# The Effect of Clearance Control on the Performance of an Oil-free Linear Refrigeration Compressor and a Comparison between using a Bleed Flow and a DC Current Bias

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**ABSTRACT**

A moving magnet type oil-free linear compressor has been designed for applications such as electronics cooling. A control system using a solenoid operated valve was developed for DC offset and axial clearance control. Operation of the linear compressor with a fixed clearance of 0.8 mm was compared with a zero DC offset operation. The fixed clearance operation requires a higher power input.

The mean piston position can be controlled by using a DC bias on the drive voltage or by using a bleed flow from the compressor body to the suction side of the compressor. Using a DC bias on the drive voltage induces a higher power loss than using a bleed flow when there is a small radial clearance between the piston and cylinder. This provides evidence that the piston control in a refrigeration system with capacity control should be for a zero DC offset using a bleed flow.

Keywords: oil-free, linear compressor, DC offset, PID control, fixed clearance, DC bias

## NOMENCLATURE

| | |
|---|---|
| A | area ($mm^2$) |
| AC | alternating current |
| BLDC | brushless direct current |
| COP | coefficient of performance |
| $c$ | radial clearance (μm) |
| $d$ | DC offset (mm) |
| D | piston diameter (mm) |
| DAQ | data acquisition card |
| DC | direct current |
| $f$ | frequency (Hz) |
| $F$ | force (N) |
| FFT | fast Fourier transformation |
| GUI | graphical user interface |
| HDAQ | high-speed DAQ |
| $I$ | drive current (A) |
| k | spring stiffness (N $mm^{-1}$) |
| L | piston length (mm) |
| LDAQ | low-speed DAQ |
| LVDT | linear variable differential transformer |
| m | mass (kg) |
| $\dot{m}$ | mass flow rate (g $s^{-1}$) |
| $P$ | pressure (bar) or power (W) |
| PID | proportional-integral-derivative |
| PR | pressure ratio |
| PV | process variable |
| PWM | pulse-width-modulation |
| R | resistance (Ω) |
| $R_g$ | specific gas constant (J $kg^{-1}$ $K^{-1}$) |
| RMS | root mean square |
| $S$ | stroke (mm) |
| SP | set point |
| $T$ | temperature (K) or period of one oscillation (s) |
| $V$ | volume ($mm^3$) |
| VCR | vapour compression refrigeration |
| $\dot{W}$ | rate of compression work (W) |
| $x$ | displacement (mm) |

### Greek

| | |
|---|---|
| $\omega$ | frequency (rad $s^{-1}$) |
| $\eta$ | efficiency |
| μ | viscosity (kg $s^{-1}$ $m^{-1}$) |
| $\gamma$ | adiabatic index |
| $\beta$ | damping coefficient (N$s$ $m^{-1}$) |

### Subscripts

| | |
|---|---|
| 1 | cylinder or bleed flow |
| a | amplitude |
| adb | adiabatic |
| b | body |
| cl | clearance |

| | |
|---|---|
| dc | direct current |
| dis | discharge |
| e | electrical |
| m | mean or mechanical |
| s | seal loss |
| suc | suction |

# 1. Introduction

## 1.1 Linear Compressor Technology

There is a need to improve the capabilities of cooling technology to dissipate increasingly high heat fluxes (approaching 200 W/cm$^2$) from electronic components in order to maintain acceptable operating temperatures, as has been indicated by Bailey et al. **[1]**. The linear compressor with clearance seal and flexure springs, which has been used for many years for space applications, is an attractive proposition for vapour compression refrigeration (VCR) systems in electronics cooling applications. Liang et al. **[2]** concluded that linear compressors offer several benefits compared to traditional compressor technologies:

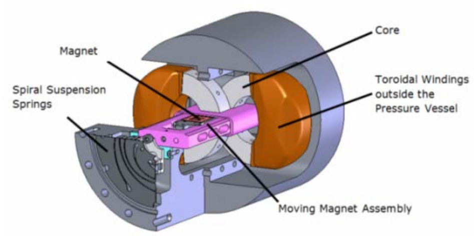
(1) A linear compressor does not have a crank mechanism to drive the piston which means less frictional loss, and thus a higher mechanical efficiency;
(2) Linear compressors can be operated without oil lubrication;
(3) A linear motor has a higher efficiency than rotary induction or brushless direct current (BLDC) motors;
(4) Linear compressors should operate at resonance to minimise the drive current and ohmic losses;
(5) Linear compressors can be compact as the size decreases with increasing frequency;
(6) To meet varying demands, a linear compressor does not need to operate in a 'start-stop' mode, but can be controlled by a simple control which varies the excitation voltage.

A number of works have been undertaken to develop linear compressors for various applications. Sunpower have been involved in these technologies for a long period **[3, 4]**. LG have licensed Sunpower's linear technology and have been marketing linear compressor systems for household refrigeration since 2002 **[5]**. Recent developments include a moving magnet design from Clever Fellows (Q drive) **[6]** and one from Infinia **[7]**. A more recent development by Embraco, is an oil-free linear compressor designed for R134a and R600a **[8]**. Embraco launched their oil-free linear compressor in 2014 with a license from Fisher & Paykel **[9]**. Bradshaw **[10]** has also built a linear compressor model that was validated by testing a prototype linear compressor that used a moving magnet motor designed by H2W Tech.

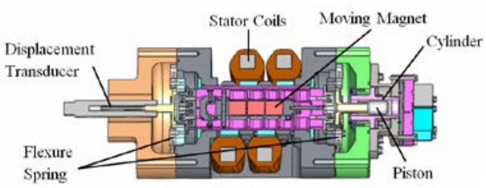
## 1.2 Oxford-type Moving Magnet Linear Compressor

Since moving coil type linear compressors are too expensive to be used in many applications **[1]**, the need for a low cost (in materials and manufacture) led to a novel moving magnet linear compressor. The new design of low-cost moving magnet linear compressor (for use with compact refrigeration heat exchangers) was completed at the University of Oxford in 2009, based on successful oil-free linear compressor technology used in space cooler applications since the 1970s. A prototype of this new linear compressor was built for experimental investigations.

