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STRUCTURAL ANALYSIS OF A LARGE VIBRATION TEST FACILITY FOR SPACE STATION

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The space station consists of several capsules. The capsules are launched individually and assembled on orbit which should undergo sine vibration testing before launching. The physical size and the weight of the capsules exceed any former satellites or spaceship in China, and the existing vibration test facilities at present are incapable of performing sine vibration test for the capsules. Therefore, a new large vibration test facility is designed and developed. The new vertical facility consists of four electromagnetic shakers, head expander, guiding system and supporting system, etc. The maximum excitation force of the facility achieves 1400kN. The structural analysis method for the design of the facility is investigated in this paper. Modal analysis of the system is performed firstly. Rocking modes are obtained, and is applied for optimizing the supporting system. The strength of the facility and the deformation of the guiding system are verified by static and dynamical loads analysis. Frequency response analysis is carried out to evaluate the dynamic performance of the facility. It is concluded that the structural analysis method can optimize the structure to improve the rationality and reliability of the facility.

Keywords: space station, vibration test facility, structure design, finite element analysis

1. Introduction

Vibration test must be conducted for capsules of space station which are launched individually and assembled on orbit. The vibration environment of the launch vehicle for spacecraft is simulated by vibration test facilities driven by electromagnetic or hydraulic shakers. The design and the manufacture quality of the spacecraft are validated by the vibration test. Driven by the development of large scale spacecraft, such as space station, the requirement of large vibration test facility has grown up. Several institutes of space technology in the world has built large vibration test facility for spacecraft, such as the Glenn Research Center of NASA constructed mechanical vibration facility of max vertical force 2135kN in 2011 for human space exploration plans [1]. The ESA/ESTEC has developed the QUAD facility providing a maximum force of 640kN by four electro-dynamic shakers [2].

The large vibration test facility usually consists of expander table, supporting structure and guiding machine etc. The structure of the vibration test facility is of complexity, and hence a lot of elaborate computations must be performed to evaluate the static and dynamical performance of the structure to insure the rationality of the design. Finite element analysis are used by Brian Ross et al to predict the moment reaction loads and cross-axis input limitation of a new large vibration test facility for the James Vebb space telescope [3]. A magnetostatic FEM analysis has been carried out to improve the design of a permanent magnet electro-dynamic vibrator [4]. Modal analysis and vibration response simulation by FEM are presented in Ref [5] with a good match to experimental results. It is shown that the FEM can provide effective technical support for the design of vibration test facility.

This paper presents the structural analysis method for the design of a large vertical vibration test facility. The facility consists of four electromagnetic shakers, head expander, guiding system and supporting system, etc. The maximum excitation force of the facility achieves 1400kN. The stiffness, strength and dynamical response characteristics of the facility are analyzed by FEM. The system performance is evaluated during the design process. It is observed that the analysis method plays a critical role for the rationality and reliability of the design of the facility.

2. Configuration of the large vibration test facility

The overall configuration of the large vertical vibration test facility with multiple exciters is shown in Fig. 1. The structure mainly consists of exciters, supporting structure, head expander, guiding bearing of the head expander and air springs etc. The four electromagnetic exciters provide 1400kN in maximum sine force. The head expander has a size of 3.8 m × 3.8 m, and is made of magnesium alloy. The guiding bearing of head expander is comprised of a pair of pre-loaded and passive bearings. The pre-load is 200kN. The supporting structure is made of steel and casting concrete. The whole facility has an enveloping volume of about 8 m × 8 m × 3.5 m (height). The main technical specifications of the facility are listed in Table 1.

