

Clearance Seal Compressors with Linear Motor Drives

Part I: Background and System Analysis

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Abstract: Linear compressors with clearance seals and flexure bearings have been used for many years for space applications, and this paper describes the development of linear compressors at Oxford. A new design of moving magnet motor is introduced and the use of flexure springs, linear motors, valves and their integration are all discussed. A ring down test indicates that the ‘viscous’ damping in the linear compressor is acceptable (in terms of the decay ratio against the number of periods). The measured motor characteristics compared well with the Vector Fields Electromagnetic Finite Element model. With a revised structural design it is predicted that the motor efficiency would be $\sim 86\%$ for the rated power (200 W shaft power @ 50 Hz) enabling the linear compressor to have a high overall efficiency. The companion paper covers the experimental evaluation of the linear compressor.

Keywords: linear compressor, clearance seal, moving magnet motor, flexure springs, valves, magnetic forces, ring down test, motor characteristics, efficiency

1 BACKGROUND

Clearance seal compressors have been developed for both Stirling cycle coolers and pulse tube coolers, a very good review of these different approaches has already been published [1].

1.1 Early Development

In the 1970’s, there was a requirement to cool an instrument on a satellite being developed to investigate the Earth’s atmosphere. This led to an Oxford designed Stirling cycle cooler, consisting of a compressor, a cold head and heat exchangers (Fig. 1). However, with the ‘cold end’ of the cryocooler at 80 K, any oil would solidify and block the heat exchanger, so the cryocooler needed to be oil free. The solution to this was provided by Dr Gordon Davey, at the University of Oxford, who adapted a ‘Pressure Modulator’ developed by Oxford’s Atmospheric Physics Department [2]. The resulting cryocooler is shown in Fig. 1 and the key features were:

- (1) Clearance Seals between piston and cylinder. If the radial clearance between the piston and the cylinder is made small enough, the resulting leakage can be tolerated. The clearance needed is 10 – 20 μm , and this requires both the piston and cylinder to have good cylindricity and be very concentric with each other.
- (2) Spiral Disk Springs are used to maintain the alignment of the piston within the cylinder and these are typically photo-etched from thin sheet, as shown in Fig. 2. The spring arms, defined by the slots, act as cantilevers ‘built-in’ at both ends. Axially the springs are compliant allowing the piston to move freely up and down in the cylinder, but radially they are stiff, so that the piston remains concentric in the cylinder.

- (3) Linear Motion: A 'loudspeaker' type moving coil, permanent magnet motor is used to drive the compressors. With the motor, piston and springs all aligned on a common axis, there are negligible transverse forces during operation.
- (4) Gamma configuration Stirling cycle design.

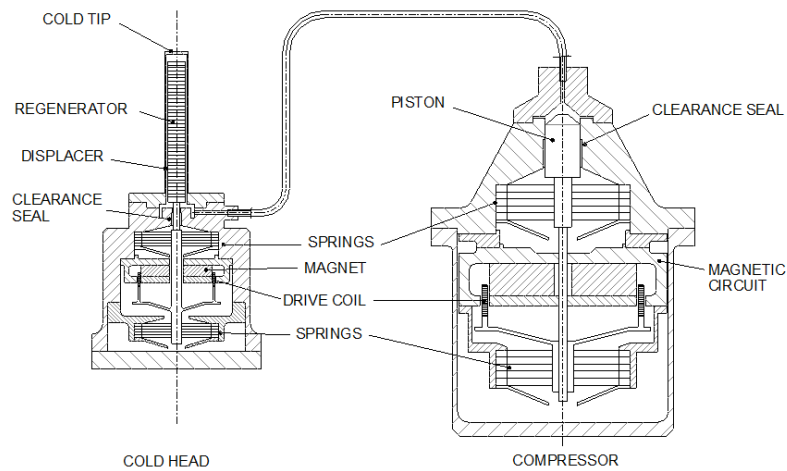


Fig. 1 Early Stirling cycle cryocooler for space



Fig. 2 A spiral disk spring (flexure)

1.2 Second Generation Clearance Seal Compressor

The first generation clearance seal machines were expensive to make and difficult to assemble, and there was a requirement for smaller, lower cost machines that would be suitable for non-space use (e.g. infrared cameras). To meet these requirements, an 'Integral' cryocooler was developed at Oxford (Fig. 3) in 1988 [3]. This is a single unit in which the displacer is integral with the compressor, and uses a moving cylinder and a fixed piston. There is only one motor; the displacer is driven by the gas forces and is designed with the resonant frequency and damping required to give the correct phase between the motor and the displacer. The second generation coolers are more robust and easier to assemble than the first generation coolers, with the cylinder (a short stiff tube) connecting the spring stacks, as opposed to a long slender shaft, with the piston cantilevered out at the end.

