Gas Spring Losses in Linear Clearance Seal Compressors

P.B. Bailey, M.W. Dadd, J.S. Reed*, C.R. Stone

Oxford University, Department of Engineering Science
Oxford, U.K.

* Now at EADS Astrium Ltd.
Stevenage, U.K.

Thomas M. Davis

Air Force Research Laboratory
Kirtland AFB, New Mexico, U.S.A

ABSTRACT

A fundamental loss mechanism in cryocoolers is associated with compression and expansion processes, and the simplest demonstration of this can be observed in a gas spring. Our understanding of these gas spring losses is largely based on work by Kornhauser, Smith and others who carried out a series of thorough investigations on conventional crank driven reciprocating compressors, where the use of normal sliding seals would minimize seal losses. The widespread use of linear clearance seals in linear compressor has raised the question of how applicable the Kornhauser correlations are to compressors with clearance seals, which have significant flow through the seals and consequent seal losses.

This paper describes experiments carried out with a clearance seal linear compressor attached to a plain gas spring volume. The static flow through the clearance seal in the compressor was measured over a range of piston positions, and this information was used to estimate the seal loss for specific strokes. Calculation of the losses from electrical power measurements was found to be unreliable due to uncertainties in the electromagnetic and other losses within the compressor. The P-V work done by the piston on the gas was measured, taking into account the small phase shifts in the instrumentation. By subtracting out the estimated seal loss, the gas spring loss was derived. Results are presented for a number of gases over a range of frequencies and strokes.

Cryocoolers 14, edited by S.D. Miller and R.G. Ross, Jr.

INTRODUCTION

Many of the loss processes present in a working cryocooler have been fairly well characterized by measurements and analysis, but experimental data has shown that these do not appear sufficient to account for the amount of power lost in working machines[1]. The work described here has been to investigate thermodynamic losses that appear to occur in miniature Stirling cycle or Pulse Tube coolers but which are not adequately explained by our existing models. These losses are large enough that a small reduction in their value would produce a significant improvement in cryocooler efficiency.

It is believed that these losses are mainly associated with the heat transfer and flow processes that occur when the cooler components are subject to a cyclic pressure variation and are not a direct result of the refrigeration process. This has suggested that there would be equivalent losses in systems that have similar pressure and flow processes but which do not have the usual mechanisms for generating refrigeration, such as a second piston or displacer. Such systems would be similar to gas springs but with components and geometries that derive from cooler designs rather than those typically selected for gas spring efficiency.

If there is no refrigeration cycle operating then all power dissipated can be attributed to the loss mechanisms including those that are of interest. The measurement of the power dissipated for differing heat transfer and flow regimes should therefore allow the correlation of loss values with variations of particular parameters in a way that would not be possible in an actual cooler.

This paper describes experiments to evaluate the magnitude of compression losses for a clearance seal compressor acting on a simple cylindrical compression space. The results are compared with correlations described in the literature. It is hoped that further work will allow this approach to be applied to more complex cooler components

GAS SPRING LOSS CORRELATIONS

A useful starting point for this subject is the 1987 paper of Kornhauser and Smith[2], which presents experimental data of loss measurements made in a reciprocating compressor over a wide range of operating conditions. This data was compared with the predictions of a number of theoretical treatments based on a one-dimensional analysis of the gas, and the conclusion drawn was that the treatments by Lee[3] and by Cooke-Yarborough[4] were the most accurate. The equation adopted by Kornhauser in his paper was Lee's.

In Kornhauser & Smith's 1993 paper[5] the work was extended to wider range of conditions including finned geometries. Lee's expression was found to correlate much of the data fairly well but an empirical modification was proposed that appears to give a significant improvement. The measurements that did not correlate well were those for high volume ratios and high molecular weight gases. It was suggested that Lee's assumptions were basically less applicable for these conditions. It was also suggested that the appendix gap (the small annular gap around the piston) might have a significant influence. It is worth mentioning that there are subtle differences (and inconsistencies) in the definition of some of the parameters used in these various correlations, however the definitions in Kornhauser's later paper do agree with Lee and Cooke-Yarborough's after some manipulation.