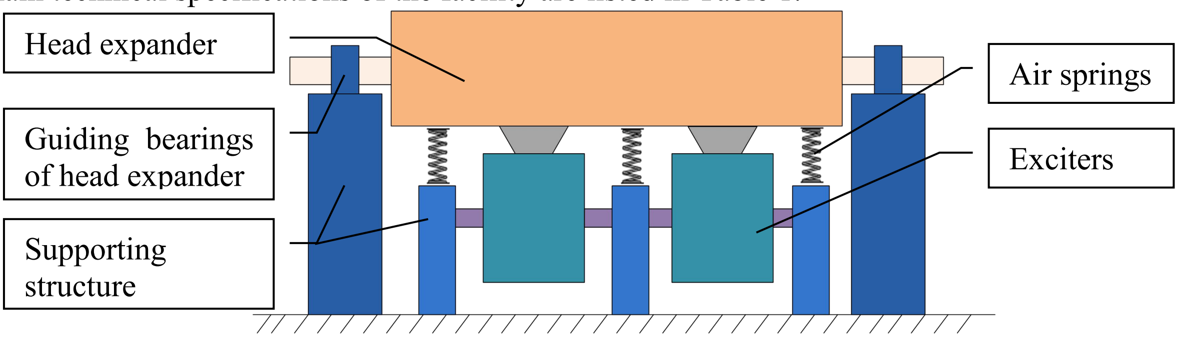


Figure 1: The sketch of the large vertical vibration test facility

Table 1: Technical specifications of the large vibration test facility

Max excitation force	1400KN
Excitation range (sine)	2~2000Hz
Max payload mass	40t
Acceleration bare table	10g
Max anti-overturning moment	350kN·m
Homogeneity bare table 2~100Hz	<±15%
Cross axis response bare table 2~100Hz	<10%

3. System modal analysis

The modal frequency of the structure is a basic and critical index to assess the dynamical performance of the vibration test facility. The vibration mode of the facility can significantly affect the quality of the test. The test state for vertical vibration is illustrated in Fig. 2. The payload is cantilevered at the expander table. The table and the payload are driven by a cluster of vertical exciters (four, for example), following an acceleration profile within a defined frequency range. Ideally, accurate cantilever boundary condition is expected for the payload. However, due to the elasticity of the facility structure, the actual vibration response of the payload is coupled with the facility. This coupling leads to a rocking mode vibration of the system which must be avoided during vibration test. For small vibration facility the structure is of high stiffness, so the coupling of the facility-payload system usually occurs outside the bandwidth of interest. But along with the system becomes more massive, the coupling shifts to the lower side of the frequency band, until entering the interesting range. In the preliminary design the analysis must focus on the vibration modes of the facility. The modes mainly related to the mass of the head expander, the stiffness of the supporting structure, and the stiffness of the guiding bearings. Detailed models of these components must be developed by FEM to predict the modes of the facility.

