

INVESTIGATION OF SEAL PERFORMANCE IN A 4- α DOUBLE-ACTING STIRLING-CYCLE HEAT-PUMP/REFRIGERATOR

D. Haywood, Ph.D. student in Mechanical Engineering

J.K. Raine, Professor in Mechanical Engineering

M.A. Gschwendtner, Postdoctoral researcher in Mechanical Engineering

Stirling Cycle Research Group, Department of Mechanical Engineering, University of Canterbury, New Zealand.
<http://www.mech.canterbury.ac.nz/research/stirling/stirling.htm>

Abstract

Concerns about the environmental impact of refrigerants used in vapour-compression heat-pumps and refrigerators, have prompted the Stirling-Cycle Research Group at the University of Canterbury to investigate the feasibility of low-cost Stirling-cycle machines that use air as the refrigerant.

Seal development is a key part of the research programme, and a series of experiments has been carried out in order to ascertain the suitability of a system using low-cost rubbing seals. Investigations have been based on a Stirling-cycle heat-pump using a wobble-yoke kinematic mechanism in a 4- α double-acting (Siemens) configuration. An automotive-type piston arrangement was employed, using Low Wear Rate Polymer (LWRP) seal-rings.

The experiments highlighted some of the weaknesses of the 4- α double-acting configuration in terms of seal behaviour. While reasonable performance was attained, the difficulty of achieving adequate sealing during pressure reversal across the piston means that other configurations may be more suitable for Stirling-cycle heat-pump/refrigerator machines.

1. Introduction

The Stirling Cycle Research Group at the University of Canterbury has been conducting research on Stirling-cycle machinery for over a decade. The aim of the group is to do fundamental research into Stirling-cycle thermodynamics, but also to use the results of this research to develop practical machinery at a prototype level, with a view to later commercial product development. Initial work has largely focused on Stirling-cycle engines, and has resulted in the production of several prototype micro-cogeneration systems, one of which (the *WhisperGen PPS16*) has been successfully commercialised¹ by Whisper Tech Ltd², a company founded by D.M. Clucas and J.K. Raine. More recently, however, the scope of the Stirling Cycle Research Group has widened to include heat-pumps and refrigerators. This has been prompted by environmental concerns about the refrigerants currently used in conventional vapour-compression systems; all of which are either toxic, flammable, ozone-depleting, or green-house gases.

Investigations are therefore being carried out into the feasibility of low-cost medium-pressure Stirling-cycle systems using air as the refrigerant (the ultimate environmentally-friendly chemical). Experimental work to date has shown these machines to be competitive with vapour-compression systems in certain specialised situations. Currently the Stirling Cycle Research Group is focusing on development for use in domestic home-heating applications (both space heating and hot water), where Stirling-cycle machines appear to have some important performance advantages over vapour-compression systems.

A heat-pump/refrigerator experimental rig, the DH1, has been developed in order to trial system performance in various different applications and situations. This paper will concentrate on the seal development programme, which aims to identify suitable methods for using low-cost rubbing seals in Stirling-cycle heat-pump/refrigerator machines.

2. Description of the DH1 heat-pump/refrigerator experimental rig

Much of the basic development work for Stirling-cycle heat-pump/refrigerator systems can be achieved using computer simulation (the Stirling Cycle Research Group uses the *SAGE* software package). However, use of an experimental rig is essential, both to demonstrate operational feasibility in an actual working machine, and also to fine-tune the system and simulation for various effects that tend to perturb machine performance from that anticipated by analysis, i.e. seal leakage, external heat-exchanger performance, complex three-dimensional flow effects (not easily modelled), kinematic mechanism behaviour, motor characteristics, etc.

