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Characteristics of a Novel Moving Magnet Linear Motor for Linear Compressor

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Abstract

Linear compressors consisting of linear motor and suspension spring have been widely used for cryogenic and refrigeration systems. This paper characterises a novel moving magnet linear motor for linear compressor including magnet spring, motor force, resonant frequency and overall performance. First, the static force of the linear motor was modelled using FEA (Finite Element Analysis) method for various excitation current and armature positions. A specific test rig was built to measure the motor force and to verify the results from FEA simulation. The magnet spring was found to be 1.3 N/mm for peak stroke. The magnet spring effect was taken account into the calculation of the resonant frequency, which mainly depends on mechanical spring and gas spring. The linear compressor using the proposed linear motor and flexure springs was instrumented for measurements of the resonant frequency at different pressure ratios and strokes. Good agreement was shown between the calculated and measured resonant frequencies for various conditions. The high efficiency novel magnet linear motor is very attractive for linear compressor.

Key words: linear compressor, linear motor, FEA, magnet spring, resonant frequency, motor force

1. Introduction

A linear motor drives a motion in a single axis by electromechanical force without any motion conversion mechanism such as crankshafts, gears and screws [1]. Compared with conventional rotary induction motors, linear motors can achieve a higher efficiency when a linear motion is required, as the absence of crank mechanism reduces the friction loss. Linear motor can be used for linear compressor along with suspension spring to align the piston within the cylinder. Linear compressors have been widely used in Stirling cryocoolers (no reed valve) as a pressure wave generator. Valved linear compressors can be used for J-T coolers and vapour compression refrigeration systems to deliver mass flow of working fluid.

Several recent investigations have been carried out to study advantages of linear compressors over those reciprocating compressors. Bansal et al. [2] summarized

characteristics of linear compressors and concluded that linear compressors are more efficient and simple with the cooling capacity modulation. Bradshaw et al. [3] compared the influences of dead volume on overall isentropic efficiency in both linear and reciprocating compressors. It was concluded that the linear compressor could operate more efficiently over a wider range of dead volumes than the crank-drive compressor. Measurements of linear compressors and rotary compressors by Park et al. [4], Liang et al. [5] and Ku [6] demonstrated higher efficiency of linear motors over rotary compressors.

Generally, there are three types of linear motors distinguished by the structure: moving coil, moving magnet and moving iron. Liang [7] summarised the pros and cons of each type. For moving coil motors, it is easier to design and control the stroke, but low power density and high cost restrict their application in household refrigerators. Moving iron motors are robust and cost effective, but their low energy conversion density is the main shortcoming [8]. The permanent magnets (PMs) of moving magnet motors are less reliable than the other two types of linear motors and their robustness is more difficult to be ensured, because of the fact that PMs are subjected to vibration stress as mentioned by Li et al. [9]. With the development of moving magnet topologies, high-energy permanent magnet material such as NdFeB (Neodymium-Ferro-Boron), and soft magnetic composites for core, high force capacity and energy-saving effect have made moving magnet motors suitable choices for linear compressors.

Among all moving magnet motors for linear compressors, the most well-known design is the one proposed by Sunpower company. Magnets assembly are coupled with piston and move in the gap between two sets of steel lamination structures and a coil [10]. It included an inner core, an outer core, a tubular moving magnet cantilevered from suspension system, and a coil mounted inside the outer core. This topology has been widely used to drive compressors in VCR (vapour compression refrigeration) systems, Stirling and pulse tube cryocoolers. A commercial moving magnet linear motor (H2W Tech) was applied into a miniature scale linear compressor by Bradshaw et al. [11]. The motor efficiency was estimated to be 41.7% according to measured heat loss. Tijani and Spoelstra [12] used a moving magnet linear motor produced by Q-drive for thermoacoustic-Stirling cooler. The electroacoustic conversion efficiency was 80%. Liang et al. [13] proposed a novel moving magnet linear motor designed for a linear compressor incorporated in a small scale refrigeration system for electronics cooling.