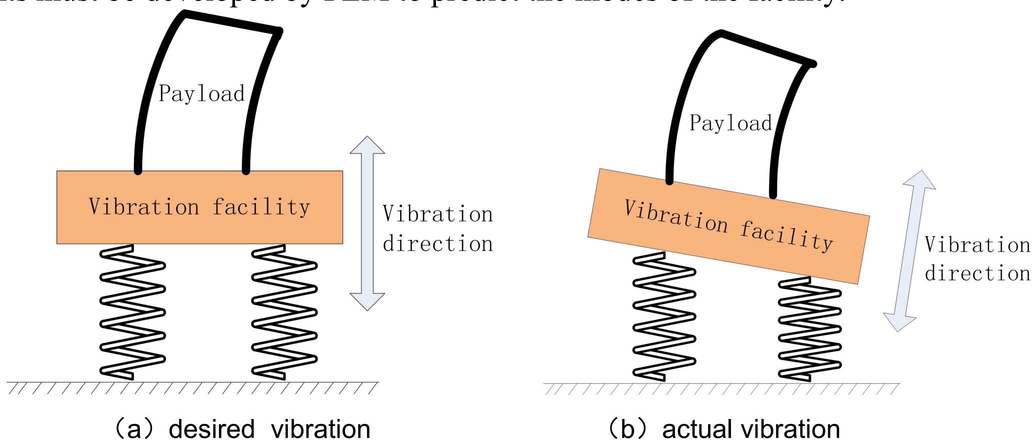


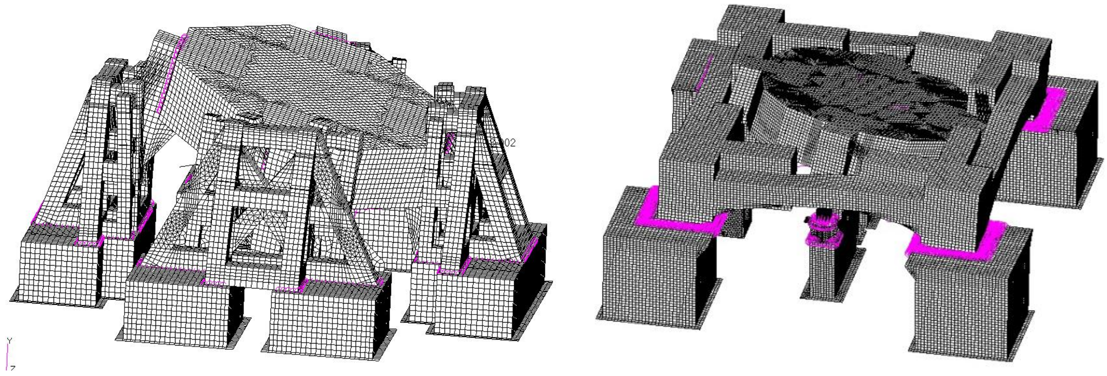
Figure 2: The effect of modes of vibration facility on test

According to the preliminary design, the head expander and the supporting structure are of plate type combined structure, and they are modelled by shell elements. The guiding bearings are modelled by spring elements based on the actual parameters. The moving components of the exciter are modelled by lumped mass element. The guiding machine of the moving components of the exciter is modelled by spring elements. All the connections of welding and bolted are simplified by ideal connections. And the material parameters are presented in Table 2.

Table 2: The material parameters of the vibration test facility

Material	Young's modulus/GPa	Poisson's ratio	Density /Kg·m^{-3}	Yield strength /MPa	Fatigue Strength of base metal /Mpa	Fatigue Strength of welding connector /Mpa
Magnesium alloy	45	0.35	1800	100	57.8	17.2~24.6
Steel	210	0.3	7800	215	/	/
Aluminum alloy	70	0.3	2700	160	/	/

Modal analysis of the preliminary design shows that the first rocking modal frequency of the facility is 36Hz, as shown in the left illustration of Fig. 3. The modal frequency is low, and the coupling of the facility with the payload will be strong. This will degrade the quality of test. It is identified by the analysis that the stiffness of the supporting structure of the head expander is inadequate. Therefore, improvement is made for the supporting structure to increase its stiffness. After the improvement the first rocking modal frequency increased to 52Hz, as shown in the right illustration of Fig. 3. Compared to the preliminary design the modal frequency increased 44% with the cost of 20% gain of mass. The improved design can reduce the coupling of the facility with the payload and improve the control accuracy of vibration test.



(a) 36Hz of preliminary design (b) 52Hz of improved design

Figure 3: The first rocking mode of the vibration facility

4. Static and dynamical loads analysis

Before the vibration test the facility statically carries the payload, hence static strength requirement for the facility must be met. The static loads are supported by head expander, air springs and supporting structure of the facility. By applying the designed static load of 400kN, the maximum stress on head expander is 5.5MPa, as shown in Fig. 4. And meanwhile, the maximum stress on supporting structure is 28.2MPa. It is observed that a good safety margin is obtained for static strength for the facility.

