Compression Losses In Cryocoolers

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ABSTRACT

Most of the loss mechanisms in Stirling and Pulse Tube cryocoolers are well documented and are relatively easy to analyse and estimate. One of these losses is related to the irreversible compression of gas in the cylinder, and the magnitude of this loss is such that it has a very significant impact on the overall system efficiency. Simple tests on a cryocooler give an estimate on the size of this loss, though the results of these tests include elements of other, known losses, such as clearance seal and pressure drop losses. The parametric variation of this 'lumped' loss suggests that it is not primarily due to these, or other known losses. Over a wide range of machines and conditions this 'lumped' loss varies with operating frequency, swept volume and pressure swing, suggesting that it is some kind of thermodynamic effect related to the area of the 'P-V' loop in the compression space.

This loss appears to be quite independent of the refrigeration cycle, and takes place in the ambient temperature part of the system as a result of cyclic pressure changes in a typical Stirling cycle geometry. The loss does depend on geometry, and is typically higher on a 'split' Stirling cycle machine than on a more compact 'integral' one.

This paper looks at the experimental data which is the basis of this loss, and compares this data with other studies carried out on compression losses in Stirling Cycle and other reciprocating machines.

INTRODUCTION

A typical Stirling cycle or Pulse Tube cryocooler consists of a compressor containing a reciprocating piston and a cold-head containing a regenerator. By supplying electrical power to the motor the piston performs work on the gas in the compression space and the cold head uses this work to transport heat from the cold end to the compression end. In the ideal case the ratio of the heat lifted at the cold end to the work done on the gas at the warm end would be equal to that of a Carnot cycle operating between the same temperatures. In practice, however, this process is subject to a number of loss mechanisms which either reduce the effective amount of work input into the cycle or place additional heat loads on the cold-end. Understanding and reducing these loss mechanisms is the key to producing an efficient cryocooler.

Many of the loss processes present in a working cryocooler have been fairly well characterized by measurements and analysis. Data has shown, however, that these do not appear sufficient to account for the amount of power lost in working machines. Indeed the discrepancy

