Design and Performance Analysis of Anti-sway & Anti-vibration System for Marine Operating Table with Inerter

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Abstract. In order to solve the low-frequency swing and vibration problems of the marine operating table, the inerter was added to the system of anti-sway and anti-vibration in this paper. The manuscript designed a simple system of anti-sway and anti-vibration for marine operating table with inerter and deduced its dynamic model. The analysis shows that the inerter can improve anti-sway and anti-vibration performance of marine operating table. Specifically, results show that all the anti-sway and anti-vibration performance that we analyzed can be improved by more than 45% when the inerter is greater than 0.5 (η>0.5). This work provides a design guidance for the anti-sway and anti-vibration design of marine operating table, which has a practical engineering significance.

1. Introduction
As an ideal modelling element, the inerter is defined to be a two-terminal mechanical device such that the applied force at the terminals is proportional to the relative acceleration between them [1]. The applications of inerter in various mechanical structures have been investigated in the past few years. In literature [2-5], we can get that inerter has been used in the field of motorcycle control systems, train and vehicle suspensions, buildings, etc. However, the inerter still has not been applied in the field of naval architecture and ocean engineering so far. In this work, the inerter is used in marine operating table which often requires in modern naval warfare to treat the wounded crews. Therefore, reducing the vibration and swing of the marine operating table has become an urgent problem to be solved. Based on this, we designed a kind of anti-sway and anti-vibration system for marine operating table with inerter and analyzed the performance of it. This paper explores the application of the inerter, laying the foundation for the practical application of it in the field of maritime.

2. System design
Marine operating table is mainly used in warships. However, most power resources of warships are
low-frequency and high-power machinery, besides, during the the voyage the warships produce low-frequency swing under the action of wind and waves, leading to the marine operating table inevitably affected by low-frequency vibration and low frequency swing, eventually reducing the efficiency of the operation. The inerter has the characteristics of preventing low-frequency vibration and passing high-frequency vibration [6]. Therefore, we considered designing the inerter to the system of marine operating table, and to improve its anti-sway and anti-vibration performance.

![Figure 1: Dynamic model of anti-sway and anti-vibration for marine operating table](image)

Figure 1 is the dynamic model schematic of anti-sway and anti-vibration system for the marine operating table designed in this paper. On both sides of the system we laid up II-ISD (Inerter-Spring-Damper) anti-vibration system [7], and in the middle we used spring-damping anti-vibration system(intermediate anti-vibration system). Such a hybrid system can be parallel to 3~4 times in the length of the marine operating table. In the following, we established a dynamic model equation for this system firstly, and then simulated the system by using MATLAB/Simulink to study its anti-sway and anti-vibration performance.

3. Establishment of dynamic model

Kinetic energy of the system shown in Figure 1:

\[ T = \frac{1}{2} m \dot{x}^2 + \frac{1}{2} J \dot{\theta}^2 + \frac{1}{2} b(\ddot{x} - \dot{\theta})^2 + \frac{1}{2} b(\ddot{x} + \dot{\theta})^2 \]  

(3.1)

And the potential energy:

\[ U = \frac{1}{2} K x^2 + \frac{1}{2} k (x - \theta l)^2 + \frac{1}{2} k (x + \theta l)^2 \]  

(3.2)

In the two formulas above:

- \( k, c, b \) —— stiffness, damping, inertia [8] of II-ISD anti-vibration system respectively;
- \( K, C \) —— stiffness and damping of intermediate anti-vibration system respectively;
- \( m \) —— the mass of marine operating table;
- \( x, \theta \) —— vibration displacement and swing angle of marine operating table;
- \( l \) —— the distance from the center of the marine operating table to both sides;
- \( J = m l^2 \) —— the moment of inertia around the center of the marine operating table.

According to the Lagrange equation:

\[ \frac{d}{dr} \left( \frac{\partial T}{\partial \dot{q}_r} \right) - \frac{\partial T}{\partial q_r} + \frac{\partial U}{\partial \dot{q}_r} = Q_r(t) \ (r=x, \theta) \]  

(3.3)

We get:
The external excitation and the damping force are all non-potential forces, and the generalized forces corresponding to the generalized coordinates are:

\[ Q_x = F_x + c(\ddot{x} - \theta) - c(\dot{x} + \ddot{\theta}) = F_x - (C + 2c)\dot{x} \]
\[ Q_\theta = M_x - cl(\ddot{x} - \theta) - cl(\dot{x} + \ddot{\theta}) = M_x - 2cl\dot{x} \]

Where \( F_x \) and \( M_x \) respectively represent excitation force from \( x \) and \( \theta \) directions.