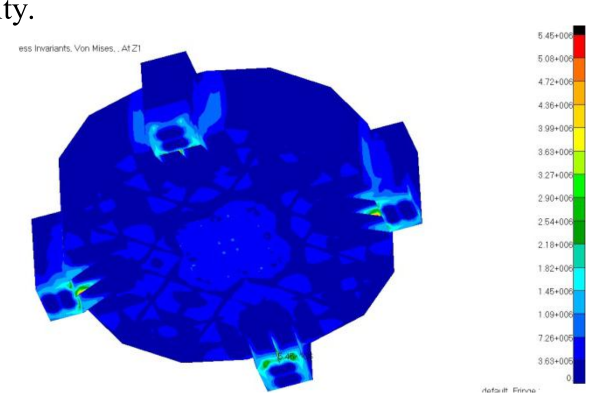


Figure 4: The static stress distribution of the head expander

During the vibration test the thrust force of the exciters transfers to the head expander and the payload, and meanwhile, high moment reaction loads are generated due to the off-centering of mass of the payload. These dynamic loads are transmitted to head expander, guiding bearings, supporting structure, and ground finally. In this case dynamical strength requirements for the facility must be satisfied. As a result of structural complexity the head expander is generally fabricated by welding process. And stress concentration occurs at the welded conjoint plates. Emphasised check must be paid to these welded connections, and fatigue strength is suggested to be applied to check the dynamical strength. In the meantime, the moment reaction loads induce elastic distortion of the head expander, which generates transverse pressing loads to the guiding bearings. And then these transverse pressing loads cause distortion of the bearings. Since the guiding bearings are critical components of the vibration test facility, and are of excessive high accuracy, inordinate distortion will cause deadlock of bearings and halt of the test. Local distortion analysis must be performed to guarantee the satisfaction of requirements of bearings distortion. The combination of maximum trust force of 1400kN and reaction moment of 350kN·m is adopted for the analysis. The maximum stress on head expander is 15.5MPa which located at the mounting surface of the bearings, as illustrated in Fig. 5. The margin of safety is 3.7 referencing to fatigue strength of the base metal. And meanwhile, the maximum stress at conjoint plates of the expander is 6.9MPa, and the margin of safety is 2.5 referencing to fatigue strength of the welding connector. And also, the maximum stress on supporting structure is 29.7MPa. The dynamical strength of the facility satisfies the design requirements.

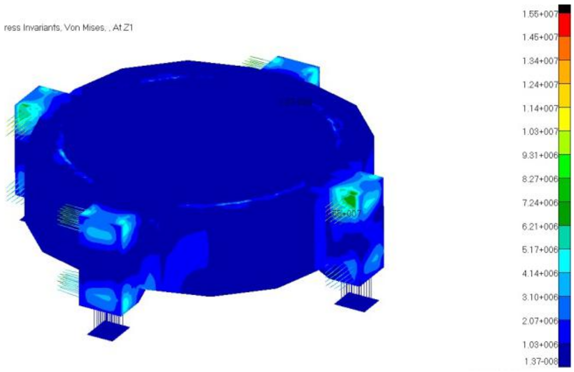


Figure 5: The stress distribution of the head expander in dynamical loads case

The distortion of the mounting surface of the bearings is also predicted in the analysis. The maximum distortion is 0.005mm which satisfies the requirement of 0.01mm, as shown in Fig. 6. The analysis result indicates that the guiding bearings can operate properly.

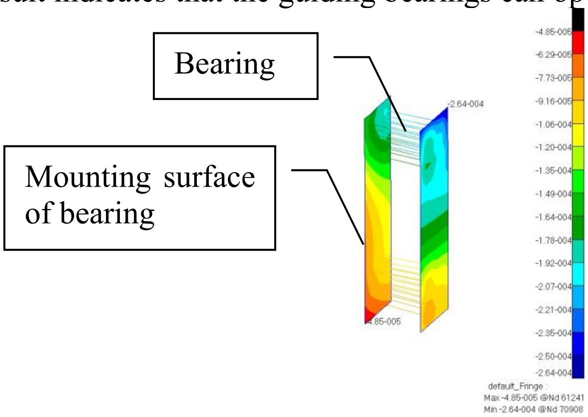


Figure 6: The distortion of the guiding bearing

5. Frequency response analysis

The frequency response characteristic is an important index to assess the transmission of excitation and the dynamical performance of the vibration test facility. The excitation force generated by exciters is firstly transferred to the head expander. Due to the structural elasticity, amplification, cross component, and inhomogeneity of vibration of the expander are produced. Along with the expander becomes larger and more complex, the modes of the expander become more intensive, and the frequency response characteristic of the expander becomes more complicated. In the frequency bandwidth of interest smooth frequency response is desired. And a good homogeneity for different spot of the table for large expander is expected. The transmissibility, the homogeneity, and the cross-axis vibration of the table are adopted to evaluate the frequency response characteristic of the facility.

The facility is driven by four synchronized exciters. By applying synchronized acceleration on the four exciters in the finite element model, the frequency response analysis is performed, and the acceleration response of the head expander is obtained. The distribution of the measuring points on the expander is shown in Fig. 7. The measuring points cover the areas of the outer ring, the middle ring, the center, the connecting position of exciter and the mounting position of the bearings. And the typical points in the central area are also illustrated.

