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Wen Wang

*Shanghai Jiao Tong University*

Xiaoliang Tai

*Shanghai Jiao Tong University*

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## Characteristic of a Miniature Linear Compressor

Wen WANG\*, Xiaoliang TAI

Shanghai Jiao Tong University,  
Shanghai, P.R. China

Tel.86-21-34206096, Fax 86-21-34206814 wenwang@sjtu.edu.cn Contact Information

### ABSTRACT

This paper shows a prototype of miniature compressor driven by moving magnetic linear actuator. Based on the prototype, the dynamics of the piston and the capacity of the compressor are analyzed, which include electromagnetic field and mechanical vibration behaviors. The theoretical results are compared with some experimental results, some problems are discussed for improving the miniature machine

### 1. INTRODUCTION

With the development of industry and economy, more and more demands of micro and meso scale cooling system occur in outdoors medical treatment, electronic cooling and engineering emergency operation, as well as in the field in aviation, aerospace and military, and so on. Micro or meso scale compressor is the core component for miniature refrigeration system with vapor compression refrigeration cycle, which is a kind of minimized machine that integrates various components, power supplies, controllers and sensors in a small volume.

The miniature compressor should employ compact structure, light weight, high efficiency, as well as low noise and vibration. Linear motor converts electrical energy into mechanical energy directly without intermediate conversion components as other compressors driven by rotary motor, so it is easier to be further reduced than a rotary motor. Therefore, the linear motor-driven piston compressor has unique advantages in efficiency and structure, many works had been done on the conventional size linear compressors including some products and prototype from some companies[Hyun Kim, etc.; N. Chen, etc.; Norihide Fujiyama, etc.]. This paper present the dynamics analysis about a prototype of miniature linear compressor.

The linear motor for minature linear compressor should satisfy some demands, such as: (1) enough actuating force for piston with compact structure; (2) the light the mover assembling piston, the better. The less weight of mover is helpful to reduce the inertia of the mover and the involved vibration, it benefits to reduce the total weight of the compressor as well; (3) all components are feasible for the manufacture; (4) it can operate reliably and efficiently.

In general, there are three kinds of linear acturators, the moving coil, the moving iron, and the moving magnet. The moving magnet linear motor is often adopted for linear compressor due to its compact structure and high operation efficiency. In this compressor, the NdFeB permanent magnet is moving reciprocately. In a limited space for miniature linear compressor, the moving magnet motor is convenient to be placed more electrical coils than the moving coil and moving iron motors, it is possible to provide enough actuating force with more compact volume.

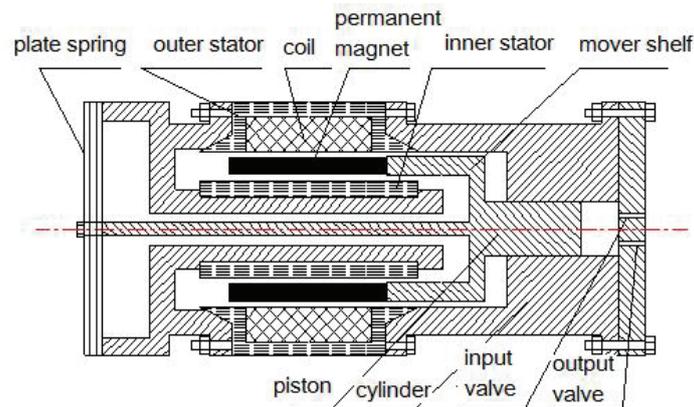


Fig.1 Schematic diagram of a miniature compressor

Fig.1 show the diagram of a miniature linear compressor driven by a moving magnet actuator, it is combined with the compressor body, the moving magnet linear motor, and the plate spring. The shelf of inner stators is made of electrical pure steel. The piston is made of alloy of aluminum, assembled with the magnet and connected with the flexible plate spring. Once the coil is actuated by alternative current, the NdFeB permanent magnet is pulled or pushed by a alternative force, and bring the mover with the motor and the piston to oscillate.

In order to optimize the magnet flux and the actuate force, the dimension of the irons, the coil, and the magnet, and the gaps were analyzed with finite element.

