Vibration Reduction In Balanced Linear Compressors

M.W.Dadd, P.B.Bailey and G.Davey

Oxford University, Oxford, UK

T.Davis and B.J. Thomlinson

Air Force Research Laboratory, Albuquerque, NM, USA

ABSTRACT

Coolers for Space applications are often powered by reciprocating compressors that use a linear compressor technology. These can deliver the requirements for long life and high reliability but have not yet produced acceptable uncompensated vibration levels at a reasonable cost. If two nominally identical compressors are mounted back to back the vibration level is reduced, but may still be too high for many applications.

Further reduction of vibration is achieved through the use of Adaptive Control systems, which are expensive and reduce the reliability of the system. If the residual vibration can be reduced by better matching of the two compressors, then cheaper, more reliable electronics can be used to achieve the desired vibration level.

Under a research and development effort with the Air Force Research Laboratory, all sources of vibration were considered but effort was concentrated on improving the matching of a compressor pair to limit the main causes of vibration. The dynamics of a compressor were modelled. The force generated by the coil was calculated from flux densities determined by a finite element analysis of the magnetic circuit. The rest of the system was modelled as a damped harmonic oscillator.

An attempt was made to reduce the residual vibration in an opposed pair of compressors by duplicating the model to simulate a compressor pair and investigating the effect of small variations of a number of parameters. These results were used to estimate the accuracy with which various parameters must be matched to achieve a certain residual vibration and this information was then used to improve the assembly of a compressor pair.

Most of the components for this compressor were already available so it was not possible to make major changes. Detailed measurements were made on all components and assemblies so that the vibration spectrum could be related to a compressor with documented manufacturing and building standards.

INTRODUCTION

The Oxford University cryogenics group has been involved in the development of space cryocoolers for many years and has produced a number of prototype coolers/compressors. A prototype balanced compressor pair (designated the Capital Compressor) that had been designed and manufactured at Oxford for a recent TRW space project, was found to have an inherently low vibration level The compressor halves had been assembled carefully to achieve true clearance seals but there had been no conscious effort to produce a low vibration machine.

The very low vibration levels required by space programmes are currently achieved by complex drive electronics that use adaptive control algorithms. Whilst effective, this approach has significant drawbacks:

- The complexity of the electronics hinders the achievement of high reliability
- The additional electronics increases cost and payload.
- A basic adaptive control approach is only able to improve on-axis vibration levels. For the reduction of off-axis levels, additional force transducers and control electronics would be required with a further impact on reliability etc.

An approach that suggested itself is the further reduction of the uncompensated vibration by improved matching of the two compressors through better matching of components and closer control of build processes. To achieve this goal it was recognised that a better understanding of the forces operating would be desirable and that methods should be found to allow build quality and component matching to be evaluated and documented. Some work has already been done on the development of a compressor model using a dynamic simulation package called Vissim. This was used to model an existing unbalanced compressor and was found to give good agreement with measured operating parameters that included out of balance forces. This work was reported in ref(1).

The project that evolved had the aim of

- Looking more closely at the forces acting in the compressors to see whether the existing model is adequate.
- Looking at what dimensional and alignment limits are required for good balance, and what the practical limitations might be.
- Documenting the assembly of a balanced compressor pair. It was hoped that some improvement might be made over previous assemblies but this might not be possible given the restriction of using largely existing components.
- Comparing the measured vibration of the assembled compressor with the values generated by the model

At the time of writing, the compressor build had not been completed so this paper will concentrate on describing the approach to modelling and build evaluation.

FORCES ACTING IN A LINEAR COMPRESSOR

The types of linear compressors used in space cryocoolers have the virtue that the systems offerees operating in them are amenable to relatively simple descriptions. Figure 1. shows the main components of a typical linear compressor. These are:

- A piston/cylinder assembly utilising a no-contact clearance seal
- A linear motor – in this case a moving coil "loudspeaker" type motor
- A suspension system comprising of two aligned sets of flexure bearings – these are usually springs with a flat spiral geometry and are known as spiral springs or flexures.

The characteristics of the suspension system are extremely important to the operation of the compressor. The geometry and alignment of the springs accurately define an axis along which the spring stiffness is low and a large movement (e.g. > 10 mm) is possible without exceeding the fatigue limit of the spring material. Perpendicular to this axis the stiffness is extremely high.