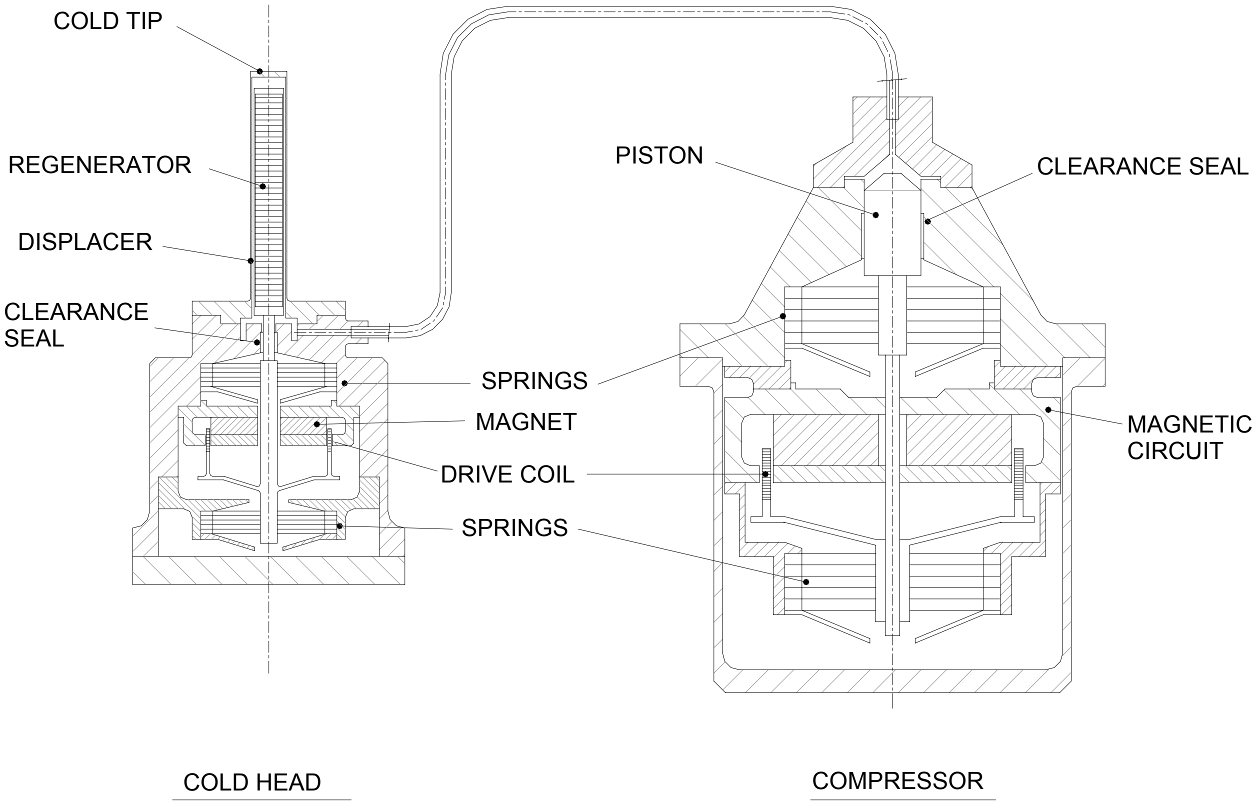

Figure 1. Schematic of original Oxford ‘split’cryocooler

is so large it seems to be the largest loss process for machines made in Oxford and has a profound impact on the efficiency and design of these machines. It would therefore be highly desirable to develop an understanding of the physical mechanisms behind these so called ‘compression losses’ and hopefully reduce them in future devices.

For convenience, this paper relates to the classic 'Oxford' type of machine, with a moving coil linear motor and clearance seal suspension system; figure 1 shows a typical machine. Compression losses are not unique to this particular 'split' configuration and are also present in 'integral' machines with the compressor and cold head close-coupled.

EXPERIMENTAL EVIDENCE

Although compression losses are observed in both types of machine, one advantage of a Stirling cycle over a pulse tube is the possibility of independently controlling the displacer and the compressor piston. One possible test is to drive the compressor whilst holding the displacer stationary. Under these circumstances there is no change in volume at the cold end of the machine, the cold end gas can do no work and hence there is no gross refrigeration. Ideally the gas should behave as a spring and no power should be absorbed. The fact that there is a significant power loss is indicative of loss processes that are not dependant on the refrigeration cycle. We will refer to this measurement of ‘gas spring loss’ as a ‘compression loss’ test.

Orlowska[1] carried out a thorough analysis of the losses in an early 'Oxford' Stirling Cryocooler, which was a machine of 3.15 cm^3 swept volume designed to give 1 Watt of refrigeration at 80 K. She carried out compression loss tests on a complete cryocooler over a range of frequency and strokes. The shaft work (taken as $W_e - i^2R$) was measured and found to be very linear with frequency (figure 2).