Fig. 1 (a) and (b) shows the key components of the moving magnet linear compressor. The linear suspension system has two sets of spiral springs to maintain the alignment of the piston within the cylinder. The accuracy of this linear system permits a radial clearance between piston and cylinder of about 10 μm to be used. This is sufficient to avoid wear, while keeping leakage losses to a minimum. The static assembly has 4 magnetic circuits consisting of coils wound on laminated and slotted cores to form an air gap. The moving assembly (3 rectangular magnets in a line within the air gap) is attached to the piston. When an alternating current is supplied to the cores, an alternating axial force is induced in the moving magnet assembly, allowing the piston to reciprocate at the same frequency. Further details of the compressor design have been reported by Bailey et al. **[1]**.




(a) Moving magnet motor design




(b) Cross section of the linear compressor and motor

**Fig. 1** Design of the 100W Oxford moving magnet compressor

Table 1 gives the design specification of the moving magnet compressor, which uses two identical compressor halves (namely Compressor 1 and 2) mounted back-to-back and in-line to minimise exported vibrations. The two compressor halves shared suction and discharge lines and electrical connections.

**Table 1** Specification of each linear compressor half

| | |
|---|---|
| Total mass of magnet/piston (kg) | 0.66 |
| Piston diameter (mm) | 18.99 |
| Total series[1] resistance of coils (Ω) | 14 |
| Peak shaft force (N) | 84 |
| Peak current at peak force (A) | 1.29 |
| Flexure stiffness (Total) (kN $m^{-1}$) | 17 |
| Maximum stroke (mm) | 14 |

[1]The four motor coils were connected with two coils in series then each pair in parallel to give a resistance of ~3.5 Ω for each compressor half.

This prototype linear compressor has been tested using gas (helium and nitrogen) and R134a. The experimental results can be found in Liang et al. **[11]** and Liang et al. **[12]** respectively. It has also been experimentally compared to a crank-drive reciprocating compressor driven by an induction motor **[2]**. An adiabatic efficiency of 42% - 60% and a motor efficiency of 71% - 89% were maintained over a fairly wide range of operating parameters, suggesting that systems using this type of compressor could have good part load efficiency. A Coefficient of Performance (COP) of 3.2 was achieved at a cooling capacity of 384 W with an evaporator temperature of 20°C and a condenser temperature of 50°C using R134a. This is assumed to be a typical CPU cooling condition.

This work focuses on the piston DC offset (or 'piston drift' in other words) which is a key issue associated with the operation of the linear compressor. The control of the piston DC offset or clearance affects the performance of the linear compressor in terms of useful stroke, motor efficiency, and volumetric efficiency. Note that the clearance (or top clearance, or axial clearance) is defined as the distance between the piston and the cylinder head at the position of minimum cylinder volume. A new approach to the piston DC offset control has been proposed and developed here. This control approach can be used for clearance volume control as well. Whether the linear compressor should operate with a fixed zero DC offset or fixed clearance has been investigated by experiments using gas.

### 1.3 Piston DC Offset

The problem of DC offset is caused by the differential pressure generated across a clearance seal which has a fluctuating pressure on one side of it (the piston-cylinder) and an almost constant pressure on the other (the compressor body). Fig. 2 shows the schematic of the linear compressor system. The pressure differential is given by

$$\Delta P_1 = P_{1,\mathrm{m}} + P_{1,\mathrm{a}} \sin(\omega t) - P_\mathrm{b} \tag{1}$$

where $P_\mathrm{b}$ is the pressure in the body.

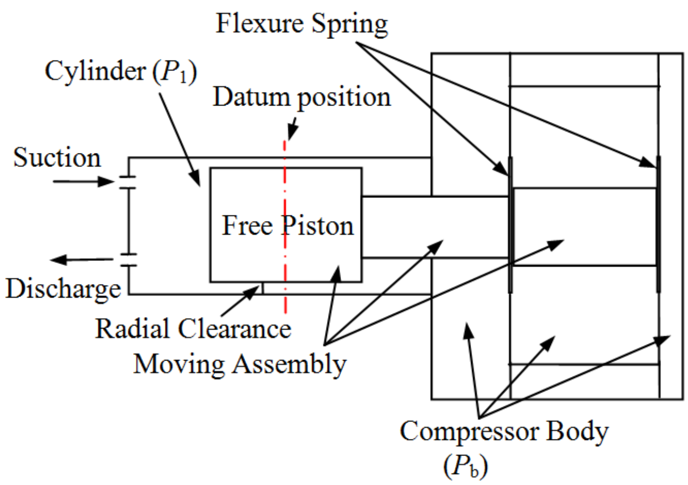



**Fig. 2** Schematic of the linear compressor system

If the body pressure $P_\mathrm{b}$ is equal to the mean working pressure $P_{1,\mathrm{m}}$, then the net volumetric flow taken round one cycle is zero. The mass flow rate, however, is proportional to the mean density, and this is higher when the piston is closer to cylinder head than when it is closer to compressor body side. Thus there will be a net mass flow from the working side to body volumes across the clearance seal, which will decrease the mean working pressure and increase the body pressure. Eventually an equilibrium point will be reached when this effect is counterbalanced by the pressure differential in the opposite direction.

If there is a difference between the mean working pressure $P_{1,\mathrm{m}}$ and the body pressure $P_\mathrm{b}$ there will be a net axial force which is only counteracted by the mechanical springs, and this will result in a shift of the mean position of the piston. Note that if the piston is not oscillating about the 'mechanical zero' of the springs, there will be a reduction in the useful stroke of the compressor. If there is no control strategy and if the piston DC offset is large, then the piston may hit the cylinder head.

Bradshaw **[10]** also reported piston drift from its datum position during operation and the linear compressor model showed that the drift decreased with the dead (clearance) volume and increased with the stroke-diameter ratio of the piston. However, the piston drift was not controlled in this work **[10]**. Spoor and Corey **[13]** derived simple expressions for the time-average mass flux through a clearance seal and predicted the piston DC offset by assuming that the piston was acting against enclosed volumes on either side. Zou et al. **[14]** employed a formula (from Choe and Kim **[15]**) to calculate the DC offset and validated this method by experiments.