The obvious axial symmetry of this type of compressor and the distinct characteristics of the suspension spring assembly suggest the division offorces acting into three types:

- Forces acting along the compressor axis where the spring stiffness is low– these will be termed "On-axis" forces.
- Forces acting perpendicular to the compressor axis where the spring stiffness is very high – these will be termed "Off-axis forces.

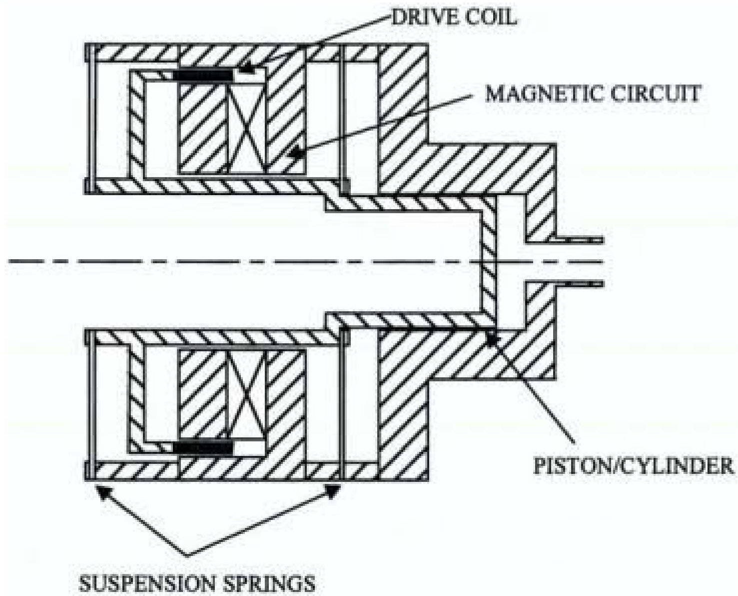

Figure 1. Schematic diagram of typical compressor.

- Forces associated with the rotation of the spiral suspension springs. The operation of these springs generally gives rise to a small rotation of the moving assembly. The forces exerted on the moving assembly, are also off-axis forces but they resolve to a zero net force and a well-defined moment acting about the compressor axis.

ON-AXIS FORCES

The On-axis forces are the dominant forces acting in the compressor. The principal forces with large magnitudes are those intrinsic to the compressors operation:
- The force generated by the coil
- The force produced by the suspension springs as they are deflected.
- The force produced by gas pressure differences acting on the piston area.

These are balanced by the inertia force acting through the centre of gravity as the moving assembly accelerates. Ideally the axis defined by the movement of centre of gravity and axes along which these forces act should coincide. The result would then be a single force acting along a single well defined axis and there would be no resulting moments. In practice there will be offsets and angular misalignments between the axes and these will result in off axis forces and moments perpendicular to the compressor axis. The radial stiffness and separation of the two sets of suspension springs determine the actual movements resulting from such misalignments. If the magnitudes of the axial forces are known then the level of misalignment that can be tolerated is readily calculated.

There are other effects that will give rise to axial forces such as:
- Windage
- Eddy current effects due to conductive components moving in the motors magnetic field
- Shear forces acting on the piston due to gas flow in the clearance seals
- Forces generated by sections of the coil leads moving in areas of stray flux.

These are much smaller in magnitude and would only have any significance if:
- they were so uncontrollable that they resulted in big differences between the two compressor assemblies.

- They were large enough and acted far enough off the compressor axis that they produced sufficient moments to cause radial deflection of the springs.

It is clear that windage and eddy currents could only have any net effect if there was fairly gross mismatching of the compressors.

The forces generated by coil leads will be very closely matched but will not be acting along the compressor axis. The shear forces acting on the piston are sensitive to the dimensions and geometry of the clearance seals, which may not be closely matched.

Frictional forces should not be present because in principal there are no contacting surfaces. The detection of vibration caused by friction would indicate a fault condition. The forces generated by friction will be very variable depending on small details of surface form etc so that it is not possible to be too specific about its form.