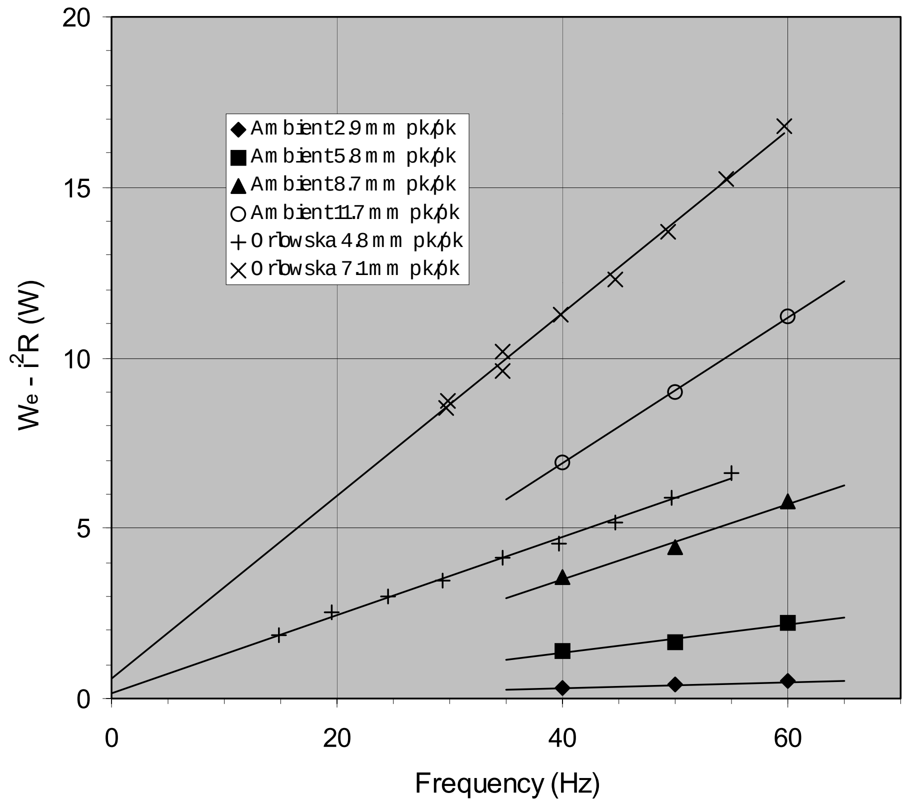

Figure 2. Compression Loss data for two Stirling coolers.

Similar results have been obtained from every other machine built and tested at Oxford. For example, further evidence was obtained from the ‘Ambient’ cooler[2]; a machine of 4.3 cm^3 swept volume designed for a domestic freezer and giving 60 Watts of refrigeration at 253 K. The results of these compression loss tests give a power input for each stroke that was approximately linear with frequency over the measured range (figure 2).

Both of these sets of results are for tests carried out on complete cryocoolers, but similar results are obtained when a compressor is attached to a dummy volume. In all cases the result of the compression loss test closely obeys

$$W_e - i^2R = kf\Delta p\Delta V \qquad (1)$$

where f is the frequency, Δp is the pressure swing (peak-to-peak), ΔV is the swept volume and k is a constant for a particular machine. The value of k was found to be between 0.1 and 0.2, tending to higher values for the traditional ‘split’ Stirling cooler, and lower values for an ‘integral’ machine. When compressors have been tested into a ‘dummy volume’, the values have been lower still.

The behaviour of machines during this test is so consistent that it has been used as an accurate diagnostic tool. If, for example, the value of k increases at higher frequencies, there is invariably a pressure drop problem, and if it increases at low frequencies, then there is excessive seal leakage.

Inclusion of the ‘compression loss’ term is part of the cryocooler design process used at Oxford. Stirling cycle machines designed in this way have a measured performance very close to that predicted by the design model, and this has been found to be true over a wide range of sizes and temperatures.

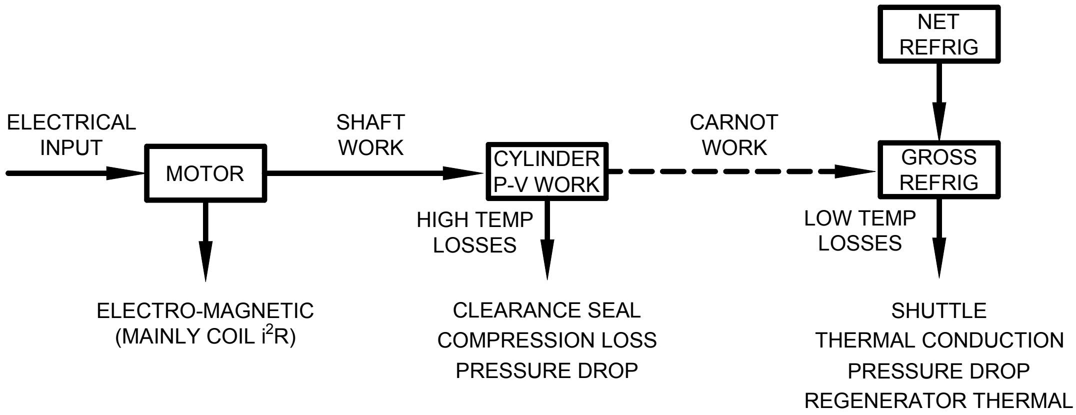

Figure 3. Schematic of Stirling Cycle Losses (simplified)

STIRLING CYCLE LOSS MODEL

There are many ways of modeling Stirling cycle machines given in the literature, mostly based on the classic Schmidt analysis. A simplified version of the model used at Oxford is given in figure 3; only the major loss mechanisms are shown.

