International Journal of Research in Engineering and Science (IJRES)
ISSN (Online): 2320-9364, ISSN (Print): 2320-9356
www.ijres.org Volume 3 Issue 11 ǁ November. 2015 ǁ PP.09-16

Static and Dynamic Characteristics Analysis of Three Coordinates manipulator based on Workbench

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Abstract: The paper studied the static and dynamic characteristics of the three coordinates manipulator which was used in reliability test system for low voltage electrical apparatus. The deformation cloud diagram and stress cloud diagram were obtained by using the finite element analysis software Ansys Workbench to carry out the static analysis, and the result showed that the mechanism satisfied the rigidity and the intensity request. The modal analysis of the mechanism which got the natural frequencies and the corresponding mode shapes of the first 8 order and compared the resonance frequency of the actual work environment was carried out. The random vibration analysis of the mechanism wad carried out on the basis of modal analysis, which got the acceleration response in the direction of x、y、z and the equivalent stress cloud diagram of the mechanism. According to the results of analysis, the vibration control of the body should be applied to reduce the random vibration response value.

Key Words: manipulator, static analysis, modal analysis, random vibration analysis, finite element analysis

I. Introduction

Modal analysis is a method to study the structure dynamic characteristics, and the basics of the structure dynamics analysis. harmonic response, random vibration analysis must be carried out after modal analysis [1]. Modal is the natural vibration characteristics of the mechanical structure, each mode has a specific natural frequency, damping ratio and mode of vibration. Modal analysis can be used to determine the natural frequency and mode shape of the structure, which is determined by the geometry and material properties of the structure and the form of constraints, and has nothing to do with the external conditions[2]. The modal analysis of the mechanism can provide the basis for the structural vibration analysis, vibration fault diagnosis and prediction and structure optimization design[3].

The research object of this paper was the simplified model of low voltage electric traction mechanism in low voltage electrical apparatus reliability test system. Main function of the mechanism is to achieve precise positioning of the low-voltage electrical contact position, and through the realization in the vibration environment of low voltage electrical appliance contact traction to realize detection of low-voltage electrical voltage and current parameters[4]. The mechanism as shown in the figure 1 was designed according to its function, which realized mechanical claw installed on the module group in the z axis move in x, y, z directions through four ball screw guide module groups.

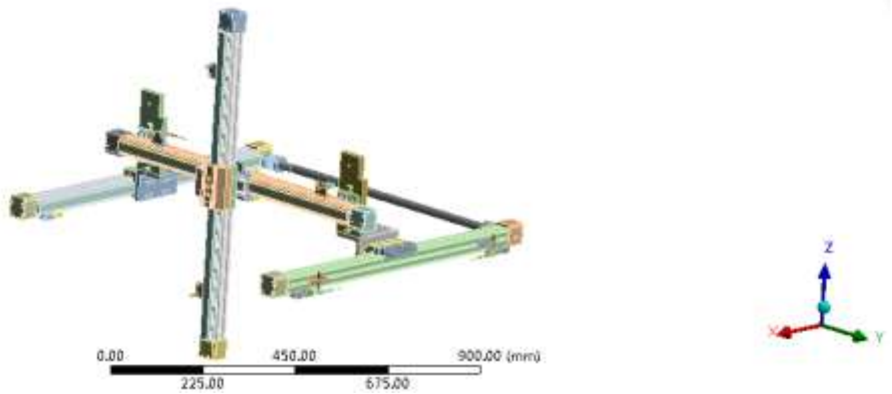

Figure 1. the simplified model of 3-DOF manipulator

II. finite element analysis

In order to improve the operation speed and reduce the computational time the 3D model of the mechanism was simplified considering some tiny characteristics have little impact on the overall performance on the mechanical arm : delete some of the smaller size of the fillet, chamfer and bolt holes features[5]. The mechanism model was built in the 3D Modeling Software Solidworks and transformed into IGES format and converted into ANSYS Workbench. The material of the mechanism was structural steel which elastic modulus E=2×105MPa and poisson's ratioμ=0.3 and the density p=7.85g/cm3. Divided the grid of the mechanism by using method control to block drawing grid. The hex dominant method is adopted to appropriately refine the grid of module between the connected parts and floor board connected X direction module and a flat. The thin slice of the parts were meshed into shell element (sweep method) which not only ensured the accuracy of the analysis results but also reduced the number of grid[6]. The rest of the other parts were divided by the method of free partition, and the total number of nodes was 1076523, the number of units was: 345874.