The DC offset can be countered in different ways, some of which have been adopted in recent studies:

(1) Increasing the mechanical spring stiffness relative to the gas spring stiffness. Van der Walt and Unger **[16]** used a mechanical spring of stiffness 70 N/mm for a piston of 0.7 kg in the Sunpower design of linear compressor, so as to have a DC offset of 0.12 mm at a stroke of 7.3 mm. They found that to hold the piston at its middle position, extremely stiff springs were needed. This method was subsequently implemented in the LG linear compressor after they licensed the Sunpower linear compressor technology. Eight cylindrical coil springs with a total stiffness of 118 N/mm were used in the commercial LG linear compressor, which dominated the system resonance reported by Bradshaw **[10]**. However, by using this approach, when the linear compressor is developed for applications with high gas forces such as air conditioning compressors, the quantities of the mechanical springs and the outline dimension of the linear compressor will be much larger, as pointed out by Zou et al. **[14]**. On the other hand, more springs with a larger dimension will not be a good option when developing a linear compressor for incorporation into a miniature VCR system for electronics cooling.

(2) Superimposing a DC bias on the AC drive voltage. Using this approach, Young and Chang **[17]** developed a device and method for controlling the piston position by detecting a phase difference between the current and stroke waveforms so that the compression clearance becomes a minimum during operation of the linear compressor. However, this approach will increase the resistive (Ohmic) loss.

(3) Adding mechanical and/or electromagnetic 'stops' to limit the motion of the piston. This will enable a safe operation of the linear compressor, preventing the piston from hitting the cylinder head.

(4) Control of the pressure in the compressor body, which typically requires a small bleed flow of gas from the compressor body to the suction side.

For the previous tests using nitrogen and R134a **[11, 12]** with the moving magnet linear compressor, the DC offset was manually maintained at a negligible level by adjusting the bleed flow valve to reduce the compressor body pressure. However, it was found that the DC offset varied more sensitively during the R134a tests because of the many interdependencies. Therefore, an automatic closed-loop control system for the DC offset is needed to achieve a fast response, appropriate stability and accuracy regardless of the operating condition of the linear compressor. In this work, a PID (Proportional-Integral-Derivative) control system for the DC offset is presented by using a bleed flow controlled by a solenoid operated valve. Fig. 3 shows the schematic of the linear compressor test loop, consisting of a main flow (in bold) and a bleed flow. The bleed flow loop connects the compressor body side to the suction line of the main flow loop. By using a solenoid valve, the bleed flow rate was controlled to vary the body pressure ($P_b$ in Equation 1) so that the piston DC offset can be controlled. The piston DC offset depends on the pressure difference between the body pressure and cylinder pressure according to Equation 1.

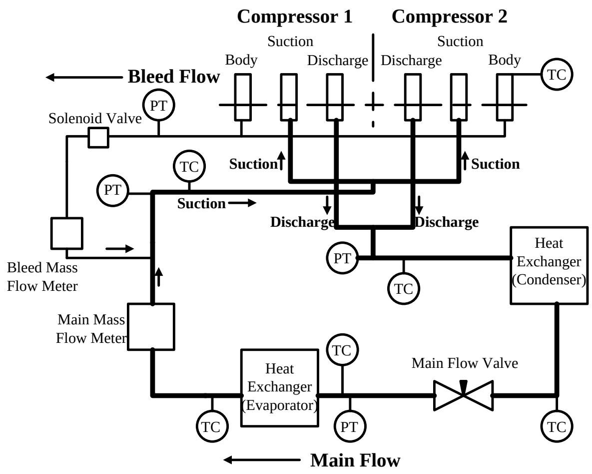



**Fig. 3** Test loop of the linear compressor consisting of the main flow loop (bold lines) and the bleed flow (PT: Pressure Transducer, TC: Thermocouple)

A fixed DC offset at zero indicates that the piston oscillates at its datum position (dash dotted line in Fig. 2) during operation. In contrast, a fixed clearance operation requires the piston DC offset to be varying so that the axial clearance volume can be kept constant. To deliver same amount of mass flow, the fixed clearance operation needs a smaller stroke than the fixed DC offset operation due to a higher volumetric efficiency. However, the influence of the fixed clearance on the motor efficiency and overall performance of the linear compressor is unknown. Using a DC current offset to control the piston position increases the Ohmic loss, while on the other hand using a bleed flow increases the compressor pumping work. This work experimentally compares two modes of operating the linear compressor, namely fixed DC offset operation and fixed clearance operation, using a bleed flow. Using a DC current bias to control the piston DC offset was compared with the bleed flow.

## 2. DC Offset Control using PID/PWM in Linear Compressor

In the linear compressor system, the process variable of the PID control is the DC offset calculated by using the piston displacement signal from a linear variable differential transformer (LVDT) displacement transducer. The set point for the DC offset is the desired value of 0 mm which means that the piston oscillates about the ‘mechanical zero’ of the springs. A PID controller determines the duty cycle of the pulse applied to the solenoid valve, which drives the DC offset towards the set point by changing the bleed flow rate to reduce the body pressure. The digital pulse train that makes up the PWM signal has a fixed frequency and varies the pulse width to alter the average flow rate through the solenoid valve. The block diagram of the PID controller for the DC offset control is illustrated in Fig. 4.

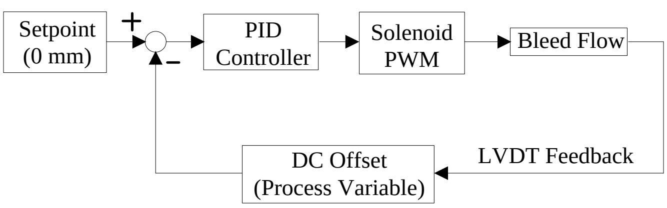



**Fig. 4** Block diagram of the PID controller for the DC offset control in the linear compressor

When the piston stroke increases, the DC offset rises pushing the piston mean position towards the cylinder head as the body pressure increases because of the gas leakage across the seal. A series of pulses, with a duty cycle being determined by the PID controller, is generated to open the solenoid valve to obtain an appropriate bleed flow so as to reduce the body pressure.

