Simulation analysis and experiment research of stiffness variable shock isolator for large-scale marine equipment

Xiang Lu¹, Xiaojie Yan¹ and Bo Wu²

¹National Key Laboratory on Ship Vibration & Noise, China Ship Development and Design Center, Zhangzhidong Road 268, 4300644, Wuhan, Hubei,China
²Zhuzhou Times New Material Technology CO., LTD. 412007, Zhuzhou, Hunan, China

*Corresponding author’s e-mail: luxiangsju@163.com

Abstract: In order to analyze working performance of shock isolators for large-scale marine equipment under heavy shock load, a certain type of stiffness variable shock isolator was studied by conducting FEA calculation and shock experiments. Good agreement between the simulation and experiments proves the efficiency of calculation method. Results show that the shock isolator can provide alterable stiffness in axial compression, and absorb enough shock energy within restricted area to achieve shock isolation and elastic limit of equipment in harmony.

1. Introduction
The ship voyages condition is so complicated that large-scale equipment on board can easily crash under heavy shock load, leaving ships unable to move or fight[1,2]. To improve anti-shock capacity of onboard equipment, shock isolators with high performance should be installed between ship body and equipment. For now, common vibration isolators are widely used as shock isolators, which can not only isolate vibration at normal voyages condition, but also transform shock energy into potential energy by elastic deformation at extreme voyages condition, thus buffing equipment[3]. Usually stiffness of common vibration isolators change little within efficient working stroke, and lower stiffness means larger working deformation and lager energy absorption[4]. However, as isolated bodies on the sea, ships have certain restrictions on the size and working area of shock isolators, so the structure and anti-shock properties of shock isolators, especially for large-scale marine equipment, should be improved.

A new type of shock isolator designed for large-scale marine equipment under heavy shock load is presented. Through FEA simulation, an empirical stiffness calculation method verified by experiments is provided. Results show that the new shock isolator provides alterable stiffness to achieve both shock isolation and elastic limit.

2. Studying object
The new shock isolator is designed with three dimensional large deformation, stiffness variation and elastic limit to satisfy the working condition of large-scale marine equipment. Unlike vibration isolators, it mainly provides shock isolation without bearing capacity. For better performance, it should be installed on both sides and bottom of equipment.

The new shock isolator is specially designed as figure 1. It consists of 4 main parts, including upper
coverplate, rubber deformation body, base and lower coverplate. The latter 3 parts are all rubber-metal vulcanized parts, which bond specialized Nitrile Rubber Buna(NRB) with high-strength metal by specific adhesive. Stiffness varies with the falling of upper coverplate when the shock isolator is axially compressed. Within 10mm of falling stroke, the working part is mainly inner rubber-metal pad. When upper coverplate touches down to the top of the base(covered with thin rubber layer), the main working parts turn into inner, outer and bottom rubber-metal pads, which provide much higher stiffness by parallel connection.

1. Upper coverplate 2. Rubber deformation body 3. Base 4. Outer rubber-metal pad 5. Rubber sealing layers 6. Inner rubber-metal pad 7. Bottom rubber-metal pad 8. Lower coverplate

Figure 1. Working principle of the shock isolator.

3. FEA Simulation

Finite element model of the shock isolator is set up and meshed for static FEA calculation. As isotropic and approximately incompressible hyper-elastic material[5], the constitutive relation of rubber material can be characterized by Mooney-Rivilin Model, which gives the relation between strain energy density $W$ and deformation tensor invariants $I_1$ and $I_2$ as follow:

$$W = C_{10}(I_1 - 3) + C_{01}(I_2 - 3) + \left( J - 1 \right)^2 / D_1$$

(1)

Where the $C_{10}$ and $C_{01}$ are called Mooney-Rivilin material constants, $J$ is called main extension ratio(for incompressible material, $J=1$), and $D_1$ is called compressibility[6]. Based on long-term accumulation of experiment data and curve fitting, material parameters of rubber-metal pads are determined as in table 1.

| Part        | $C_{10}$ | $C_{01}$ | $D_1$ | Density (g/mm$^3$) |
|-------------|----------|----------|-------|--------------------|
| Inner part  | 0.470    | 0.12     | 0.001 | 1.25               |
| Outer part  | 0.362    | 0.09     | 0.001 | 1.25               |

Each Rubber-metal vulcanized part is considered as a whole, regardless of bonding strength. Materials of metal parts are listed in table 2.

| Part            | Elasticity modulus (MPa) | Poisson ratio |
|-----------------|--------------------------|---------------|
| Central pillar  | $2.09 \times 10^5$       | 0.295         |
| Others          | $2.06 \times 10^5$       | 0.28          |

As for boundary conditions, bottom plane of lower coverplate is fixed. Axial and radial displacement is applied on the rigid central point of upper coverplate. Forces are calculated as displacement increases.

Figure 2~4 show strain contours on the cross sections of rubber-metal pads and overall static stiffness curves with increasing displacement $D$. The maximum strain in each condition is below the safety value 1.5. As in figure 2, strain of outer rubber-metal pad changes a lot between (a) and (b), also slope of curve rises sharply right when displacement exceeds 10mm. In addition, there is no force until displacement exceeds 5mm. This is because that rubber sealing layers in the gap have very low stiffness, and conductive outer channels are designed on the face of metal parts, which rubber layers
will fully fill in after compression.