III. Static characteristic analysis of 3-DOF Manipulator

Static analysis is the analysis of the structure performance under the static load. Static analysis must be considered in most of the structure design, which is mainly to consider the deformation of the structure under the static load and constraint force, stress and strain distribution etc, is the basis of further structure dynamics and other mechanical analysis. According to Darren Bell's principle[7], the dynamic problem can be transformed into a static problem by means of adding the inertia force, which is to be converted to the equilibrium problem of the elastic body, namely:

$$F+FN+(-ma)=0$$

Toke the static analysis of the mechanical arm, fixed the 16 faces connected the two parallel modules in X axis direction and the platform, and exerted 300N tension on the plane where the gripper was installed on the Z axis module, which simulated the tension mechanical claw drag the low-voltage electrical contact. According to the theory of structural mechanics, the module in the direction of Y axis get maximum deformation when the module in the direction of Z axis lands in the middle of module in the direction of Y axis. Therefore the static analysis of the mechanical arm was carried out in the position, and the deformation map and the stress map were obtained as shown in Figure 2 and Figure 3.

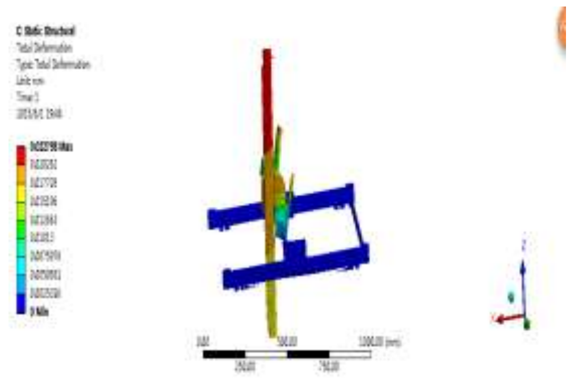

Figure2.the deformation map of static analysis

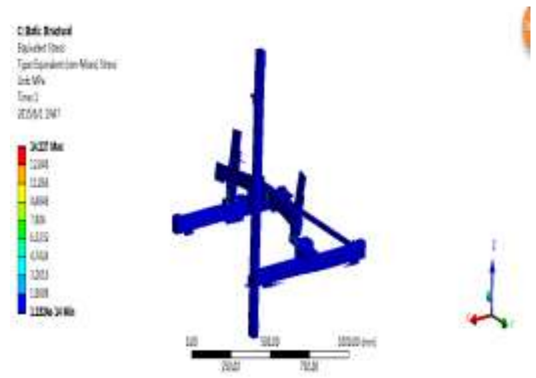

Figure3.the stress map of static analysis

As is shown in Fig 2, the maximum deformation of the mechanism lied in the upper part of the module in the direction of Z axis and the deformation is 0.034 mm, the allowable defection of the flexural member is L/500 (L is the span of flexural member)equals to 1.8 mm which is much less than the allowable defection of the member. Therefore, the design of the mechanism is in accordance with the rigidity requirement. As is shown in Fig 3, the maximum stress of the mechanism produced at the boundary constraints where stress concentration produced was 25.078MPa, which was much lower than the yield strength of structural steel, 250MPa.

IV. Dynamic characteristic analysis of three-DOF mechanical arm

4.1 Modal analysis of 3-DOF mechanical arm

The core content of modal analysis is to identify the modal parameters of the structure. the differential equations of motion for structure:

$$[M]\left\{\ddot{q}\right\}+[C]\left\{\dot{q}\right\}+[K]\{q\}=\{F(t)\} \tag{1}$$