An example of the DC offset control with a set point of 0 using nitrogen as working fluid is also shown in Fig. 5, when the linear compressor was operated at a stroke of 10 mm and had an initial DC offset of -1.0 mm (note that negative displacement means towards the cylinder head from the datum position). A series of pulses with a frequency of 40 Hz and changing duty cycle were generated to reduce the DC offset to 0 mm after 50 seconds.

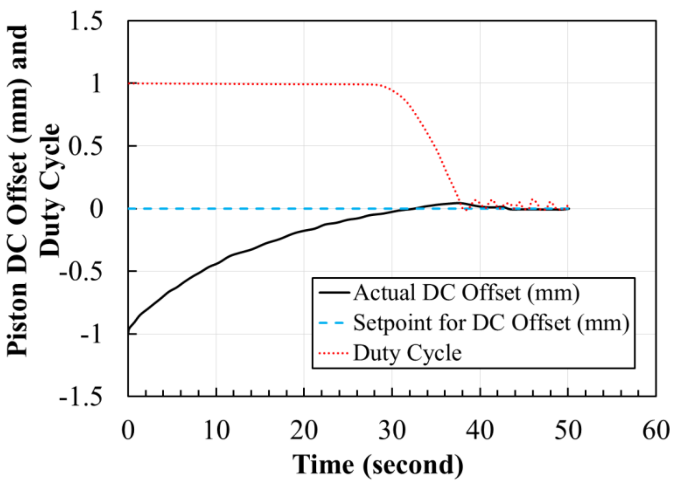



**Fig. 5** DC offset variation in response to the PID controller using solenoid valve (stroke of 10 mm)

A series of control system tests have been conducted by changing the pulse frequency from 1 to 10 Hz for a test condition of a pressure ratio of 4 and a stroke of 10 mm. The settling time remains same for different frequencies when the PID gains are fixed. The only difference for the different operating frequencies is the duty cycle at steady-state which varies from 4.4% at 10 Hz to 6.8% at 1 Hz. Therefore, the solenoid valve for the DC offset control can operate at a frequency of 1 Hz, and this should increases the durability.

## 3. Fixed Clearance Operation of the Oil-free Linear Compressor

### 3.1 Experimental Setup

The clearance is defined as the distance between the maximum displacement of the piston towards the cylinder head and the position for minimum cylinder volume. Since a negative displacement means towards the cylinder head from the datum position, the clearance can be expressed as

$$x_{\mathrm{cl}} = 7.57 - x = 7.57 - \frac{S}{2} + d \tag{2}$$

where $S$ is the stroke and $d$ is the DC offset from the piston displacement $x$. The mechanical zero position is 7.57 mm from the cylinder head.The previous PID control system for the DC offset control can be also used for the clearance control simply by changing the process variable from being the DC offset to being the clearance. As shown in Table 1, the maximum displacement of the piston towards the cylinder head was set at 7.0 mm which means the minimum clearance allowed for fixed clearance operation is 0.57 mm (Equation 2). So as to avoid dangerous excursions a clearance of 0.8 mm was chosen to be the set point and this provides comparison with a fixed zero DC offset.

The control system and associated instrumentation for the linear compressor experiments is shown in Fig. 6, which shows the power flow, together with the instruments to control and measure the compressor performance. The drive frequency was adjusted on the signal generator so that the resonance could be identified and applied to the linear motor, while the amplitude control was used to set the piston stroke (Peak-to-Peak value). A power amplifier finally amplified the analogue signal in order to drive the linear compressor. A capacitance box was employed to provide power factor correction, as this reduces the voltage required from the amplifier.

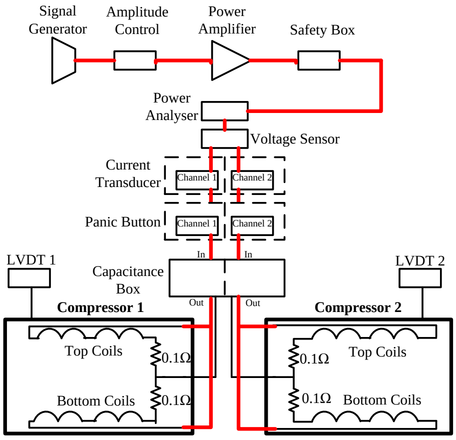



**Fig. 6** Compressor test and power supply instrumentation

### 3.2 Test Conditions

Table 2 lists the test conditions for the linear compressor operated with a fixed clearance of 0.8 mm and a fixed DC offset of 0 respectively using nitrogen. The linear compressor was initially filled to 7.0 bar (fill pressure) for both types of operation. Both tests were conducted at resonance. With comparable test conditions (pressure ratio, stroke, and suction pressure), the performance of the linear compressor can then be compared between the fixed clearance operation and the fixed DC offset operation. It is worth noting that strokes smaller than 10 mm were not considered for these tests as they cannot achieve a clearance of 0.8 mm, particularly for low pressure ratios (smaller than 2.5).

**Table 2** Experimental conditions for the linear compressor operated with a fixed clearance of 0.8 mm in comparison with a fixed DC offset of 0, respectively

| | Fixed Clearance | Fixed DC offset |
|---|---|---|
| Gas | nitrogen | |
| Pressure ratio | 2.5, 3.0, 3.5 | |
| Suction pressure (bar) | 4.4, 3.9, 3.5 | |
| Stroke (mm) | 8, 9, 10, 11, 12, 13, 13.5 | |
| Fill pressure (bar) | 7.0 | |
| Clearance (mm) | 0.8 | varying |
| DC offset (mm) | varying | 0 |
| Mass flow rate (g $s^{-1}$) | 0.36 – 0.96 | 0.18 – 1.90 |
| HDAQ sampling rate (kHz) | 5 | |
| HDAQ sampling number | 5000 | |
| LDAQ sampling rate (kHz) | 2 | |
| PWM frequency (Hz) | 40 | |
| Power factor | > 0.9 | |

## 4. Results and Discussion

### 4.1 Mass Flow

Fig. 7 shows the clearance variation with the stroke for operation with a fixed clearance of 0.8 mm and a fixed DC offset of 0 at different pressure ratios for one compressor half. As the two compressor halves shared bleed, suction and discharge lines and electrical connections, the PID control system for the clearance control can only be based on the displacement signal from one compressor. The fixed clearance operation is very accurately controlled at 0.8 mm for both compressors, with an average error of 1%. As expected, the clearance for the fixed DC offset operation decreases linearly with the increasing stroke. With negligible DC offset, the clearance approaches 0.8 mm when the stroke rises to about 13 mm. When the stroke is 10 mm, a DC offset of 1.8 mm is required to achieve a clearance of 0.8 mm.

