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Method of analyzing vibration characteristics of armature system of hermetically sealed electromagnetic relay*

REN Wan-bin†1, CUI Li[2], LUO Fu-biao[2], ZHAI Guo-fu[1]

([1]School of Electrical Engineering and Automation, Harbin Institute of Technology, Harbin 150001, China)

([2]Guilin Aerospace Electronics Company, Guilin 541002, China)

†E-mail: renwanbin@yahoo.com.cn

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Abstract: Published research is minimal on vibration characteristics of hermetically sealed electromagnetic relay (EMR) exposed to mechanical environment. The vibration characteristics of armature system, link contact system with electromagnetic system will cause EMR malfunction. The nonlinear dynamics model of armature systems was studied by considering electromagnetic attraction force and opposite mechanical force in this paper. Angular displacements of armature under different sinusoidal vibration conditions are solved in order to obtain the failure mode result from armature system. Vibration tests showed the presented analyzing method is suitable for EMR. The conclusions are instructive for increasing vibration resistance of armature systems of EMR, and are significant for reliability design of switch apparatus.

Key words: Hermetically sealed electromagnetic relay (EMR), Armature system, Vibration characteristics, Analyzing method

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INTRODUCTION

Standard hermetically sealed electromagnetic relay (EMR) filled with nitrogen (N_2) by metal housing, has the advantage of high reliability and long life under severe working conditions, so it has the optimal protection style. Generally, hermetically sealed EMR could be divided into three main sections, which are electromagnetic system, contact system and housing case. When using relays in stationary devices exposed to external vibrations, these additional loads cause periodical oscillations to all non-movable and movable parts within the relay, which in turn result in change of contact pressure and armature confining force, hence affecting the reliability of the relay, that is to say, causing normally closed contact break off or normally opened contact clog, and seriously making armature position move inversely and mechanical component to shatter. The complexity and rigor of working environment demands that electromagnetic relay must have the ability to withstand vibrations and shocks from mechanical environment. General mechanical factors include vibration, shock, swing, centrifugal acceleration, explosion, earthquake and other mechanics of machinery.

Statistics showed that 70% of EMR failures result from contact system, whose vibration characteristics were investigated completely (Ройзен, 1979; Chambega, 1996; Zhai *et al.*, 2004; Ren and Zhai, 2004; Ren *et al.*, 2006). But some normally closed contacts are separated by armature bouncing during severe vibration conditions, so armature vibration characteristics analysis is also important for increasing the mechanical environment resistance ability of EMR.

Up to now, research mainly focuses on how to increase magnetic system sensitivity and reduce power consumption, so many new electromagnetic structures and optimization results were presented (Shoffa *et al.*, 2004; Zhai *et al.*, 2002; 2003), but relevant anti-vibration research is minimal. The movable armature vibration characteristics of relay

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and affecting factors should be investigated (Wang, 1998). The multi-segment linear body-spring- damping vibration model of typical balanced armature relay was established and the analytical solutions of the vibration response were presented (Xu *et al.*, 1995). The natural frequency of some clapper relay's armature was analyzed by using Rayleigh method (Xie and Wang, 2003). In this paper, armature motion characteristics of typical differential magnetic system in sinusoid vibration condition were researched to determine armature vibration failure mode, so that the method can provide theoretical foundation for the mechanical, electrical and magnetic integrated parameter optimization design of hermetically sealed EMR.

MATHEMATICAL MODEL OF VIBRATION CHARACTERISTICS OF ARMATURE SYSTEM

Description of modelling

(1) In general, there are three mechanical excitation directions for EMR being researched in work. But the direction concording with armature motion is the worst case according to vibration test and analysis. So this direction is taken as the most dangerous direction in this paper.

(2) Armature with rectangular, short and flat cross-section was taken as a rigid body.

(3) The fore and after two lateral sides movement of armature were restricted by a pedestal body. There is enough permanent magnet attractive force at the magnet pole shank, where the armature is supported. So the rotation around the staff is the armature's only degree of freedom. Considering the relay vibration resistance in pick-up status is better than in release status, the equipoise of armature in release status is defined as the start point of angle θ, and clockwise is the forward direction.