At the heart of the DH1 heat-pump/refrigerator experimental rig is a wobble-yoke (a kinematic mechanism developed by D.M. Clucas³ at the Stirling Cycle Research Group), that provides sinusoidal movement of four pistons at 90° phase spacing. The remainder of the system is based around a number of interchangeable modules, including cylinder packs (which incorporate the regenerator and heat-exchangers), piston packs, and transfer passage plates. This gives maximum flexibility to the rig allowing manipulation of cylinder bore, regenerator size and shape, piston and cylinder length, compression ratio (and amount of dead volume), heat-exchanger configuration, transfer passage geometry, operating speed, and cylinder pressure. The rig is currently set up in a 4- α double-acting configuration (also known as the Siemens arrangement⁴) as shown in Figure 2.1., but can easily be set-up in other configurations as well.

For the purposes of machine performance evaluation, a temperature-stabilised coolant and heatant liquid is circulated through the appropriate heat-exchangers, causing the Stirling-cycle heat-pump/refrigerator machine to operate continuously between two fixed temperatures. A dedicated data acquisition system then provides information on operating temperature, refrigeration and heating effects, input power, refrigeration and heating coefficients of performance, and mechanical efficiency, in real-time as the experiment is running. The temperatures at which the data are obtained are chosen in order to simulate the heat-exchanger temperatures of various environments that the machine could be operating in, e.g. indoor and outdoor heat-exchanger surface temperatures for a domestic heat-pump. Error analysis for the data is by a pseudo-Monte Carlo method.

3. Seal development programme at the Stirling Cycle Research Group

A number of different research programmes are currently being undertaken using the DH1 experimental rig (including substantial work on regenerator optimisation), however the seal development programme has produced some very interesting results while still in progress.

In the initial phase of research the Stirling Cycle Research Group has chosen to concentrate on rubbing seals (rather than clearance seals) as offering the most promise for use in low-cost heat-pump/refrigerator systems. There are three main reasons for this:

- (i) Simplicity of manufacture – the generous tolerances and straightforward shape (in comparison to a clearance seal system involving gas bearings) appears to make the piston packs in a rubbing seal system considerably easier to manufacture.
- (ii) Low cost – the low cost of materials in a rubbing seal system (combined with simplicity of manufacture) helps to minimise overall component costs, e.g. some of the completed

piston packs used in the DH1 cost less than US\$10.00 each, even when manufactured as one-off items.

- (iii) Demonstrated longevity – low wear rate polymer (LWRP) rubbing seals have successfully run for tens of thousands of hours in test engines at Whisper Tech Ltd.