Substitute (3.4) ~ (3.5) into (3.3):

\[ M\ddot{q} + C\dot{q} + Kq = F \]

Where \( M = \begin{bmatrix} m + b & 0 \\ 0 & (m + b)l^2 \end{bmatrix} \); \( C = \begin{bmatrix} C + 2c & 0 \\ 2lc & 0 \end{bmatrix} \); \( K = \begin{bmatrix} K + 2k & 0 \\ 0 & 2kl^2 \end{bmatrix} \); \( F = \begin{bmatrix} F_x \\ M_x \end{bmatrix} \); \( q = \begin{bmatrix} x \\ \theta \end{bmatrix} \).

\[ \dot{q} = \begin{bmatrix} \dot{x} \\ \dot{\theta} \end{bmatrix}; \quad \ddot{q} = \begin{bmatrix} \ddot{x} \\ \ddot{\theta} \end{bmatrix}. \]

Making \( |K - \omega^2 M| = 0 \), we get the natural frequencies of \( x \) and \( \theta \) directions:

\[ \omega_x = \sqrt{\frac{K + 2k}{m + b}} \cdot \sqrt{\frac{1}{1 + \eta}} \cdot \sqrt{\frac{K + 2k}{m}} \]
\[ \omega_\theta = \sqrt{\frac{2k}{m + b}} \cdot \sqrt{\frac{1}{1 + \eta}} \cdot \sqrt{\frac{2k}{m}} \]

Where \( \eta = b / m \) represents inertance-mass ratio. Obviously, the greater the \( \eta \), the smaller the natural frequencies (\( \omega_x \) and \( \omega_\theta \)). Specifically, when \( \eta = 0.5 \), \( \omega_{x,\eta}(\eta = 0.5) \approx 0.82\omega_{x,\eta}(\eta = 0) \); when \( \eta = 1 \), \( \omega_{x,\eta}(\eta = 1) \approx 0.71\omega_{x,\eta}(\eta = 0) \). Reasons are mainly the existence of the inerter, which is equivalent to increasing the mass of the system. Qualitatively, the lower the natural frequencies, the better the low-frequency anti-vibration effect.

4. Performance Analysis

In order to more intuitively analyze the system mentioned in this paper, we used an aculation example to specific analysis. Table 1 shows some basic parameters of a marine operating table.

| Parameter | Value     | Parameter | Value     |
|-----------|-----------|-----------|-----------|
| \( m \)   | 200kg     | \( l \)   | 26cm      |
| \( k \)   | 25kN/m    | \( K \)   | 30kN/m    |
| Damping ratio \( \zeta(c) \) | 0.1       | Damping ratio \( \zeta(C) \) | 0.1       |
| \( F_x \) | 4sin(30\pi t) (kN) | \( M_x \) | 8sin(75\pi t) (kN·m) |
We substituted \( F, M, C \) and \( K \) with specific data and expressed equation (3.6) as:

\[
\ddot{q} = -M^{-1}Cq - M^{-1}Kq + M^{-1}F
\]  

(4.1)

By simulating equation (4.1) in MATLAB/Simulink, we got the response of vibration displacement and acceleration, and also the vibration intensity.

The vibration displacement reflects the accuracy of the marine operating table, and the acceleration reflects its work safety. So we wanted to use the response of vibration displacement and acceleration to evaluate our design.

Figure 2 shows the displacement response of each degree of freedom with different \( \eta \); Figure 3 shows the acceleration response of each degree of freedom with different \( \eta \).

From Figure 2 and Figure 3, we can conclude that in the directions of \( x \) and \( \theta \), the larger the \( \eta \), the smaller the amplitude of the vibration displacement and acceleration; specifically, when \( \eta = 0.5 \), compared with \( \eta = 0 \), the amplitude in the \( x \) and \( \theta \) directions decrease by about 50% and 45% respectively; when \( \eta = 1 \), they are decreased by about 70% and 60% respectively.

To explain why the two kinds of amplitude reduction rates are similar, for no matter it is vibration displacement or acceleration, they both have the same amplitude amplification factor, so they should also have the same amplitude under the same inerterance.

As the inerterance increases, the amplitude should have been linearly reduced, but the dynamic characteristics of the overall system have changed (the natural frequency is reduced), so although the inerterance increases, the reduction rate of amplitude gets smaller on the contrary. This explains why the amplitude reduction rate is not a linear decline.

**Figure 2:** Displacement response of each degree of freedom with different \( \eta \)

**Figure 3:** Acceleration response of each degree of freedom with different \( \eta \)
5. Conclusions
In this work, we designed an anti-sway and anti-vibration system for marine operating table with inerter. Based on this, dynamic model of this system was established and its natural frequencies were analyzed. More specifically, we analyzed the response of a marine operating table. All of them achieved better results. Main conclusions are summarized below:

1) Inerter can reduce the natural frequencies of the system; specifically, when $\eta=0.5$, $\omega_{n,0}(\eta = 0.5) \approx 0.82\omega_{n,0}(\eta = 0)$; when $\eta=1.0$, $\omega_{n,0}(\eta = 1) \approx 0.71\omega_{n,0}(\eta = 0)$.

2) The system of marine operating table with inerter has a smaller displacement and acceleration response compared to $\eta=0$. Specifically, when $\eta=0.5$, the sway and vibration of marine operating table is reduced by more than 45%; when $\eta=1$, it is reduced by more than 60%.

In summary: The anti-sway and anti-vibration performance of marine operating table employing inerter has been improved.

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