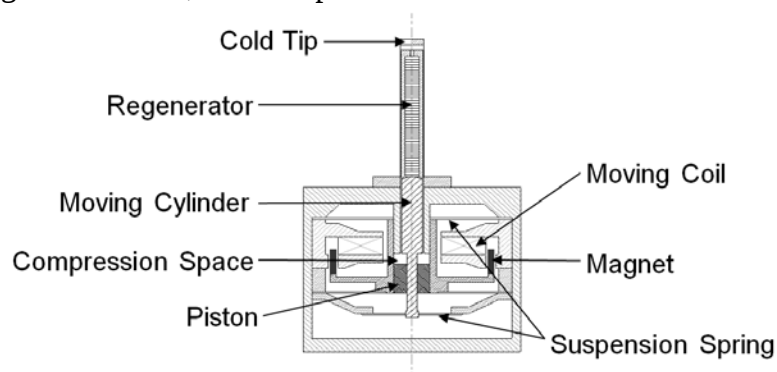


Fig. 3 ‘Integral’ clearance seal compressor embodied in a Stirling cycle cooler which has a single motor and pneumatically driven displacer

1.3 Third Generation Clearance Seal Compressor

A requirement for a compressor to drive a cold head, but with tight restrictions on the diameter and length of the compressor led to the design of a new moving coil motor, with improvements to the spiral disk springs. The original design developed into a back-to-back configuration with two identical compressors acting on a common cylinder space, and this was used as the basis of the High Efficiency Compressor [4]. This is incorporated into a pulse tube cryocooler system by Northrop Grumman Aerospace Systems (NGAS) as shown in Fig. 4. The pulse tube is similar to a gamma configuration Stirling engine, except a gas column acts as the displacer. A comparison of the specific cooling performance (kg/W) plotted against the cold-head temperature (K) shows that the NGAS coolers have the best specific performance in the range of 10-90 K [5].

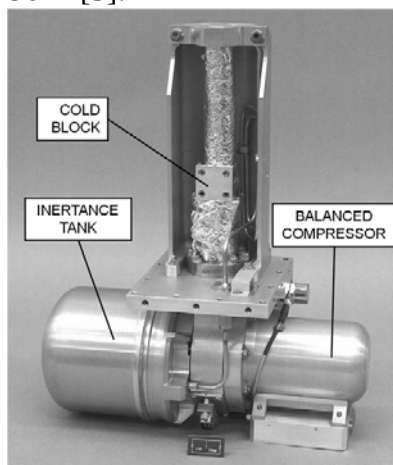


Fig. 4 HEC cryocooler with pulse tube cold head and balanced compressor (with acknowledgement to NGAS)

1.4 Valved Compressors

By adding valves, linear compressors can be used to create a ‘DC’ flow through a system, rather than being a ‘pressure wave generator’ required by Stirling and pulse tube systems. One application for this is in conventional vapour compression refrigeration, but in cryogenics valved compressors are also needed for Joule-Thomson (J-T) coolers [6].

2 MOVING COIL AND MOVING MAGNET LINEAR MOTORS

2.1 Moving Coil Compressor Design

All the compressors described so far use a moving coil similar to those used in loudspeakers. A static magnetic circuit produces a high radial magnetic field in an annular air gap. The drive coil is positioned in the air-gap and when it is energised with a current it develops an axial force. The application of an oscillating current to the motor causes the piston to reciprocate, as in a conventional crank driven machine, but without any side forces. The piston is engaged in a cylinder with a small radial clearance of about 10 microns. This radial gap is sufficient to avoid contact that would generate wear, but it is also small enough to keep leakage losses down to acceptable levels. The design has already been shown in Fig. 1.

Outside of space applications, cost has been the major obstacle to extending the use of this technology. This is partly due to the low production numbers, but there are also aspects of the compressor design that are not favourable for low cost mass production. The moving coil linear motor itself has significant drawbacks that are discussed next.

Returning to Fig. 1 it will be seen that there is a static magnetic circuit, and this produces a high magnetic field in the annular air gap containing the drive coil. For acceptable moving masses the magnetic flux in the gap needs to be ~ 1 Tesla, with a flux path length that might be 3 to 5 mm long. The air gap can be considered as a critical part of the motor because the gap contains a large amount of magnetic energy and the power of the motor depends on how well the air gap is used. The amount of magnet material required for moving coil motors is typically much higher than for moving magnet motors. Another aspect of a moving coil motor that results in complexity and added cost is the requirement for various components to transmit electrical current from the external power source to the moving drive coil [7].

