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Dynamics analysis of vibration process in Particle Impact Noise Detection

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Abstract: Particle Impact Noise Detection (PIND) test is a reliability screening technique for hermetic device that is prescribed by MIL-PRF-39016E. Some test conditions are specified, although MIL-PRF-39016E did not specify how to obtain these conditions. This paper establishes the dynamics model of vibration process based on first order mass-spring system. The corresponding Simulink model is also established to simulate vibration process in optional input excitations. The response equations are derived in sinusoidal excitations and the required electromagnetic force waves are computed in order to obtain a given vibration and shock accelerations. Last, some simulation results are given.

Key words: Particle Impact Noise Detection (PIND), Vibration, Electromagnetic force
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INTRODUCTION

Military Electronic Components are the indispensable fundamental components in national defense electronic systems. Space relays which are kinds of automatic manipulative components to navigation, ignition control, signal transmission, source transfer and attitude regulating are applied in satellite, launch vehicle, guided missile, nuclear weapon, aircraft, warcraft, weapon carrier, torpedo, radar, etc. "Devine Boad-4" airship uses above one thousand relays and redundant design of double relay group is used to improve reliability. It is quite evident that the reliability of the space relay decides the reliability of the aerospace system (Zheng and Fang, 2001; Huang, 1993).

To avoid using relays with particulate contaminants in aerospace applications, a reliability screening test, i.e. Particle Impact Noise Detection (PIND), is used to detect particulate contaminants before delivery. Though PIND test exerts huge effect on safeguard reliability, misjudgments and skipped discretions in PIND test often occur. Moreover, the relay which has passed PIND test cannot pass the next time PIND test, which makes the degree of confidence of PIND test dubious (Ding *et al*., 2004; Roettjer, 2005). So it is necessary to research PIND test theory.

Vibration test is the most important part of PIND test. In this paper, after the dynamics model of vibration test is established, the corresponding electromagnetic force is computed to obtain the specific vibration test condition, and Simulink model of vibration test is also established. Simulation results showed that accurate vibration test conditions can be obtained by using the computed electromagnetic force.

DYNAMICS ANALYSIS OF VIBRATION PROCESS IN PIND

The existing PIND testers produced by Spectral Dynamics Corporation and Model 4501 PIND testers are applied widely in the world. Fig.1 shows the vibrator structure of Model 4501 PIND tester. Vibrator is kind of electromagnetic structure, with acoustic and acceleration sensors fixed on top of the armature, a stainless hammer is fixed below the armature, the armature holds to equilibrium position and a magnet which is fixed on an anvil is under vibrator.

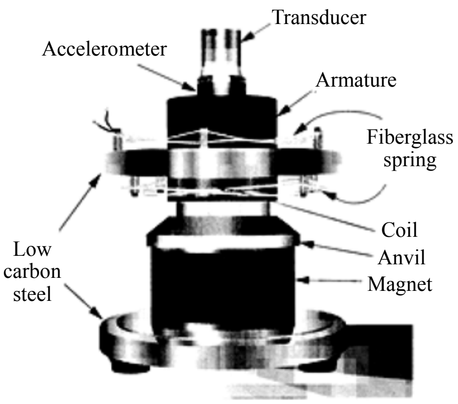

Fig.1 Structure diagram of Model 4501

During vibration test process, the armature and sensors move up and town under the alternating electromagnetic force.

The fiberglass spring can be regarded as a linear damping spring, so vibration test process is equivalent to unidirectional mass-spring system. Fig.2 show the mass-spring system, M is the mass of armature, C is the damping coefficient, K_1 is the stiffness of the fiberglass spring and F is the electromagnetic force. When F is equal to zero, the fiberglass spring fixes the armature in equilibrium position by counteracting gravitation. The constant fore and its corresponding steady displacement can be neglected in analysis.