(4) θ_1 is defined as the critical angle of armature disengaging from right yoke; θ_2 is defined as the critical angle of armature contact with movable spring; θ_3 is defined as the critical angle of N.C. spring group breakout; θ_4 is the critical angle of N.O. spring group closing; θ_5 is the critical angle of armature contact with yoke; θ_6 is the termination angle of armature.

(5) The friction torques between armature and staff, push pole and movable spring are ignored. Moreover, the energy loss between components caused by collision is also ignored.

The restraints between armature and yoke, spring are unilateral, thus the complete armature motion could be divided into several regions with motion direction being different at each region. The whole dynamic system, which includes different elastic components and damping components, has varied natural frequency.

Electromagnetic attractive force and mechanical opposite force

Polarized magnetic system usually includes armature, yoke, permanent magnet, coil, iron core and so on. The mathematical model for determining hold force and pick-up force of armature using magnetic circuit theory is presented here. The equivalent magnetic circuit for polarized relay electromagnetic system (shown in Fig.1) is shown in Fig.2.

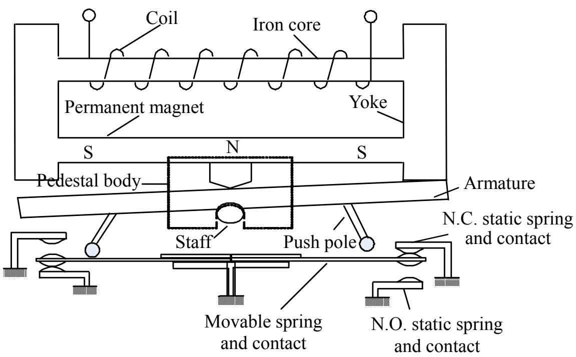

Fig.1 Some differential-type polarized electromagnetic relay structure

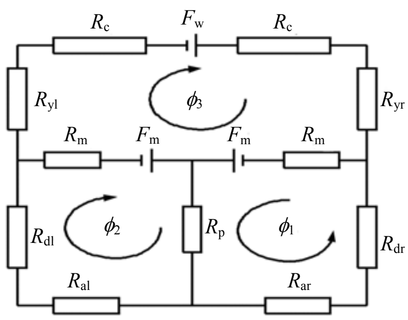

Fig.2 Equivalent magnetic circuit

The magnetic circuit equations set can be written as:

$$
\begin{aligned}
&(R_{al}+R_p+R_m+R_{yl}+R_{dl})\phi_1+R_p\phi_2+R_m\phi_3=F_m,\\
&R_p\phi_1+(R_{ar}+R_{dr}+R_{yr}+R_m+R_p)\phi_2-R_m\phi_3=F_m, \quad (1)\\
&R_m\phi_1-R_m\phi_2+(2R_m+R_{yl}+R_{yr}+R_c+R_f)=F_w,
\end{aligned}
$$

where ϕ_1, ϕ_2, ϕ_3 are the magnetic flux values of each circuit; R_{al}, R_{ar} are the magnetic reluctances of left and right armature segment; R_{dl}, R_{dr} are the magnetic reluctances of left and right air gap; R_{yl}, R_{yr} are the magnetic reluctances of left and right yoke segment; R_c is the magnetic reluctance of iron core; R_p is the magnetic reluctance of support position; F_w is the magnetic potential of coil.

From Eq.(1), obtain the values of ϕ_1, ϕ_2 and ϕ_3, then the magnetic flux corresponding to every air gap is solved. Substitute them into Maxwell equation for electromagnetic torque of air gap, finally the torque for armature disengaging from yoke support can be expressed as:

$$M_0=F_{mR}l_R-F_{mL}l_L=M_R-M_L, \tag{2}$$

where F_{mR}, F_{mL} is permanent magnet attractive force of right and left pole-face respectively; l_R is the distance between center of right pole-face and armature rotation shaft; l_L the distance between center of left pole-face and armature rotation shaft.

In addition, the relation between mechanical opposite force and deflection of spring can be calculated according to material mechanics (Ройзен, 1979).

Motion equations

If exciting displacement is described by $\Lambda(t)=A\sin\Omega t$, A is the exciting amplitude, Ω is the exciting angular frequency, then the armature motion can be classified into six cases.