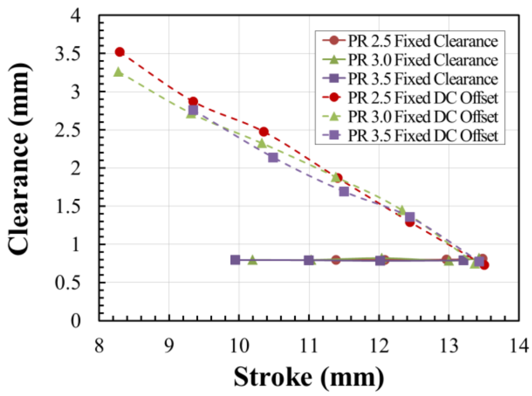



**Fig. 7** Actual clearance plotted against stroke for operation with a fixed clearance of 0.8 mm (solid lines) and fixed DC offset of 0 (broken lines) at different pressure ratios (PR: Pressure Ratio)

Fig. 8 plots the bleed mass flow rate as a function of the stroke for the two modes of the linear compressor. For the fixed DC offset operation, the bleed flow rate increases with an increasing stroke due to a higher net mass flow from the cylinder volume to the compressor body volume across the clearance seal. A maximum bleed flow of 0.07 g $s^{-1}$ is required for a stroke of 13 mm and a pressure ratio of 3.5. The bleed flow rate for the fixed clearance operation also increases with the stroke but for a smaller stroke it turns out to be lower than the fixed DC offset operation.

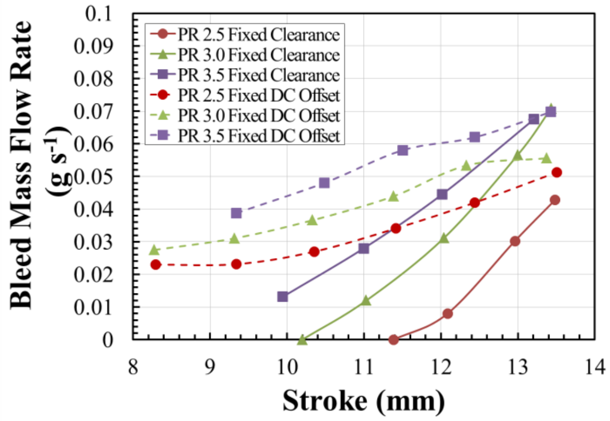



**Fig. 8** Bleed mass flow rate against stroke for operations with fixed clearance of 0.8 mm (solid lines) and fixed DC offset of 0 (broken lines) at different pressure ratios

### 4.2 Power Input

The input-RMS current plotted against the mass flow rate at each pressure ratio is shown in Fig. 9, for the two modes of the linear compressor operation. The current increases very linearly with an increasing mass flow rate regardless of the type of operation. The fixed clearance operation requires a higher current than the fixed DC offset operation. The main reason for this is that when the linear compressor operates at a fixed small clearance, the motor force constant becomes smaller (as the piston does not oscillate about the 'mechanical zero' of the springs), which means that to achieve a similar shaft power to compress the gas, more current is required for the fixed clearance operation, and the Ohmic losses will increase. Also, the magnetic field saturates at a lower current when the piston position is more distant from the datum position. Furthermore, the gas leakage across the clearance seal for the fixed clearance operation is higher than for the fixed DC offset operation. To compress the same amount of gas, more current (power) is needed.

Fig. 9 also shows the power input for comparison between the two types of operation. The power input varies linearly with the mass flow rate for the fixed clearance operation, similar to the fixed DC offset operation. However, the gradient appears to be constant with different pressure ratios for the fixed clearance operation, while it increases noticeably with an increasing pressure ratio for the fixed DC offset operation.

The increases in the current and power input required are not desirable as the motor efficiency will be reduced when the linear compressor operates with a fixed small clearance. It is worth mentioning that these effects are a consequence of the particular design of Oxford-type moving magnet motor. Other motor designs may have different characteristics, and these increases in power may be less (or even insignificant). However, this work provides a framework for such investigations regardless of the motor design.

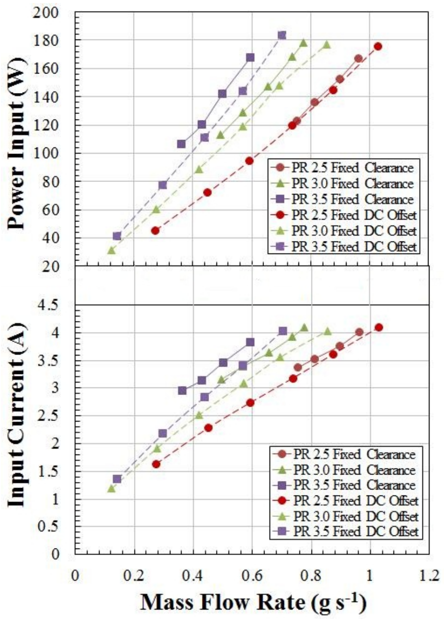



**Fig. 9** Power input and current (RMS) as a function of mass flow rate for operation with a fixed clearance of 0.8 mm (solid lines) and a fixed DC offset of 0 (broken lines) at different pressure ratios

### 4.3 Motor and Adiabatic Efficiencies

The electrical efficiency (or motor efficiency) of the linear compressor is given as

$$\eta_{\mathrm{e}} = \frac{\dot{W}_{\mathrm{shaft}}}{P_{\mathrm{in}}} \tag{3}$$

where $P_{\mathrm{in}}$ is the input power into the compressor.

