Stirling-Cycle Heat-Pumps and Refrigerators – a Realistic Alternative?

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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. Such machines theoretically have the highest efficiency possible for any practical thermodynamic system, and thus provide a tempting alternative to traditional vapour-compression technology. This paper outlines the working principles of Stirling-cycle heat-pumps and refrigerators, and describes some of the work performed at Canterbury University. Some of the heat-pump development programme results are also presented, and briefly discussed.

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 these systems has been successfully commercialized (Clucas and Raine 1997 159-168) by Whisper Tech Ltd (Whisper Tech 2002 1), a company founded by D. Clucas and J. Raine. Whisper Tech Ltd is now a multi-million dollar company that designs and manufactures Stirling micro-cogeneration systems in Christchurch, New Zealand.

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 (also known as reverse-Rankine-cycle) systems; all of which are either toxic, flammable, ozone-depleting, or green-house gases.

Investigations are therefore being carried out into the feasibility of Stirling-cycle heat-pumps and refrigerators which incorporate some of the ground-breaking technology that has been developed for engine applications. Primarily this work has concentrated on low-cost medium-pressure Stirling-cycle systems that use air as the refrigerant (the ultimate environmentally-friendly chemical).

2) WORKING PRINCIPLES OF STIRLING-CYCLE HEAT-PUMPS AND REFRIGERATORS

In its simplest form the Stirling-cycle refrigerator or heat-pump uses exactly the same components as the Stirling-cycle engine (see Figure 1.); albeit for slightly different purposes.

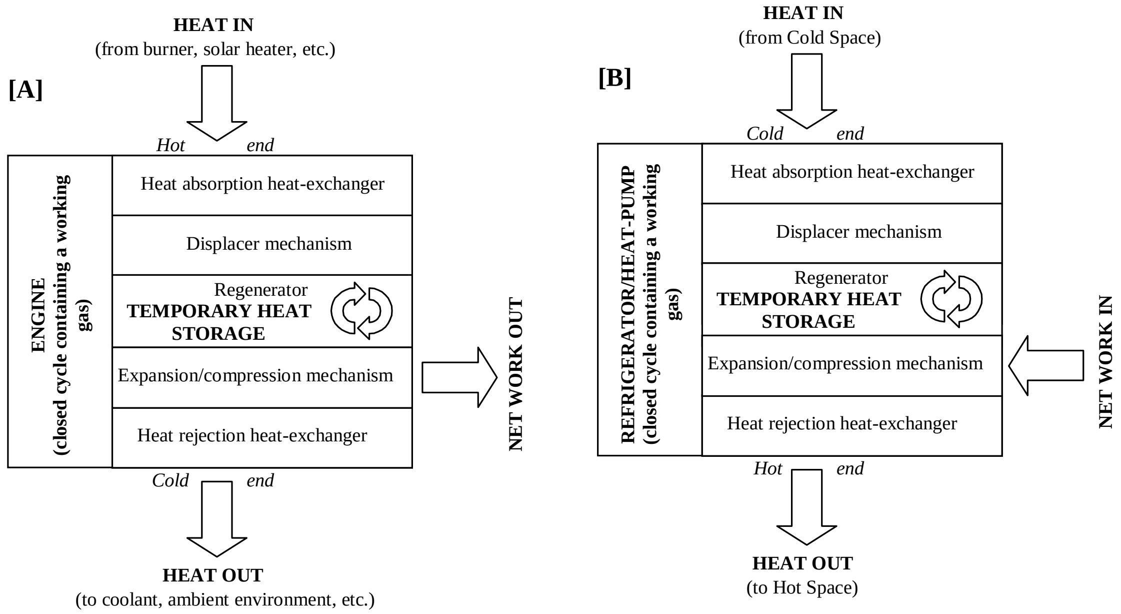

Figure 1. Stirling-cycle machine block diagrams: (A) Engine (B) Refrigerator or heat-pump.

There are five main components in a Stirling-cycle machine, as shown in Figure 1.

(a) *Working gas* – the Stirling Cycle is a closed cycle and the various thermodynamic processes are carried out on a working gas that is trapped within the system.