Case 1: $0<\theta\le\theta_1$, the armature does not disengage from yoke. Taking it as individual body, then the motion equation is

$$I\ddot{\theta}+C_y l_R^2\dot{\theta}-M_0+k_y l_R^2(\theta_1-\theta)-mAL_a\Omega^2\sin\Omega t=0, \tag{3}$$

where k_y, C_y are elastic coefficient and damping coefficient of yoke respectively; I is armature moment of inertia; m is the armature mass; L_a is the armature length.

Case 2: $\theta_1<\theta\le\theta_2$, armature disengages from yoke, but does not yet contact N.C. spring group, then the motion equation is

$$I\ddot{\theta}-M_0-mAL_a\Omega^2\sin\Omega t=0. \tag{4}$$

Case 3: $\theta_2<\theta\le\theta_3$, armature contacts movable spring, which is still restrained by N.C. static spring, then motion equation is

$$\begin{aligned}&I\ddot{\theta}+N(C_{jd}+C_d)l_s^2\dot{\theta}-M_0\\&+N(k_{jd}+k_d)l_s^2(\theta-\theta_2)-mAL_a\Omega^2\sin\Omega t=0,\end{aligned} \tag{5}$$

where N are the numbers of switched contact; k_d, C_d are elastic coefficient and damping coefficient of movable spring respectively; k_{jd} and C_{jd} are elastic coefficient and damping coefficient of N.C. static spring respectively; l_s is the distance between movable contact and armature rotation shaft.

Case 4: $\theta_3<\theta\le\theta_4$, N.C. spring group breaks out, but does not yet contact N.O. static spring, then motion equation is

$$\begin{aligned}&I\ddot{\theta}+NC_d l_s^2\dot{\theta}-M_0+Nk_{jd}l_s^2(\theta_3-\theta_2)\\&+Nk_d l_s^2(\theta-\theta_3)-mAL_a\Omega^2\sin\Omega t=0.\end{aligned} \tag{6}$$

Case 5: $\theta_4<\theta\le\theta_5$, N.O. spring group closes, but does not yet contact left yoke, then motion equation is

$$\begin{aligned}&I\ddot{\theta}+N(C_d+C_{jh})l_s^2\dot{\theta}-M_0+Nk_{jd}l_s^2(\theta_3-\theta_2)\\&+Nk_d l_s^2(\theta-\theta_3)+Nk_{jh}l_s^2(\theta-\theta_4)-mAL_a\Omega^2\sin\Omega t=0,\end{aligned} \tag{7}$$

where k_{jh} and C_{jh} are elastic coefficient and damping coefficient of N.O. static spring respectively.

Case 6: $\theta_5<\theta\le\theta_6$, armature contacts left yoke

$$\begin{aligned}&I\ddot{\theta}+N(C_d+C_{jh})l_s^2\dot{\theta}+NC_y l_L^2\dot{\theta}-M_0\\&+Nk_{jd}l_s^2(\theta_3-\theta_2)+Nk_d l_s^2(\theta-\theta_3)+Nk_{jh}l_s^2(\theta-\theta_4)\\&-k_y l_L^2(\theta-\theta_5)-mAL_a\Omega^2\sin\Omega t=0.\end{aligned} \tag{8}$$

The armature's angular displacement response in the above six cases can be solved by using numerical computing method, when displacement crosses over the adjacent cases, the last current state of forward case can be chosen as initial conditions of next state. Therefore, the armature's transient response (including displacement and velocity) is determined accurately when the exciting frequency and acceleration are given. With the same argument, this analysis is suitable for other armature structure.

EXAMPLES

Taking differential-type polarized EMR (shown in Fig.1) as example, define horizontal position of armature as θ=0. By calculation, the armature position in release condition is θ=–0.244 rad, and in pick-up condition is θ=0.244 rad for the balanced structure. The permanent magnet attractive force characteristic curve and mechanical opposite force characteristic curve are shown in Fig.3, the attractive torque in release condition is M=6×10^{-4} N·m; while the yoke rigidity k_y=2.4×10^5 N/m is obtained by using testing and analyzing system (Liang *et al.*, 2004), so the critical angle expression of armature disengaging from yoke is

$$\theta_1 = -4M/(kL_a^2) = -0.02431 \text{ rad}. \tag{9}$$

The diagrammatic shadow region in Fig.3 shows that the contact deformation between armature and yoke, and the critical angle θ_2~θ_5 is indicated clearly; detailed parameters are listed in Table 1.