Many previous studies have proved that when a linear compressor is operated at its resonant frequency, system efficiency reaches maximum. Some researchers have investigated the modelling of resonant frequency for linear compressors. The non-linear characteristic of gas pressure makes it difficult to derive the exact gas spring stiffness. A simple linearization approach to obtain the approximation of gas stiffness was presented by Cadman and Cohen [14], which is widely adopted by Bradshaw [15] and

Liang et al. [16]. The experimental results by Liang [16] showed the feasibility of this approach. A new solution to the linearization of the gas pressure force was proposed by Dainez et al. [17]. Based on the same effect on compressor of the gas pressure and equivalent forces, an equivalent damping force, an equivalent spring force, and an equivalent continuous force were used to replace the nonlinear gas force. However, in most of the studies mentioned above, the influence of magnet spring was not taken into account. As presented by Chen et al. [18], when zero current was supplied, magnetic force was zero at central position, but this increased greatly with the displacement. Xia et al. [19] further considered the effect of position-dependable magnetic force without excitation current as ‘magnet spring’. Simulations and tests were conducted to analyse the resonant frequency of the linear compressor for Stirling cryocooler. The results showed that magnet spring existed and influenced the resonant frequency for Sunpower type moving magnet linear compressor.

This work introduced a novel moving magnet linear motor for linear compressor and presented the characteristics of the linear motor. A FEA (Finite Element Analysis) model was built to study the static motor force and magnet spring. Measurements from static motor force and compressor were conducted to validate the FEA simulation. The resonant frequency was measured for wide range of operating conditions and calculated with consideration of magnet spring effect. The configuration and specifications of the linear motor are shown in Fig. 1 and Table 1 respectively. The proposed linear motor is composed of four laminated cores wound by coils and three moving magnets. Other combinations are possible as long as the number of cores is one greater than the number of magnets. The slotting of cores forms an air gap where magnets are placed. Coils on the one side of magnets are in serial, and in parallel with coils on the other side. Polarisation of magnets is alternative. Two identical linear motors have been prototyped for a VCR system. Note that the eddy current loss shown in Table 1 is high because the linear motor is not an optimized design.

Fig. 1 Configuration of the proposed moving magnet linear motor

Table 1 Design specifications for the moving magnet linear motor

Total mass of moving assembly (kg)	0.66
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Magnet material	NdFeB
Number of magnet	3
Dimension for each magnet (mm)	18×18×12.7
Number of core	4
Number of turns for each coil	620
Resistance of total motor coil (Ω)	3.5
Peak shaft force (N)	84
Peak current at peak force (A)	1.29
Maximum stroke (mm)	14
Core loss (W)	1.5
Eddy current loss (W)	10
Operating frequency (Hz)	30-50

2. Simulations

2.1 Computational Method

In a magneto-static solution, the magnetic field is produced by DC currents flowing in coils and by permanent magnets. There is no time variation effect included in a magneto-static solution, and objects are considered to be stationary. The Maxwell 3D Magneto-static module is capable of solving static electromagnetic problem for stationary objects when the magnetic field is produced by permanent magnets and by DC current flowing in conductors. The equation solved computed by 3D Magneto-static is derived from Ampere's Law, which is

$$\nabla \times \vec{H} = \vec{J} \quad (1)$$

where H is the magnetic field density, B is flux density, and J represents current density.

The Maxwell's equation can be written as

$$\nabla \cdot \vec{B} = 0 \quad (2)$$

where B is flux density.

The constitutive relationship for linear material is:

$$\vec{B} = \mu_r \mu_0 \vec{H} + \mu_0 \vec{M}_p \quad (3)$$

where μ_0 is the permeability of vacuum and μ_r is the relative permeability, $\vec{M}_p$ is the permanent magnetization.

For nonlinear material, Maxwell module provides an advanced setup option allowing a solution based on a previously computed demagnetization operating point

$$\vec{B} = \vec{B}_0 + [\tilde{\mu}](\vec{H} - \vec{H}_0) \quad (4)$$

$$[\tilde{\mu}] = \frac{\partial}{\partial \vec{H}} \vec{B} = [\Delta\tilde{\mu}] + [\mu] \quad (5)$$

Where $\vec{B}_0$ and $\vec{H}_0$ are the previous field solution, $[\Delta\tilde{\mu}]$ is a general full tensor, $[\mu]$ is magnetic permeability matrix.

In Maxwell 3D Magneto-static solver, magnetic field $\vec{H}$ is considered to be composed of the following components:

$$\vec{H} = \vec{H}_p + \nabla\varphi + \vec{H}_c \quad (6)$$

where φ is the magnetic scalar potential, $\vec{H}_p$ is a particular solution constructed by assigning values to all the edges in the mesh in such a way that Ampere's law holds on all contours of all tetrahedral faces in the mesh, $\vec{H}_c$ accounts for the permanent magnets.