(b) *Heat-exchangers* – two heat exchangers are used to transfer heat across the system boundary. A *heat absorbing heat-exchanger* transfers heat from outside the system into the working gas, and a *heat rejecting heat-exchanger* transfers heat from the working gas to outside the system. For example, on a refrigerator the heat absorbing heat-exchanger would transfer heat from the cold space into the working gas, and the heat rejecting heat-exchanger would transfer heat from the working gas to the ambient environment.

(c) *Displacer mechanism* – this moves (or displaces) the working gas between the hot and cold ends of the machine (via the regenerator).

(d) *Regenerator* – this acts both as a thermal barrier between the hot and cold ends of the machine, and also as a "thermal store" for the cycle. Physically a regenerator usually consists of a mesh material (household pot scrubbers have even been used in some engines), and heat is transferred as the working gas is forced through the regenerator mesh. When the working gas is displaced from the hot end of the machine (via the regenerator) to the cold end of the machine, heat is "deposited" in the regenerator, and the temperature of the working gas is lowered. When the reverse displacement occurs, heat is "withdrawn" from the regenerator again, and the temperature of the working gas is raised.

(e) *Expansion/compression mechanism* – this expands and/or compresses the working gas. In an engine this mechanism produces a net work output. In a refrigerator or heat-pump a net work input is required to move the heat from a low to a high temperature regime (in accordance with the Second Law of Thermodynamics).

A Stirling-cycle machine can be constructed in a variety of different configurations. For example, the expansion/compression mechanisms can be embodied as turbo-machinery, a piston-cylinder, or even using acoustic waves. Most commonly, Stirling-cycle machines use a piston-cylinder, in either an α, β, or γ configuration. An α-configuration machine uses two pistons which displace and expand/compress the gas at the same time. β and γ-configuration machines have a separate displacer-piston and expansion/compression piston (usually called a power-piston).

In an ideal Stirling-cycle refrigerator or heat-pump the components of the machine interact to produce four separate thermodynamic processes. These processes are illustrated using a simplified α-configuration machine in Figure 3., and are shown on pressure-volume and temperature-entropy diagrams in Figure 2. It should be noted that for the ideal Stirling Cycle the heat-exchangers, regenerator, and transfer passages are assumed to have zero volume.

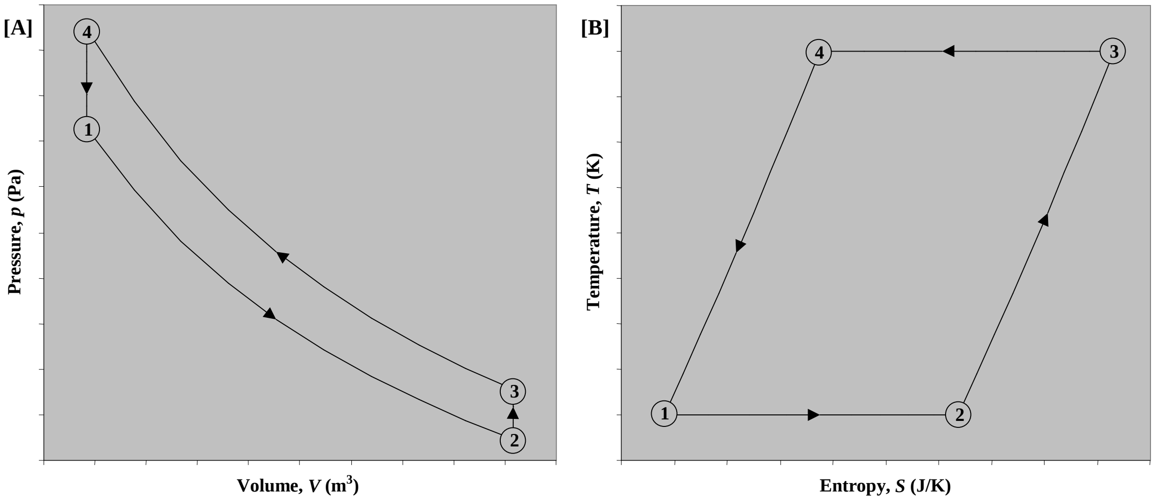

Figure 2. Thermodynamic processes in the ideal Stirling-cycle refrigerator or heat-pump. (A) Pressure-volume diagram. (B) Temperature-entropy diagram.