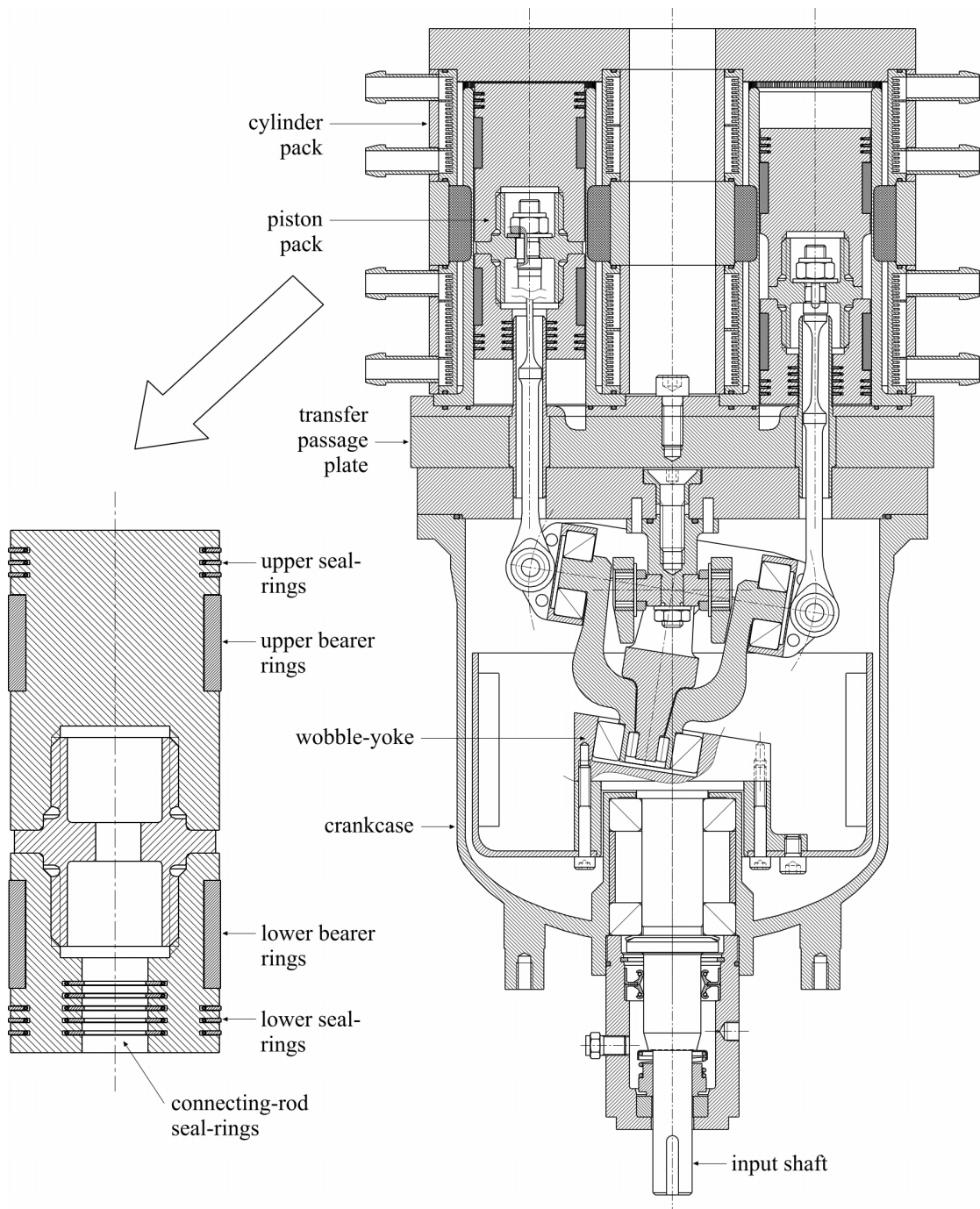


Figure 2.1. Cross-sectional general assembly drawing of DH1 heat-pump/refrigerator experimental rig in a 4- α double-acting configuration (some components removed for clarity).

It is worth noting, however, that although LWRP rubbing seals have provided very satisfactory performance in Stirling-cycle engines, there are still some reasonable grounds for doubting their performance in other contexts. Primarily, this relates to seal friction and leakage in terms of the efficiency/power compromise for the machine. In engines

(particularly in cogeneration situations), power output tends to be more important than efficiency. However, in heat-pump and refrigeration applications, the efficiency (or coefficient of performance) is of far more importance than the sheer quantity of heating or refrigeration effect. It is quite possible that the efficiency losses due to seal friction and leakage, which are acceptable in an engine, will be highly unacceptable in the context of a heat-pump or refrigerator.

4. Description of piston packs used in the DH1 experimental rig

At first glance, the piston packs used in the DH1 seem similar in layout to those used in many automotive applications; consisting of a cylindrical piston with gapped rings fitted into circumferential grooves. However, unlike automotive-type systems the materials used for both piston and rings are polymeric, with seal pre-energisation being achieved via stainless steel backing-springs. Further differences lie in the separation of the bearing and sealing functions, and in the use of seal-rings at both ends of the piston in order to minimise appendix gap losses. A drawing of the piston pack is shown in Figure 2.1.