The diagram shows 'compression loss' as a separate term from the clearance seal and pressure drop losses. Experimentally, the 'Compression Loss' test gives a lumped value for many of the high temperature losses, including windage, pressure drop, heat transfer and clearance seal losses. To separate out these losses, it is useful to look at the parametric dependency of each one.

Seal Loss

Caused by leakage through the clearance seal, the loss is conventionally given by an analytical solution for laminar flow through an idealized seal as

$$W_{seal} = \frac{\pi D t^3 (\Delta p)^2}{96 \mu L_{seal}} \qquad (2)$$

Thus for a given machine, the seal loss is a function only of the pressure swing, and is independent of frequency. Figure 4 compares the measured compression loss of the ambient cooler at 11.7 mm stroke with the theoretical (and ideal) seal leakage loss. It should be noted that imperfections in the piston, cylinder and alignment can increase the magnitude of this loss, but there is no evidence that such imperfections will change its functional dependence.

Pressure Drop Losses

The power lost is proportional either to velocity squared (for laminar flow) or to velocity cubed (turbulent flow). In a complicated geometry, the total loss will be the sum of individual pressure drop terms; laminar flow will only be present in elements such as long pipes and heat exchangers. In practice the flow entry and exit terms (essentially turbulent) often dominate. The $\frac{1}{2}\rho v^2$ terms are further complicated by the variation of density through a cycle, but with a low volumetric compression ratio, the density change is much less than the velocity change. The theoretical pressure drop term for the ambient cooler is plotted on figure 4. The calculated values plotted here are for the machine operating as a cryocooler with the displacer moving. In a 'compression loss' test the displacer is stationary, and so the mass flow through the regenerator and heat exchangers will be significantly less, hence the curve as plotted can be regarded as an upper bound for the pressure drop loss for these tests.