The shaft power $\dot{W}_{\mathrm{shaft}}$ is the experimentally determined power required to run a compressor and was calculated from the measured current and displacement, ignoring all the frictional losses of the compressor. Therefore, the shaft power can be written as

$$\dot{W}_{\mathrm{shaft}} = \frac{1}{T}\int_0^T F_{\mathrm{motor}} \cdot \dot{x}\mathrm{d}t \tag{4}$$

where $F_{\mathrm{motor}}$ is the motor force and $\dot{x}$ is the velocity of the shaft which is the derivative of the armature position $x$, and $T$ is the period of one oscillation.

$$\eta_{\mathrm{adb}} = \frac{\dot{W}_{\mathrm{adb}}}{P_{\mathrm{in}}} \tag{5}$$

The theoretical adiabatic power $\dot{W}_{\mathrm{adb}}$ can be rewritten as **[18]**

$$\dot{W}_{\mathrm{adb}} = \frac{\gamma}{\gamma-1}\dot{m}R_{\mathrm{g}}T_{\mathrm{suc}}\left[(PR)^{\frac{\gamma-1}{\gamma}} - 1\right] \tag{6}$$

where $\gamma$ is the adiabatic index (1.402 for nitrogen).

The motor efficiency and adiabatic efficiency are plotted in Fig. 10 for the two types of operation of the linear compressor. As expected, operation of a fixed clearance led to a generally lower motor efficiency than the fixed DC offset operation for a wide range of mass flow rate and pressure ratios, due to more power input being required (shown in Fig. 9). When the linear compressor operates with a fixed clearance of 0.8 mm, the motor efficiency appears to remain almost unchanged at 73%. In contrast, the fixed DC offset operation has an increasing motor efficiency with a decreasing stroke, due to a lower Ohmic loss. Within a mass flow rate range of 0.2-1.0 g $s^{-1}$, the maximum motor efficiency is 82% for the fixed zero DC offset operation and 74% for the fixed clearance operation. For a mass flow rate ranging from 0.2 g $s^{-1}$ to 1.0 g $s^{-1}$, the adiabatic efficiency for the fixed clearance operation is about 4% lower than the fixed DC offset operation on average. Overall, the adiabatic efficiency decreases with an increasing pressure ratio.

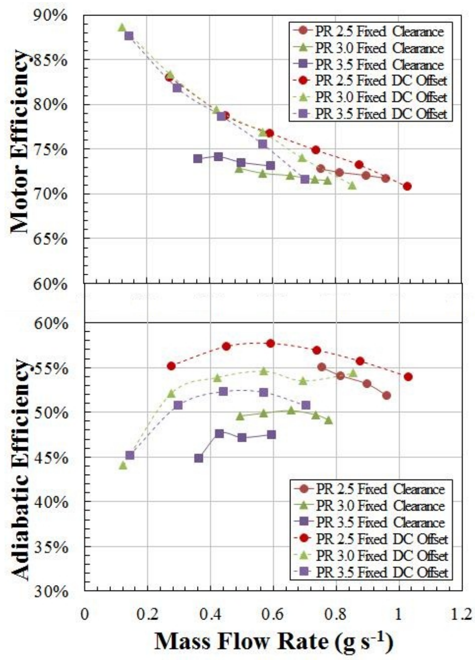



**Fig. 10** Motor efficiency and overall adiabatic efficiency against mass flow rate for operation with a fixed clearance of 0.8 mm (solid lines) and a fixed DC offset of 0 (broken lines) at different pressure ratios

### 4.4 Design Point

Table 3 gives the compressor performance data for both modes of operation at a specific test condition with a stroke of 10 mm and a pressure ratio of 3.0, which can be considered as the design point of this prototype. Compared to the fixed DC offset operation with a comparable condition (suction pressure, voltage, frequency, etc.), the fixed clearance operation leads to a 6% lower motor efficiency and a 4% lower adiabatic efficiency.

**Table 3** Performance comparison between the two types of operations of the linear compressor for a condition with a stroke of 10 mm and a pressure ratio of 3.0 using nitrogen

| | Fixed clearance (0.8 mm) | Fixed DC offset (0 mm) |
|---|---|---|
| Suction pressure (bar) | 3.8 | 3.9 |
| Mass flow rate (g $s^{-1}$) | 0.49 | 0.42 |
| Bleed flow rate (g $s^{-1}$) | 0 | 0.04 |
| Resonant frequency (Hz) | 37.5 | 38.5 |
| Power input (W) | 113 | 89 |
| Total current (A) | 3.2 | 2.5 |
| Voltage (V) | 36.0 | 35.4 |
| Adiabatic efficiency | 50% | 54% |

From the results above, it can be seen that the fixed clearance operation of the linear compressor led to stable performance for the different test conditions. However, the experimental comparison between the two types of operation indicates that the proposed fixed clearance operation for strokes smaller than 13 mm leads to a lower efficiency.

## 5. Using a Bleed Flow Compared with Superimposing a DC Current Bias

The DC offset in this work was controlled using a bleed flow. As has been discussed in Section 1.3, the piston DC offset can be countered in other ways as well. Increasing the mechanical spring stiffness does not seem to be desirable due to an increase in size and its effect on the resonant frequency. Superimposing a DC bias on the AC drive voltage will increase the Ohmic loss. It is worth comparing the extra Ohmic loss using a DC bias to the loss of compressor output when using a bleed flow. This section presents a model of the DC current required to keep the piston DC offset at zero so as to be compared to the method of using a bleed flow which has been adopted so far in this work.

### 5.1 Effective Radial Clearance

Although oil-free operation demonstrates a lot of benefits, the absence of oil will increase the seal leakage which causes a compressor work loss. The leakage across the radial clearance seal can be described as a steady flow through an annulus, as shown in Fig. 11. Note the detailed analysis has been given in Liang et al. **[19]**.

