Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2008

Analysis of the No-Load Characteristic of the Moving Coil Linear Compressor

Yingbai Xie
North China Electric Power University

Xiuzhi Huang
North China Electric Power University

Liangming Gui
North China Electric Power University

Zhouxuan Xu
North China Electric Power University

Follow this and additional works at: https://docs.lib.purdue.edu/icec

Analysis of the No-Load Characteristic of the Moving Coil Linear Compressor

Yingbai XIE [1], Xiuzhi HUANG[2] *, Liangming GUI[3], Zhouxuan XU[4]

[1,2,3,4]Department of Power Engineering, North China Electric Power University,
Baoding, Hebei China
Tel: 86-312-7522882 Fax: 86-312-7522440

[1]xieyb@ncepu.edu.cn

[2]huang_xiuzhi@hotmail.com

[3]guiliangmin2006@163.com

[4]steffine@sina.com

ABSTRACT

The linear compressor is driven by a linear motor. The efficiency of the whole unit is higher than that of the traditional compressor. A moving coil linear compressor is taken for an example to find its no-load characteristic. The open loop and closed loop transfer functions of the system in no-load condition are obtained derived from the equation of system motion. The Matlab software is applied to analyze the stability, time domain and frequency domain of the system. Result indicates that the moving coil linear compressor is almost stable at no-load stage, and the characteristic of staring is relative fast, but the overshoot is relative high, and the damping ratio should be increase to lower the overshoot.

1. INTRODUCTION

A linear compressor is a piston-type compressor in which the piston is driven by a linear motor, rather than by a rotary motor coupled to a conversion mechanism as in a conventional reciprocating compressor (Unger, R.Z., 1998). Linear motors are simple devices in which axial forces are generated by currents in a magnetic field. Because all the driving forces in a linear compressor act along the line of motion, there is no sideways thrust on the piston, substantially reducing bearing loads and allowing the use of gas bearings or low viscosity oil. The linear compressor is now proven in a variety of hardware. Its efficiency, modulation, oil-free option, and features that should make it compete successfully with conventional compressor over a wide range of applications (Unger, R.Z., van der Walt, N.R., 1996).

The changeability if the piston stroke is one of the characters of the linear compressor. It can make the compressor easy start at differential pressure and adjust to the change of the load. The change of the piston stroke can lead to the change of the pressure ratio, clearance and dead point of the compressor. This paper takes the moving coil linear compressor for example, by founding motion equations to analysis stability, time and frequency of control system.

2. DYNAMIC MODEL

A schematic moving coil linear compressor is showed in figure 1. It uses permanent-magnet to excite. When alternate current flows though the coil, at the function of magnetic field, the coil generates alternate axial force

which makes the piston do reciprocating motion to compress the gas. This compressor has many characteristics such as simple construction, compact, high efficiency, lower starting current flow and so on.
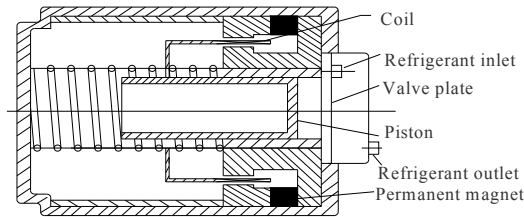

Figure 1 The construction of moving coil linear compressor

The motion equation of the system showed in figure 1 is

$$M \frac{d^2X}{dt^2} + C \frac{dX}{dt} + KX = BIL - \Delta p \tag{1}$$

Where M is the mass of the moving coil and piston, C is damping constant, K is spring effect, B is magnetic induction intensity, L is the length of coil.

In equation (1) ΔP is the change of the gas which belongs to disturb variable. Because we only do research on the no-load characteristic of the system, take no account of it. This system is a single input and single output (SISO) LTI system, in which the current flow I is input variable and the displacement X is output variable. Using Laplace transform, the open loop transfer functions of the mechanical system can be obtain

$$G(s) = \frac{X(s)}{I(s)} = \frac{BL}{Ms^2 + Cs + K} \tag{2}$$

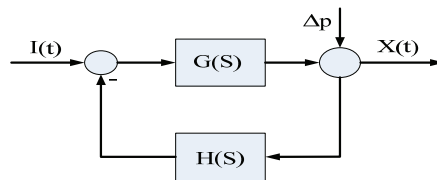Figure 2 is closed loop transfer functions block diagram of the system.

Figure 2 The block diagram of vibrating system

According to the construction characters and optimizing computer results of linear compressor, choose the parameters M=0.3 kg, C=0.7 N.s/m, K=7.365 kN/m, B=0.25 T, L=42m.