Where: [M], [C], [K] respectively mean the quality matrix, the stiffness matrix and damping matrix of the system, {q}, {F(t)} respectively refers to the system displacement response vector of all points and the external excitation vector of system. Free vibration is happened excluding the effect of damping, which means: [C], {F(t)} are 0, and the differential equations of motion is changed:

$$[M]\left\{\ddot{q}\right\}+[K]\{q\}=\{0\} \tag{2}$$

Assume the solution of equation is harmonic motion:

$$\{q\}=\{Q\}e^{jwt} \tag{3}$$

Where :the elements in {Q} represent the amplitudes of each point in the system. Then turn equation (3) into equation (2), get:

$$\left([K]-\omega^2[M]\right)\{Q\}=(0) \tag{4}$$

In order to make the equation with non-zero solution, then:

$$\left|[K]-\omega^2[M]\right|=0 \tag{5}$$

[M] and [K] are positive definite matrices. Expand equation (5) and get N eigenvalues $\omega_r^2 (r = 1,2 \cdots N)$, the square root ω_r is the natural frequency of the system. In order of size: $\omega_1 < \omega_2 < \cdots \omega_N$, are respectively called the 1 order, 2 order... N order natural frequency. Turn each eigenvalue into equation (5) and get corresponding vector $\{\Phi\}_r$, which is the so-called modal(mode shape), and also known as the vibration shape because it is the deformation shape of system modal vibration.

The block Lanczos mode extraction method, which is mainly applicable to large symmetrical characteristic value problem in addition can well handle the vibration type of rigid body, and solve for the large and medium sized model, is a powerful function method[8] , was used to carry out modal analysis of 3-DOF mechanical arm. The 3-DOF manipulator is a continuous and uniform distribution of mass and elasticity, which have infinite modes in theory. However, the low order modes' natural frequency was low, which was closer to the working frequency band and played a more important role in the structure's vibration mode. The first 8 order modes of the structure were extracted consequently, and the natural frequency and vibration modes were obtained. The first 8 order modes shapes and natural frequencies of the 3-DOF manipulator were displayed in Table 1.

Table 1. The first 8 order modes shapes and natural frequencies of the 3-DOF manipulator

order	Natural frequency/Hz	Modal description
1	67.609	Modules in the direction of Y、Z axis swing along the X axis
2	107.84	Module in the direction of Y axis swing along the X axis
3	139.62	Modules in the direction of Y、Z axis swing along the Y axis
4	157.47	Modules in the direction of Y、Z axis swing along the Z axis
5	178.74	Modules in the direction of Y、Z axis swing along the Y axis
6	256.79	The module in the direction of Y twist axial
7	304.64	The X axis direction module has a first-order bending, and the lower end of the Z axis module is bent.
8	333.08	first-order bending of Z axis direction module in the ZY plane, Y axis direction module swing along the Y axis direction

The table indicated that the first 5 order mode shapes of the 3-DOF mechanism were modules swung along the X、Y、Z direction, mode shapes were more and more complex with the improvement of the order number that mode shapes of 6、7、8 order with local bending and torsion.

The vibration excitation of vehicle low-voltage electric was mainly caused by three reasons: the vibration caused by wheel imbalance, the outbreak vibration of engine under idle speed and common speed, the unbalance vibration of transmission shaft. When the speed is about 85km/h, frequency of the excitation caused by wheel imbalance generally lower than 11 Hz[9]; when passenger car engine is at idle speed for 700r/min, outbreak frequency is almost 35Hz[10]; and unbalance vibration frequency of transmission shaft at a speed of 50~80km/h is about 33~68Hz according to the relevant research data[11]. The first order natural frequency of the traction mechanism is 67.609Hz, so the design of the mechanism needs to be improved to obtain good dynamic characteristics.