This paper refers to two expressions for the heat transfer loss, the first, which was obtained theoretically by Lee and Cooke-Yarborough, will be referred to as 'Lee', and which gives the compression loss, W_c, as

$$W_c = \frac{\pi}{2}\, p_o V_0 \left(\frac{p_a}{p_0}\right)^2 \frac{(\gamma - 1)}{\gamma} \frac{1}{y} \left(\frac{\cosh(y)\sinh(y) - \sin(y)\cos(y)}{\cosh^2(y) - \sin^2(y)}\right) \tag{1}$$

where

$$y = \left(\frac{Pe_{\omega}}{8}\right)^{\frac{1}{2}} \tag{2}$$

and Pe_{ω} is the Peclet number based on the angular frequency and a conventional hydraulic diameter

$$Pe_{\varpi} = \frac{\omega D_h^2}{4\alpha_0} \tag{3}$$

$$D_h = \frac{4V_o}{A_o} \tag{4}$$

where
p_0 = pressure at mid-stroke
V_0 = cylinder volume at mid-stroke
p_a = pressure amplitude
γ = ratio of specific heats
ω = angular frequency
α_o = thermal diffusivity of gas at mid-stroke
A_o = cylinder surface area at mid-stroke

It should be noted that this expression does not take into account mass exchange of the compression volume with any other parts of the system. Additionally the cylinder wall is assumed uniform and isothermal. Finally, the pressure swing is assumed to be sinusoidal and relatively small compared to the mean pressure.

The second expression (a modification of Lee's expression) is given in the 1993 Kornhauser paper as a better fit to their data, and will be referred to in this paper as 'Kornhauser'. Based on equation 1 above, it replaces the parameter y with y' which is defined as:

$$\begin{aligned} y' &= 1.20 y^{0.86} \\ y' &= 0.49 Pe_{\omega}^{0.43} \end{aligned} \tag{5}$$

All of the experimental data from these early papers is derived from crank-driven compressors with conventional seals. In clearance seal compressors, such as those used on cryocoolers, there is a significant flow through the seal and it might be expected that this flow would influence the overall loss values.

DESCRIPTION OF EXPERIMENTS

Test equipment

The test rig was based on a linear compressor previously used as part of a Stirling cycle domestic freezer, and consisted of a clearance seal piston/cylinder assembly driven by a linear motor (figures 1 & 2). At maximum stroke the top of the piston reaches the end of the cylinder. Mounted on top of this assembly is a short cylindrical volume and a cylinder head which make up the fixed volume 'gas spring'.

Experimental Approach

Measurements of thermodynamic losses in a machine of this type can be derived either from electrical measurements of the input power to the compressor, or from measurement of the 'P-V' work done by the piston on the gas in the compression space.

Measurement of electrical power can more easily be made with high accuracy, but there are several mechanical and electromagnetic loss mechanisms within a compressor, and some of these are difficult to characterize accurately. In addition, the compressor is a resonant device,

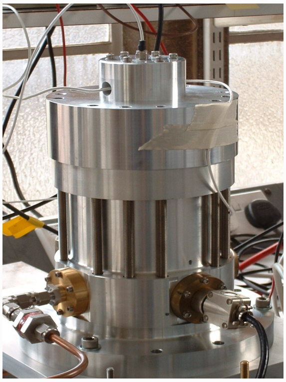

Figure 1. Test compressor

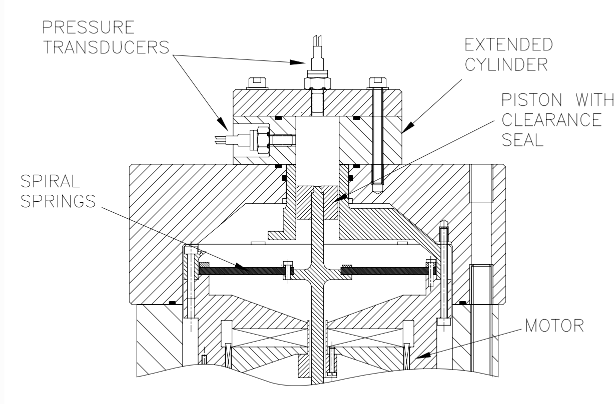

Figure 2. Section through test compressor.

and the linear motor is inefficient when operating away from the resonant frequency. Under these conditions most of the electrical power input is dissipated as 'i^2R' loss in the drive coil, and the calculated thermodynamic work done would be a small (and inaccurate) difference between two large numbers. If the frequency of testing was limited to 'near resonant' conditions, only a very limited range of data would be obtained. For these reasons, the '*P-V*' work was derived from measurements of pressure and piston displacement.