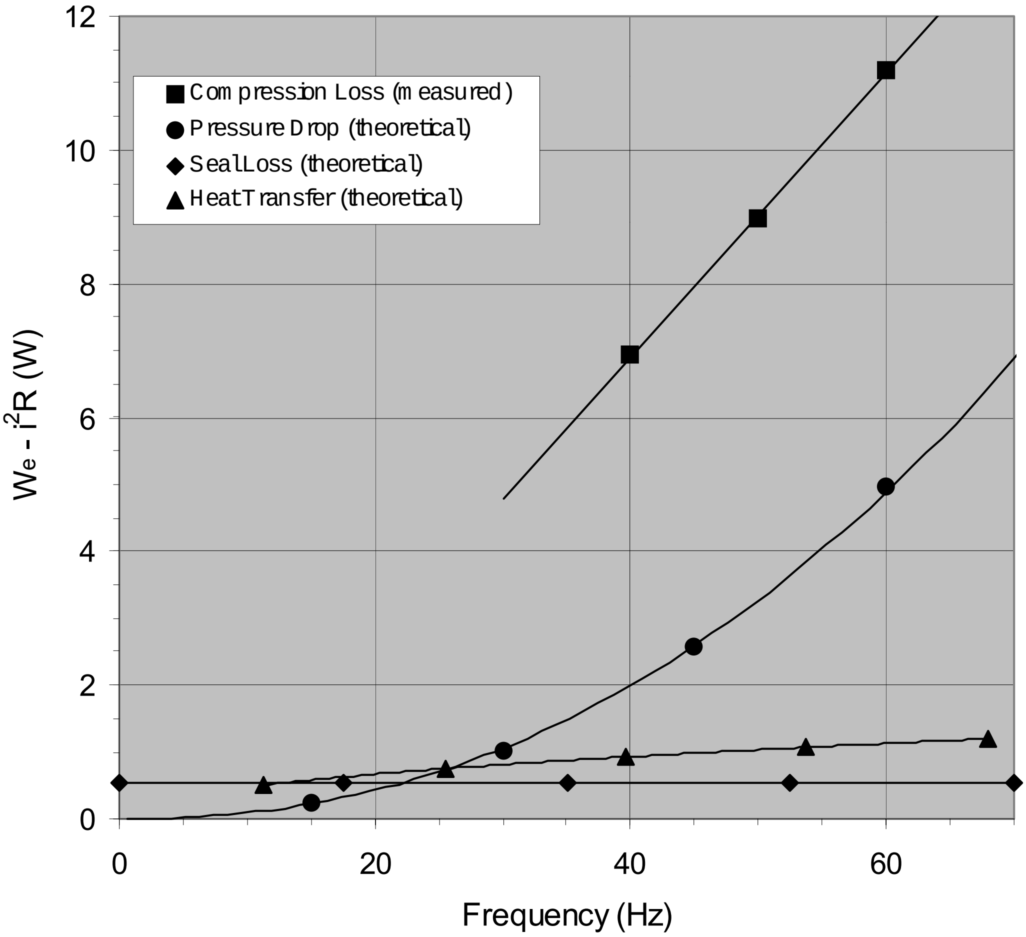

Figure 4. Measured Compression Loss for the Ambient cooler at 11.7 mm pk/pk stroke, compared with theoretical values for the expected loss mechanisms.

Heat Transfer Losses.

In machines of the size and operating frequency considered here, there is insufficient heat transfer to keep the compression and expansion processes in the compressor cylinder isothermal. Instead, a more general polytropic process must be assumed, and significant temperature swings are probable.

Various authors have produced correlations for the losses expected in the geometry of a simple cylindrical gas spring. Kornhauser & Smith[3] described experiments with a closed, conventionally sealed piston, suggesting that the losses are small and the processes are nearly adiabatic. More recently this work has been extended to include gas inflows into the cylinder[4], and concludes that there are increased losses compared to the closed geometry. In addition to flows to and from the cylinder, there is the possibility of effects due to flow through the clearance seal. Hence the equivalent losses in a complete Stirling cooler are difficult to predict and there is little scope for determining such values other than by measurement. The authors are aware of others attempting to model these losses using computer based numerical methods, but without conclusive results.

Figure 5 shows data derived from Kornhauser & Smith[3], plotted non-dimensionally, together with equivalent values taken from the ambient cooler results; the Kornhauser & Smith data is also plotted in figure 4.

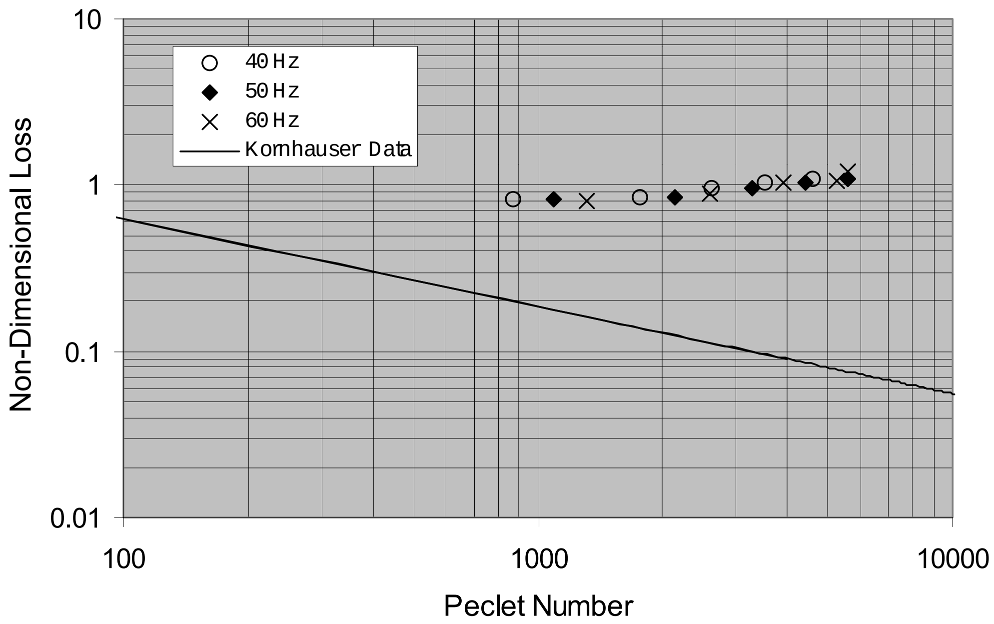

Figure 5. Compresssion Losses for the Ambient Cooler plotted non-dimensionally as a function of Peclet number. This is compared with an approximation to data from Kornhauser and Smith (ref 3, fig. 1) for the loss in a gas spring. The Peclet number is based on mean piston velocity, and the non-dimensional loss is based on the work of adiabatic compression.