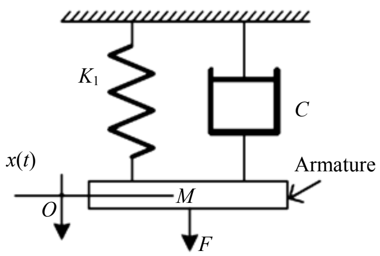

Fig.2 The mass-spring system in vibration process

Positive direction is down and steady equilibrium position is zero point. Let $x(t)$ be the armature displacement, which can be derived as

$$M\ddot{x} + C\dot{x} + K_1 x = F. \quad (1)$$

If electromagnetic force has sinusoidal waveform, with F=$F_A$cos(ωt), Eq.(1) becomes

$$M\ddot{x} + C\dot{x} + K_1 x = F_A \cos(\omega t), \quad (2)$$

where F_A is the amplitude of electromagnetic force and ω is angular frequency of electromagnetic force.

Transforming Eq.(2)

$$\ddot{x} + 2r\omega_1\dot{x} + \omega_1 x = \omega_1^2 A\cos(\omega t), \quad (3)$$

where ω_1 is the undamped intrinsic angular frequency of the vibration system, equal to $\sqrt{K_1/M}$ rad/s, r is the ratio of damping, equal to $C/(2M\omega_1)$, A is the steady displacement by electromagnetic force F_A, equal to F_A/K_1 m.

The general solution of Eq.(3) is

$$x(t) = X\cos(\omega t - \varphi). \quad (4)$$

Substituting Eq.(4) into Eq.(3)

$$\begin{aligned} &X[(\omega_1^2 - \omega^2)\cos\varphi + 2r\omega_1\omega\sin\varphi]\cos(\omega t) \\ &+X[(\omega_1^2 - \omega^2)\sin\varphi - 2r\omega_1\omega\cos\varphi]\sin(\omega t) \\ &= \omega_1^2 A\cos(\omega t). \end{aligned} \quad (5)$$

Since Eq.(5) is always true for arbitrary time, the corresponding factors of cos(ωt) and sin(ωt) should be equal on either side:

$$\begin{cases} X[(\omega_1^2 - \omega^2)\cos\varphi + 2r\omega_1\omega\sin\varphi] = \omega_1^2 A; \\ X[(\omega_1^2 - \omega^2)\sin\varphi - 2r\omega_1\omega\cos\varphi] = 0. \end{cases} \quad (6)$$

Solving Eq.(6)

$$\begin{cases} X = \dfrac{A}{\sqrt{(1-(\omega/\omega_1)^2)^2 + (2r\omega/\omega_1)^2}}; \\ \varphi = \arctan\dfrac{2r\omega/\omega_1}{1-(\omega/\omega_1)^2}. \end{cases} \quad (7)$$

Eq.(7) certainly shows that Eq.(4) is the solution of Eq.(3), where X and φ are determined by Eq.(7). Some results can be obtained:

(1) The response is still sinusoidal signal produced by sinusoidal excitation and is determined by X and φ.

(2) The response frequency is the same with electromagnetic force.

(3) The amplitude X is directly proportional to

the steady displacement of electromagnetic force, i.e.,

$$\begin{cases} X = |H(\omega)| A; \\ |H(\omega)| = \dfrac{1}{\sqrt{(1-(\omega/\omega_1)^2)^2 + (2r\omega/\omega_1)^2}}, \end{cases} \tag{8}$$

where $|H(\omega)|$ is the steady displacement of system response and A is the amplification factor.

SIMULATION MODEL OF VIBRATION PROCESS IN PIND

To analyze the relations between arbitrary forms of electromagnetic forces and vibration responses, Simulink model of Eq.(1) is established. Fig.3 is Simulink model of vibration process which includes four parts: (1) electromagnetic force excitation F; (2) fiberglass spring stiffness part K_1x; (3) fiberglass spring damping part $C\dot{x}$; (4) anvil buffer part $K_2(x-X_S)$ (when analyzing shock process). The sum of parts (1), (2) and (3) is $M\ddot{x}$ of Eq.(1). $M\ddot{x}$ divided by M is acceleration a and speed v and displacement x can be obtained by integrating acceleration a.

In terms of Model 4501, simulation parameters can be obtained: M=2 kg, C=50, $K_1=4\times10^4$, $K_2=6.8302\times10^8$, X_S=6.5 mm, gravitational acceleration g=9.81 m/s^2. So $\omega_1=\sqrt{K_1/M}$ = 141.4214 rad/s and $r=C/(2M\omega_1)$=0.0884.