The position of the piston was measured using a commercial non-contacting LVDT, allowing the gas volume to be determined as a function of time. Provision was made for pressure transducers to be fitted flush with the sides of the cylinder and in the cylinder head as shown. Two pressure transducers were installed in order to check whether varying gas temperature would affect the accuracy of the transducers. One transducer was an Entran transducer that was compensated to minimize thermal effects, the other was a Druck transducer which did not have equivalent compensation. It was found that the two transducers gave very similar results and the conclusion drawn was that for the range of measurements made, the varying gas temperature did not affect the performance of the transducers. A third pressure transducer recorded the pressure in the body of the compressor to measure the small pressure variation in the pressure vessel surrounding the linear motor.

An important consideration when measuring a relatively small power, (as is the case here) is that the shape of the '*P-V*' loop is long and thin, and the calculated area of the loop is very sensitive to phase shifts in the electronics of any instrumentation used. Prior to these tests, the phase shift in the instrumentation was carefully measured, and data from the pressure and displacement measurements was digitally corrected to allow for this phase shift. Note that the resonant frequency of the pressure transducer diaphragms was expected to be much higher than the operating frequency of the compressor and hence there was not expected to be any significant mechanical phase shift in the measurement.

Analysis

The total power input into the gas volume is made up of two components. The major component is the compression loss in the gas that we are interested in. In addition to this there is also a seal loss due to leakage through the clearance seal:

$$W_{in} = W_c + W_s \tag{6}$$

$$W_{in} = f \oint pdV \tag{7}$$

Where W_c is the 'compression loss', W_s is the loss due to flow through the clearance seal, and W_{in} is the "pdV" work done by the piston on the gas in the compression space, which is a function of the frequency f, and the measured values of pressure p and cylinder volume V.

The seal loss cannot be measured independently and was calculated using measurements of the clearances seal flow properties together with pressure difference across it. The Reynolds numbers for the leakage flows that occur through clearance seals are generally very low and well within the laminar flow regime. To verify this a CFD model was run to check that the flow was laminar, taking into account the real pressure swing (which was not perfectly sinusoidal), an estimated temperature swing in the gas, and the edges of the piston. It was found that the flow was always laminar over the whole range of operating conditions.

Given the laminar nature of the flow, the instantaneous mass flow rate dm/dt through the clearance seal is given as a function of the time-varying pressure difference across the seal Δp by the equation

$$\frac{dm}{dt} = \frac{\pi D \rho_m r^3(t)}{12\mu L} \Delta p(t) \tag{8}$$

Where D is the piston diameter, ρ_m is the mean gas density, μ the mean viscosity, L the seal length, and r is the effective radial clearance of the seal, which is function of piston position, and hence time.

For this compressor, the leakage past the seal had been experimentally determined as a function of piston position, and from this the effective radial clearance had been calculated (see figure 3).

To evaluate Ws, data for the stroke and for the pressure either side of the seal was used to estimate the instantaneous seal leakage, and this was integrated round a cycle to give an estimate of the work lost pumping gas through the seal.

To enable a wider variation of Peclet number to be explored, 4 different gases were used in the testing – helium, neon, nitrogen and krypton. With these, and by varying the frequency between 2 and 75 Hz, the stroke between 1.6 and 12.1 mm, and the fill pressure between 0.25 and 15 bar, a range of Peclet numbers from 3 to 19000 was investigated.

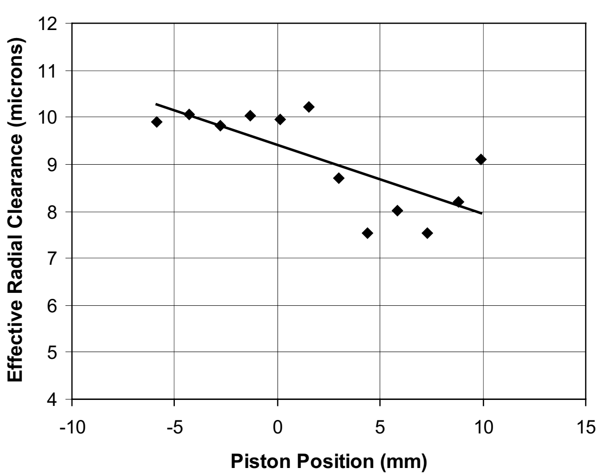

Figure 3. Effective radial clearance of seal as a function of piston position.