Finally moving coil designs are not well suited to larger machines where moving mass and motor efficiency become more critical issues. For high efficiency (e.g. 90+ %), conductor losses need to be reduced. This can be achieved in moving magnet motors by increasing the size of the stationary stator coils. In a moving magnet motor the moving mass and hence resonance is unaffected, while in a moving coil motor the coil mass has to increase significantly. The resulting increase in moving mass is very disadvantageous to the design of a moving coil motor, making it more difficult to achieve resonance at high operating frequencies.

2.2 Recent Developments

Proprietary designs already exist that address some of these requirements:

- (1) Sunpower [8] has a moving magnet design which is compact and has evolved with Sunpower's gas bearing technology.
- (2) Clever Fellows (Q drive) [9] has a moving magnet design with multiple coil and magnet assemblies. It has evolved with Clever Fellows own flexure geometry. The motor geometry is not well suited to spiral spring designs due to the spring rotation.
- (3) Infinia [10] have a number of designs that can be either moving magnet or moving iron. These are used in conjunction with Insignia's own spiral spring designs.

A more recent development has been made by the Brazilian company Embraco, who have announced a new oil-free linear compressor, designed for R134a and R600a [11]. A patent published in 2005 shows a compressor that has significant similarities with the Sunpower design [12].

3 OXFORD MOVING MAGNET MOTOR

3.1 Magnetic Design

Since moving coil compressors are too expensive to be used in many applications, the need for low cost in materials and manufacture led to a novel low-cost moving magnet linear motor which is shown in Fig. 5. This moving magnet motor has transverse flux paths between the static part and the moving magnet assembly.

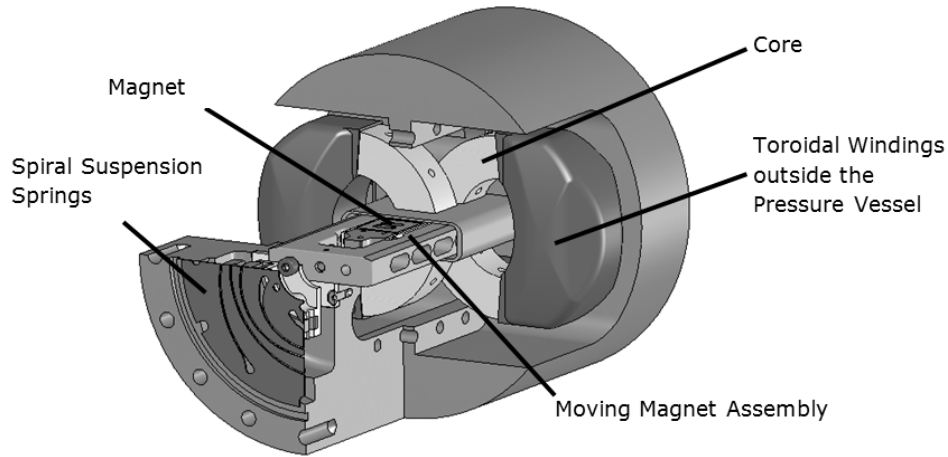


Fig. 5 Key elements in the design of the 100W Oxford moving magnet motor

The static part of the motor consists of a number of magnetic circuits and each circuit has a laminated core with a coil wound around it. The cores are slotted and the sides of the slot form pole pieces that define a rectangular air gap. The cores are positioned so that all the air gaps are aligned along the motor axis. The disposition of neighbouring cores alternates so that the coils do not obstruct each other.

The moving assembly consists of a number of rectangular magnets arranged in a line; they occupy the air gap created by the static assembly. The polarisation of the magnets (Fig. 6) alternates and is arranged so as to induce fields that circulate around the adjacent cores. The magnet assembly in particular is very simple and robust, but nonetheless, the magnet utilisation is good, the moving mass is low, and high efficiencies are readily attained.

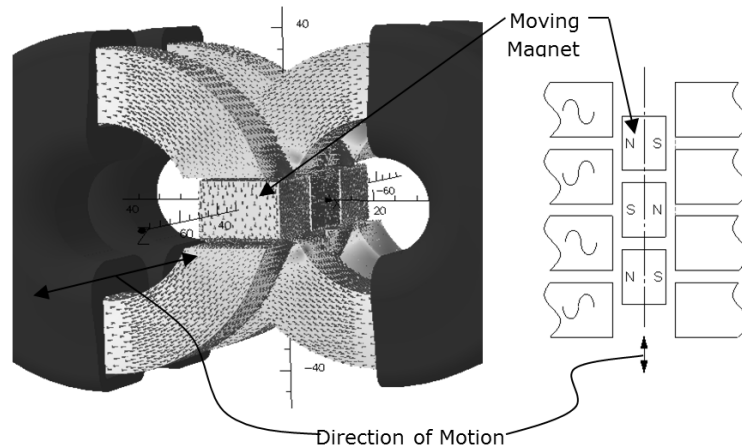


Fig. 6 Magnetic circuit and flux distribution for a design with 4 cores and 3 magnets

Axial movement of the magnet assembly has the effect of varying the magnetic flux through the cores, which induces voltages in the coils; the polarity being opposite for neighbouring cores. If an alternating current of appropriate polarity is applied to the coils then an alternating axial force is developed as desired.