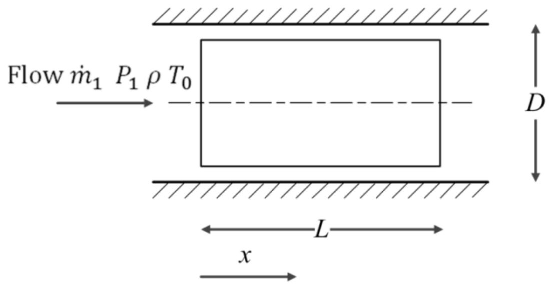



**Fig. 11** Flow through a clearance seal, with radial clearance *c* **[19]**

By assuming a perfect gas and a concentric piston in the cylinder, the leakage rate is

$$\dot{m}_1 = \frac{\pi D c^3}{12\mu L}\frac{(P_1 + P_\mathrm{b})}{2R_\mathrm{g}T_\mathrm{suc}}(P_1 - P_\mathrm{b}) \tag{7}$$

where $c$ is the radial clearance, and μ is the viscosity.

The measured bleed flow rate shown in Fig. 8 is the mean value of leakage rates in a cycle. Therefore, the equation above can be rewritten as

$$\dot{m}_\mathrm{bleed} = \frac{\sum \frac{\pi D c^3}{12\mu L}\frac{(P_{1,\mathrm{i}} + P_{\mathrm{b,i}})}{2R_\mathrm{g}T_\mathrm{suc}}(P_{1,\mathrm{i}} - P_{\mathrm{b,i}})}{T} \tag{8}$$

Where $T$ is the period of one oscillation, $P_{1,\mathrm{i}}$ is the instantaneous cylinder pressure which can be inferred from the piston dynamics (see next section) and $P_{\mathrm{b,i}}$ is the instantaneous body pressure from measurement.

Therefore, the effective radial clearance $c$ can be inferred from Equation 8. Fig. 12 shows the calculated effective radial clearance against strokes at different pressure ratios when the compressor was operated with a fixed zero offset. The effective clearance increases with increasing stroke, averaging at 11.8 µm. This might be due to a change in eccentricity of the piston, as a fully eccentric piston has 2.5 times the flow of a concentric piston. The previous steady-state flow test indicated an effective clearance of 12.5 µm, and this value is used here for a power loss calculation.

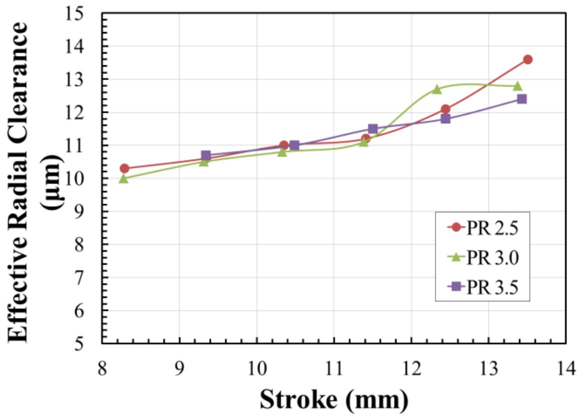



**Fig. 12** Effective radial clearances for tests with a fixed zero DC offset

If the pressure $P_1$ is assumed to be sinusoidal (which it is to a first approximation), then the loss can then be written as

$$\dot{W}_1 \approx \frac{\pi D c^3}{24\mu L}\left(P_1^2 - P_\mathrm{b}^2\right)\left[\frac{(P_1 - P_\mathrm{b})}{P_1}\right] \tag{9}$$

### 5.2 DC Current Model

Fig. 13 illustrates the forces acting on the piston assembly comprising the motor force $F_\mathrm{motor}$ (electromotive force), the mechanical (flexure) spring force $F_\mathrm{spring}$, the cylinder gas pressure force $F_\mathrm{cylinder}$, the body gas pressure (motor space) force $F_\mathrm{body}$, and the damping force (or drag force) due to friction between the cylinder wall and the piston $F_\mathrm{damping}$.

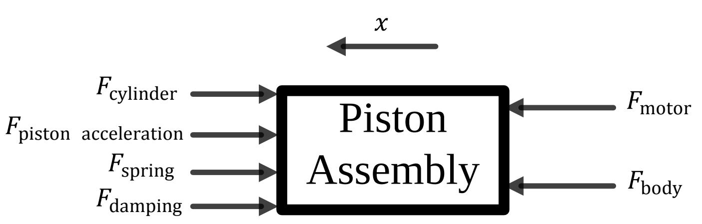



**Fig. 13** Free body diagram of the piston assembly in the linear compressor

The cylinder gas pressure force can be inferred from the piston dynamics as

$$F_\mathrm{cylinder} = F_\mathrm{motor} - F_\mathrm{piston\ acceleration} - F_\mathrm{spring} - F_\mathrm{damping} + F_\mathrm{body} \tag{10}$$

This can be expanded further to give

$$P_1 = (F_\mathrm{motor} - m\ddot{x} - k_\mathrm{m}x - \beta\dot{x})/A + P_\mathrm{b} \tag{11}$$

where the damping coefficient $\beta$ is readily calculated as the flow in the clearance seal can be treated as laminar and $k_\mathrm{m}$ is the mechanical spring stiffness.

Assuming similar piston dynamics when superimposing a DC bias on the AC drive voltage to keep the piston DC offset at zero, the extra motor force required will be the difference between the previously measured body pressure $P_\mathrm{b}$ (from fixed zero offset test using bleed flow) and the inferred body pressure $P_\mathrm{new,b}$ when there is no bleed flow. The extra motor force can be written as:

$$F_{\mathrm{extra,motor}} = (P_{\mathrm{new,b}} - P_{\mathrm{b}})A \tag{12}$$

The bleed flow from Equation 8 can be simplified as

$$\dot{m}_{\mathrm{bleed}} = \frac{kc^3 \sum({P_{1,\mathrm{i}}}^2 - {P_{\mathrm{new,b}}}^2)}{T} \tag{13}$$

When there is no net flow, the steady-state body pressure can be inferred as

$$P_{\mathrm{new,b}} = \frac{\sqrt{\sum {P_{1,\mathrm{i}}}^2}}{T} \tag{14}$$

Therefore, the motor force required using a DC bias is

$$F = F_{\mathrm{motor}} + F_{\mathrm{extra,motor}} \tag{15}$$

A 3-D map (Force-Current-Displacement) has been generated for the linear motor so that the current values can be found by interpolation. Therefore, the DC bias $I_{\mathrm{dc}}$ on the drive current can be calculated.