## 2. ELECTRO-MAGNET FIELD ANALYSIS IN THE LINEAR MOTOR

The magnet flux in the gaps is affected together by the permanent magnet and the alternative magnet from the coil. The magnet fields are varied when the permanent magnets reciprocates.

### 2.1 Model for Electro Magnet Field

Since the linear motor is symmetrical, the magnet flux is axis symmetrical also. The govern equation for magnetic potential  $A$  is expressed as,

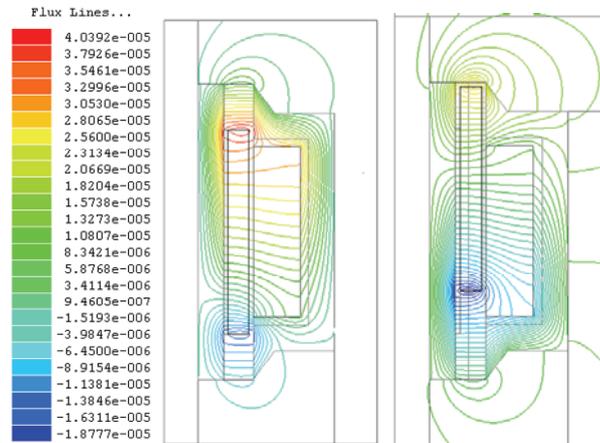
$$\left. \begin{aligned} \frac{\partial}{\partial z} \left[ \nu \frac{\partial (A_\theta)}{\partial z} \right] + \frac{\partial}{\partial r} \left[ \frac{\nu}{r} \frac{\partial (rA_\theta)}{\partial r} \right] &= -J_\theta \\ \Gamma_1 : A_\theta &= A_{\theta 0} \\ \Gamma_2 : \frac{\nu}{r} \frac{\partial (rA_\theta)}{\partial n} &= -H_t \end{aligned} \right\} \quad (1)$$

Where,  $\Gamma_1$  and  $\Gamma_2$  are two kinds of boundary. The reluctivity  $\nu = 1/\mu$ ,  $\mu$  is permeability, the permanent magnet is N45H, its relative permeability  $\mu_0 = 1.04$ .  $A_\theta$  is the magnet potential on  $\theta$  direction,  $A_{\theta 0}$  is the value at the environment far away,  $A_{\theta 0} = 0$ .  $H_t$  is the magnet field intensity at the circle direction.

### 2.2 Electromagnetic Field of Linear Motor

Figure 2(a), (b) show the distribution of magnetic flux when the magnet is at the balanced position and one end position respectively. In operation, the magnet flux can be thought as even. Once the mover reaches one end, the excitation coil and the magnet lead partial magnetic flux a little distortion, then, the actuating force is not just along the axis. In principle, the overall force on the mover should be on the axis because of the the structure of symmtetic cylinder. On the balance position, the axis force reach the maximum, once the mover moves to one end, the force is

reduced gradually. On the end, the current phase converts, the direction of axis force on the mover turn over, push the mover together with the spring force and the chamber gas, the mover accelerates and moves to another end.



(a) The mover is at the middle position (b) the mover is at the end position  
Fig.2 The magnetic line of force distributions for different positions of permanent magnet

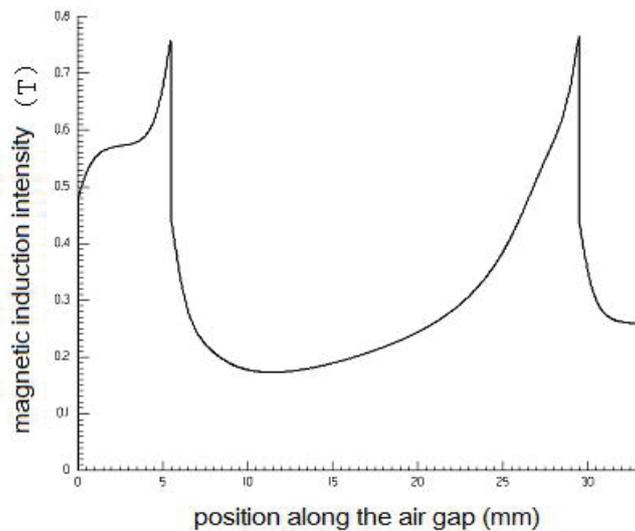


Fig.3 The magnetic induction intensity distribution along the center line of air gap

The linear motor is very important, its structure and related parameters should be optimized according to the requirements from dynamics, weight and volume. Figure 3 shows the profile of the magnet induction intensity along the center line of air gap, there are two peaks on the ends of the magnet, which is the source of the drive force. The maximum force from the linear motor is calculated as about 52N.