The motor shown in Fig.5 and Fig.6 has three magnets and four cores but other combinations are possible as long as the number of cores is one greater than the number of magnets. The two basic components of this design are a slotted magnetic circuit and a rectangular magnet. The magnetic circuits can be made in much the same way as

conventional toroidal transformers – the core is strip-wound to reduce eddy currents and the coil is wound using a toroidal winding machine.

The integration of the motor with the linear suspension system is illustrated in Fig.5. The compressor has a one-piece compressor body to which all the main static components are attached. As can be seen, in this design the coil assemblies are outside the pressure vessel as the magnetic circuits are attached externally to the compressor body in appropriate cut-outs.

The pressure containment is achieved by enclosing the moving magnet assembly in a rectangular tube, which is positioned within the air gap and sealed around both ends into the compressor body. The pressure forces acting on the tube can be adequately supported by the surrounding components, that is, the magnetic circuits and the compressor body.

The two spring assemblies are attached to the ends of the compressor body. The inner connection between the two spring assemblies is made by a beam contained in the rectangular tube, which also houses the magnets. The piston is cantilevered from the moving magnet assembly and the cylinder is fixed to the end of the compressor body. The first application of this design is in a valved compressor, in which the cylinder head has conventional reed valves mounted on a valve plate.

There are two identical compressor ‘halves’ mounted in line, and the two balanced halves operate in opposite directions from one another to virtually eliminate vibration. The two compressor halves share suction and discharge lines and electrical connections; in the subsequent discussion they will be referred to as Compressor 1 and Compressor 2.

3.2 Flexure Spring Stiffness

The main disadvantage of moving magnet motors compared with moving coil motors is the presence of off-axis reluctance forces (which are present even when there is no energisation of the coils) and these have the capacity to upset the linearity of the movement. The off-axis motor forces are resisted by the radial stiffness of the suspension system, and their relative magnitude is important in determining the viability of a compressor design.

A Finite Element (FE) magnetic circuit analysis was used to evaluate the principal off-axis force, which is the side force in a direction perpendicular to the pole piece faces. When the magnet assembly is aligned in the centre of the air gap the side force is zero, as would be expected from symmetry. As the magnet assembly is moved off centre, it is attracted to the nearest pole piece, with the force increasing linearly in the direction of the movement away from centre. From the FE model, this force was calculated to be 200 kN/m (0.2 N/ μ m), and to be independent of the axial position of the magnet assembly.

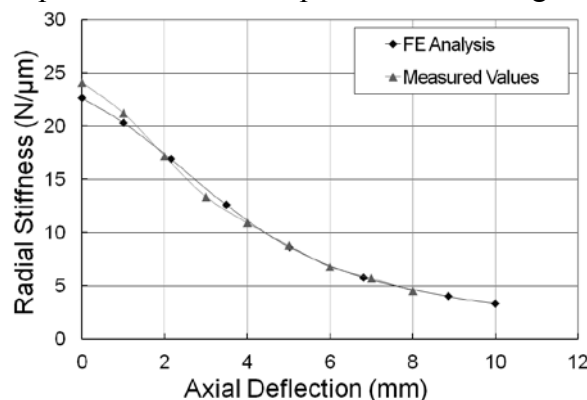


Fig. 7 Radial stiffness of suspension (the measured values have been corrected for the test rig compliance)

Using Lusas FE software, the radial stiffness of the suspension system was estimated by analysing the behaviour of a spiral spring. To verify the calculated values, some simple

measurements were made on a test assembly using a force meter and a ‘Mu-Checker’ displacement probe; the data points were corrected to account for other compliant parts of the system. The results in Fig. 7 (two sets of six springs) show the radial stiffness decreases as the axial deflection increases. So to keep the radial deflection below $1\text{ }\mu\text{m}$, the radial force should be less than 5 N. If the magnetic forces increase by $0.2\text{ N}/\mu\text{m}$, then this implies that provided that the magnet assembly is no more than $25\text{ }\mu\text{m}$ from the centre of the air gap, the alignment will be maintained.

The axial stiffness of the suspension system was also analysed. Fig. 8 shows the axial stiffness calculated using experimental data in comparison with results from Lusas FE model. It is seen that the two methods give very similar results.