Once the magnetic scalar potential is obtained, Maxwell's equations and the constitutive relationship are then used to obtain the magnetic field and magnetic flux density. The advantages of this formulation over other existing ones includes saving considerably computational resources, the absence of a gauge due to excellent numerical stability, and lowering cancellation errors [20].

2.2 Mesh and Flux Density Distribution

The simulation was carried out with the following assumptions:

- 1) The influence of the strut used to place and support three magnets on the flux distribution can be neglected so that the model can be simplified;
- 2) Coils are wound uniformly around cores, and current distributes uniformly at each conductor.

In order to implement a magneto-static analysis, a three-dimensional FEA model was constructed in ANSYS Maxwell [20]. Fig. 2 shows the FEA mesh of the proposed linear motor. Maxwell adopts an adaptive iterative process to produce optimal mesh in which the mesh in high error density is refined. It can be seen that the mesh is very dense in the air gap between magnets and cores. Relatively fine mesh was also used in the magnets region where the shaft force is integrated. Length-based mesh refinement inside objects was applied to cores, magnets and air gap between them. Desired calculation accuracy was ensured when the maximum element length for cores, magnets and gaps were restricted smaller than 0.1 mm, 0.05 mm and 0.01 mm respectively.

Fig. 2 FEA model of the moving magnet linear motor in ANSYS Maxwell

Current multiplied by the number of coil turns was applied to each winding through a section perpendicular to its flow path. The direction of current was opposite for windings at each side of magnets. Two types of default boundary conditions were defined: natural boundary condition and Neumann boundary condition. Magnetic field is continuous across object interfaces according to natural boundary condition. For Neumann boundary condition, magnetic field is tangential to the outer boundary and flux cannot cross it. Insulating boundary conditions was set between the surfaces of windings and cores as there was insulating cover for each coil. The nonlinear residual error for the 3D Magnetostatic solver was set to be 0.001.

Fig. 3 shows the flux density distribution of the linear motor when 1.5 A was applied to each coil at armature position 0 mm. As can be seen, flux density distribution in three cores is similar while the flux density in the other core is very small. Two adjacent electrified windings create opposite magnetic fields, and the polarities of two adjacent magnets are opposite as well. Magnetic fields are composed of two parts, magnetic fields created by electrified windings and magnetic fields created by magnets. When the armature is at a specific position, three magnets form three closed magnetic circuits with three cores with magnetic field in the same direction and air respectively. As the amplitude of excitation current, the material and size of magnets are all same, three cores that form magnetic circuits with magnets have similar flux density distribution. The fourth core forms magnetic circuits with air. As air is of large magnetic resistance, the core forming magnetic circuits with air has very small flux density. However, as moving magnet assembly oscillates, the flux density on the fourth core will vary accordingly. All of the other three coils and coils form the magnetic circuit with one magnet and small air gap. Large flux density is shown at the slotting corners of cores that are close to the magnet in its circuit, while very small flux density can be found at the slotting corners on the other side.

Fig. 3 Magnetic flux density distribution of the linear motor with a current of 1.5 A at armature position of 0 mm

2.3 Magnet Spring Force Results

Even when no current is applied to coils, there is magnetic force (shaft force) applied on magnets. The position-dependable interaction of permanent magnets and cores when zero current is applied to the motor can be referred as magnet spring [19]. The corresponding shaft force is magnet spring force. FEA simulations were carried out at different armature positions to investigate the magnet spring force.

The shaft force at different positions varying from -8mm to 8mm is shown in Fig. 4. It can be seen that when armature is at its central position, the shaft force applied on magnets equals to zero. However, with the increase of the displacement, the shaft force also increases. The direction of the magnet spring force is the same as its displacement direction, which means the magnet spring force tends to move the armature further from its central position if the motion is not constrained. The magnet force is distributed symmetrically along the central position of armature. When the armature position is 7 mm, the magnet spring force is 9.0 N. This is corresponding to a magnet spring rate magnitude of 1.28 N/mm.