### 3. OPERATION CHARACTERISTIC

#### 3.1 Dynamic Equations of The Linear Compressor

The oscillation of the mover in the linear compressor can be simplified as a resonance system with a resistance. The mover is suffered by electromagnet force, elastic force, resistance, friction, and the pressure from gas. Suppose the position of the piston is  $x(s)$  when the piston is at static balance. The input voltage is consumed at the coil resistance and the induction voltage, The dynamic equations are,

$$\begin{cases} m \frac{d^2x(t)}{dt^2} + C \frac{dx(t)}{dt} + k[x(t) - x(s)] = F_e(t) + F_m(t) + F_g(t) \\ U(t) = iR + L_0 \frac{di}{dt} + K_0 \frac{dx}{dt} \end{cases} \quad (4)$$

Where,  $m$  is the mover mass,  $k$  is the rigidity of the spring,  $C$  is the mechanical resistance,  $R$  is the coil electrical resistance,  $F_e(t)$  is the electromagnet force,  $F_m(t)$  is the friction,  $F_g(t)$  is from the gas pressure in the chamber that is a non linear force, and related to the position of piston and the process index of gas. Friction  $F_m(t)$  is nonlinear as well that is related to the materials of the piston and the chamber, their roughness, and the pressure where they touch each other. Since the friction is difficult to describe accurately, its effect is added into the mechanical resistance.

### 3.2 Frequency Characteristics

The linear compressor is a typical vibration system, which employs a nature frequency, the performance of linear compressor will vary with the actuating frequency of the motor

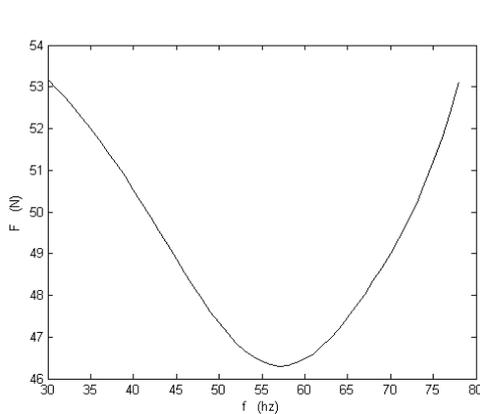


Fig.4 Variation of the magnetic force with the input current Frequencies

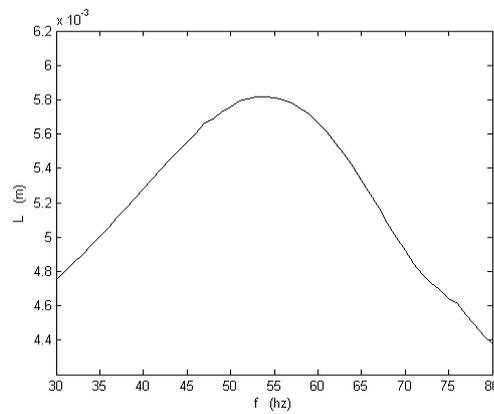


Fig.5 Variation of the piston stroking with the input current frequencies

Fig. 4 and 5 show the waves about the driven force and the amplitude of the piston, according to the results, when the actuating frequency is about 56Hz, the required driven force is the lowest, and the piston can reach the biggest amplitude, about 5.82mm.

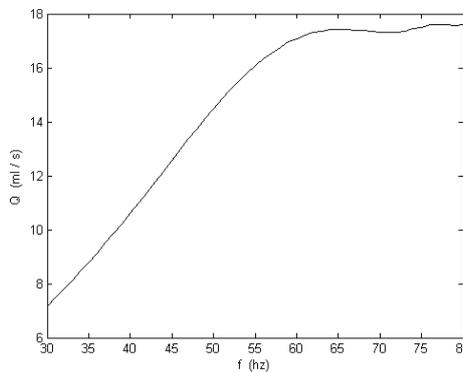


Fig.6 Variation of the air displacement with the input current frequencies

Fig 6 shows the variation of gas displacement with the actuating frequency. When the actuating frequency is less than 56Hz, the gas displacement capacity increases with the actuating frequency, when the frequency is higher than 56Hz, the gas displacement is changed slightly with the frequency. That is, increasing the actuating frequency is not always a effective method for increasing gas displacement.