CORRESPONDING ELECTROMAGNETIC FORCE OF VIBRATION TEST CONDITIONS

Zhang *et al.*(2004a; 2004b) obtained the best vibration test conditions which are the given vibration acceleration and the corresponding best vibration frequency under the constant rated power. But they did not explain how to achieve the given acceleration and the best vibration frequency.

MIL-PRF-39016E specifies that excitations must be sinusoidal waveform in vibration test of PIND, so electromagnetic force F is sinusoidal waveform. Vibration frequency is acquired exactly easily in vibration test and is equal to excitation frequency. But it is difficult to obtain the exact acceleration amplitude which is determined by system parameters and excitation. To get the ideal vibration test conditions, it is necessary to study the corresponding electromagnetic force excitation.

Vibration acceleration can be obtained by differentiating Eq.(4) for sinusoidal electromagnetic excitation:

$$a(t) = \ddot{x}(t) = -X\omega^2\cos(\omega t - \varphi). \tag{9}$$

The amplitude of vibration acceleration is obtained by Eq.(7),

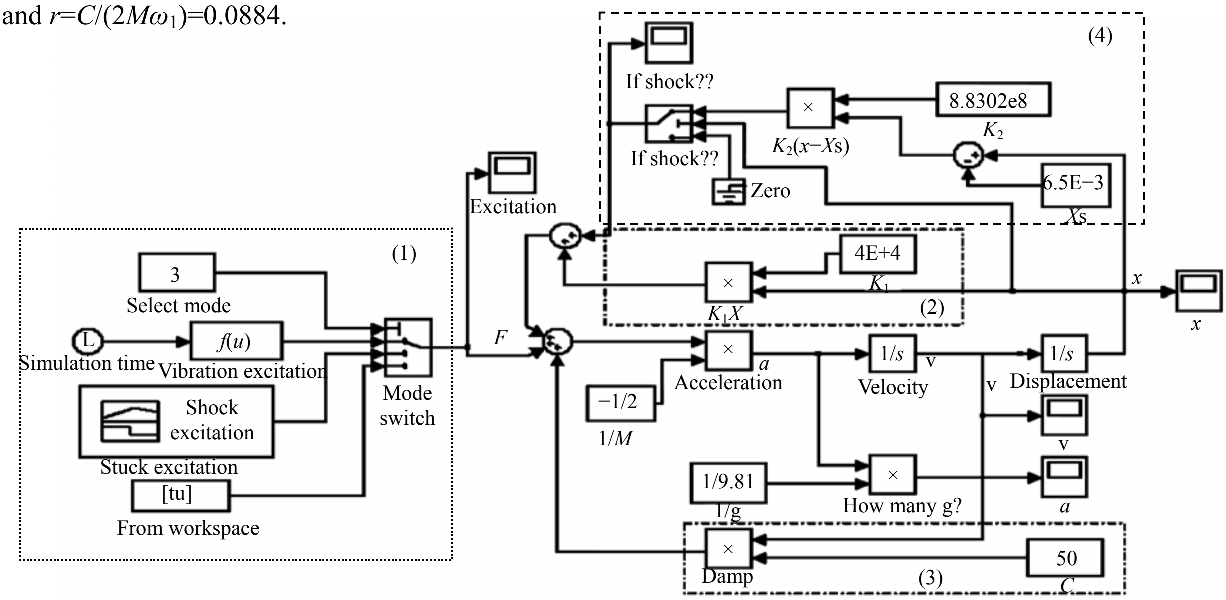

Fig.3 Simulink model of vibration process

$$a_{\rm m}=\frac{A\omega^2}{\sqrt{(1-(\omega/\omega_1)^2)^2+(2\xi\omega/\omega_1)^2}}. \quad (10)$$

After transforming

$$F_{\rm A}=\frac{K_1a_{\rm m}\sqrt{(1-(\omega/\omega_1)^2)^2+(2\xi\omega/\omega_1)^2}}{\omega^2}. \quad (11)$$

Eq.(11) is the amplitude of electromagnetic force under the given vibration acceleration and frequency.

SIMULATION

To testify the validity of Eq.(11), some simulations are given by using Simulink model of vibration process.