1 → 2 : Isothermal expansion – the low-pressure working gas expands isothermally at cold end temperature, hence absorbing heat from the cold space (via the heat absorbing heat-exchanger) and doing work to the expansion piston.

2 → 3 : Isochoric displacement – both pistons move together to transfer all the working gas isochorically through the regenerator to the hot end of the machine. Heat is delivered to the gas as it passes through the regenerator, thus raising the temperature of the gas to that of the hot space. As the temperature rises, the gas pressure increases significantly.

3 → 4 : Isothermal compression – the compression piston does work to the gas and compresses it isothermally at hot end temperature, hence rejecting heat to the hot space (via the heat rejecting heat-exchanger). Because the gas is at high pressure, more work is required for compression than was obtained from the gas during expansion (in 1→2). The cycle therefore has a net work input.

4 → 1 : Isochoric displacement – both pistons move together to transfer all the working gas isochorically through the regenerator to the cold end of the machine. Heat is absorbed from the gas as it passes through the regenerator, thus lowering the temperature of the gas to that of the cold space. As the temperature reduces, the gas pressure drops significantly, and the system returns to its initial conditions (at 1).

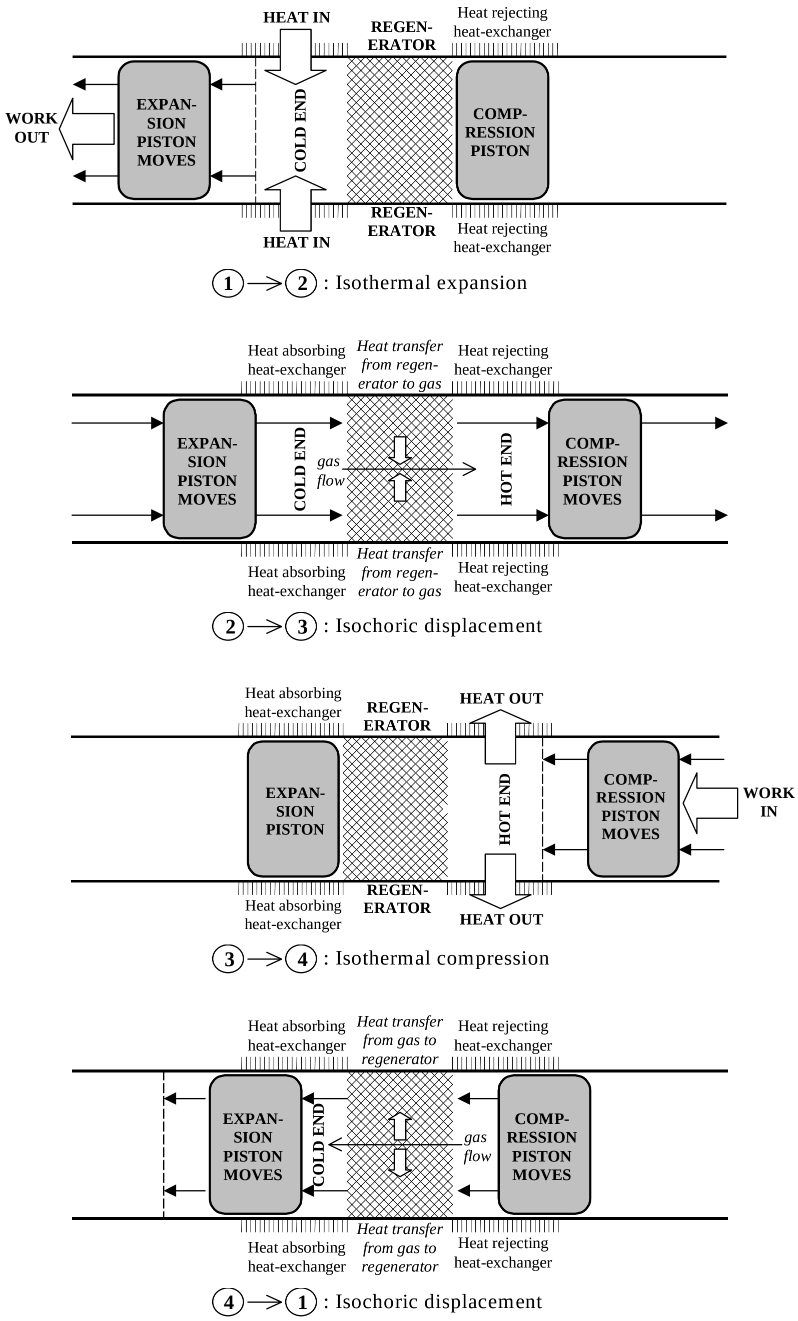

Figure 3. Thermodynamic processes in the ideal Stirling-cycle refrigerator or heat-pump as shown on a simplified α-configuration machine.

3) ANALYSIS OF THE STIRLING CYCLE

The work, heat transferred, and co-efficient of performance (COP) of an ideal Stirling-cycle heat-pump or refrigerator can be readily evaluated from first principles using the pressure-volume and temperature-entropy diagrams shown in Figure 2. i.e. from the definitions of:

work, $W = -\oint p\mathrm{d}V$ (J); heat transfer, $Q = \int T\mathrm{d}S$ (J); and COP, $\chi = \frac{-Q}{W}$;

then the following formulae can be developed for an ideal Stirling-cycle machine:

$$W = mR\ln\left(\frac{V_2}{V_1}\right)(T_H - T_L)$$

$$Q_H = -mRT_H \ln\left(\frac{V_2}{V_1}\right),\; Q_L = mRT_L \ln\left(\frac{V_2}{V_1}\right)$$

$$\chi_{H\,STIRLING} = \frac{T_H}{T_H - T_L},\; \chi_{L\,STIRLING} = \frac{T_L}{T_H - T_L}$$

where m is gas mass (kg), R is specific gas constant (J/kgK), and where the subscripts H and L denote the high and low temperature isotherms respectively.