There is a economical frequency zone for linear compressor operation. When the operation frequency is close to the nature frequency of the mover, the driven force is near isochronous with the piston velocity. Therefore, the operation efficiency is the best in this zone. In this paper, the optimum frequency is about 56Hz with this prototype, the optimum frequency is related to the rigidity and the mass of the mover.

### 3.3 Operation Characteristics of Linear Compressor

Figure 7 shows the simulated track of the piston, the oscillation quickly reaches a steady state, the steady amplitude is about 5.8mm, and the related gas displacement is about 17ml/s. In the startup, there is a current peak, 2.84A, the push force from the motor is 16.85N/A. In steady operation, the velocity of the mover is 0.92m/s.

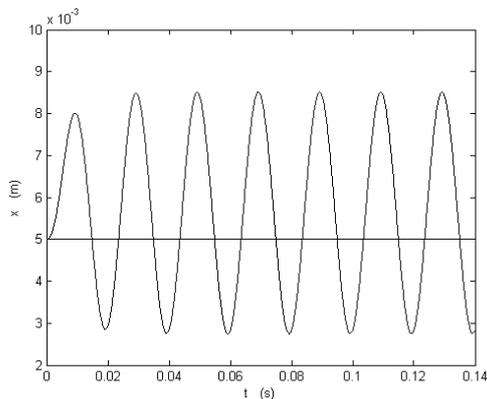


Fig.7 Variation of the piston displacement with the time

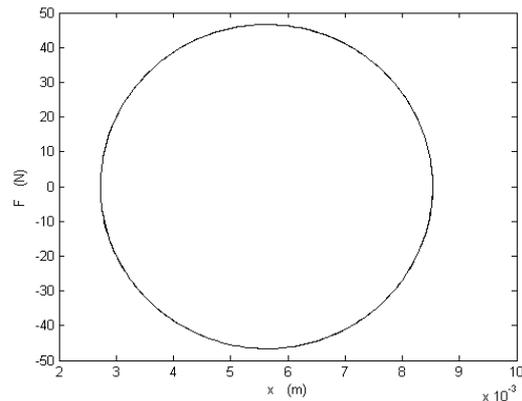


Fig.8 Variation of the magnetic force with the piston displacement

Fig 8 shows the driven force with the displacement of the piston. It is apparent that their phase difference, is about  $90^\circ$ , it is helpful to obtain the maximum driven force with high efficiency.

### 3.4 Clearance volume and Leakage in Compressor

#### 3.4.1 The influence of clearance volume

The clearance volume and the leakage play important roles on the performance of compressor, their effects are more sensitive in miniature compressor than in normal scale compressor. The clearance volume comes from three aspects: (1) the necessary space to avoid collision between the piston to the cylinder cover; (2) the piston's random excursion of equilibrium position in operation; (3) the piston's excursion of equilibrium position by load. If the excursion was 0.6mm, the clearance volume would be changed by 10%.

If the length of the clearance volume in cylinder was increased from 0.3mm to 1.5mm, the ratio of clearance volume to piston stroke would decreased from 0.95 to 0.77. In this prototype, the designed clearance length is 0.5mm, and the ratio is 0.924, the theoretical displacement capacity of the compressor is 17ml/s, the loss from the clearance volume is 1.3ml/s.

#### 3.4.2 The influence of leakage

The leakage between the piston and the cylinder is inevitable since the gap between the piston and the cylinder is necessary for the piston gliding. If the sealing relies on the gap, the narrower the gap, the less the leakage, but the more the friction. The narrow gap demands high decision on the cylinder surface manufacturing and the mover assembling. Besides, the gap should satisfy the clearance from thermal expansion.