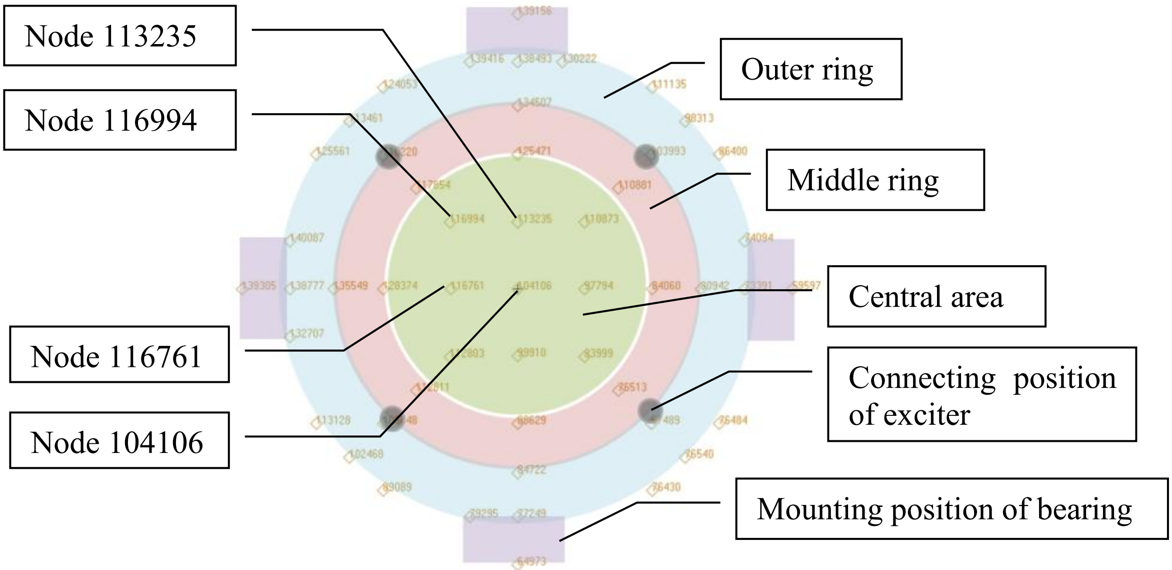


Figure 7: The measuring position in frequency response analysis

The vibration transmissibility of the central area of the expander is shown in Fig. 8. It observed that the acceleration is amplified at 225Hz and 310Hz. This amplification is mainly caused by the local first and second bending modes of the expander, as shown in Fig. 9. It is necessary that the response amplification of the expander occurs outside 100Hz for test of large spacecraft.

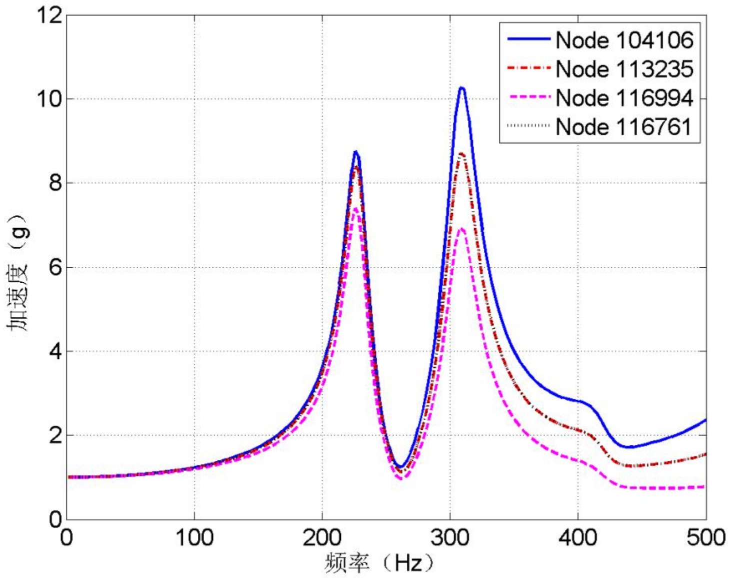


Figure 8: The vibration transmissibility of the central area of the head expander