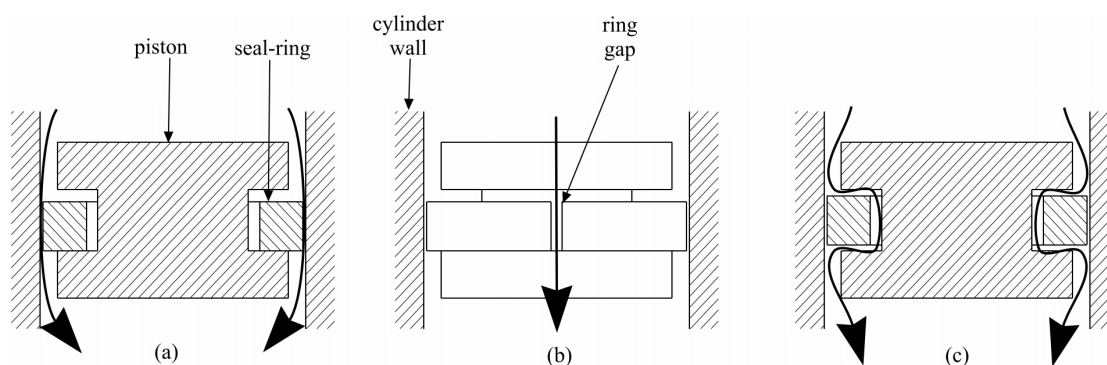


Figure 4.1. Simplified diagram of leakage paths through seal-rings: (a) wall gap leakage (b) ring gap leakage (c) pressure reversal leakage.

A comprehensive analysis of piston pack performance* in the DH1 experimental rig (and the development of seal simulation software) is the subject of another paper currently being written by the authors. However, for the purposes of clarity a brief explanatory overview of seal behaviour will be given here.

The overall mechanisms for seal friction and leakage in the piston packs are the combination of several different sub-processes. The sliding movement of the piston within the cylinder produces frictional losses in two ways:

- (i) Side-load friction – this is the result of piston side-loading (very small in the case of a wobble-yoke mechanism) and occurs between the bearer-rings and cylinder wall.
- (ii) Sealing friction – this is a consequence of seal loading both by pre-energisation and gas pressure, and occurs between the seal-rings and cylinder wall.

Gas leakage through the piston pack occurs via three main pathways (see Figure 4.1.):

- (i) Wall gap – as the piston slides within the cylinder a certain amount of leakage occurs between the rubbing face of the seal and the cylinder wall.
- (ii) Ring gap – leakage takes place through the ring gap whenever there is a pressure difference across the piston.

* There is not a great deal in the literature about seal behaviour in 4- α double-acting Stirling-cycle machines, however a description of the relationship between gas leakage and movement of a piston ring within a groove (in the context of the Stirling Cycle) is given by Tanaka and Yamashita⁵. Further useful information can be found in classic papers on automotive seal behaviour such as Namazian and Heywood⁶.

- (iii) Pressure reversal – quite large amounts of leakage occur when the pressure across the piston is reversed. This takes place twice per cycle in the 4- α double-acting layout, and causes the ring to jump from one side of the ring-groove to the other, momentarily producing a very significant leakage path and allowing a “puff” of gas to pass behind the seal.

It is extremely important to understand the interconnectivity of the separate cycles in the 4- α double-acting layout. This means that the leakage path from the cylinders is not only to the crankcase (as in single-acting layouts), but is also between the different cycles themselves, i.e. leakage from any given cycle may not only affect the work produced by that cycle, but may also interfere with the performance of the two adjacent cycles.

This can lead to some very counterintuitive effects for those used to thinking in terms of single-acting machine configurations. For example, benchmark seal testing was performed on the DH1 experimental rig with no seal-rings fitted at all, i.e. the only sealing (per se) was via the large clearance gap between the piston and cylinder. According to single-acting piston behaviour, it would be expected that the compression work would be significantly reduced, leading to a comparable reduction in heating effect. In fact, the heating effect changes only moderately, while the net input power required to drive the cycle increases by over three hundred percent.