OFF-AXIS FORCES

Off axis forces can only register a net effect because of some deviation from symmetry either required in the design or as a consequence of imperfections in the compressor build. Because of the high radial stiffness only large forces will produce significant displacements. There is also the possibility that smaller forces may have enough leverage to cause a significant angular deflection. Some possible forces that were considered are:

- Unbalanced gas forces acting on the sides of the piston
- Forces generated by current carrying leads interacting with the motor flux
- Forces generated by deviations from linear movement of the suspension system

The forces acting on the sides of the piston are large and are determined by the detailed geometry of the clearance seal. Defects in the clearance seal geometry can lead to significant imbalance and misalignment of these forces. The radial gap in a clearance seal is typically around 10 microns for reasonable seal efficiency. With machining tolerances typically around +/- 1 micron and a similar tolerance on alignment it is clear that this effect merits careful consideration.

The forces generated by the current leads are an inevitable part of a practical design. Their magnitudes are readily estimated, and are likely to be insignificant.

Off–axis Forces generated by inaccuracies in the linear movement of the suspension system need to be considered but it is likely that the linearity required for the clearance seal will automatically ensure that they are not significant.

MOMENTS ABOUT COMPRESSOR AXIS

The magnitude of the rotation produced by the springs will be determined by certain aspects of the spring geometry, principally the spiral arm length and curvature. The manufacturing processes used allow these parameters to be controlled to tight limits, typically better than +/- 0.1%. It is clear that provided that the axes of the compressor halves are well aligned and that the compressor strokes are well matched, any residual moment is likely to be insignificant.

RESONANCES

In addition to the forces already described there remains the possibility that mechanical assemblies have resonant modes that may be excited at particular frequencies. Small out of balance forces, whether on-axis or off-axis, could be amplified to the point that they become a problem. The general approach to this problem is to keep the resonant frequencies of components as high as possible by designing the moving assembly for high stiffness. The use of materials that have some intrinsic damping can also be considered although the opportunities to do this are limited by the need for other mechanical properties. The moving assembly/suspension spring system does present a particular instance where resonances may be a problem. The stroke required of the springs limits their stiffness to relatively low values.

There are a number of vibration modes that should be considered:

- Resonance of the spring arms – the inner and outer ends being nodes and the centre being an anti-node.
- Torsional resonance of the moving assembly about the compressor axis
- Torsional resonance of the moving assembly perpendicular to the compressor axis
- Radial resonance of the moving assembly with respect to the compressor axis.

These modes of resonance were investigated in the Capital compressor being built and the resonant frequencies were determined to be at least 400 Hz.

A STRATEGY FOR GOOD BALANCE

The aim for a well-balanced compressor is that each compressor half should only produce an On-axis force and that these forces should equal so that when they are aligned back to back they cancel out To approach this goal the design has to be effective, the compressor components need to be well matched and the build quality has to be adequately controlled.

The overall strategy that has been adopted is:

- Close matching of the amplitude of On-axis forces.
- Alignment of On-axis forces to avoid generating unnecessary couples.
- Minimising of Off-axis forces
- Design of suspension system with high radial stiffness to minimise deflections.
- Mechanical design that avoids assembly resonances in the operating frequency range
- Design which includes damping (where possible) to minimise amplitudes of resonances

IMPROVING THE BALANCE OF THE CAPITAL COMPRESSOR

The above has described in general terms the effect of different forces and their possible sources without detailed reference to their magnitudes and real significance. For the Capital compressor the Vissim based model described below was used to calculate the principal On-axis forces for typical operating conditions. The effects of mismatching certain parameters were investigated. Also measurements were made of the torsional and radial stiffness of the compressor's suspension system. This information was used as the basis for deciding which of the effects described were likely to have any real impact on the overall balance.

For the Capital compressor, areas open to some improvement were identified as:

- Better matching of principal On axis forces – reducing differences in axial offsets of coils, better matching of moving masses, motor parameters i.e. no of turns in coil, field in air gap
- Reducing radial offsets between principal On axis forces
- Improving geometry of clearance seals by closer control of component manufacture and alignment
- Improving angular alignment of compressor halves with respect to each other

ON-AXIS FORCES AND SPRING STIFFNESS FOR CAPITAL COMPRESSOR

Maximum values of principal On-axis forces	100 N
Minimum radial stiffness of suspension system	760,000 N/m
Torsional stiffness perpendicular to compressor axis (about C.O.G.)	780 Km/radian