(a) Strain contour (D=10mm)  (b) Strain contour (D=30mm)  (c) Static stiffness curve

Figure 2. Simulation at axial compression condition.

(a) Strain contour (D=10mm)  (b) Strain contour (D=20mm)  (c) Static stiffness curve

Figure 3. Simulation at axial extension condition.

(a) Strain contour (D=10mm)  (b) Strain contour (D=20mm)  (c) Static stiffness curve

Figure 4. Simulation at radial condition.

Based on above curves, with empirical dynamic-static ratio and shock-dynamic ratio set respectively as 1.4 and 2.0, the shock stiffness is estimated in Table 3.

| Load condition        | Displacement (mm) | Static stiffness (kN/mm) | Shock stiffness (kN/mm) |
|-----------------------|-------------------|--------------------------|-------------------------|
| Axial compression     | U<10              | 0.67                     | 13.16                   |
|                       | 10≤U<50           | 4.7                      |                         |
| Axial extension       | 0≤U<50            | 1.05                     | 2.94                    |
| radial                | 0≤U<50            | 4.85                     | 13.58                   |

FEA results show that design of structure makes stiffness of the shock isolator alterable according to its deformation in axial compression, which can limit excessive displacement elastically. Also, shock stiffness is high in axial compression and radial direction, but low in axial extension.

4. Shock experiments

Prototypes of the shock isolator are installed in uniforms under the drop hammer of LC-4 impact machine as shown in figure 5. Prototypes and drop hammer will hit the anvil together after gravity drop. Mass and falling height of drop hammer can be increased until shock isolators reach the maximum deformation.
Accelerometers installed on the top of drop hammer will collect and transmit acceleration data to computer processing system. By processing data of displacement $x$ and force $F$, dynamic stiffness curve $F(x)$ is given to characterize hysteresis of rubber-metal parts. Typical curves of certain falling heights $H$ and masses $m$ are shown in figure 6. The shock stiffness rises obviously when displacement reaches around 10mm under small axial compression shock load, but under large shock load, the impact speed gets so high that stiffness variation can’t keep up. Also, Rubber-metal parts get strongly rigidified with large deformation when displacement reaches around 40mm, protecting the shock isolator from excessive deformation.

$$\int_{0}^{D_m} W F(x) \, dx =$$

According to time-domain vibration wave-forms, using free decay oscillation method, defining $A$ as attenuation rate of neighbor peaks, it can be calculated that the ideal damping coefficient $\xi = \ln A / 2\pi$, and equivalent damping ratio $\eta = \xi / 2\sqrt{mK_e}$. Calculation results are listed in table 4.

| Loading direction | Axial compression | Axial extension | Radial |
|-------------------|------------------|----------------|--------|
| Hammer mass $\ (t)$ | 4.82             | 4.99           | 5.05   |
| Falling height $\ (cm)$ | 35               | 1.2            | 45     |

Figure 6. Dynamic stiffness curves at different conditions.
Results verify the alterable shock stiffness in axial compression. Under large shock load, the shock isolator has high shock stiffness and strength in axial compression and radial direction, which provides adequate elastic limit capacity and shock energy absorption. However, shock stiffness and strength are low in axial extension, which makes axial extension unstable and excessive. This can be avoided by symmetrical installation of same shock isolators on both sides of equipment during practical use.

FEA calculation results match the experiment results well, with deviations less than 20%. As a conclusion, the FEA calculation method is applicable and accurate to fulfill engineering requirements.

5. Conclusions
An empirical FEA calculation method has been applied to study stiffness characteristics of a new type of stiffness variable shock isolator. Shock stiffness calculation results were then verified by shock experiments in each direction. Conclusions are as follow:

(1) This shock isolator has alterable stiffness in axial compression. Within 10mm of working stroke, low stiffness is provided for better vibration and shock isolation. Beyond 10mm of working stroke, high stiffness is provided for elastic limit.

(2) This shock isolator provides high shock stiffness, shock strength and damping coefficient in axial compression and radial direction, ensuring enough shock energy absorption within restricted working area under large shock load, thus protecting the equipment from secondary impact.

(3) This shock isolator may have excessive and unstable deformation in axial extension. To avoid rigid impact accident of equipment, symmetrical installation should be required for practical use on board.

(4) With the validity of empirical rubber constitution model and stiffness ratio coefficients, the FEA calculation method can meet the requirement of engineering application.

References
[1] Gaberson, H. A. (2000) Classification of violent environments that cause equipment failure. Sound and Vibration, 34(5): 16-23.
[2] Scavuzzo, R. J. (2000) Naval shock analysis and design (shock and vibration monograph series). Falls Church: The shock and vibration information analysis center.
[3] Ai, Wei., Jin, L. A., Chi, W. (2015) The status and prospect of ship vibration isolation technology. Marine technology, 2015(1):4-8.
[4] Han, Lu., Meng, X. S., Yan, M., Zhu, H. (2017) Analysis of the shock response of a single-layer vibration isolation system with a limiter. Noise and Vibration Control, 37(5):29-32.
[5] Wei, S., Xu, M. H., Huang, C. (2014) A design method of rubber vibration isolator. Development and Application of Materials, 29(2):11-16.
[6] Ren, J., Zhong, J. L., Ma, D. W. (2014) An accurate modeling method based on cord/rubber composite material micromechanics. Acta Materiae Compositae Sinica, 31(6): 1516-1524.