Fig. 4 Shaft force variation with armature position without excitation current

2.4 Resonant Frequency Model

For linear compressor, damping constant can be ignored and the resonant frequency of the mass-spring system can be described as follow:

$$f = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \quad (7)$$

where f is the resonant frequency, m is moving mass (magnet and piston assembly), and k is the total spring stiffness.

The total spring stiffness of the compressor consists of mechanical spring (flexure spring) k_{me} , gas spring k_g and magnet spring k_{mag} which can be described as

$$k = k_{me} + k_g + k_{mag} \quad (8)$$

As shown in Table 1, the total moving mass is 0.66 Kg and the mechanical spring stiffness is 16.3 N/mm. As has been reported by Pollak et al. [21], for low pressure ratio system, the linearized gas spring force was divided by the stroke of the compressor, which would yield estimated gas spring stiffness in the compression chamber.

$$k_g = \frac{(P_d - P_s)A}{S} \quad (9)$$

where P_s and P_d represent suction pressure and discharge pressure respectively, A represents piston area, and S is the stroke of linear compressor (peak-to-peak value of armature/piston displacement).

The magnet spring stiffness can be calculated by the following formula:

$$k_{ma} = \frac{F_{ms}}{x} \quad (10)$$

where F_{ms} represents magnet spring force, and x represents the armature/piston displacement. The magnet spring force tends to move the magnets towards regions of higher flux, so that there is always an attractive force pulling the magnets.

3. Instrumentation

3.1 Static Motor Force Measurement

Static motor force measurements were conducted at different armature/piston positions and different excitation currents. Fig. 5 shows the schematic diagram of static motor force measurement. There is a 10 μm clearance between piston and cylinder wall to reduce the friction loss and to have least gas leakage. The four coils were connected in series, and a DC current through a power amplifier was supplied to the linear motor. A Hall Effect current transducer was utilized to measure the DC current. A linear variable differential transformer (LVDT) displacement transducer was used to measure the armature/piston position. Helium gas was charged into the linear compressor to maintain a certain armature position through the suction port or the body port, depending on the direction of the DC current flow. A pressure gauge was used to measure the gas pressure when the armature/piston position was settled.

Fig. 5 Schematic diagram of static motor force measurement (PG: pressure gauge, CT: current transducer, c: radial clearance, LVDT: linear variable differential transformer)

The armature/piston position for testing ranges from -7 mm to 7 mm in a step of 1 mm. For each armature/piston position, the DC current was changed from -1.8 A to 1.8 A in a step of 0.2 A. For each operating condition (excitation current), the pressure regulator was adjusted manually to keep the armature/piston at its desired position. Note that the

force measured is a resultant force of axial mechanical spring force and static motor force. The static motor force F_m can be estimated according to

$$F_m = P * A - k_{me} * x \quad (11)$$

where P is the gauge pressure (either body pressure or suction pressure).

Instruments for the measurement of static motor force are listed in Table 2. Note that there is phase shift for the LVDT modulator which caused error of displacement measurements.

Table 2 List of instruments for static motor force measurements

Instruments	Model	Accuracy
Current transducer	LA LEM 25-NP	±0.5 %
LVDT	Lucas Schaevitz	± 0.025 mm
LVDT modulator	ATA-101	Phase shift: 5 degrees for 50 Hz
Pressure gauge	DRUCK 1400	±0.15 %
Capacitor	EPCOS B32361	±5 %
DC Power amplifier	22AH370 /00 /15 /29	Not applicable
Helium gas	BOC	Not applicable
Vacuum pump	Gardner Denver–WZ-79204-05	Not applicable

3.2 Linear Motor/Compressor Measurement using Helium

The proposed linear motor/compressor has been instrumented for measurement using gas. The linear compressor has a maximum stroke of 14 mm, piston diameter of 19 mm, and maximum shaft power output of 100 W. Two identical compressors were mounted back-to-back with shared discharge and suction lines. The detail of the test rig has been reported by Liang et al. [16]. Table 3 lists the main instruments for the test rig.