This surprising outcome is the result of the manner in which the various cycles interact in the 4- α double-acting configuration, giving rise to a slight inward leakage of gas at the start of compression, and a maximum pressure difference across the pistons driving an outward leakage of gas just prior to the expansion part of the cycle. Therefore leakage does not begin to occur until the majority of the compression work has already been performed, while on the other hand, very significant leakage occurs right at the start of the expansion process, greatly reducing the exergy of the working gas. The consequence of these effects is that compression work remains largely unchanged, while the expansion work is severely reduced. Because of the low work ratio of the Stirling Cycle this has a very profound proportional effect on the amount of input work required to drive the machine.

5. Some results from the seal development programme

As mentioned earlier in this paper, there were a number of concerns about the use of rubbing seals in the context of a Stirling-cycle heat-pump/refrigerator (see Section 3). It was felt that the sealing required to produce an acceptably high indicated coefficient of performance in the 4- α double-acting configuration, might also produce unacceptably high losses in seal friction; leading to severely reduced mechanical efficiency and low overall brake thermal performance. This was based on the assumption that multiple seals would be required, and the resulting trade-off between gas leakage and frictional losses in this context, i.e. that gas leakage is reduced by increasing the number of seals, while conversely, frictional losses are increased.

A series of basic experiments using a range of piston seal configurations was carried out in order to investigate these concerns. The experiments were performed using the DH1 experimental rig in a heat-pump configuration, with heat-exchanger surface temperatures chosen to simulate that of a system operating with a 20°C indoor temperature, and two outdoor temperature set-points at 5°C and –5°C. Five separate tests were performed for each set-up in order to ascertain consistency of results, with the final data representing the mean values obtained.

For the purposes of this paper, only some of the results are shown, namely those experiments consisting of piston packs with 0, 1, 2, 3, 4, and 5 seals. Seal-rings were fitted to both ends of the piston for all tests using more than one seal. The data for mechanical efficiency have been modified to eliminate the effects of the crankcase shaft seal, as this would not be employed in an actual heat-pump system.

Graphs of heating effect and indicated input power with variation in the number of piston seals are shown in Figure 5.1., and graphs of heating coefficient of performance and mechanical efficiency are shown in Figure 5.2. Mean value trend lines are shown for clarity but should not be used for interpolation; the error bars represent 95% certainty.

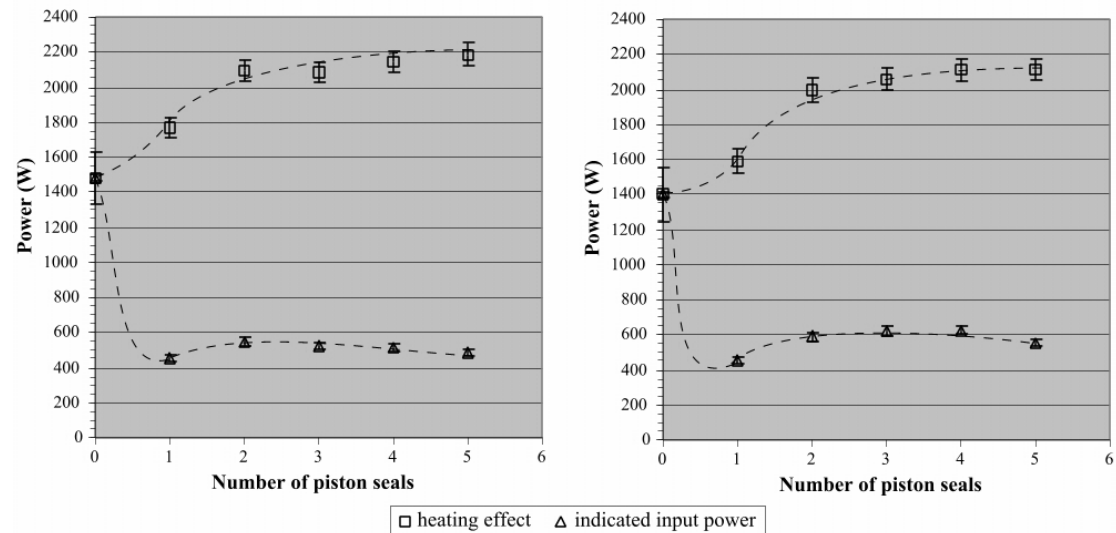


Figure 5.1. Heat-pump heating effect and indicated input power with variation in the number of piston seals at simulated outdoor temperatures of 5°C (left) and -5°C (right).