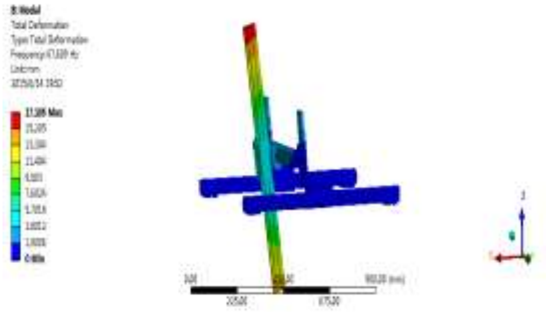

Figure 4. the first order mode of the mechanism

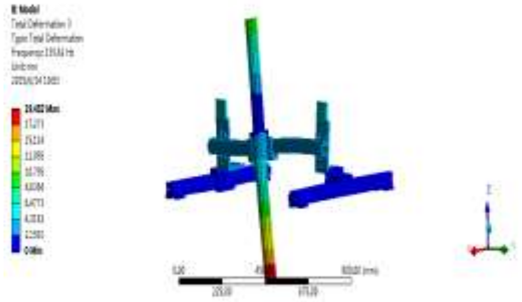

Figure 5.the third order mode of the mechanism

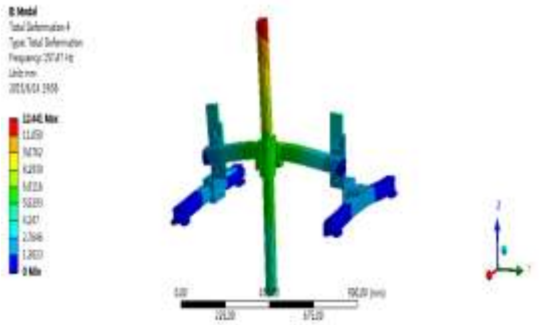

Figure 6.the fourth order mode of the mechanism

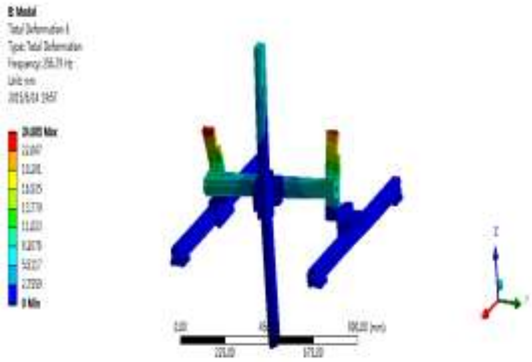

Figure 7.the sixth order mode of the mechanism

4.2 Random vibration analysis of 3-DOF mechanical arm

The random vibration analysis is a technology combines the result of the modal analysis with a known spectrum to calculate the displacement, velocity, acceleration and stress response of the model. The random vibration response mean array of the traction mechanism is:

$$\{\mu_x(t)\} = E[\{x(t)\}] = [\tilde{P}]\int_{-\infty}^{+\infty}[h(\tau)]\,d\tau[\tilde{P}]^{\tau}\{\mu_F\} = [\tilde{P}][H(0)][\tilde{P}]^{\tau}\{\mu_F\} \tag{6}$$

Where: $\{\mu_x(t)\}$ means the average vector of the traction mechanism response; $x(t)$ refers to the time domain response vector; $[\tilde{P}]$ refers to regular modal matrix; μ_F refers to the average vector of excitation; $h(\tau)$ means impulse response function matrix; $H(0)$ means frequency response function matrix when ω equals to 0. Frequency response function matrix is a diagonal matrix:

$$[H(\omega)] = \mathrm{diag}(H_j(\omega)) \qquad (j = 1,2,3,\cdots\cdots,\mathrm{n}) \tag{7}$$

Where: $H_j(\omega)=\frac{1}{\omega_j^2-\omega^2+2i\xi_j\omega_j\omega}$, ω_j、ξ_j respectively mean jth order frequency and corresponding damping ratio.

The random vibration response correlation function matrix of the traction mechanism is:

$$[R_x(\tau)]=\mathrm{E}[\{x(t)\}\{x(t+\tau)\}^T]=\begin{bmatrix} R_{x_1x_1}(\tau) & \cdots & R_{x_1x_n}(\tau) \\ \vdots & & \vdots \\ R_{x_nx_1}(\tau) & \cdots & R_{x_nx_n}(\tau) \end{bmatrix}$$

$$=\iint_{-\infty}^{+\infty}[\tilde{P}]\,[h(\varepsilon_1)][\tilde{P}]^\tau[R_F(\tau+\varepsilon_1+\varepsilon_2)]\iint_{-\infty}^{+\infty}[\tilde{P}]\,[h(\varepsilon_1)][\tilde{P}]^\tau\,\mathrm{d}\varepsilon_1\mathrm{d}\varepsilon_2 \tag{8}$$

Where: $[R_F(\tau)]=\begin{bmatrix} R_{F_1F_1}(\tau) & \cdots & R_{F_1F_n}(\tau) \\ \vdots & & \vdots \\ R_{F_nF_1}(\tau) & \cdots & R_{F_nF_n}(\tau) \end{bmatrix}$, the diagonal elements are autocorrelation function and the non diagonal elements are cross-correlation function in $[R_F(\tau)]$ and $[R_x(\tau)]$.