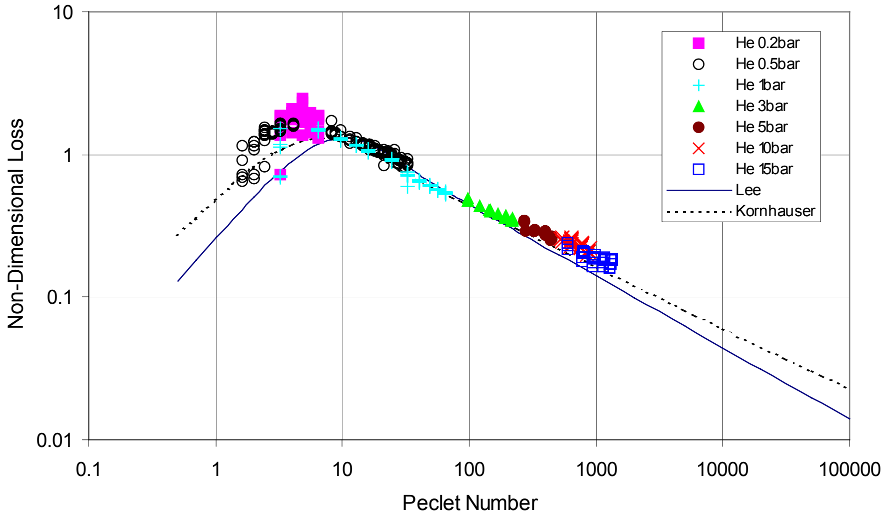

Figure 4. Non-Dimensional loss as a function of Peclet number for helium

EXPERIMENTAL RESULTS

For convenience, the measured gas spring losses are plotted non-dimensionally (L_{ND}) with the denominator equal to the ideal adiabatic work of compression for the same volume ratio r_v

$$L_{ND} = \frac{W_c}{\dfrac{p_0 V_0}{\gamma - 1}\left(\dfrac{1+r_v}{2}\right)^{\gamma-1}\left(1 - r_v^{1-\gamma}\right)} \tag{9}$$

The gas spring losses measured for helium are plotted in Figure 4. The results generally give good support to Kornhauser's modification of Lee's equation. The results show little scatter for the range of Peclet Numbers between 10 and 1000 and there is little evidence for any dependence on pressure. The scatter for the results for Peclet numbers between 1 and 10 is greater but measurement errors were becoming much more significant and the scatter is within the expected range of error.

The data for nitrogen (shown in figure 5), and for neon and krypton are similar, but all exhibit a deviation from Kornhauser's equation for Peclet numbers above 1000 (it was not possible to get data for helium for Pe>1000).

The results for all the gases are shown in Figure 6. It will be seen that altogether the results give good support to Kornhauser's modified equation. The spread of the results is really quite small given the range of pressures, frequencies and gases used.

The lowest Peclet number results (<10) were only obtained for helium and neon. Even allowing for the scatter on the helium results there does appear to be a difference in the maximum values of the non-dimensional loss although the Peclet numbers do correspond. One possible explanation for this is that the boundary layer depth for helium at these frequencies will be large compared to the size of the cylinder. In addition, these Peclet numbers are in the 'transition' regime between adiabatic and isothermal behavior, and the choice of regime may be sensitive to other factors. Being at the extremity of the data range, more results are really required to demonstrate that it is a real effect.

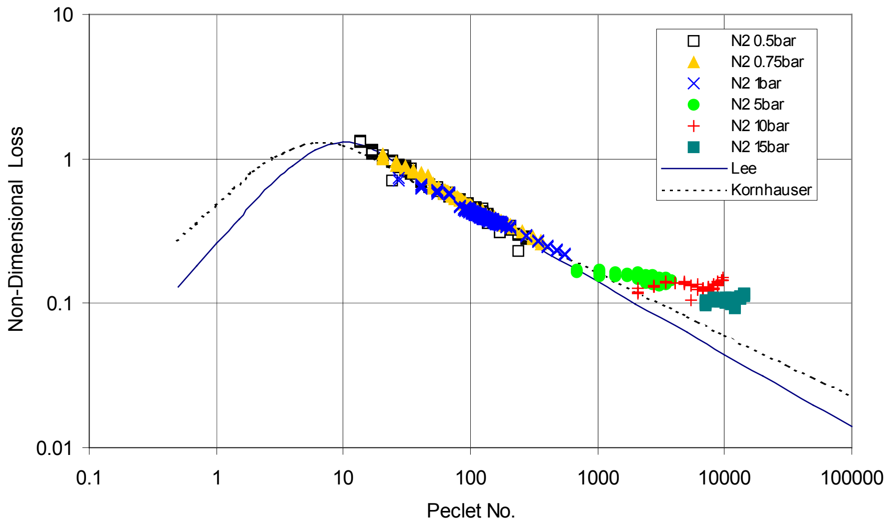

Figure 5. Non-Dimensional loss as a function of Peclet number for nitrogen.