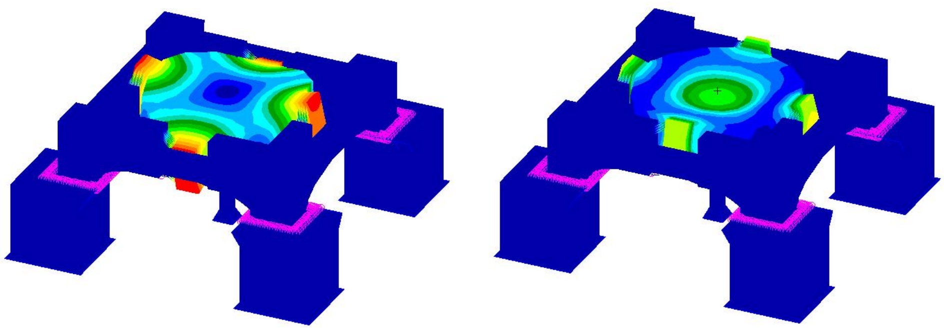


Figure 9: The first and second local modes of the head expander

By comparing the vibration transmissibility of the measuring points of other areas with those of the central area, the degree of homogeneity of vibration magnitude of the expander is calculated. According to the results, the degree of homogeneity of all the measuring areas in the frequency range of 2~500Hz is less than 12.3%, which is superior to the requirement of class-A of national standard 'Verification Regulation of digital electrodynamic Vibration Testing System'. The calculation of degree of homogeneity is followed as

$$N = \frac{|a_{\Delta\max}|}{a} \times 100\% . \tag{1}$$

where a indicates acceleration magnitude of the central measuring point, and $|a_{\Delta\max}|$ is the maximum difference between the central measuring point and the other measuring points in the same test.

The cross-axis vibration characteristic of the expander is shown in Fig. 10. It is observed that there is no obvious cross-axis vibration in the range of 2~100Hz. For the different measuring areas the cross-axis vibration of the central area is the mildest, and the cross-axis vibration of the mounting position of bearing is the most intense.

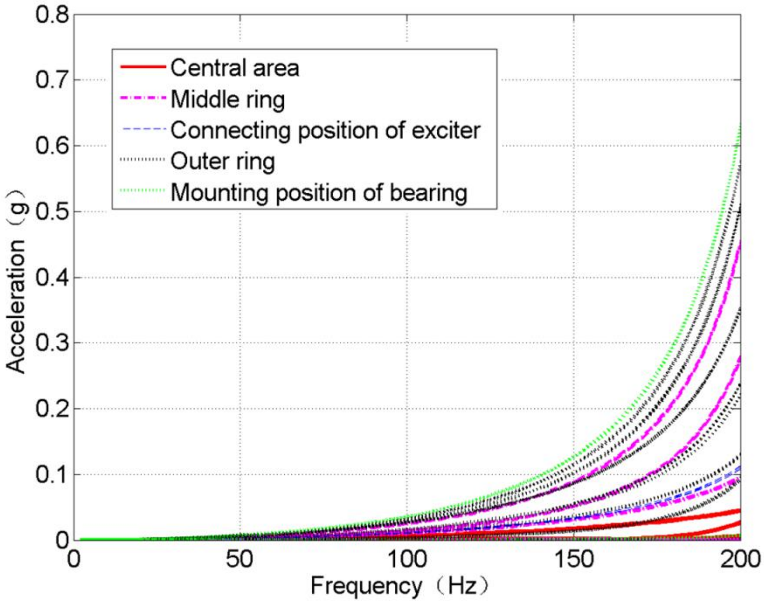


Figure 10: The cross-axis vibration characteristic of the head expander

According to the analysis, the ratio of cross-axis vibration of the central area is calculated as 0.05%, and the ratio of the mounting position of the bearings is 3.5% in 2~100Hz. And hence a satisfied cross-axis vibration performance is achieved. The calculation of ratio of cross-axis vibration is followed as

$$T = \frac{\sqrt{a_x^2 + a_z^2}}{a_y} \times 100\% . \tag{2}$$

where a_x and a_z indicate the acceleration components in the two horizontal directions respectively, and a_y is the acceleration component in the vertical direction which is the excitation direction.

6. Conclusions

The structural analysis method for the design of a large and complex vibration test facility is investigated in this paper. The facility is dedicated to vibration test of space station capsules, and is the largest vibration test facility at the BISEE to date. Comprehensive and detailed analyses are performed to verify the rationality of the design, and to improve the performance of the structure. The results show that the performance satisfies the design requirements. The facility had been manufactured and accepted by test. It is concluded that the analyses effectively support the development of the facility.

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