Power spectral density function matrix for random vibration response is:

$$[S_x(\tau)]=\begin{bmatrix} S_{x_1x_1}(\tau) & \cdots & S_{x_1x_n}(\tau) \\ \vdots & & \vdots \\ S_{x_nx_1}(\tau) & \cdots & S_{x_nx_n}(\tau) \end{bmatrix}=[\tilde{P}][H^*(\omega)][\tilde{P}][S_F(\omega)][\tilde{P}][H(\omega)][\tilde{P}]^T \tag{9}$$

Where: $[H^*(\omega)]=[H(-\omega)]$.

Mean square matrix of random vibration response is:

$$[\Psi_x^2]=\mathrm{E}[\{x(t)\}\{x(t)\}^T] \tag{10}$$

In the formula, the relationship between the power spectral density function matrix and the mean square matrix、the correlation function matrix is:

$$[\Psi_x^2]=[R_x(0)]=\int_{-\infty}^{+\infty}[S_x(\omega)]d\omega \tag{11}$$

The research object of this paper is the traction mechanism in the reliability detection system of low voltage electrical apparatus. Taking into account the mechanism is placed in cabinet on vibration table in vibration, which transfers to the mechanism through the vibration table, cabinet, platform, changes displacement in the vertical direction, these are random excitation on the mechanism. If this excitation were too large, it would affect the operation of the mechanism, but also cause fatigue damage to the body. Therefore, it is necessary to carry out the random vibration analysis of the mechanism.

Added PSD displacement on 16 fixed surfaces of the mechanism to simulate the vibration transmitted to the mechanism, inputted the tabular data as shown in figure 8 for load data and set the displacement direction as z axis. 1 sigma acceleration nephograms and stress nephogram were shown in Figure 9 to Figure 12.