Table 3 List of instruments for the measurements of linear compressor

Instruments	Model	Quantity	Accuracy
Data acquisition card	NI USB-6341	2	Not applicable
Current transducer	LA LEM 25-NP	2	±0.5 %
Voltage attenuator	Fylde 261HVA HV	1	±0.5 %
Voltage isolation amplifier	Fylde 4600A	1	±0.5 %
Power analyser	PM100	1	±1.5 %

LVDT	Lucas Schaevitz	2	± 0.025 mm
Pressure gauge	DRUCK 1400	3	± 0.15 %
Capacitor	EPCOS B32361	2	± 5 %
AC amplifier	Vonyx VXA-2000	1	Not applicable
Pressure transducer	Druck PMP1400	4	± 0.15 %
Thermocouple	K-type	8	$\pm 1.5^{\circ}\text{C}$
Mass flow meter	Hastings HFM-201	1	± 1 %
Solenoid valve	Asco Solenoid Valve H262602DOD1	2	Not applicable
Oscilloscope	RS Pro IDS1000 Series IDS1072AU	3	Not applicable

A series of measurements have been conducted at different pressure ratios, strokes and drive frequency using Helium. Table 4 lists the test conditions for linear compressor measurements using helium. Two NI USB-6341 DAQ (data acquisition) cards which has a maximum sampling rate of 500 kHz were used to collect real time data for a low-speed DAQ system (LDAQ) and a high-speed DAQ system (HDAQ). The DAQ card has 16 analogue input channels and 2 analogue output channels. The pressure ratio was changed by regulating the throttling/expansion valve. A sinusoidal voltage signal was generated by LabVIEW through analogue output channel of DAQ card. The signal was then amplified by an AC power amplifier to drive the linear motor. A PID controller was built in LabVIEW to control the armature/piston stroke by adjusting the voltage amplitude. The driving frequency of the voltage signal was kept at the resonant frequency of the linear compressor.

Table 4 Test conditions for the linear motor/compressor measurements using helium

Gas	Helium
Pressure ratio	1.5, 2.0, 2.5, 3.0, 3.5
Stroke (mm)	6, 8, 9, 10, 11, 12, 13
Fill Pressure (bar)	3.5, 7, 8
Operating frequency (Hz)	28.5-41

The resonant frequency for various pressure ratios and strokes was found experimentally. By manually adjusting the driving frequency of the voltage signal in LabVIEW, the frequency at which the simplified motor efficiency reached the maximum

was considered to be the resonant frequency for the specific operating point. The simplified motor efficiency was calculated by:

$$\eta = \frac{P - I^2 R}{P} \quad (12)$$

where P is the electric power into to the linear motor measured by the power analyser, I is the current measured by the current transducer, R is the total resistance of the windings.

Fig. 6 shows the simplified motor efficiency varying with drive frequency at a pressure ratio of 2.0 using helium for various strokes. The resonant frequency for 13 mm, 10 mm, and 6 mm is 39 Hz, 34 Hz, and 32.5 Hz respectively. As the minimum step for adjusting the drive frequency is 0.5 Hz, there is error in resonant frequency from measurements. However, it can be seen that the motor efficiency curve is sufficiently flat for frequencies within 1 Hz difference from the resonant frequency. Therefore, the measured resonant frequency has sufficient accuracy, which can be used to validate the calculations.

Fig. 6 Simplified motor efficiency varying with drive frequency at pressure ratio of 2.0 using helium

3.3 Interpolation of Dynamic Motor Force

During the linear compressor operation, the motor force varied continuously with the armature/piston position and instantaneous excitation current. Measuring the dynamic motor force directly was difficult. Interpolating dynamic motor force was conducted to reveal the dynamic motor force characteristic. The measurements of static motor forces at different armature position with various excitation current provides a force-current-displacement 3-D map which can be used for the interpolation. Fig.7 gives the Linear

motor Force-Current-Displacement 3-D map according to the static motor force measurement (Section 3.1). The shaft force at different armature positions with various excitation current can be interpolated accordingly.

Fig. 7 Linear motor Force-Current-Displacement 3-D map from the static motor force measurement