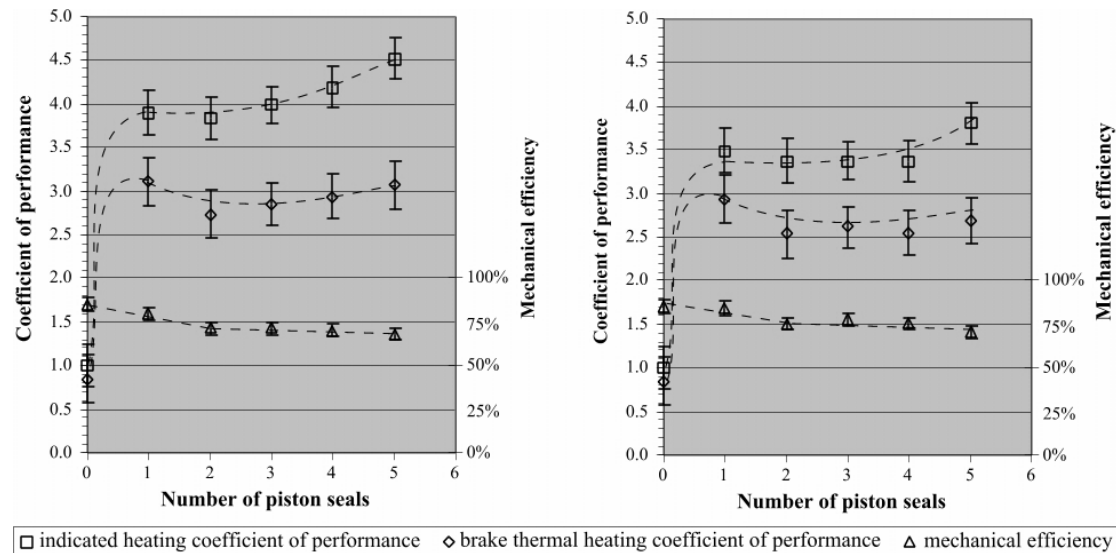


Figure 5.2. Heat-pump indicated heating coefficient of performance, brake thermal heating coefficient of performance, and mechanical efficiency with variation in the number of piston seals at simulated outdoor temperatures of 5°C (left) and -5°C (right).

Inspection of Figure 5.1. and Figure 5.2. reveal a number of interesting trends:

- (i) Mechanical efficiency – this declines quite sharply between zero and two seals, but then appears to decrease only moderately as extra seals are added.
- (ii) Indicated heating coefficient of performance – the subtle performance changes and the span of the error bars make it difficult to comment on this data, however for the 5°C results there is a significant increase in coefficient of performance over the range of the tests, i.e. between one seal and five seals. The changes in performance are less marked for the –5°C set of results.
- (iii) Heating effect – this increases significantly between one and two seals, but then appears to make only moderate gains (if any) as further seals are added.
- (iv) Indicated input power – this shows a small increase between one and two seals, and then an apparent decrease (or possibly a levelling off) with additional seals.

Examination of the mechanical efficiency data suggests that the trend exhibited is mostly likely due to uneven seal-ring loading, with the seals nearest the high pressure end of the piston taking the brunt of the sealing force. This hypothesis was confirmed by inspection of the piston pack, which showed the seal rings at the extreme ends of the piston to be most heavily worn, thus indicating highest loading. Additional seals are therefore only lightly loaded, causing less friction, and consequently less deterioration of mechanical efficiency.

