Stability Analysis and Experimental Study of Portable Missile Control Device Test

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Abstract. According to the stability requirement of the missile launcher in operating the portable missile launching control device, a structure is designed. Firstly, the structural mechanics theory method is used to analyse the deflection of the outrigger. The force bending of the sector plate is analysed and the bending displacement of the rigid frame after the column is simplified is analysed. Then the main structure of the portable missile launching device is given and the stability of the outrigger, base and column system is analysed and simulated by ANSYS and ADAMS simulation software respectively. Finally, the stability of the equipment was tested to verify the calculation and analysis of the stability of the device. The feasibility and rationality of the design of the portable missile launching device was confirmed.

1. Introduction  
From the 1970s Germany began to study air defense missiles [1], to the current fourth generation of air defense missiles [2], countries are developing their own national air defense weapons and equipment. Among the air defense missile equipment of various countries, portable air defense missiles occupy a very important position [3]. Due to the excellent economic and portability of portable air defense missiles, it is favored by various countries [4]. In the live-fire field test, Garee M J used the Man-Portable Air-Defense System (MPADS) as the weapon of choice for experimental data sources [5]. Since the beginning of 1975, China has conducted in-depth research on portable air defense missiles [6]. After years of development and research, there have been many developments and breakthroughs.

In the development of missiles, the testing of missile performance plays an extremely important role [7]. During the entire testing process, the launch test is at the center, because it is the best way to test the comprehensive performance of the missile [8]. In practice, the test data available in the development of missile systems and in most cases in the test phase is usually limited, and the data obtained is failure and discontinuity [9]. Liu K et al. established a reliability model for portable air defense missile weapon systems, starting from the specific requirements of reliability design [10], combining the components of the portable air defense missile weapon system and the characteristics of use. Most of topics in military performance modeling have been successfully applied to simulators [11].

In 2007, Harbin Engineering University Wang Li-quan and Yin Tie-hong analyzed and calculated the force during the lifting process of the crane, and studied the load stability of the lifting process [12]. In 2010, Sun Hai-jian of Northeastern University conducted a detailed analysis of the stability of
In 2013, Zhao Rui-xue of Xuzhou Heavy Machinery Co Ltd used modern multi-body dynamics and Monte Carlo method to calculate the truck crane anti-overturning stability [14]. In 2018, Zhai Hao-xing of Heze City Product Inspection and Testing Institute's Iridium focuses on the analysis of the anti-overturning stability of the lifting machinery under various load conditions [15].

The stability of the portable missile control device is very important. It is necessary to analyze the stability of the device by using ANSYS and ADAMS simulation software. This is both targeted and of certain research value.

2. Static analysis of missile launching control device

In the portable missile launch control test equipment, the base supports all the weights of the upper column and the launching mechanism. The force analysis of the base determines the failure or not. Modeling the mechanical model of the outrigger can simplify the model of the outrigger into a simply supported beam subjected to two concentrated forces. The specific force diagram is shown in figure 1.

![Figure 1. Schematic diagram of Simplified beam](image)

2.1. Analysis of the bending process of the outrigger

Since the maximum static force of the launcher base needs to be 100kN, in the process of analysing the outriggers, it is assumed that the force is applied to the centre of the two stress points, then the force G is all assumed by the beam shown in figure 1 [16]. G is distributed to P1 and P2 according to the principle of force distribution, so the positive pressure at the point of action is

\[ P_1 = P_2 = \frac{G}{2} = 50kN \] (1)

According to the bending principle of the beam, the counterforces of the concentrated forces P1 and P2 at the two support points M and N are obtained.

\[ R_x = \frac{(2a + b)G}{2l} = 43406.6N \] (2)
\[ R_y = \frac{(2l - 2a - b)G}{2l} = 56593.4N \] (3)

Using the superposition method to calculate the deflection of the beam under two concentrated forces, and looking at the deflection of the calculated amount of superposition method [17], the deflection curve equation of the beam is obtained.

\[ v = v_1 + v_2 \]

The deflection curve equation is regarded as the superposition of the displacements under the action of P1 and P2 alone, so there is:

\[
\begin{align*}
  v_1 &= -\frac{P_1(l-a)x}{6EI_xl}(l^2 - x^2 - (l-a)^2), 0 \leq x \leq a \\
  v_2 &= -\frac{P_2(l-a)}{6EI_xl} \left( \frac{1}{l-a} (x-a)^3 + \left[l^2 - (l-a)^2\right]x - x^3 \right), a \leq x \leq l
\end{align*}
\] (4)
\[
\begin{align*}
    v_1 &= -\frac{P(l-a-b)x}{6EI_x} \left( l^2 - x^2 - (l-a-b)^2 \right), 0 \leq x \leq a+b \\
    v_2 &= -\frac{P(l-a-b)}{6EI_x} \left[ \frac{1}{l-a-b} \left( x-a-b \right) + \left( l-a-b \right)^2 + \left( l-a-b \right) x - x^2 \right], a+b \leq x \leq l
\end{align*}
\]

(5)

In the equation, \( EI_x \) is the bending rigidity of the beam, 304 stainless steel is used as the material of the beam, and the cross-sectional dimension is \( y=36 \text{mm} \), and the cross-sectional shape is as shown in figure 2.

![Figure 2. Schematic diagram of the cross section of the leg](image)

\[ EI_x = E \times \int_A y^2 \, dA = 1.024 \times 10^{12} \text{N/mm} \]

The deflection of the beam is calculated by the equations (4) and (5), and the values of the lengths are substituted, and the deflection curve of the beam can be obtained as shown in figure 3.