(1) Vibration test conditions are 20g of acceleration and 40 Hz of frequency. The amplitude of electromagnetic force is computed by Eq.(11):

$$F_{\rm A}=\frac{K_1a_{\rm m}\sqrt{(1-(\omega/\omega_1)^2)^2+(2\xi\omega/\omega_1)^2}}{\omega^2}=270.9808\ \text{N}.$$

Simulation time is 0.8 s and Fig.4 shows electromagnetic excitation and its system responses. Fig.4a is electromagnetic force under 40 Hz of frequency and 270.9808 N of amplitude. Fig.4b shows displacement response. Fig.4c shows speed of armature. Vibration acceleration is displayed in Fig.4d with its amplitude being 20.0012g. The relative error is

$$E=\frac{20.0012-20}{20}\%=0.006\%.$$

(2) Vibration test conditions are 10g of acceleration and 100 Hz of frequency. The amplitude of electromagnetic force is computed by Eq.(11):

$$F_{\rm A}=\frac{K_1a_{\rm m}\sqrt{(1-(\omega/\omega_1)^2)^2+(2\xi\omega/\omega_1)^2}}{\omega^2}=186.4239\ \text{N}.$$

Simulation time is 0.5 s and Fig.5 shows electromagnetic excitation and its system responses. Fig.5a is electromagnetic force under 100 Hz of frequency and 186.4239 N of amplitude. Fig.5b shows

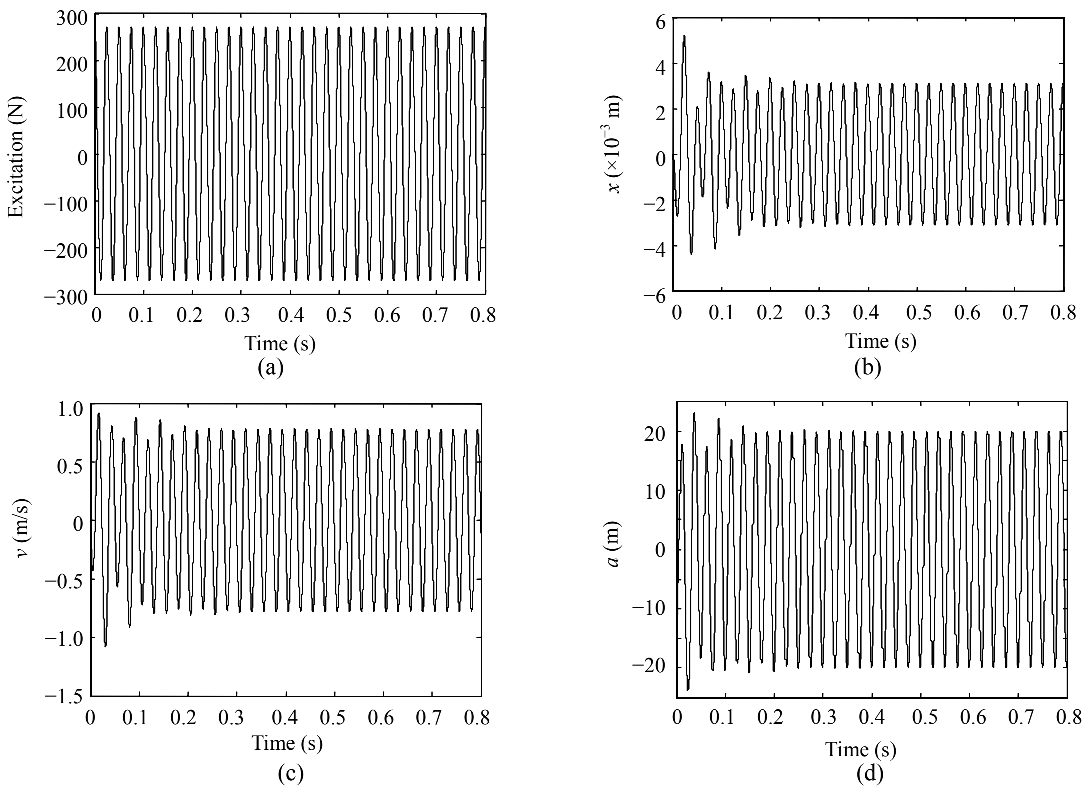

Fig.4 Electromagnetic force at 20g of vibration acceleration
(a) Electromagnetic force; (b) Displacement; (c) Speed; (d) Acceleration