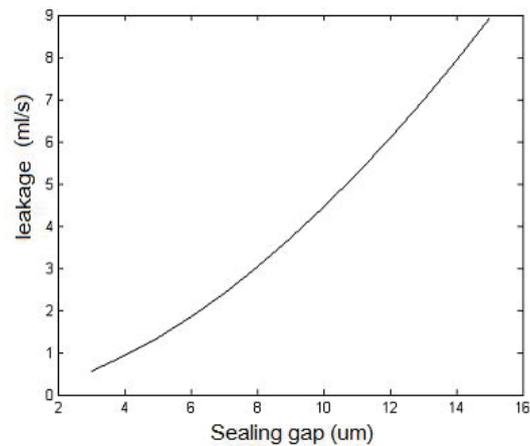


Fig.9 Variation of the leakage with the seal clearance

Based the prototype, the leakage relation with the gap is calculated as shown in figure 9. If the gap increased from 0.003mm to 0.015mm, the leakage would increased from 0.57ml/s to 8.94ml/s, the ratio of leakage to theoretical capacity would increase from 3.4% to 53%.

#### 4. EXPERIMENT ON A PROTOTYPE

Figure 10 show the photo of the prototype of the linear compressor, the diameter of the cylinder is 8mm, the outer diameter of the compressor is 40mm, and its length is 120mm, the mass of the mover is about 65g, the motor coil is about 600 cycles.



Fig 10 Prototype of miniature linear compressor

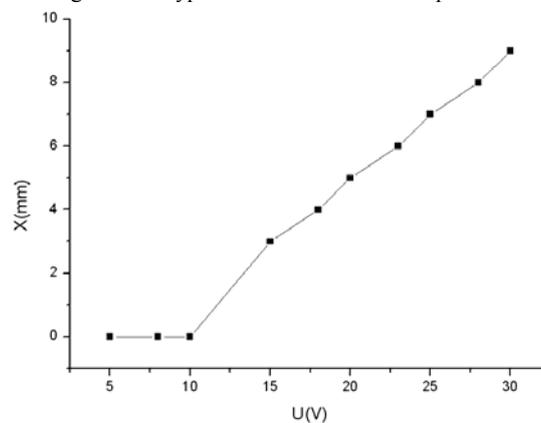


Fig.11 Variation of the no-load piston displacement with the input voltage

Fig. 11 show the piston displacement without load. In experiment, the piston hardly move until the driven voltage is more than 10V, according to above simulation, the friction force is about 23N, which is more than design value. Therefore, the piston displacement is less than the designed value, while the driven voltage is about 25V, the displacement is 7mm in test, much less than 11mm the design object.

According to dynamics calculation, the friction force is a important role on the piston oscillation amplitude in the prototype. In Fig. 12, is the friction is increased from 5N to 35N, the displacement of the piston is reduced from 9.4mm to 2.8mm.