![Figure 3. Deflection curve of the simplified beam of the leg](image)

2.2. Force analysis of the column

According to the design specifications of the portable missile control test equipment, the force of the column is complicated. Firstly, the gravity of the launching mechanism is directly applied to the column. In addition, many components, such as operator seats, cooling cylinders, etc., are attached to the column by clamps. The force diagram of the column part is shown in figure 4.

![Figure 4. Schematic diagram of the force on the column](image)
Firstly, the effect of the force on the seat on the column system is calculated. The positive pressure exerted on the seat by the operator can be considered to be applied to a rigid frame of reduced diameter \[18\].

The BC segment is subjected to a concentrated force, and the distance between the point on the BC segment and the force point C is \(x_1\), and the bending moment of the BC segment is:

\[ M(x_1) = F_1 x_1 \]  
\[ (6) \]

At the same time, the partial derivative of the bending moment to \(f\) is:

\[ \frac{\partial M(x_1)}{\partial f} = 0 \]  
\[ (7) \]

Similarly, suppose there is a point on the AB segment, which is a distance \(x_2\) from the point B. Find the bending moment of the AB segment under the original force and the imaginary force and its derivative to \(f\) is as follows:

\[ M(x_2) = F_1 a + f x_2 \]  
\[ \frac{\partial M(x_2)}{\partial f} = x_2 \]  
\[ (8) \]

Applying the Cartesian theorem \[19\], the horizontal displacement of the B section is calculated in the case where the rigid frame is only subjected to the force \(F_1\) in Figure 5.

\[ \delta_b = \int_{1}^{2} \frac{M(x)}{EI} \frac{\partial M(x)}{\partial f} dx \]
\[ = \frac{1}{EI_1} \left[ \int_{0}^{1} (F_1 a + f x) \cdot x_1 dx + \int_{1}^{2} F_1 x_1 \cdot 0 dx \right] \]
\[ = \frac{1}{EI_1} \left( \frac{F_1 a b^2}{2} + \frac{f b^3}{3} \right) \]  
\[ (9) \]

It can be seen that since the heights of the two points are different, a simple superposition operation cannot be performed. The transformation can be performed by the similarity of the triangle, and the displacement expression caused by the positive pressure at the seat to the column system is:

\[ \delta_b^* = \frac{F_1 a b l}{2EI_1} \]  
\[ (10) \]

Therefore, it can be concluded that the bending displacement of the column system when subjected to the above various forces is:

\[ \delta = \sqrt{\left(\delta_b^* + \delta_o^*\right)^2} = \frac{1}{2EI_1} \sqrt{\left(F_1 a l^2\right)^2 + \left(F_1 a b l^2\right)^2} = 0.323\text{mm} \]

3. Simulation results of the column

In the ANSYS Workbench, the force of the column system is analysed. The analysis is performed solely by the force applied at the seat, applying a fixed constraint to it and the weight of the device. After the mesh is divided and analysed, the deformation of the column system of the test equipment is obtained. The cloud map is shown in Figure 5.
Figure 5. Column system deformation cloud map

It can be seen from the deformation cloud diagram of the column system that the displacement deformation at the seat clamp is relatively large, but the maximum displacement is only 1.65 mm, which is an acceptable deformation. At the same time, the tip of the column will be scratched by about 1 mm due to the force. The amount of deflection of the deflection relative to the launching mechanism is small, so the bending resistance of the column system can be considered to meet the requirements.

Firstly, the position of the overturning position needs to be determined. Since the base of the device is in a central symmetrical form, only a range of 60° is required to represent the force overturning of the entire device. The ADAMS model of the column system is as shown in figure 6.

Figure 6. ADAMS model diagram of the column system

The constraint and force are applied to the simplified model of the column, and the simulation is performed to obtain the displacement change of the corresponding point as shown in figure 7.

Figure 7. The displacement curve of the feet

It can be seen from figure 7 that the displacement of the base of the outrigger is always constant within a certain sometime, indicating that the heavy moment is greater than the overturning moment, so that no overturning occurs.

4. Device structure and experimental verification

The main content of the stability test of the column is to verify whether the column will cause the device to overturn under the pressure. According to the description and calculation in Section 2.2, pressure is applied at the joint between the seat and the launching mechanism to observe whether there is displacement at the foot. The test chart of the whole device is shown in figure 8.
For the overturning of the column and the base system, the main verification method is whether the foot displaces, so the displacement of the foot opposite the force position is measured, and the result is the height of the foot is 9.78mm when the force is not applied, and the height of the foot after the force is 9.97mm, the difference between the two is 0.19mm. Due to the unevenness of the ground, there may be unevenness and offset of the force of the base, so the movement of 0.19mm can be regarded as almost no overturning of the base and it can work normally. The correctness of the calculation and analysis of the overturning stability of the column in Section 2.3 was verified.

5. Conclusion

In this paper, the stability of the device is calculated and analyzed in detail. It is determined that the stability of the portable missile launching device meets the requirements. It provides a basis for the design and inspection of portable missile control devices. The test of the impact process needs to be carried out in the future; in addition, since the conditions for the field launch are not available, the experiment of the impact of the real impact on the overall stability of the launching mechanism and the device will be improved in the future.

Acknowledgments

This paper was funded by NSFC (Contract name: Research on ultimate bearing capacity and parametric design for the grouted clamps strengthening the partially damaged structure of jacket pipes), (Grant No: 51879063) and (Contract name: Research on analysis and experiments of gripping and bearing mechanism for large-scale holding and lifting tools on ocean foundation piles), (Grant No: 51479043). The views expressed here were the authors alone.

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