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Research on low-frequency mechanical characteristics of the MR dampers in ship isolators

YAO Xiong-liang（姚熊亮）, TIAN Zheng-dong（田正东）, SHEN Zhi-hua（沈志华）, GUO Shao-jing*（郭绍静）

College of Shipbuilding Engineering, Harbin Engineering University, Harbin 150001, China

Abstract: A new isolator composed of a steel rope spring and a magneto-rheological (MR) damper was designed and a study on low-frequency mechanical characteristics of MR dampers in isolators was carried out. It used the characteristics of the MR damper, such as fast response, controllable damping, small energy consumption, wide dynamic scope, and great adaptation. The relationships between MR damping forces and influencing factors were analyzed based on experimental data. The results show that damping force is not only related to structural dimensions, but also closely related to controllable current and vibration frequency. Finally, the empirical formula for damping forces was corrected, and the relationship between correction coefficients and factors analyzed.

Keywords: ship isolator; magnetorheological fluid; magnetorheological fluid damper; low-frequency; mechanical characteristics

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1 Introduction

With the characteristics of high damping force, great adaptation to temperature, fast response, low energy dissipation, simple structure and continuously controllable damping, the magnetorheological fluid damper (MR damper) has come to be new vibration reduction equipment with high performance and intelligence in the domain of car, machine, bridge and architecture, etc., and it has been applied to practical project, which shows great application prospect [1-4]. At present, researches indicate that the mechanical characteristics of MR damper are affected greatly by adscititious magnetic field, displacement amplitude and exciting frequency; dynamic constitutive relation of MR fluid is very complicated after rheological change and the dynamic damping force of MR damper shows strongly non-linear relationship.

Although MR damper has been applied to engineering greatly [5-8], there are still no corresponding researches in aspects of ship vibration reduction. The low-frequency vibration reduction is a difficult point in the development of ship isolators. If a new isolator composed of a damper and a style of vibration absorber can improve the low-frequency vibration reduction effect, it will make great sense to improve ship concealment, vitality and fighting strength.

So a new ship isolator system composed of steel rope and MR damper is presented in this paper, the purpose is to reconstruct the passive vibration absorber of ship foundation and strengthen the low-frequency vibration reduction effect of isolator system. A series of characteristic curves of MR damper are obtained through vibration experiment on isolator system in this paper. According to the experimental results, the relations of damping force and its influencing factors are analyzed and some useful conclusions are obtained, which is of great significance to practical application of MR damper on ship.

2 Experiment plan

The ship isolator is designed to study the mechanical characteristics of MR damper in this paper. It makes sense to practical application of isolator. The new ship isolator composed of traditional vibration absorber and power component in parallel is presented as shown in Fig.1, and the dynamic characteristics of it is controllable, so the vibration reduction effect will be improved in a wide frequency range (especially in low-frequency range). The detailed design of the ship isolator in this paper is composed of steel wire springs

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***Corresponding author Email:** guoshaojing1984@163.com

and MR dampers in parallel.

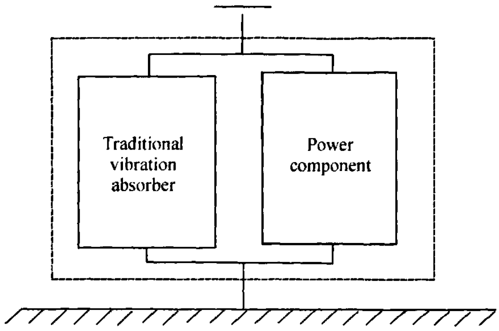

Fig.1 Sketch map of ship isolator

3 Experiment model

The isolator system model is composed of many parts as shown in Fig.2 (1-Lower structure, which mainly simulates the bottom deck; 2-Steel wire spring; 3-Lateral restriction structure, which mainly restricts the motion of upper structure in horizontal direction and resists the torque of equipment running; 4-Upper structure, which bears the mass of large scale equipment; 5-MR damper; 6-Controllable mass; 7-falsework). The experiment model is shown in Fig.3.

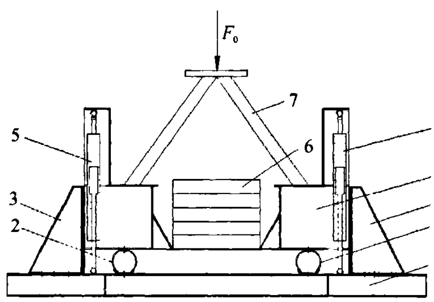

Fig.2 Lateral sketch of the system model

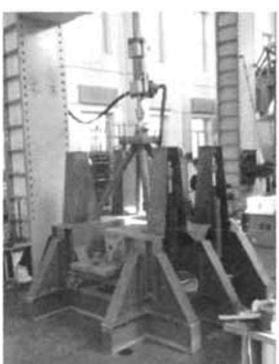
Fig.3 Experiment model

In order to describe the mechanical characteristics of MR damper in the isolator system, the low-frequency vibration responses (damping force, velocity, acceleration and displacement) of isolator are tested by changing the exciting frequency, current, controllable mass and exciting force F_0.