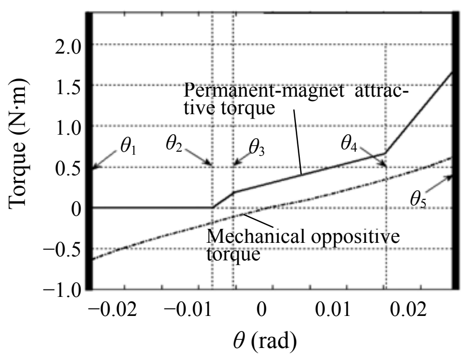

Fig.3 Permanent attractive torque and mechanical opposite torque

Table 1 The parameters of some relay

Parameters	Value
Armature mass m (kg)	1.1×10^{-4}
Armature moment of inertia I (m^4)	5×10^{-8}
Rigidity of movable spring k_d (N/m)	306.3
Rigidity of N.C. contact static spring k_{jd} (N/m)	2018.7
Rigidity of N.O. contact static spring k_{jh} (N/m)	789.9
Initial pressure between N.C. contacts F_0 (N)	0.033
Critical angle θ_2 (rad)	–0.00815
Critical angle θ_3 (rad)	–0.00517
Critical angle θ_4 (rad)	0.01512
Critical angle θ_5 (rad)	0.02431
Damping coefficient of yoke C_y (N·s/m)	0.005

The phase-plane diagram of armature (exciting frequency f=500 Hz, acceleration A_{cc}=10g, initial velocity $\dot{\theta}=0$) is shown in Fig.4 showing that the motion process is stable, and that the maximum of angular θ_{max}=–0.02445<θ_1, so that armature keeping contact with yoke can be determined, that is, the vibration characteristics meet the requirement of product specification.

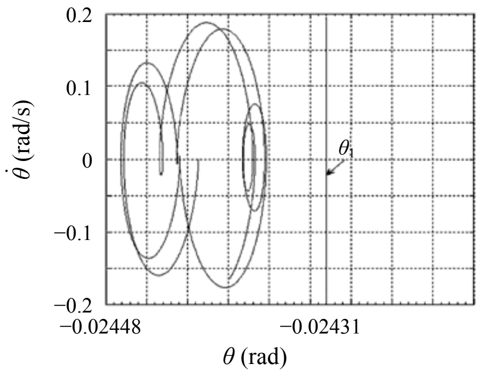

Fig.4 10g/500 Hz vibration condition

The phase-plane diagram of armature (exciting frequency f=1320 Hz, acceleration A_{cc}=20g, initial velocity $\dot{\theta}=0$) is shown in Fig.5 indicating the armature motion angle crosses the critical angle θ_1 abruptly and that amplitude variation increases gradually; that the maximum reached –0.00679 rad after the static balanced position is passed many times, so that the armature disengages from the yoke and collides with movable spring.

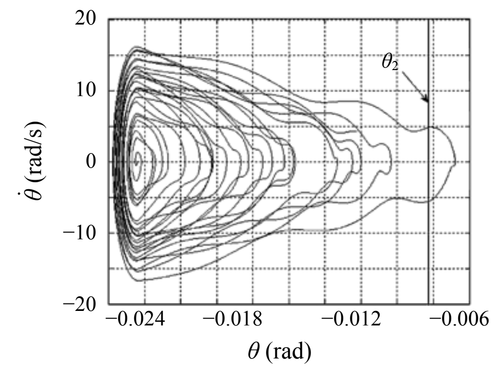

Fig.5 20g/1320 Hz vibration condition

When armature damping changes to 0.05 N·s/m, the maximum of response reduced sharply (shown in Fig.6). The collision between armature and yoke occurred periodically, and armature will not contact with movable spring. Being a dangerous case, it should be avoided. Fig.7 shows the phase-plane diagram of 20g/2000 Hz.