Inspection of the expressions for Stirling Cycle COP reveals the interesting fact that:

$$\chi_{H\,STIRLING} = \chi_{H\,CARNOT} \text{ and } \chi_{L\,STIRLING} = \chi_{L\,CARNOT}$$

or, in other words, that the Stirling-cycle heat-pump or refrigerator has the maximum ideal COP possible under the Second Law of Thermodynamics.

Unfortunately, practical Stirling-cycle heat pumps and refrigerators differ from the ideal cycle in several important aspects:

(i) The regenerator and heat-exchangers in practical machines have non-zero volume. This means that the working gas is never completely in either the hot or cold end of the machine, and therefore never at a uniform temperature.

(ii) The piston motion is usually semi-sinusoidal rather than discontinuous, leading to non-optimal manipulation of the working gas.

(iii) The expansion and compression processes in practical Stirling-cycle machines are polytropic rather than isothermal. This causes temperature fluctuations in the working gas and leads to adiabatic and transient heat transfer losses.

(iv) Fluid friction losses occur during gas displacement, particularly due to flow through the regenerator.

(v) Other factors such as heat conduction between the hot and cold ends of the machine, seal leakage and friction, appendix gap effects, and friction in kinematic mechanisms all cause real Stirling-cycle machines to differ from ideal behaviour.

The combination of these factors means that the Stirling Cycle differs from its ideal embodiment more than any other thermodynamic cycle; making it extremely difficult to develop an accurate mathematical model to predict system performance. On the other hand, the vast number of critical parameters that can be varied on a practical machine mean that experimental optimization is enormously time-consuming (and expensive).

Fortunately, cheap computing power has meant that machine simulations can be built up from an analysis of processes at a very detailed level. Initial modeling is done at a sub-component level, with the various processes being evaluated using standard thermodynamic and fluiddynamic formulae. These sub-components are assembled mathematically to form components, and the components are assembled to form a complete Stirling-cycle machine. The equations resulting from such simulations do not, of course, have closed-form solutions; but they can be solved using numerical iteration methods.