4. Results and Discussions

4.1 Comparison of Static Motor Force between FEA Simulation and Measurement

Fig. 8 shows the comparison of static motor force varying with excitation current at armature position of -2 mm and -4 mm between the FEA simulation and measurement. Note that negative displacement means moving towards the cylinder head. Generally, the FEA simulation results are in very good agreement with the measured results. It can be seen that motor force increases as current increases. The variation of motor force with current at a fixed armature/piston position includes linear part and nonlinear part. The difference between the FEA simulation and the measurements in linear part can be ignored. In the linear part, the slope of force variation is the force constant of this motor. As shown in Fig. 8 (a) and (b), the force constant for the position -2 mm is about 63 N/A while the force constant for the position -4 mm is about 62 N/A. The force constant decreases to 60 N/A when the armature position is 6 mm away from the centre. At the maximum position (± 7 mm), the force constant is 50 N/A. With the increase of the excitation current after 1.2 A for the armature position of -2 mm, saturation of motor force occurs, which is due to the nonlinear permittivity of the core. The positive saturation current is smaller than the negative saturation current. In Fig. 8 (a), the measured positive saturation occurs at a current of 1.2 A, while motor force does not saturate even when negative current reaches -1.8 A. It can also be seen that the more

distance that the moving assembly moves away from the datum position, the earlier saturation will occur. The difference between the FEA simulation and the measurements in non-linear part is caused by the hysteresis characteristic of core, which has not been considered in the FEA simulation. As the hysteresis loop of soft material is very narrow, its influence on the linear part is small while the influence on saturation analysis is more obvious. Consider that the linear motor is operated in the linear part in most cases, the largest difference of nonlinear part is about 18%, FEA results are acceptable.

Fig. 8 Shaft force varying with current at different armature positions; (a) -2mm (b) -4 mm

4.2 Voltage, Current, Displacement from Measurements

Fig. 9 (a) shows the measured voltage and current in two cycles for a pressure ratio of 2.0, a stroke of 6 mm, and a drive frequency of 31.5 Hz. It can be seen that there is a phase difference between the voltage and current. The voltage signal leads the current signal about 40° phase angle, which means that the inductance in linear motor has not been balanced by the capacitor. This is due to the limited range of capacitance box. The power factor for this test condition is 0.76. The amplitude of voltage is 85 V, and the amplitude of current is about 2.1 A. Note that the measured current is for two linear motors in parallel, so that the real power consumed by each linear motor is about 68 W. A 23% voltage input can be reduced if a matched capacitor is used.

Fig. 9 (b) shows the measured displacement and the interpolated motor force in two cycles for this particular test condition. The interpolated motor force is shown to vary as a sinusoidal function over a cycle. As can be seen, phase angle difference between displacement and shaft force is about 70° . It indicates that the drive frequency is lower than the resonant frequency of linear compressor because Chun has reported that the angle between displacement and current should be smaller than 90° when drive frequency is lower than resonant frequency [22]. Fig. 6 shows the measured resonant frequency is 33.5 Hz for this condition while the drive frequency is 31.5 Hz. The amplitude of interpolated shaft force is 73 N. The shaft force leads the displacement about 90° as expected. This is because shaft force should be in phase with velocity according to Faraday law of induction.

(a)

(b)

Fig. 9 Measured voltage, current, displacement and interpolated shaft force in one cycle for a pressure ratio of 2.0, a stroke of 12 mm and a drive frequency of 31.5 Hz; (a) measured current and voltage (b) measured displacement and interpolated shaft force

4.3 Gas Spring and Resonant Frequency

Fig. 10 shows the measured pressure-volume (P - V) diagram for a stroke of 13 mm and a pressure ratio of 3.5 using helium in which the suction and discharge pressures are also plotted. Four processes (suction, compression, discharge, and expansion) are clearly illustrated. There is significant pressure drop across the reed valves when they open. This will account for the overall efficiency of the linear compressor. The thermodynamic power calculated according to the area of the enclosed P - V loop for this condition is 63 W. This is lower than the shaft power (product of shaft force and piston velocity) 79.5 W. The difference between the two is due to the gas leakage across the 10 μm radial clearance gap and heat transfer through the cylinder wall. It can also be seen from Fig. 9 that the gas force looks very linear for the low pressure ratio.