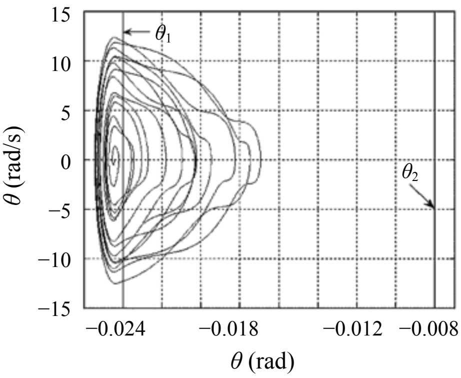

Fig.6 20g/1320 Hz vibration condition and C_y=0.05

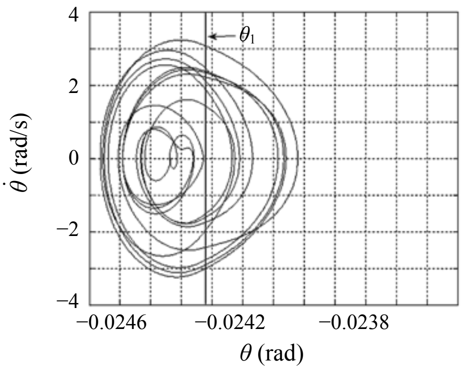

Fig.7 20g/2000 Hz vibration condition

High-frequency vibration test was conducted by method 204D in Mil-std-202; the frequency ranges was 10~2000 Hz and acceleration was 20g. Vibration-test direction was parallel to the armature motion, which was the dangerous case. When armature was in release state, normally closed contact group breaks off, corresponding exciting frequency was 1200~1400 Hz. That concords with the above calculation results.

CONCLUSION

Dynamic model of armature system was built in order to analyze vibration characteristics of hermetically sealed EMR. The typical nonlinear feature and failure mode were verified by phase-plane diagram and varied oscillation period. By comparison of calculation results and test results, the validity of this analysis method is justified.

References

Chambega, J., 1996. A Qualitative Analysis on the Effect of External Vibrations on the Performance of Relays. Proceedings of the 4th IEEE AFRICON Conference, Part 2, p.1035-1039. [doi:10.1109/AFRCON.1996.563040]

Liang, H.M., Zhang, Y.C., Zhai, G.F., 2004. Testing apparatus of static attractive force and spring force characteristics of micro electromagnetic relay. *Electromechanical Components*, **4**:3-9 (in Chinese).

Ройзен, В.з., 1979. Miniature Sealed Electromagnetic Relay. Posts & Telecom Press, Beijing, p.70-79 (in Chinese).

Ren, W.B., Zhai, G.F., 2004. Research on Contact Displacement Response of Switching Electrical Apparatus in the Condition of Vibration. Proceedings of the First International Conference on Reliability of Electrical Products and Electrical Contacts, p.72-75.

Ren, W.B., Zhai, G.F., Cui, L., 2006. Contact vibration characteristic of electromagnetic relay. *IEICE Transactions on Electronics*, **E89-C**(8):1177-1181. [doi:10.1093/ietele/e89-c.8.1177]

Shoffa, V.N., Chicheryukin, V.N., Ivakin, B.F., 2004. A mathematical model and a procedure for designing the complex magnetic system of a polarized relay. *Electrical Technology Russia*, **75**:32-40.

Wang, L.Z., 1998. The effective method for increasing vibration resistance of high power relay. *Relay in Aerospace*, **1**:18-20 (in Chinese).

Xie, W.J., Wang, S.J., 2003. Analysis of natural frequency of clapper style electromagnetic system. *Electromechanical Components*, **1**:9-14 (in Chinese).

Xu, H., Gao, Y.N., Yang, M., 1995. Research on the vibration resistance of the armature system of a small-sized electromagnetic relay. *Journal of Xi'an Jiaotong University*, **4**:90-97 (in Chinese).

Zhai, G.F., Liang, H.M., Wang, H., Wang, L.Z., 2002. Research and analysis on the shape of the permanent magnet torque characteristic curve of polarized magnetic system. *Proceedings of the CSEE*, **11**:110-114 (in Chinese).

Zhai, G.F., Liang, H.M., Wang, H., 2003. Research on the parameters optimum design of polarized magnetic system based on orthogonal design. *Proceedings of the CSEE*, **10**:158-163 (in Chinese).

Zhai, G.F., Ren, W.B., Xu, F., 2004. Analysis method for vibration acceleration of electromagnetic relay. *J. of Vibration Engineering*, **1**:66-71 (in Chinese).