Other Loss Mechanisms.

There are several other active loss mechanisms that should be taken into account: electro-magnetic losses (eddy current, magnetic hysteresis), mechanical friction, windage, thermal loss due to flow mixing, heat transfer loss 'behind the piston', etc. In various machines these losses have been measured and usually found to be small.

COMPRESSION LOSS MECHANISM

The major loss mechanisms identified for these machines have well known parametric dependencies. The results of 'compression loss' tests are inconsistent with any of these, and there are three possible explanations for this.

Firstly, it is possible that two of the mechanisms combine characteristics to give the observed behaviour which is a loss per cycle, proportional to the compressor 'P-V' area. For instance, the heat transfer loss (which decreases with increasing frequency) could combine with the pressure drop loss (increasing with frequency) to give a loss per cycle. There are two arguments against this. The expected theoretical magnitudes of these individual losses are much smaller than the observed compression loss. In addition, it seems extremely unlikely that, given the wide range of machines which exhibit this phenomenon, the two losses always *exactly* summate to give a 'loss per cycle'.

The second explanation is that it is possible that one of the other known loss mechanisms which we have judged to be insignificant is in fact much larger than we think. However, none of them appear to have the correct parametric dependency, and experimentation has so far indicated that they are small in magnitude.

The third reason, and in our view the most likely, is that there are one or more loss mechanisms active that are not properly understood or accounted for. The functional dependence gives the appearance of a thermodynamic phenomenon, with a magnitude proportional to the area of the compressor 'P-V' loop, and proportional to frequency.

CONCLUSION

A 'compression loss' is always observed in 'Oxford' type Stirling and pulse tube cryocoolers, and this is significantly higher than that predicted by existing loss mechanisms. It is postulated that this is either due to existing models of loss mechanisms being incorrect, or due to one or more unknown losses occurring in these machines. In either case data indicates behavior similar to a thermodynamic cycle, with the loss proportional to the work done on the gas in the compression space.

Work is currently under way at Oxford to try to determine the nature of these losses, in the hope that a fuller understanding of them may lead to an increase in cryocooler efficiency.

ACKNOWLEDGEMENTS

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NOMENCLATURE

D	=	Piston Diameter
f	=	Frequency
i	=	Current (rms)
k	=	Numeric constant
L_{seal}	=	Length of Clearance Seal
R	=	Coil Resistance
t	=	Seal Radial Clearance
W_e	=	Work in (electrical)
W_{seal}	=	Work lost in the Clearance Seal
Δp	=	Pressure Swing (peak-to-peak)
ΔV	=	Swept Volume
μ	=	Gas viscosity

REFERENCES

1. Orlowska, A.H., “An Investigation of some Heat Transfer and Gas Flow Problems Relevant to Miniature Refrigerators”, *DPhil Thesis*, University of Oxford, UK (1985), figure 5.6, page 5-21.
2. Green, R.H., Bailey, P.B., Roberts, L., Davey, G., “The Design and Testing of a Stirling Cycle Domestic Freezer”, *Applications for Natural Refrigerants,* International Institute of Refrigeration, Aarhus, Denmark (1996), pp. 2.11–2.15.
3. Kornhauser, A.A., Smith, J.L., "The Effects of Heat Transfer on Gas Spring Performance", *Proc. 26th Intersociety Energy Conversion Engineering Conference,* American Nuclear Society, La Grange Park, Ill., USA (1991), Vol 5. pp 180-185.
4. Catelmi, F.J., Gedeon, D., Kornhauser, A.A., "An Analytical Model for Turbulent Compression-Driven Heat Transfer", *ASME Journal of Heat Transfer,* Vol. 120 (1998), pp 617-623.