With only a few of the seal-rings significantly loaded, it therefore seems surprising that indicated coefficient of performance reaches its peak value with five seals fitted to the piston pack. It appears likely, however, that this is due to the effects of pressure reversal leakage. During most of the piston movement it would seem that only two seals are required to control wall gap and ring gap leakage, but at the point of pressure reversal (as the seals move from one side of the ring groove to the other) an extra, and much larger, leakage path is opened. Additional rings serve to increase flow resistance during the pressure reversal process, thus reducing gas leakage and increasing coefficient of performance. The more pronounced leakage effects shown by the –5°C set of results are probably a reflection of the greater difference in working gas internal energy across the piston pack.

The apparent trends exhibited by the heating effect and indicated input power between two and five seals are the anticipated outcome of improved sealing, although some of the increase in heating effect may be due to greater seal friction. The large improvement in heating effect (nearly 400W) between one and two seals, is probably due (at least in part) to the reduction of appendix gap losses through use of seals at both ends of the piston. The reduced value for indicated input power with one seal is intriguing, and the causes of this are still under investigation.

The span of the error bars makes definitive statements about brake thermal heating coefficient of performance rather difficult, with the gain in indicated performance from reduced leakage appearing to balance out the extra seal friction quite evenly. There is perhaps a tenuous indication that superior brake thermal coefficient of performance is attained with only one seal, although at the cost of a severely reduced heating effect. On the other hand, only slightly lower coefficient of performance is attained with five seals, producing the maximum value of heating effect recorded. Intermediate numbers of seals seem to produce less satisfactory results. It is, of course, interesting to speculate about how performance will change when even more seals are fitted to the piston pack.

It should be noted that these experiments were performed when the DH1 experimental rig was fitted with engine-style regenerator packs. Further performance gains are expected as a result of the regenerator optimisation programme for heat-pump/refrigerator applications. However, the results obtained to date are, at least, in the ball-park of comparable vapour-compression machine performance.

6. Conclusions

The results obtained suggest that reasonable heating effect and coefficient of performance may be achieved by Stirling-cycle heat-pumps fitted with low-cost rubbing seals.

From the experiments performed, three specific conclusions could be drawn with regard to rubbing seals in the 4- α double-acting configuration:

- (i) Mechanical efficiency reduced as the number of seal-rings was increased, however, for high seal numbers this effect was tempered by the fact that only a few seals carried the majority of the pressure load.
- (ii) Indicated heating coefficient of performance was highest with the maximum number of seals fitted to the piston pack. This was attributed to the reduction of seal leakage, and in particular, a decrease in leakage during pressure-reversal across the piston.
- (iii) The change in brake thermal heating coefficient of performance was relatively subtle with variation in the number of seal-rings, as any gains in indicated performance tended to be counteracted by reduced mechanical efficiency.

It has not escaped our notice that elimination of pressure reversal across the piston could result in significant reductions in leakage while requiring fewer seal-rings, and may lead to improved brake thermal coefficient of performance. Development work on other four-piston configurations (which are less susceptible to pressure reversal leakage effects) is currently underway at the Stirling Cycle Research Group.

7. References

1. D.M. Clucas and J.K. Raine, "The WhisperGen: a strategy for commercial production", *Proceedings of the 8th International Stirling Engine Conference and Exhibition*, pp 159-168.
2. <http://www.whispertech.co.nz>
3. D.M. Clucas and J.K. Raine, "A New Wobble Drive with particular application in a Stirling Engine", *Journal of Mech. Eng. Sci., Proc. of Inst. of Mech. Engrs., Part C*, vol. 208 (1994), pp 337-346.
4. C.D. West, *Principles and applications of Stirling Engines*, Van Nostrand Reinhold Co., New York, 1986, p.64.
5. A. Tanaka and I. Yamashita, "Relation between gas leakage and movement of piston ring within groove", *Proceedings of the 4th International Stirling Engine Conference*, pp 271-275.
6. M. Namazian and J.B. Heywood, *Flow in the Piston-Cylinder-Ring Crevices of a Spark-Ignition Engine: Effect on Hydrocarbon Emissions, Efficiency, and Power*, SAE Technical Paper Series 820088, Detroit, 1982.