At the highest Peclet numbers (>1000) all of the gases (except helium for which there is no data) show higher loss values than predicted by either of the analytic expressions.

The most likely reason for this deviation is the error in estimation of the seal loss. At high Peclet numbers, the seal loss becomes a significant part of the total *'P-V'* work done on the gas. It is known that the dynamic seal loss can be higher than that predicted by the simple theory and the data from static flow measurements – these effects are also pressure dependent, and will result in the seal loss increasing at higher fill pressures.

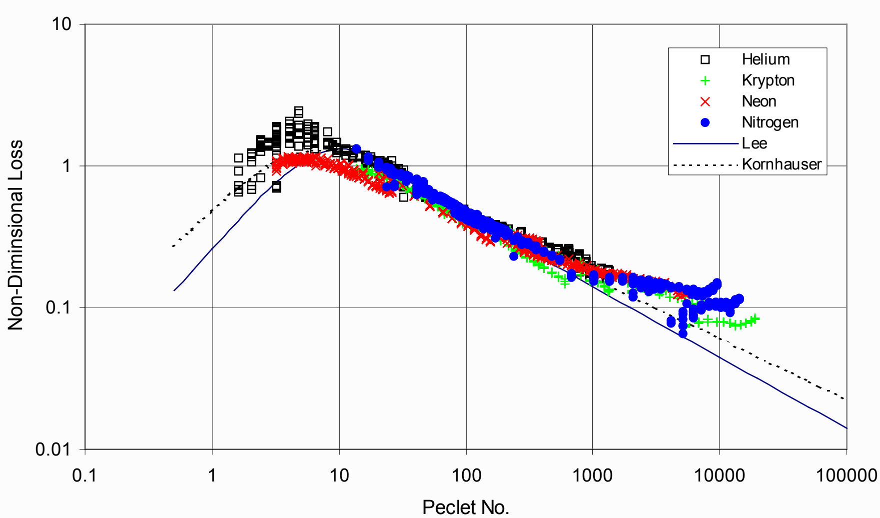

Figure 6. Non-Dimensional loss as a function of Peclet number for all gases.

CONCLUSIONS

An important conclusion is that these expressions, derived for conventional sliding seal compressors, are valid for clearance seal machines, once the seal pumping loss has been allowed for. Other conclusions that can be drawn from these results are:

- For Peclet numbers in the range 10 to 1000, Kornhauser's modified version of Lee's equation gives good results and can be used in applications involving typical clearance seals.
- There is evidence for a significant deviation from Lee's expression for all but the lightest gases (i.e. helium and hydrogen) where the Peclet number is above 1000. However, this deviation could be due to inaccuracies in the estimation of the clearance seal loss, and more investigation is needed to investigate this further.
- Data in the Peclet number range from 1 to 10 gives significant scatter. This region is significant in terms of real Cryocoolers, and more work is required to determine the nature of compression losses in this area.

ACKNOWLEDGMENT

This effort was sponsored by the Air Force Office of Scientific Research, Air Force Material Command, USAF, under grant number FA8655-04-1-3016. The U.S. Government is authorized to reproduce and distribute reprints for Government purpose notwithstanding any copyright notation thereon. The views and conclusions contained herein are those of the authors and should not be interpreted as necessarily representing the official policies or endorsements, either expressed or implied, of the Air Force Office of Scientific Research or the U.S. Government.

We acknowledge the strong support of Thom Davis of AFRL and Barrett Flake of EOARD, and would especially like to thank Jaime Reed (now of EADS Astrium) who carried out the experimental program.

REFERENCES

1. Reed, J.S., Davey, G., Dadd, M.W., Bailey, P.B., "Compression Losses in Cryocoolers," *Cryocoolers 13*, Kluwer Academic/Plenum Publishers, New York (2001), pp. 353-362.
2. Kornhauser, A.A., Smith, J.L., "A Comparison of Cylinder Heat Transfer Expressions Based on prediction of Gas Spring Hysteresis Loss," *Fluid Flow in Heat Transfer and Reciprocating Machinery*, ASME 1987, pp. 89-96.
3. Lee, K.P., "A Simplistic Model of Cyclic Heat Transfer Phenomena in Closed Spaces," 18th IECEC, 1983.
4. Cooke-Yarborough E.H. and Ryden, D.J., "Mechanical power losses caused by imperfect heat transfer in a nearly-isothermal Stirling engine," SAE, 1985.
5. Kornhauser A.A., Smith, J.L., "The Effects of Heat Transfer on Gas Spring Performance," Journal of Energy Resources Technology, Vol. 115, ASME, 1993, pp. 70-75.