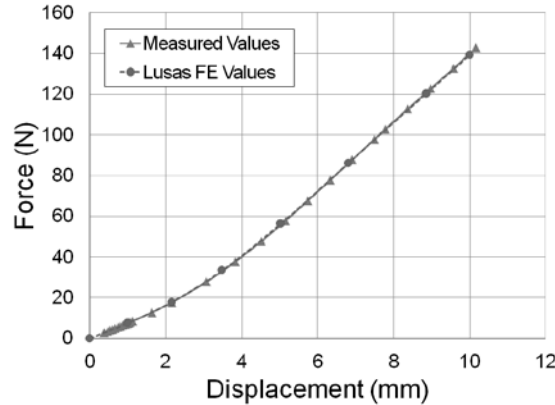


Fig. 8 Axial force/displacement for the suspension system

The material selected for the suspension spring exhibits a “fatigue limit”. Typically it is martensitic stainless steel (such as Sandvik 7C27Mo2). The flexure springs are designed so that the maximum stress is safely below the fatigue limit. A typical batch acceptance test is 10^7 cycles at 25% over-stroke and predicted spring arm reliability is typically higher than 0.9998. The S-N curve is shown in Fig. 9.

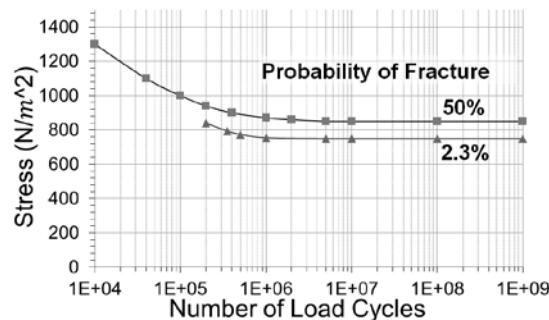


Fig. 9 S-N curve of Sandvik 7C27Mo2 for the suspension spring

3.3 Motor Characteristics

The moving magnet has been modelled using the Vector Fields (VF) FE Magnetic Circuit Analysis software to evaluate its characteristics. The critical characteristic for its operation as a motor is the variation of the force constant (i.e. force/current) with both current and axial position. To verify these values from the VF model, measurements were taken to observe the force varying with current at certain armature positions from -6 mm to 6 mm (Note that -6 mm is towards the Top Dead Centre from zero position where there is no shaft force acting). The force characteristic for Compressor 1 at two axial positions is shown in Fig.10. With these characteristics it is possible to calculate the shaft power for the measured current input and armature motion.

From Fig. 10 it can be seen that the magnetic field reached saturation when the current increased to about 1.2 A for -4 mm and 1.0 A for -6 mm armature displacements. Note that the force is not zero when the current is zero. This is because of the reluctance force which tends to move the magnet towards regions of higher flux so that there is always an attractive force pulling the magnet. The reluctance increases with the displacement.

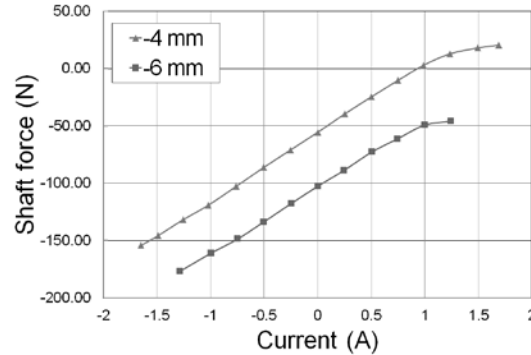


Fig. 10 Shaft force varying with current at armature displacements of -4 mm and -6 mm (the slope gives the force constant, see Fig. 11)

To avoid saturation, the force constant (the slope of the line in Fig. 10) was calculated using the measurement points below a current of 1 A. Similar calculations were done for both compressor halves at the other displacements. A comparison of the force constants between the VF FE model and the experimental calibration is shown in Fig. 11. The measurement results of the two compressor halves (1 and 2) are very close and agree well with the model, which confirms the accuracy of the modelling. However, in order to identify whether the current reached the saturation point of the cores, it is better to use a 3D map (Force-Current-Displacement) with interpolation to calculate the shaft power than to use the characteristics curve of Fig. 11.

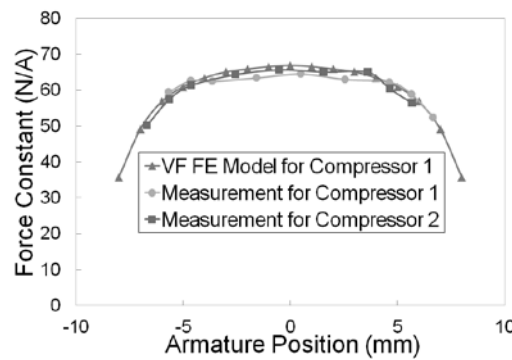


Fig. 11 Motor force constant variation with armature position

3.4 Imbalance of Current in Motor Coils

In the current design of moving magnet motor, there are 4 coils in each half, arranged as 2 coils in series that are connected in parallel to the power supply. It should not be assumed that the currents in the two halves of the compressor are equal as the resistance and inductance of each coil might not be the same due to manufacturing tolerances and differences in the operating conditions of the two halves. In particular, if there are differences in the leakage flows in the two compressor halves, then the mean position (DC offset) will be different in each compressor half. Therefore, the current in the two halves needs to be measured separately. Most importantly, precise measurement of current is required if an accurate value of force is to be determined from the Force-Current-Displacement Map.