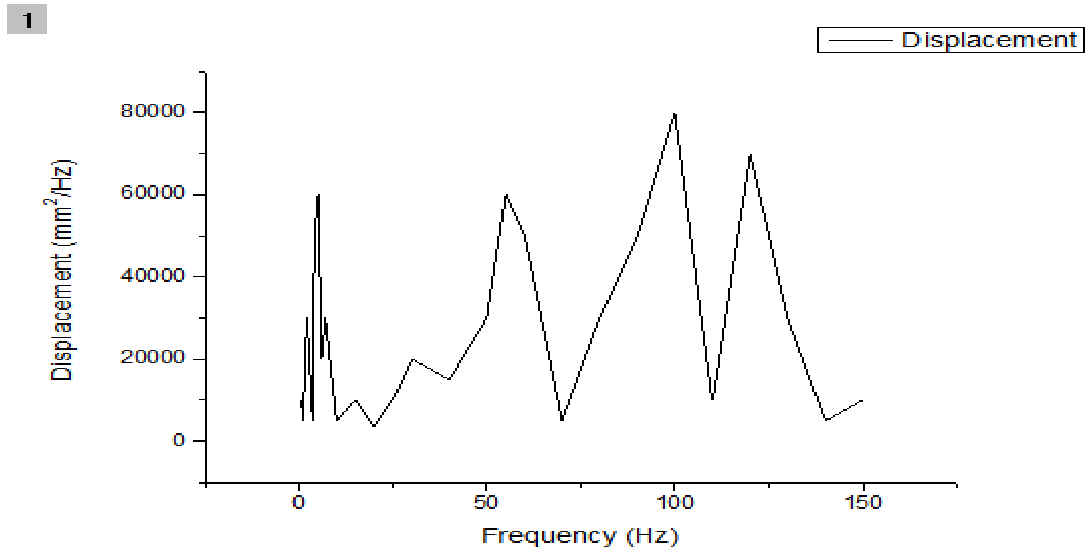

Figure8. Random vibration spectrum

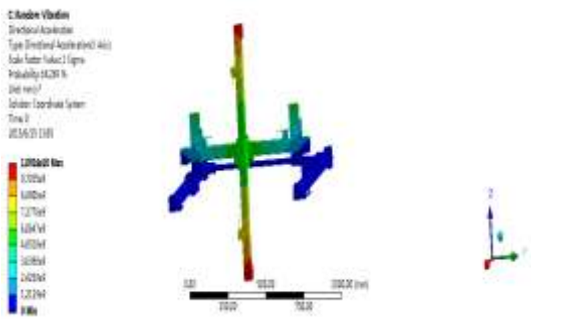

Figure9. acceleration nephogram in the direction of x

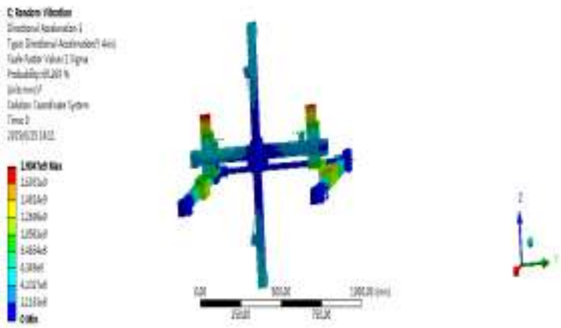

Figure10.acceleration nephogram in the direction of y

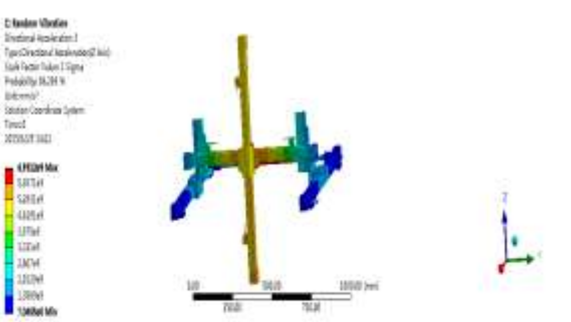

Figure11.acceleration nephogram in the direction of z

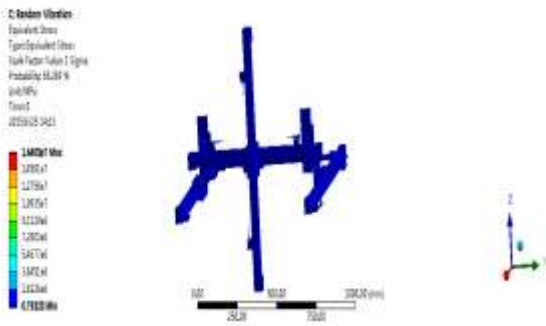

Figure12.stress nephogram of random vibration

According to the analysis results, the maximum acceleration of traction mechanism in x, y, x axis were respectively 1.09×1010mm/s2, 1.9×109mm/s2, 6.6×109mm/s2, the maximum equivalent stress was 1.6×107MPa. The acceleration of the mechanism in the direction of x was maximal and acceleration response、stress of the mechanism under given random excitation were large. The coupling effect of multi axis vibration is more complex and the natural frequencies are low[12], the random vibration response value can be effectively reduced through vibration control of the mechanism.

V. Conclusion

The paper researched low voltage electric traction mechanism in low voltage electrical apparatus reliability test system and carried out its finite element analysis through simplifying small features and refining local grids which not only accelerated computing speed but also ensured the accuracy of results.

The static analysis of the mechanism was carried out, and the deformation and stress of the mechanism are obtained, which indicated that the mechanism meet the intensity and stiffness requirements;

The modal analysis of the mechanism was carried out, and the natural frequencies and corresponding vibration mode of the first 8 orders were obtained, which indicated the first order natural frequency is 67.609Hz within the area of the unbalance vibration frequency of transmission shaft at a speed of 50~80km/h , 33~68Hz. Therefore the design of the mechanism should be improved to obtain better dynamic performance;

The random vibration analysis of the mechanism, which obtained the acceleration response values in the direction of x、y、z and the stress are large, indicated the vibration control should be applied to the mechanism to reduce the random vibration response value, was carried out on the basis of modal analysis.

VI. Acknowledgment

This work was supported by the the Local capacity-building projects in Shanghai Universities and Colleges [Grant no: 14110501200].

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