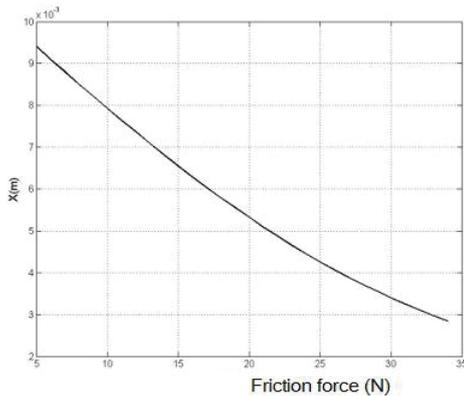


Fig.12 Variation of the load piston displacement with the friction force

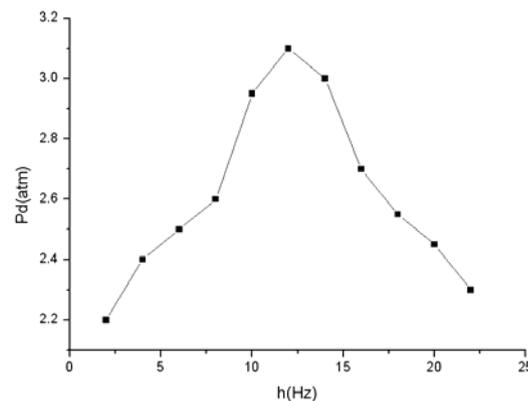


Fig.13 Variation of the load discharge pressure with the power frequency

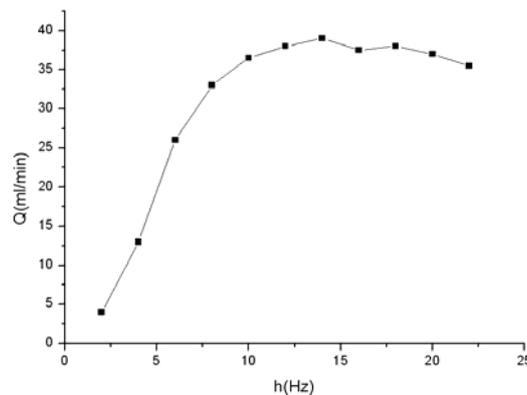


Fig.14 Variation of the load air displacement with the power frequency

The variation of driven frequency bring about the variation of its working capacity as well. Figure 13 and fig.14 show the profiles of experimental discharge pressure and displaced gas quantity. In the test, the driven voltage was about 26V, the charging pressure was  $2 \times 10^5$ Pa. Once the frequency is about 12Hz, the discharging pressure reaches the peak, about  $3.1 \times 10^5$ Pa, and the ratio of pressure rising is 1.55. According to figure 14, once the frequency is more than 12Hz, the gas displacement almost keep in 38ml/min. The discharging pressure and gas displacement do not reach the designed values, this is due to too large friction and leakage.

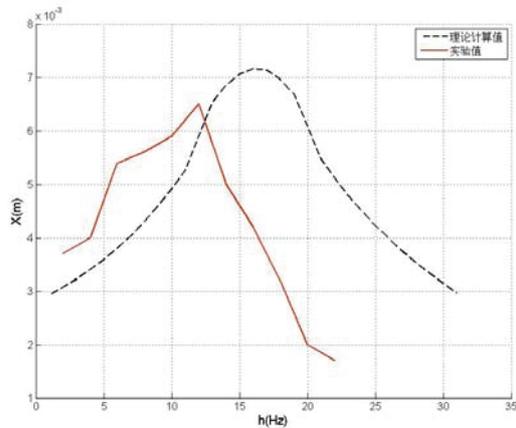


Fig.15 Contrast diagram of piston displacement between theory calculation and experimental results

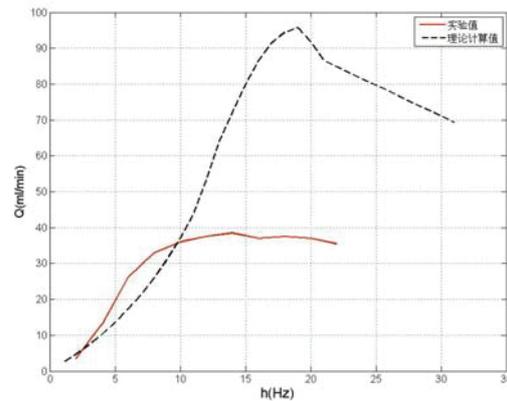


Fig.16 Contrast diagram of air displacement between theory calculation and experimental results

The experimental piston displacement is compared with the theoretical calculation in Fig. 15. Their variation trends are same with each other. However, the optimum frequency is 16Hz in theoretical calculation, but it is 12 Hz in experiment, the maximum piston displacement is 7.2mm in theoretical calculation, and it is 6.5mm in experiment.

In fig16, the theoretical and experimental gas placements are compared, there is a apparent difference, according to above discussion, it is because of the too much leakage and friction, the later leads reduction of the piston displacement and the frequency of the maximum piston displacement.

## 5. SUMMARY

In this paper, a prototype of miniature linear compressor is designed, the discussion about electromagnetic dynamics and gas dynamics with simulation. the feasibility and optimum about the design and operation have be analysis theoretically.

In experiment, the gas displacement is about 38ml/min, and the ratio of pressure rise is about 1.55 when the charging pressure is 2 bar. The characteristics of the linear compressor is similar with the simulated results, however, the displacement capacity and efficiency did not reach the expected values due to unideal friction and leakage.

Through this work, a series factors are analyzed about the linear compressor with simulation and experiment, and the experimental results verify partial design ideas, and display some problems which is helpful to improve it and further study.

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