Since, in each compressor half, the coils are not all in series, (the ‘top’ two coils are in series, and the ‘bottom’ two coils are in series) it is possible that there is a current imbalance between each pair of coils in parallel and that this will affect the force. Matched 0.1Ω resistors were installed in series with the coils so that the current could be measured separately. Static tests were performed with a deliberate imbalance in current between the top and bottom coils, but with a constant total current.

A total current of 1 A, 2 A and 3 A was applied, with the imbalance increased in steps of 0.2 A up to a difference of 1 A; the current at the rated operating point is 2 A rms. The results are given in Fig. 12. For 1 A, 2 A and 3 A for one compressor half, the maximum discrepancy of the force constant is 0.01%, 0.1% and 1.2% which could be ignored. The force constant is reduced for 2A and 3A of total input current because saturation is reached at about 1.3 A. It can be concluded that any discrepancy in current between the top and bottom coils will not make much difference to the motor force. Furthermore, when the compressor was operated there was no significant difference between the current flowing to the top and bottom coils. It is noted that this imbalance issue does not need to arise in a commercial machine as the coils could be wound so that they would all be connected in series – this is just a matter of choosing the appropriate wire diameter and number of turns.

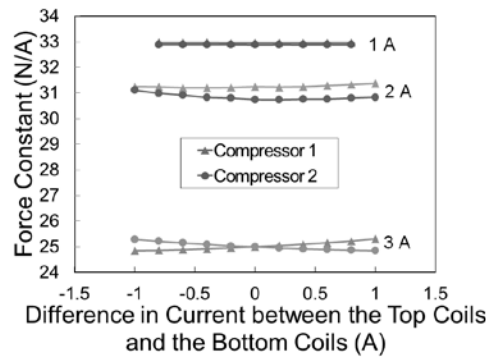


Fig. 12 The impact of current imbalance between the top coils and bottom coils on the Force Constant for the compressor motor

3.5 Valve Design

Compressors have valves at the inlet and the outlet, and the only difference between the suction valve and discharge valve is usually their size. The suction valve is larger because the low-pressure suction valve has a larger volume flow rate than the high-pressure discharge valve. The two valves for this oil-free compressor are mounted on a common valve-plate and face the top of the piston. Both valves are ‘reed’ valves photo-etched from martensitic stainless steel sheet. The relevant dimension for both discharge and suction valves are given in Table 1 together with the resonant frequency and stiffness calculated by using FEM.

Table 1 Key dimensions for both the discharge and suction reed valves

	Discharge valve	Suction valve
Thickness (mm)	0.2	0.3
Length (clamp line to centre of seat) (mm)	14	20
Length (clamp line to tip) (mm)	18	24
Width of arm (mm)	4	not applicable
Max valve lift (mm)	1.05	1
Resonant frequency (Hz)	405	246
Stiffness (N/mm)	0.561	0.361
Orifice diameter (mm)	5	5

3.6 Ring Down Test

The ring down test is used to measure the decay response of the compressor, and to assess if there is any evidence of friction. The compressor is evacuated, and the ring down test is completed by recording the decay of the piston displacement in response to a step change in coil current. Fig. 13 shows the logarithmic decrement in displacement for the piston in compressor 1. The displacement amplitude decays exponentially. The decay is predominantly due to eddy current losses with a component of magnetic hysteresis (losses in the spring are negligible, and there is no ‘windage’ loss as the compressor has been evacuated). For the eddy current loss, the damping force is proportional to the velocity and hence a viscous damping behaviour is expected. If the ring down test was performed on an unbalanced compressor then compressor movement would also dissipate energy and interpretation of the decay would be less clear.

Furthermore, Fig. 13 gives the decay ratio (the ratio of the peak amplitude at successive oscillations) for successive oscillations. It is easily seen that the ratio remains constant (between 0.92 and 0.945) for more than 30 periods. Therefore, there is no evidence of Coulombic friction which would come from rubbing surfaces and produce a linear variation of decay ratio.