3. ANALYSE OF SYSTEM PERFORMANCE

All the characters of a system lies on the closed loop transfer functions, stability lies on the pole, and dynamic performance lies on the pole and zero. According to figure 2, the closed loop transfer function can be obtained from the open loop transfer function

$$\Phi(s) = \frac{G(s)H(s)}{1+G(s)H(s)} \tag{3}$$

Where H(s)=1, from the normalized form of closed loop transfer function

$$s^2 + 2\zeta\omega_n s + \omega_n^2 = 0 \tag{4}$$

We can get the closed loop characteristic root s and important characteristic parameter such as damping ratio ξ=0.00745, undamped oscillation frequency ω_n=157rad/s and so on.

3.1 Time analyse
3.1.1 Stability analyse
Pole determines the inherence moving attribute of the system. Its position determines the stability and rapidity of moving modality. When the pole has negative real part or is a negative real number, the corresponding modality must be convergent. Through computer this system has a pair of conjugate complex (pole) of which the real part is negative $s_{1,2}$= -1.17±157i. Since the two root of the system both has negative real part, we can estimate this system is steady.

3.1.2 Time domain response
Figure 3 is step response of the system. The pole y=0.00241, delay time t_d=0.00722s, rise time t_r=0.0201s, pole time t_p=0.0204s, adjusting time t_s=3s, overshoot σ %=95.9%. This system belongs to second-order oscillation segment. Dynamic course analyse:
because 0＜ξ＜1

$$s_{1,2} = -\zeta\omega_n \pm j\omega_n \sqrt{1-\zeta^2} \tag{5}$$

Characteristic equation has a pair of conjugate complex roots of which the real part is negative. Its roots correspond a pair of conjugate complex pole at the left of s plane. The characteristics of this periodic damping second-order system are
(1) The overshoot is function of damping ratio, and is independent of oscillating frequency. The smaller damping ratio is, the bigger oscillating frequency is;
(2) The smaller damping ratio is, the smaller rise time is;
(3) The response speed of system is relative to the angular frequency ω_n of undamped free oscillation. The bigger ω_n is, the higher response speed is. According to the computing results, undersize of damping ratio can lead overshoot over. So it must be adjusted.
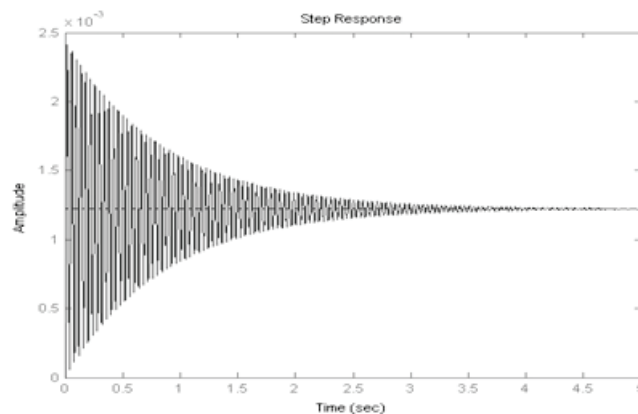

Figure 3 Step response of the system

3.2 Root locus analyse

Figure 4 is the root locus diagram. The root locus starts from the two symmetrical poles at unreal axis to infinite distance with growing of closed loop gain k_g. As both the closed loop poles are in the left part of s plane, the system is stability. But closed loop pole are both complex pole. Its unit-step response is underdamped oscillation response, and the bigger k_g. (closed loop gain) is, the bigger overshoot is.
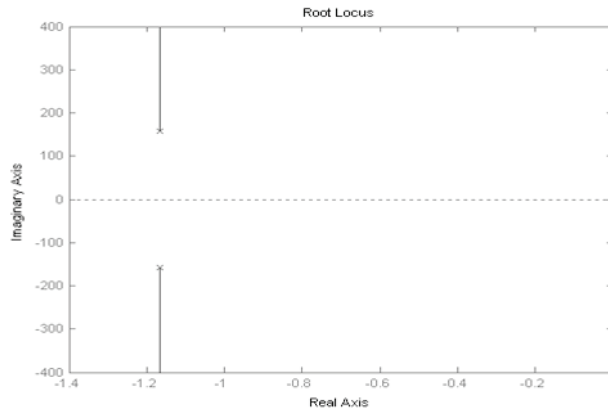

Figure 4 Root locus diagram of the system

3.3 Frequency analyse

Frequency characteristic is the frequency related to the input and output complex sign ratio at steady state when the linear system or segment effected by sine function. It attributes the dynamic law of system.