If a maximum of 1 micron radial movement of any of point on the piston is set as a criteria then maximum allowable values can be set for Off-axis forces and moments perpendicular to compressor axis. With the peak value of the principal forces, maximum angular misalignments and offsets can also be defined:

Maximum Off-axis force	0.76 N
Maximum moment perpendicular to compressor axis	0.023 N.m
Maximum angular misalignment for On axis forces	0.0076 radians
Maximum offset for On axis forces.	0.23 mm

MODEL USED TO INVESTIGATE PRINCIPAL ON-AXIS FORCES

The model used to evaluate the effects of build mismatches is briefly described below. The differential equation defining the motion for the On-axis forces described above is:

$$m\ddot{x} + k_1 x + P_d(x,t)A = F(x,t) \quad (1)$$

x is the displacement of the piston assembly from its rest position
m is the moving mass, k_1 is the spring rate of the suspension springs
P_d is the pressure difference acting across the piston over an *area* A.
$P_d = P - P_0$ where P is cycle pressure, P_0 is pressure in compressor body
$F(x,t)$ is the force generated by the driving coil.

The variation in P_d is primarily generated by the cycle pressure. This is determined by the thermodynamic and pressure drop processes occurring in the refrigeration cycle and cannot be simply described. An approximate model, that is simple and useful, can be developed by treating the gas force as a spring/ damper combination The resulting differential equation is that for a damped harmonic oscillator:

$$m\ddot{x} + c\dot{x} + kx = F(x,t) \quad (2)$$

c is effective damping constant for the gas, k is total spring rate, $k = k_1 + k_2$, k_2 is the effective spring rate of the gas.

The force generated by the coil is given by

$$F(x,t) = i(t)\int B(x)\frac{dl}{dx}dx \quad (3)$$

$\frac{dl}{dx}$ is the conductor length per axial length of coil, $i(t)$ is the current through the coil

The differential equation defining the behaviour of the moving coil motor as an electrical system is:

$$V(t) = E(t) + R\,i(t) + L\frac{di}{dt} \quad (4)$$

$V(t)$ is the applied voltage, $E(t)$ is the back emf generated by the coil, R is the coil resistance, L is the coil inductance.

$E(t)$ is given by:

$$E(t) = \dot{x}\int B(x)\frac{dl}{dx}dx \quad (5)$$

The values of integral $\int B(x)\frac{dl}{dx}dx$ that are required for Eqs. (2) and (4) are accessed in Vissim as a "Look up" table.

The look up table values were calculated in a spread sheet using flux distribution defined by:

$$B(x) = B_0 \qquad \text{in air gap} \quad (6)$$

$$B(x) = a + \frac{b}{x^n} \qquad \text{in fringing fields} \quad (7)$$

These equations were derived empirically using values generated by a finite element analysis of the magnetic circuit.

SOME RESULTS OBTAINED FROM THE MODEL

The work described in Dadd et al.[1] showed that model values and values measured on a particular unbalanced compressor were close for all the main parameters i.e. instantaneous values of forces, currents and voltage inputs. The only value that had to be adjusted to obtain good agreement was the coil inductance. A similar comparison will be made with the Capital compressor when its build has been completed.

The model was used to investigate how the mismatch of particular build parameters would

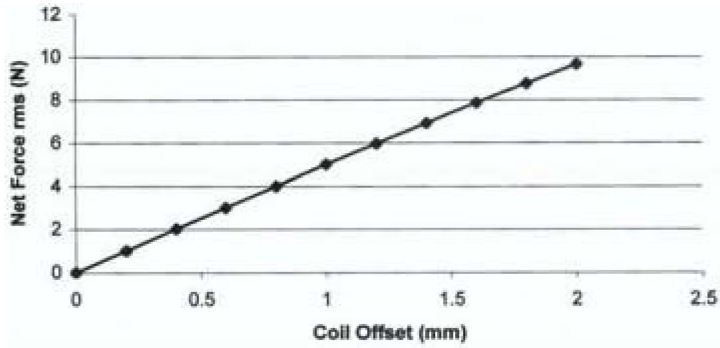

Figure 2. Variation of net force with axial offset of coil on one side.