Fig. 10 P - V diagram of the linear compressor for a stroke of 13 mm and a pressure ratio of 3.5 using helium

Fig. 11 shows the comparison of resonant frequency between calculation and measurement at various strokes and pressure ratios using helium. Close agreement has been achieved. The maximum error between the calculated and measured results is only 4.6%. As the drive frequency was adjusted by a minimum step of 0.5 Hz, an error lower than 1 Hz is reasonable. The change of resonant frequency is mainly induced by the change of gas spring rate as shown in Equation (7). For a fixed pressure ratio, with the increase of stroke, the resonant frequency decreases due to lower gas spring stiffness. The pressure ratio affects the resonant frequency greatly. Larger pressure ratio leads to higher resonant frequency due to higher gas spring stiffness according to Equation 8. When the stroke is fixed at 8 mm, resonant frequency increase by about 10% as pressure ratio varies from 1.5 to 3.0. Table 4 gives the calculated spring rate for various stroke at a pressure ratio of 3.0. For the design point (pressure ratio of 3.0 and stroke of 13 mm), the gas spring stiffness is 16.78 N/mm, mechanical spring stiffness is 16.3 N/mm, and magnet spring is -1.16 N/mm. Previous work by other researchers [23] adopted coil springs to achieve very high mechanical spring stiffness. This can have two effects: resonant frequency can be dominated by the mechanical spring so that the range of drive frequency is narrow; piston offset will be much smaller. However, this is not desirable when size and weight are concerned.

Table 4 Spring stiffness for various stroke at a pressure ratio of 3.0

Stroke (mm)	Mechanical spring stiffness (N/mm)	Gas spring stiffness (N/mm)	Magnet spring stiffness (N/mm)

8	16.3	26.61	0.60
9	16.3	23.78	0.68
10	16.3	21.78	0.78
11	16.3	19.85	0.89
12	16.3	18.12	0.98
13	16.3	16.78	1.16

Fig. 11 Comparison of resonant frequency between calculation and measurement at different strokes and pressure ratios (PR) using helium

4.4 Capacitance

In order to reduce the voltage input, a capacitor is usually employed in series with the linear motor. Fig.11 shows the electric circuit of the linear motor. When it is operated at resonant frequency, the voltage of the impedance associated with motor force will be in phase with current as displacement leads current 90° , so the circuit then becomes a typical RLC circuit. The matching capacitance is dependent on the inductance and drive frequency as follow

$$C = \frac{4\pi^2 f^2}{L} \quad (13)$$

The total inductance of two linear motors is 0.141 H. However, as shown in Fig. 11, resonant frequency varies for different operation conditions.

Fig. 12 shows the matching capacitance for various strokes and pressure ratios. It can be seen that the trend of capacitance is opposite from that of resonant frequency. With the increase of stroke or decrease of pressure ratio, the required matching capacitance

increases. When a stroke of 12 mm is fixed, the required matching capacitance increases by 72 μF when pressure varies from 1.5 to 3. For a pressure ratio of 2.0, the matching capacitance varies from 144 μF to 182 μF . The capacitor needs to be adjustable for a large range if it is expected to reduce the voltage input under various test conditions particularly when the gain of the AC power amplifier is not sufficiently high.

Fig. 12 Electric circuit diagram of the linear motor

Fig. 13 Capacitance compensation at different strokes and pressure ratios using helium

5. Conclusions

A FEA model was established to analyse the static motor force and magnet spring of a novel linear motor for a linear compressor. A test rig was built to verify the simulated results. Another test rig was built to measure the performance of the linear motor/compressor system. The resonant frequency of the linear compressor was measured for comparison with calculated values with consideration of the magnet spring effect. Key findings are as below:

- (1) Saturation current reduces as armature position increases, and it is also influenced by current direction. The force constant of the linear motor does not vary with armature position when it is within 5 mm from the centre.
- (2) Equivalent magnet spring force increases with armature positions. It tends to drive armature further from central position. Magnet spring force reaches 9 N when armature position is 7 mm.
- (3) Taking the magnet spring into account, the resonant frequency was calculated. The maximum error between the measured and calculated resonant frequency is only 4.3%.
- (4) The dynamic motor force was interpolated based on measured displacement, measured static force varying with excitation current and positions. Resonant frequency increases with pressure ratio.
- (5) The matching capacitance for the linear motor varies in a large range with various pressure ratios and strokes when it is operated at resonant frequency.