### 5.3 Power Loss using Bleed Flow and DC Current Bias

Using a bleed flow will cause a power loss across the clearance seal and a pumping loss. The power loss across the seal can be calculated as follow

$$\dot{W}_{\mathrm{s}} = \dot{W}_1 / \eta_{\mathrm{e}} \tag{16}$$

where $\dot{W}_1$is the seal loss calculated using Equation 9 and $\eta_{\mathrm{e}}$ is the motor efficiency calculated using Equation 3.

The pumping loss can be calculated as

$$\dot{W}_{\mathrm{p}} = \dot{V}_1 \cdot \Delta P \,/ \eta_{\mathrm{adb}} \tag{17}$$

where $\dot{V}_1$ is the volume rate of the bleed flow, $\Delta P$ is the pressure difference between discharge and suction lines, and $\eta_{\mathrm{adb}}$ is the adiabatic efficiency calculated using Equation 5.

The pumping loss can be expanded as

$$\dot{W}_{\mathrm{p}} = \dot{m}_1 \frac{R_{\mathrm{g}} T_{\mathrm{suc}} (P_{\mathrm{dis}} - P_{\mathrm{suc}})}{P_{\mathrm{suc}} \eta_{\mathrm{adb}}} \tag{18}$$

where $\dot{m}_1$ is the bleed flow calculated using Equation 7.

Additionally, a 12 V DC power supply was used to drive the solenoid valve. The resistance of the solenoid is 12 Ω. With a 10 Hz pulse, the steady-state duty cycle is 4.4%, meaning the solenoid is turned on for 4.4% of each cycle. The electrical power is calculated to be 0.53 W, which is negligible compared with the seal loss and pumping loss.

The total power loss for using a bleed flow to control the piston DC offset therefore is

$$\dot{W}_{\mathrm{bleed}} = \dot{W}_{\mathrm{s}} + \dot{W}_{\mathrm{p}} \tag{19}$$

Using a DC bias on the drive voltage (current) will induce a power loss across the clearance seal and an extra Ohmic loss. The power loss across the seal can be calculated using Equation 16 as well. The extra Ohmic (copper) loss can be calculated as

$$\dot{W}_{\mathrm{dc}} = {I_{\mathrm{dc}}}^{2}\mathrm{R} \tag{20}$$

where R is the total resistance of the coils which equals to 1.75 Ω.

The total loss for using a DC bias is

$$\dot{W}_{\mathrm{bias}} = \dot{W}_{\mathrm{s}} + \dot{W}_{\mathrm{dc}} \tag{21}$$

### 5.4 Results

The total power loss due to using two approaches (bleed flow and DC bias) to control the piston DC offset has been calculated for different radial clearance (7.5, 12.5 and 17.5 µm). Fig. 14 plots the total power loss against the stroke at a pressure ratio of 3.5. In general, the power loss using a DC bias increases faster than when using a bleed flow as stroke increases. Higher radial clearances lead to a higher power loss regardless of the approach used for controlling the piston DC offset. For a radial clearance of 17.5 µm, using a bleed flow will cause a much higher pumping loss leading to a much higher total loss compared to using a DC bias. However, the difference decreases as strokes increases (4 W at full stroke). For a 12.5 µm radial clearance, power losses for using two approaches are close to each other, particularly at higher strokes. When the radial clearance is 7.5 µm, the loss due to using a bleed flow hardly changes, averaging at 1.9 W while using a DC bias causes a linearly increasing power loss reaching 10 W at full stroke.

In linear compressors, the radial clearance between the piston and the cylinder should be made small enough to tolerate the resulting leakage. This indicates that a bleed flow is more desirable for being used to control the piston DC offset than a DC bias.

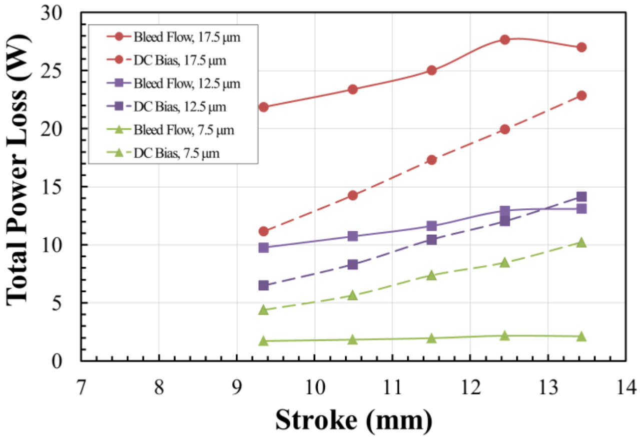



**Fig. 14** Total power loss using bleed flow compared with using a DC bias on the voltage to maintain a zero piston DC offset at a pressure ratio of 3.5

### 6. Conclusions

This paper summarises different methods of controlling the DC offset. This led to a PID control system being used for DC offset control; this comprises a PID controller in LabVIEW, a PWM controlled solenoid valve, and a LVDT displacement transducer. Operation of the linear compressor with a fixed clearance of 0.8 mm using nitrogen was then carried out for comparison with the fixed DC offset operation. The test conditions cover pressure ratios of 2.5, 3.0, and 3.5 and strokes of 9-14 mm.

The experimental results show that:

(1) At full load, the performance of two modes is of course very close to each other.
(2) As the stroke decreases, the fixed clearance operation requires a higher current and subsequent power input than the fixed DC offset operation, and the difference increases with a decreasing stroke, as a result of which the motor efficiency and adiabatic efficiency are reduced. This indicates that the fixed clearance operation is not desirable;
(3) In general, when the linear compressor operates with a fixed clearance of 0.8 mm, the motor efficiency and adiabatic efficiency have a very small change for a stroke ranging from 9 mm to 13 mm.
(4)

A comparison between using a DC bias on the drive voltage (current) (as opposed to using a bleed flow) will induce more power loss than using a bleed flow when the radial clearance is below 12.5 μm.

Therefore, it can be argued that the linear compressor in this work operates better with a controlled zero DC offset than with a controlled clearance volume. This provides evidence that the piston control in a refrigeration system with capacity control should be for a zero DC offset using a bleed flow.

## 7. Acknowledgements

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