Figure 5 is Bode diagram of system (the upper figure is magnitude, the nether figure is phase). Its pole $M_r=0.0821$, harmonic frequency $\omega_r=156.7$rad/s. The expression of magnitude is

$$L(\omega) = 20\lg \frac{1}{\sqrt{(1-\frac{\omega^2}{\omega_n^2})^2 + 4\xi \frac{\omega^2}{\omega_n^2}}}$$

(6)

when $\omega \ll \omega_n$, $L(\omega) \approx -58$dB;
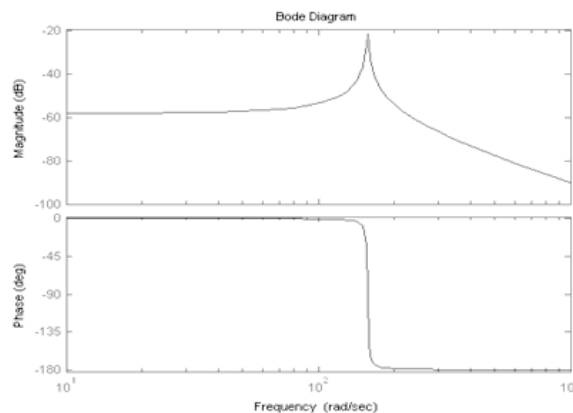when $\omega \gg \omega_n$, $L(\omega) \approx -40\lg\omega/\omega_n$.

Figure 5 Bode diagram of the system

We can see from Bode diagram, the frequency of this oscillation segment is ω_r. The magnitude characteristic reaches the max at harmonic frequency, and the pole depends on damping ratio. If the harmonic frequency of system is over, it can cause the overshoot of dynamic response over. It can influence stability of system.

The expression of phase is

$$\varphi(\omega) = -\arctan\frac{2\xi\dfrac{\omega}{\omega_n}}{1-\dfrac{\omega^2}{\omega_n^2}} \tag{7}$$

when $\omega=0$, $\varphi(0)=0°$;
when $\omega=\omega_n$, $\varphi(\omega_n)=-90°$;
when $\omega\to\infty$, $\varphi(\omega_n)=-180°$.

Damping ratio can influence the change rate of $\varphi(\omega)$ at the neighborhood of $\omega=\omega_n$. The smaller the damping ratio is, the bigger the change rate is.

Figure 6 is Nyquist diagram. Because that the number of pole of transfer function G(s) at s plane is zero, and Nyquist diagram does not enclose point (-1,j0). According to Nyquist criterion, the number of pole of closed loop system at the right of s plane is zero. So the closed loop system is steady.
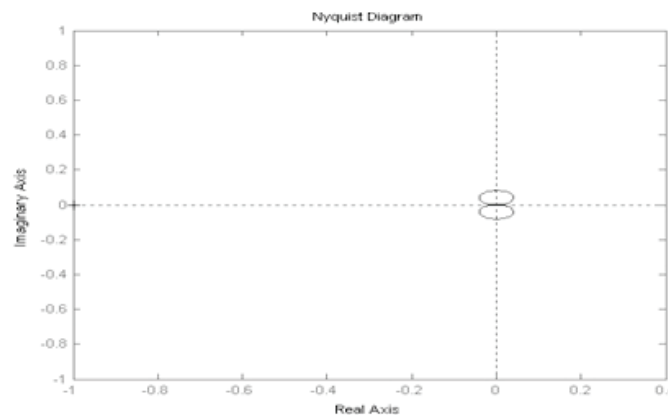

Figure 6 Nyquist diagram of the system

4. CONCLUSIONS

According to the chrematistics of moving coil linear compressor, found mathematic model of system. The Matlab software is applied to analyze the stability, time domain and frequency domain of the system.
(1) According to stability analyse, Nyquist diagram, we can conclude that the moving coil linear compressor is almost stable at no-load stage;
(2) According to time-domain analysis, root locus diagram and Bode diagram, we can get that the overshoot is relative high, and the damping ratio should be increase to lower the overshoot.

REFERENCES

Huang, B.J, Chen, Y.C., 2002, System Dynamics and Control of a Linear Compressor for Stroke and Frequency Adjustment, *Journal of Dynamic Systems, Measurement, and Control*, vol. 5, no. 124: P. 176-181.

Tae, W.C., Jung, R., 2004, Analysis and control for Compressor System Driven by PWM Inverter, *The 30th Annual Conf. of the IEEE Industrial Electronics Society*, p. 263-267.

Unger, R.Z., 1998, Linear Compressors for Clean and Specialty Gases, *Proc. of Int. Compressor Engineering Conf. , Purdue University, West Lafayette*, p. 334-339

Unger, R.Z., van der Walt, N.R., 1996, Linear Compressor for Non-CFC Refrigeration, *Int. Appliance Technical Conf.,Purdue University, West Lafayette,* P. 44-60

Xie, J.F., 2005, Theory and Experimental Research of the Moving-magnetic Linear Compressor, *Dr. Thesis of Zhejiang University*, P. 7-9.