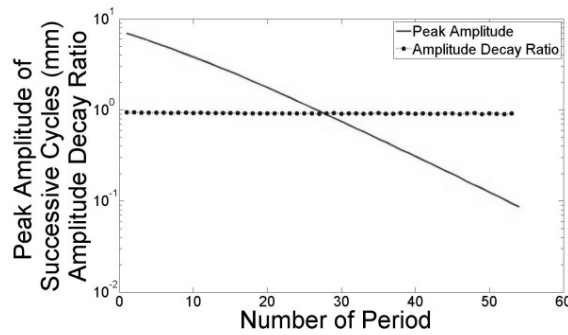


Fig. 13 Displacement amplitude decay for compressor 1

4 PARASITIC LOSSES

4.1 Seal Leakage

Although oil-free operation demonstrates a lot of benefits, the absence of oil will increase the seal leakage which causes a power loss. The leakage can be described as a steady flow through an annulus in Fig. 14.

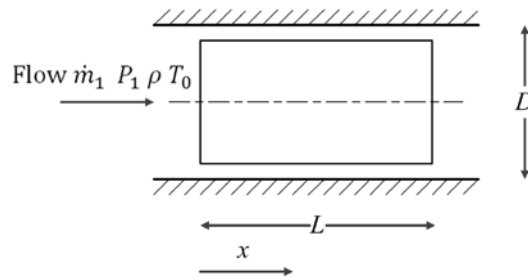


Fig. 14 Flow through clearance seal, with radial clearance c

A perfect gas enters at pressure P_1 , density ρ , gas constant R and temperature T_0 . The mass flow rate is $\dot{m}_1$ and is assumed constant.

$$\dot{m}_1 = \rho V \quad (5)$$

Note that the pressure (and velocity/volumetric flow rate for a gas of viscosity μ) does not change linearly with clearance c . The equation can be rewritten for a seal of length L as [13]

$$\dot{m}_1 = \frac{\pi D c^3}{12 \mu L} \frac{(P_1 + P_2)}{2 R T_0} (P_1 - P_2) \quad (6)$$

where the second term is the mean density of the gas (mean of the inlet and outlet densities), and the last term is the pressure difference across the seal.

The power loss due to leakage can be found by integrating

$$\dot{W}_1 = \int_0^L V dP \quad (7)$$

The power loss then becomes

$$\dot{W}_1 \approx \frac{\pi D c^3}{24 \mu L} (P_1^2 - P_2^2) \left[\frac{(P_1 - P_2)}{P_1} \right] \quad (8)$$

Typically the seal has a fluctuating pressure on one side, and a nominally constant pressure on the other. To a first approximation, the fluctuating pressure can be assumed to be a sine wave of frequency ω , amplitude P_{1a} and mean P_{1m}

$$P_1 = P_{1m} + P_{1a} \sin(\omega t) \quad (9)$$

Therefore, the power loss becomes

$$\dot{W}_1 \approx \frac{\pi f D c^3}{24 \mu L} \oint \left[(P_{1m} + P_{1a} \sin(\omega t))^2 - P_2^2 \right] \left[1 - \frac{P_2}{P_{1m} + P_{1a} \sin(\omega t)} \right] dt \quad (10)$$

As the mean cylinder pressure P_{1m} is very close to body pressure P_2 , the power loss due to seal leakage then can be simplified as

$$\dot{W}_1 \approx \frac{\pi f D c^3}{24 \mu L} \oint \frac{[P_{1a}^2 \sin(\omega t)(2P_2 + P_{1a} \sin(\omega t)) - P_2^2]}{P_2 + P_{1a} \sin(\omega t)} dt \quad (11)$$

4.2 DC Offset

The problem of DC offset is caused by the differential pressure generated across a clearance seal which has a constant pressure on one side of it and a fluctuating pressure on the other.

On the body side of the seal, the pressure is essentially constant, whereas the working side of the seal experiences a fluctuating pressure. The pressure differential is given by

$$\Delta P_1 = P_{1m} + P_{1a} \sin(\omega t) - P_0 \quad (11)$$

where P_0 is the pressure in the body.

A detailed analysis of the DC Offset could be made with a computer model based on the Schmidt analysis [14] which eliminates errors due to the variations in the length of the seal and the non-sinusoidal pressure waveform, but it cannot model variations in the effective clearance of the seal, as these are significant but cannot easily be measured.

If there is a difference between the mean cycle pressure P_{1m} and the body pressure P_0 there will be a net axial force which is counteracted by the mechanical springs, and will result in a shift of the mean DC position of the piston. Note that if the piston is not oscillating about the 'mechanical zero' of the springs, there will probably be a reduction in the useful stroke of the compressor.

4.3 Copper Loss

The Copper Losses result from resistive heating and so are also referred to as the " I squared R loss":

$$P_{Cop} = I_{rms}^2 R \quad (12)$$

where P_{Cop} is the copper loss, I_{rms} is the RMS value of input current and R is the resistance of the coils.