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Reference

- [1] J. F. Gieras, Z. J. Piech, and B. Z. Tomczuk, *Linear Synchronous Motors*, 2nd ed. Boc Raton, FL, USA: CRC Press, 2012.
- [2] Bansal, P., Vineyard, E., Abdelaziz, O., 2011. Advances in household appliances – a review. *Appl. Therm. Eng.* 31, 3748–3760.
- [3] Bradshaw, C.R., Groll, E.A., Garimella, S.V., 2012. Linear compressors for electronics cooling: energy recovery and the useful benefits. *Proceedings of International Compressor Engineering Conference, Purdue*, 1134, pp. 1–10.
- [4] Park, K., Hong, E., Lee, H.-K., 2002. Linear motor for linear compressor. *Proceedings of International Compressor Engineering Conference, Purdue*, C12-1.
- [5] Liang, K., Stone, C.R., Hancock, W., Dadd, M.W., Bailey, P.B., 2014. Comparison between a crank-drive reciprocating compressor and a novel oil-free linear compressor. *Int. J. Refrigeration* 45, 25–34.
- [6] Ku, B., Park, J., Hwang, Y., Lee, J., 2010. Performance evaluation of the energy efficiency of crank-driven compressor and linear compressor for a household

refrigerator. Proceedings of International Compressor Engineering Conference, Purdue, 1218, pp. 1–8.

[7] Liang, K., 2017. A review of linear compressor for refrigeration. *Int. J. Refrigeration* 84, 253–273.

[8] Wang, J.-B., Howe, D., Lin, Z.-Y., 2007. Comparative studies on linear motor topologies for reciprocating vapor compressors. *IEEE International Electric Machines & Drives Conference*, Antalya, pp. 364–369.

[9] Li, X., Xu, W., 2017. Comparative study of transversal-flux permanent magnetic linear oscillatory machines for compressor. *IEEE transactions on Industrial Electronics* 65, pp. 7437-7466.

[10] Unger, R., Novotny, S., 2002. A high performance linear compressor for CPU cooling. *Proceedings of International Compressor Engineering Conference*, Purdue, C23-3.

[11] Bradshaw, C.R., Groll, E.A., Garimella, S.V., 2011. A comprehensive model of miniature-scale linear compressor for electronics cooling. *Int. J. Refrigeration* 34, pp. 63-73.

[12] Tijani, M.E.H., Spoelstra, S. 2008. Study of a coaxial thermoacoustic-Stirling cooler. *Cryogenics* 48, pp. 77-82.

[13] Liang, K., Dadd, M.W., Bailey, P.B., 2013a. Clearance seal compressors with linear motor drive e part i: background and system analysis. *Proc. Inst. Mech. Eng. J. Power Energy* 227 (3), pp. 242-251.

[14] Cadman, R.V., Cohen, R., 1969b. Electrodynamic oscillating compressors: Part 2 Evaluation of specific designs for gas load. *ASME J. Basic Eng.* 91(4), pp. 656-663.

[15] Bradshaw, C.R., 2012. A Miniature-scale Linear Compressor for Electronics Cooling. (Ph.D. Thesis). Purdue University.

[16] Liang, K., 2014. A Novel linear Electromagnetic-Drive Oil-Free Refrigeration Compressor (Ph.D. thesis). Oxford University.

[17] Dainez, P.S., Oliveira, J., Nied, A., Cavalca, M.S.M., 2014. A linear resonant compressor model based on a new linearization method of the gas pressure force. *Int. J. Refrigeration* 48, pp. 201–209.

[18] Chen, N., Tang, Y.J., Wu, Y.N., 2007. Study on static and dynamic characteristics of moving magnet linear compressors. *Cryogenics* 47, pp. 457–467.

[19] Xia, M., Chen, X., 2010. Analysis of resonant frequency of moving magnet linear compressor of Stirling cryocooler. *Int. J. Refrigeration* 33 (4), pp. 739–744.

[20] Ansys Maxwell, ANSYS, Inc. Available on <http://www.ansys.com/>.

- [21] Pollak, E., Soedel, W., Cohen, R., and Friedlaender, F.J., 1979. On the resonance and operational behavior of an oscillating electrodynamic compressor. *Journal of Sound and Vibration* 67(1), pp. 121-133.
- [22] Chun T.-W., Ahn, J.-R., Yoo, J.-Y., Lee C.-W. 2004. Analysis and Control for Linear Compressor System Driven by PWM. The 30th Annual Conference of the IEEE Industrial Electronics Society, Busan.
- [23] Lee, H.-K., Song, G.-Y., Park, J.-S., Hong, E.-P., Jung, W.-H., Park, K.-B., 2000. Development of the linear compressor for a household refrigerator. *Proceedings of International Compressor Engineering Conference, Purdue*, pp. 31–38.