4 Experimental work condition

The mechanical characteristics of MR damper under simple harmonic exciting load $F_0 = f_0\sin(\omega t)$ are measured through vibration experiment. Work conditions are set as follows: the amplitude of exciting force f_0 is 24 000 N; the control mass is 1 t and 2 t respectively; the exciting frequencies are 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14 and15 Hz and the control currents i are 0, 0.25, 0.5, 1.0, 1.5, 1.75 and 2.0 A, respectively.

5 Analysis and results

5.1 The relationship between damping force and current of MR damper

A large number of experimental data were analyzed and the damping force characteristic of MR damper was concluded in this paper. Three dampers (numbered A, B, C) are mainly studied. The relationship between damping force of MR damper and current is shown in Fig.4 on the condition that the control mass is 1t, the exciting frequency is 1 Hz and the control current varies from 0 to 2 A.

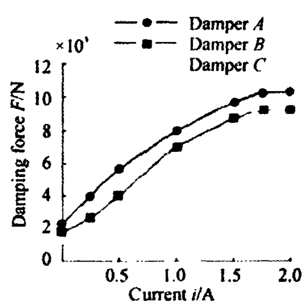

Fig.4. Force-current curves of MR damper

From Fig.4, it can be seen that the damping force variation trends of three dampers are almost the same and the damping force increases with increasing of the control current. And the trend gradually tends slow and reaches a saturation value in the end. This phenomenon corresponds well with the saturation of inner magnetic filed.

5.2 The relationships between damping force and displacement and velocity of MR damper

The relationships between damping force and displacement and velocity of damper A are shown in Figs.5 and 6 at frequency 2 Hz, exciting force 24 000N and current 1 A.

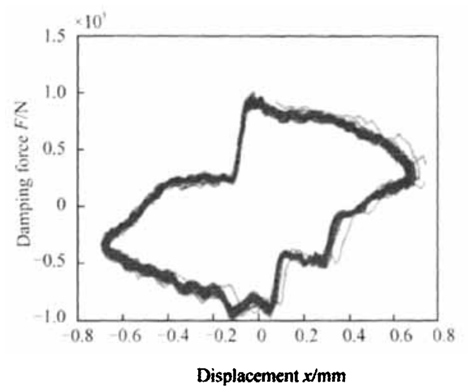

Fig.5 Force-displacement curves of MR damper

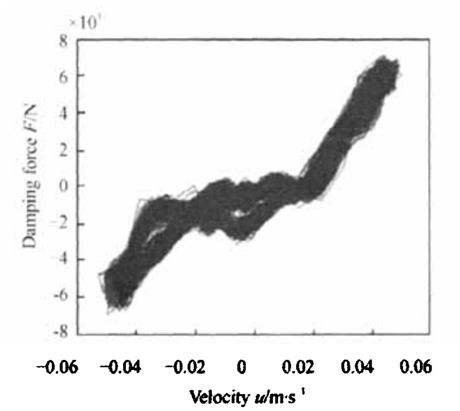

Fig.6 Force-velocity curves of MR damper

From Fig.5 it can be seen that the damper stagnant curve is rather full, the maximum displacement is obvious and the damping force has contractive trend in the first and third quadrants. The reason is that the joint part between shock isolator and damper has gap. The damper is connected by hinges in this paper. When it exports force, there is a tiny move, which causes connection error. And the damper must overcome this error while reverse loading. Fig.6 shows that damping force and velocity have tight relation. The relation in low velocity zone is nonlinear and in high velocity zone is linear. And the damping force increases with the increasing of velocity.

5.3 Correction of empirical formula

In order to describe the mechanical characteristics of MR damper, the parameters of dampers are given in Table 1 and Fig.7. The viscosity coefficient η is 0.65 in this paper [9].

Table 1 Structure dimension of MR damper

Gap length h/mm	Inner diameter of cylinder D/mm	Shaft diameter of piston d/mm	Effective length of piston L/mm
2	100	40	40

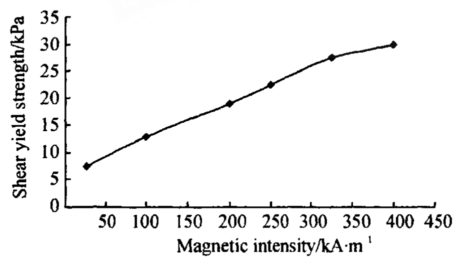

Fig.7 Relation between shear yield strength and magnetic intensity

The relations of damping force and its influencing factors are obtained by analyzing the experimental data, and the damping force is calculated through the empirical formula[10]:

$$F = \frac{12\eta L A_p}{\pi D h^3} A_p + \frac{3 L A_p}{h} \tau_y \operatorname{sgn}(u), \tag{1}$$

where $A_P = \pi(D^2 - d^2)/4$ when frequency is 1 Hz, i=0.25 A, control mass is 1t, it can be found that the damping orces obtained according to empirical formula are quite different from the experimental data. These are mainly manifested in that the velocity slope (viscosity coefficient) and intercept (coulomb force) obtained according to empirical formula are different from experimental data (seen in Fig.8), which shows that empirical formula can't express the damping force of MR damper well. Therefore, the slope correction coefficient C_S and the intercept correction coefficient C_P are introduced to correct the empirical formula.