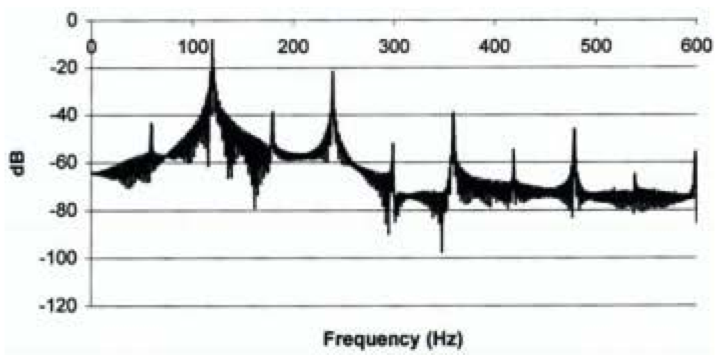

Figure 3. Vibration Spectrum for coil offset of 0.2 mm.

effect the balance such parameters could be moving mass, inductance, magnetic flux distribution etc. As an example, one finding is the dependence on the mean position of the coil in the magnetic circuit. This can vary because of the build up of tolerances in the compressor build. Figure 2. shows how the residual force varies with the changes in offset for one of the compressors. Figure 3. shows the frequency spectrum for the case where one coil is centred and the other is offset by 0.2mm. It will be seen that the residual force is becoming quite significant i.e. 1% of value for each half and that it is dominated by the second harmonic.

MEASUREMENTS AND DOCUMENTATION OF COMPRESSOR BUILD

An important and time-consuming part of the work described in this paper was the measurement and documentation of various parameters during the build. These measurements included:

- Component masses
- Coil characteristics – resistance, No of turns and Inductance
- Magnetic flux distribution in gap
- Geometrical and dimensional features of individual components e.g. concentricity
- The geometrical features e.g. alignments of the compressor assemblies as they are built up.

The purpose of these measurements was twofold. Firstly to establish a build specification on which to base any modelling or interpretation of measurements. Secondly to actually identify areas where mismatches were significant or where components were not manufactured to adequate standards.

Many of these measurements e.g. mass, coil resistance etc required only ordinary laboratory equipment and will not be discussed further. However the measurement of geometrical features and alignment required specialised metrology equipment. Many of these measurements also needed to be taken during the compressor build when cleanliness is very important.

To enable inspection measurements to be made in a clean environment a TalyRond 100 was installed in the clean room. The essential features of this type of instrument are:

- An accurate turntable incorporating centering and levelling adjustments
- Vertical and horizontal columns allowing accurate positioning of probes
- A number of sensitive displacement probes capable of determining surface forms e.g. roundness, concentricity, surface roughness etc.

Whilst taking these alignment measurements it became clear that the process could be helped by having specific measurement features e.g. easily accessible reference surfaces. This was not possible with the capital compressor but may be considered in future designs.

CONCLUSIONS

Although it is not possible to comment with the advantage of actual vibration measurements, some conclusions can be made:

- The existing model appears to be adequate for describing the principal On-axis forces
- The magnetic properties of the magnets and magnetic circuit components are variable. Good matching of these components is helped if they are chosen from within a single batch.
- The matching of the on axis forces requires matching of coil offset as well as other parameters
- The offsets between the axes of the principal forces can cause significant moments if their alignment is not adequately controlled
- The clearance seal geometry needs to be closely controlled to avoid significant unbalanced forces acting perpendicular to the compressor's axis.
- Future compressor designs could usefully incorporate specific features to facilitate measurements on build alignment

In many ways good balance is synonymous with high build quality – i.e. both are concerned with good alignments, absence of friction and the minimising of resonance. Low vibration compressors require a high build standard: a high build standard may be demonstrated by low vibration measurements. Assessing build quality is an important issue because of its effect on reliability. The pursuit of low vibration may therefore have added benefits.

ACKNOWLEDGEMENTS

The Capital compressor was designed and built under a contract with TRW, who also made compressor components available for this work.

REFERENCES

1. Dadd, M.W., Davey, G., Lion Stoppato, P.F., Bailey, P.B., "Vibration Reduction in Balanced Linear Compressors in the 17th International Cryogenic Engineering Conference", ICEC 17, Institute of Physics Publishing, Bristol (1998), pp.127-131.