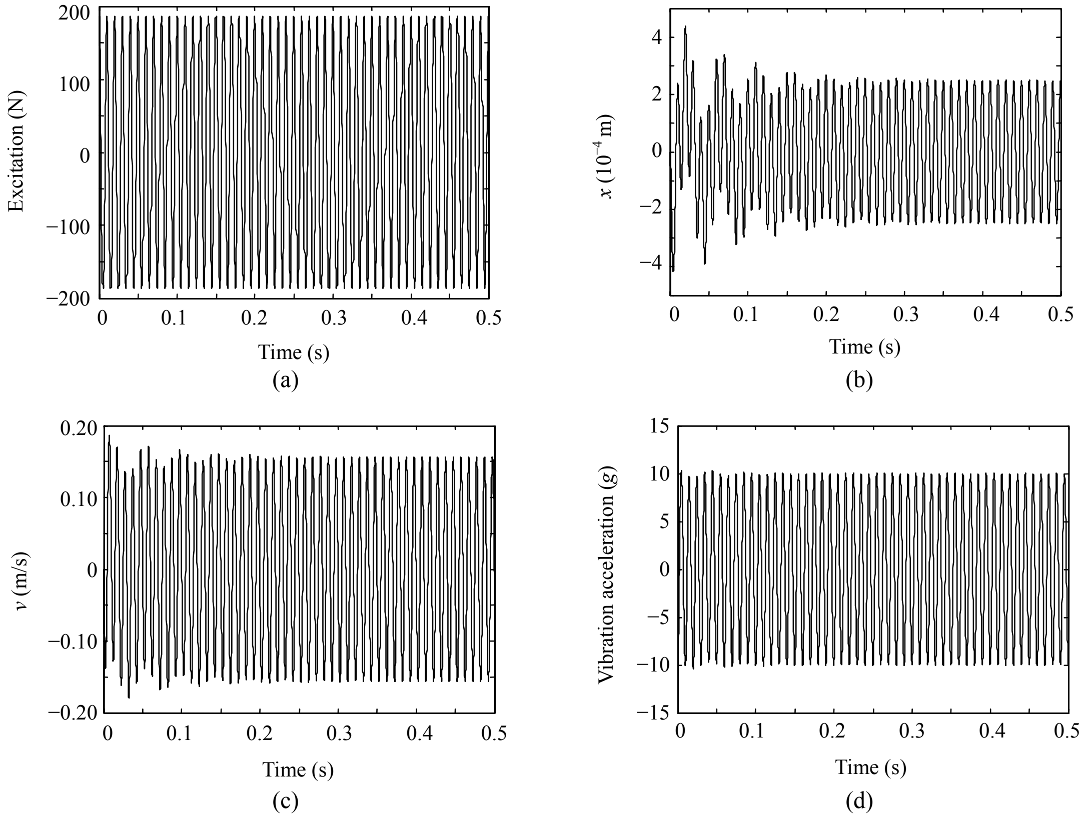

Fig.5 Electromagnetic force at 10g of vibration acceleration
(a) Electromagnetic force; (b) Displacement; (c) Speed; (d) Acceleration

displacement response. Fig.5c shows speed of armature. Vibration acceleration is displayed in Fig.5d with its amplitude being 10.0015g. The relative error is

$$E = \frac{10.0015 - 10}{10}\% = 0.016\%.$$

From the above two simulations, the exact vibration acceleration can be obtained by Eq.(11) with its relative error being less than 1‰.

CONCLUSION

Particle Impact Noise Detection (PIND) test is a reliability screening technique for hermetic components. But MIL-PRF-39016E did not specify how to obtain these conditions.

This paper establishes the dynamics model of vibration process of PIND based on first order mass-spring system. The corresponding Simulink model is established to simulate vibration under optional input excitations. The response equations are also derived in sinusoidal wave excitations and the required electromagnetic force waves are computed in order to obtain a given vibration and shock accelerations. Last, some simulation results are given.

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