4.4 Core Loss

Some of the power that would ideally be transferred through the compressor is lost in the core, resulting in heat and sometimes noise. The principal losses are a Hysteresis Loss and an Eddy Current Loss. Manufacturers of core materials generally do not give a breakdown of the loss into these components. Instead total core loss is presented as a function of peak flux density and frequency.

The core material for the compressor is Unisil 27M4 (New designation: M089-27N). The specific total loss against peak magnetic polarization is given in [15]. This typical curve is obtained with thickness of 0.27 mm, density of 7.65 g/cc, and a frequency of 50 Hz. The specific loss for the compressor peak flux of 1.6 T is ~ 1 W/kg. The total mass of the cores for each compressor half is 1.5 kg, hence total core loss for the compressor pair is ~ 3 W.

4.5 Additional Eddy Current Losses

There were two other eddy current losses occurring in the compressor in addition to the eddy current losses occurring in the motor cores.

The pressure containment tube traverses the flux path between the moving magnets and the cores. As the magnets move, the flux linkage in the tube material varies and eddy currents are induced. These losses are the dominant losses when the ring down test is performed. These losses were estimated using a simple model giving a value of ~ 2 W for the 25 Hz operation in vacuum. As the loss is proportional to the square of the frequency this loss is estimated to be ~ 8 W for the design operation of 200 W at 50 Hz (i.e. 4%). Although not a large loss this is big enough to warrant further work to reduce it. The main possibilities are reducing the wall thickness and/or using a lower conductivity material – e.g. titanium alloy.

The other eddy current loss that was identified is due to stray AC fields from the stator coils inducing eddy currents in the aluminium structure. This loss was a consequence of the relatively massive build of the prototype and the use of a single aluminium block to aid construction. The loss was diagnosed by measuring the power loss in the motor when the armature was prevented from moving. It was easy to demonstrate that AC fields were coupling with the aluminium body and that when shielding was used the loss was significantly reduced. It was not possible to evaluate this loss accurately for the operating conditions but it was estimated that at the design point it could produce a loss of $\sim 10 - 15$ W. This loss could be reduced by changes to the design compressor structure and also the choice of materials.

4.6 Prediction of Motor Efficiency

According to the loss analysis above, for the design point of a 200 W power input (100W per compressor “half”) at 50 Hz, the mean resistive (‘copper’) loss will be 20 W while the total core losses are ~ 3 W. The additional eddy current losses are ~ 20 W reducing the motor efficiency by $\sim 10\%$. It is anticipated that these losses can be reduced to nearer 2% of the shaft power. Overall, considering all these motor losses, the prediction of motor efficiency is 78%. However, the additional eddy current losses for the prototype of motor can be reduced, and after the design is revised the motor efficiency will be approximately 86%, which provides a potentially high overall efficiency for the linear compressor.

5 CONCLUSIONS

Linear Compressors are well established in space applications, and three generations of moving coil compressor design have been described here. These designs have proven the use of oil-free operation by having a ‘clearance seal’ between the piston and cylinder. Central to achieving this is a highly linear motion that is obtained from flexure springs. The

compressors are either used without valves (for Stirling cycle or pulse tube coolers), or with valves (in Joule-Thomson or vapour compression cycle machines).

Moving coil linear motors are inherently expensive and do not scale well to larger applications, and this has led to the design of a novel moving magnet motor that makes efficient use of the magnetic materials. The approach to its design has been fully detailed, along with measurements to validate the modelling of the components.

The experimental results on this linear compressor show that:

- (1) The shaft force calibrated via experiments agreed well with the results from a Vector Fields Finite Element model. The saturation currents were also found for each set of displacements from -6 mm to 6 mm;
- (2) The impact of current imbalance for two set of coils in parallel on the magnetic force can be ignored. But it is still important to monitor the current value for each compressor half;
- (3) The ‘viscous’ damping in the linear compressor is acceptable (in terms of the decay ratio against the number of periods);
- (4) The relevant loss analysis was done indicating the potential high efficiency for this linear compressor architecture;

Compared with conventional compressors, linear compressors can have their output modulated by control of the piston displacement. There is a clear scope for further experiments to study the clearance seal compressor in order to evaluate the thermodynamic performance using gases (nitrogen and helium) and refrigerants (R134a). The oil-free operation allows refrigeration compressors to have much more compact evaporators and condensers, since there is no risk of oil-fouling. Flow rate control by compressor amplitude modulation also means that there is no need for stop-start operation in which temperature gradients have to be re-established.

Although the existing compressor demonstrates the potential for high efficiency, it has a number of shortcomings that will be addressed in subsequent developments. These include: minimizing eddy current losses in the pressure containment tube and compressor structure, increasing the power density, and evolving the design into one that is better suited to mass production.

The companion paper reports on the instrumentation used with linear compressors, and the results when the compressor is tested with gas.

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