper has a good attenuation effect on the residual vibration after the impact.

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基于MR阻尼器的船用隔离器系统试验研究

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摘　要：随着船舶技术的发展，现代船舶对船用隔离器系统提出了越来越高的要求，即船用隔离器系统应同时具备低频减振和高频抗冲击的能力，这是传统的船用减振器系统所无法做到的. 为了解决此问题，本文提出了一种新的船用隔离器系统，该系统由钢丝绳弹簧和磁流变阻尼器相并联组成. 文中对该船用隔离器系统的减振和抗冲击性能进行了模型试验研究。减振试验的激振力频率为 1-15 Hz，力幅为 2.94-11.76 kN；冲击试验的最大冲击输入加速度为 20 g，脉宽为 10 ms，减振试验和冲击试验均采用 MTS 液压加载系统来进行. 试验结果表明，该船用隔离器系统具有较好的减振效果，使用了 MR 阻尼器后系统得共振峰值被明显的削弱；在冲击试验中，冲击响应的衰减速度随着 MR 阻尼器的阻尼增加而明显加快，但是 MR 阻尼器再冲击瞬间的出力特性明显与低频振动情况下不同，MR 阻尼器的出力表现为受控制电流强度影响不大.

关键词：MR 阻尼器；减振；抗冲击；隔离器