The correction empirical formula is

$$F = C_S \frac{12\eta L A_p}{\pi D h^3} A_p u + C_P \frac{3 L A_p}{h} \tau_y \operatorname{sgn}(u). \tag{2}$$

According to the correction Eq.(2), the variation phenomena of C_S and C_P with different frequency ratios λ and control currents are shown in Fig.9 and Fig.10 separately.

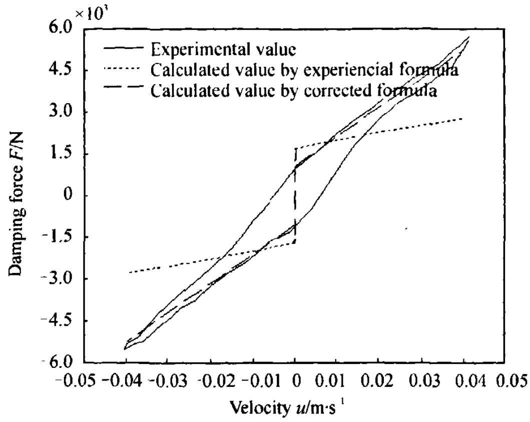

Fig.8 Relation between damping force and velocity

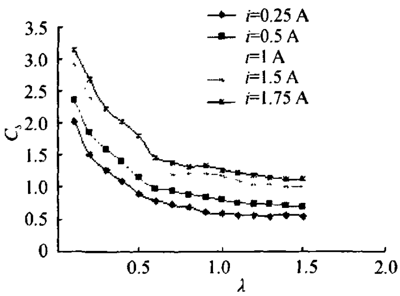

Fig.9 Relation between C_S and frequency ratio λ

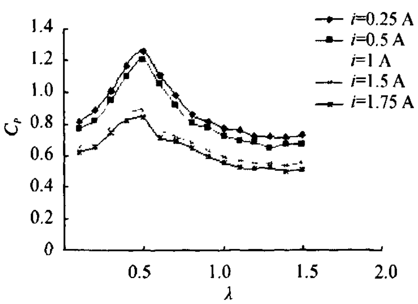

Fig.10 Relation between C_P and frequency ratio λ

In order to better express the actual damping force of MR damper with the correction formula, another example is calculated with the exciting force 24 000 N and the exciting frequency 3 Hz. The results show that the correction formula of damping force can reflect the actual damping force well as shown in Fig.11.

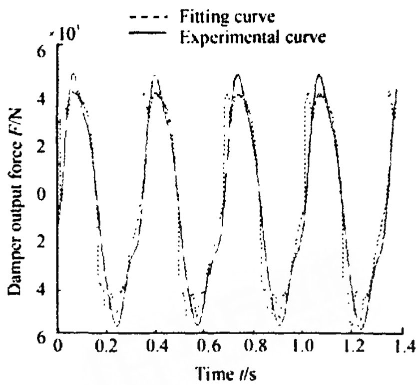

Fig.11 Comparison curves of the experiment with the calculated result

6 Conclusions

In this paper, the main conclusions are drawn as follows:

1) The damping force of MR damper increases slowly with increasing of the control current, but the trend abates gradually and reaches a saturation value in the end.

2) The damping force of MR damper is not only related to the structure dimension, vibration velocity and the controllable current, but also related to the exciting frequency.

3) The damping force of MR damper obtained according to empirical formula is quite different from the experimental data, which may be resulted from the bubble and the sedimentation of the MR fluid.

4) The correction formula can describe the damping force better than the empirical formula. It shows that the correction coefficient C_S increases with increasing of current and the variation reduces obviously when the current is near saturation. The correction coefficient C_S decreases fast in low frequency, while in high frequency, C_S decreases very slowly, seeming not to vary with frequency; while C_P decreases with increasing of current, and it increases quickly to a peak value in low frequency, but decreases slowly in high frequency.

5) Correction coefficient C_S and C_P have great

influence on the damping force of MR damper and the correction formula can better describe the damping force of MR damper.

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YAO Xiong-liang was born in 1963. He is a professor at the College of Shipbuilding Engineering, Harbin Engineering University, majoring in the ship structural mechanics, prediction and control of ship vibration, structural response under underwater impulsive load. He has published more than 90 papers in journals and conference proceedings and 4 books for graduates and undergraduate students.

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作者： YAO Xiong-liang， TIAN Zheng-dong， SHEN Zhi-hua， GUO Shao-jing
作者单位： College of Shipbuilding Engineering,Harbin Engineering University,Harbin 150001,China

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