This “brute force” approach has been used successfully in several software packages that are now commercially available (Gedeon 1996, Thomas 2002 67-74). These incorporate hundreds (or even thousands) of different input variables to represent the geometry and materials of a Stirling-cycle machine. While this software is extremely expensive, the results produced are comparatively accurate. Therefore much of the optimization process can be performed in software, thus greatly reducing the experimental refinement required.

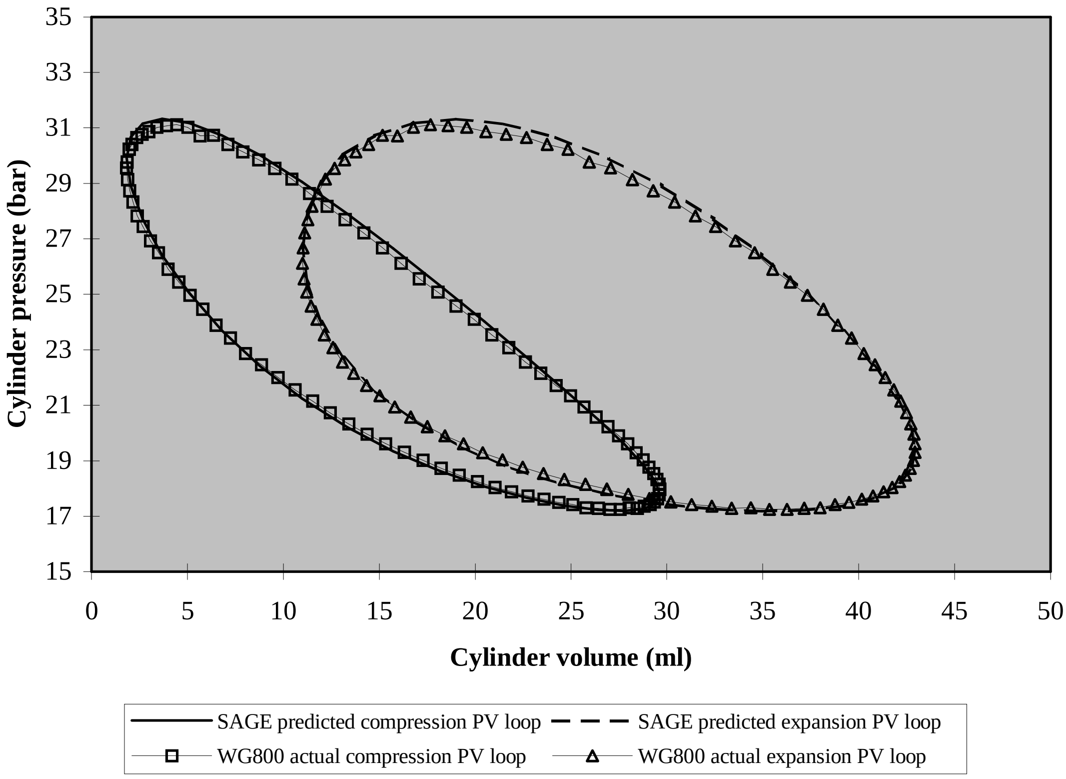

Figure 4. Actual and *SAGE* predicted WG800 cylinder pressure with variation in volume.

Some typical results from an analysis of a Whisper Tech Ltd engine using such a software package are shown in Figure 4. Further information about the Stirling Cycle, its practical limitations, and analysis can be found in West (West 1986) and Wurm (Wurm et al 1991 75-86).

4) STIRLING CYCLE RESEARCH AT THE UNIVERSITY OF CANTERBURY

A number of different research programmes are currently being undertaken by the Stirling-Cycle Research Group at the University of Canterbury. These include a Stirling-cycle heat-pump development programme, and a Stirling-cycle refrigerator development programme. Additionally there are two main areas of specific component research: the seal modeling and development programme, and the regenerator development programme.

Apart from the system simulation and seal modeling work, which are mathematical analysis-based projects, the majority of the research performed by the Stirling-Cycle Research Group is experimental in nature. Two main experimental rigs are used for this work: the DH1 heat-pump/refrigerator experimental rig, and the MAG1 regenerator test rig.

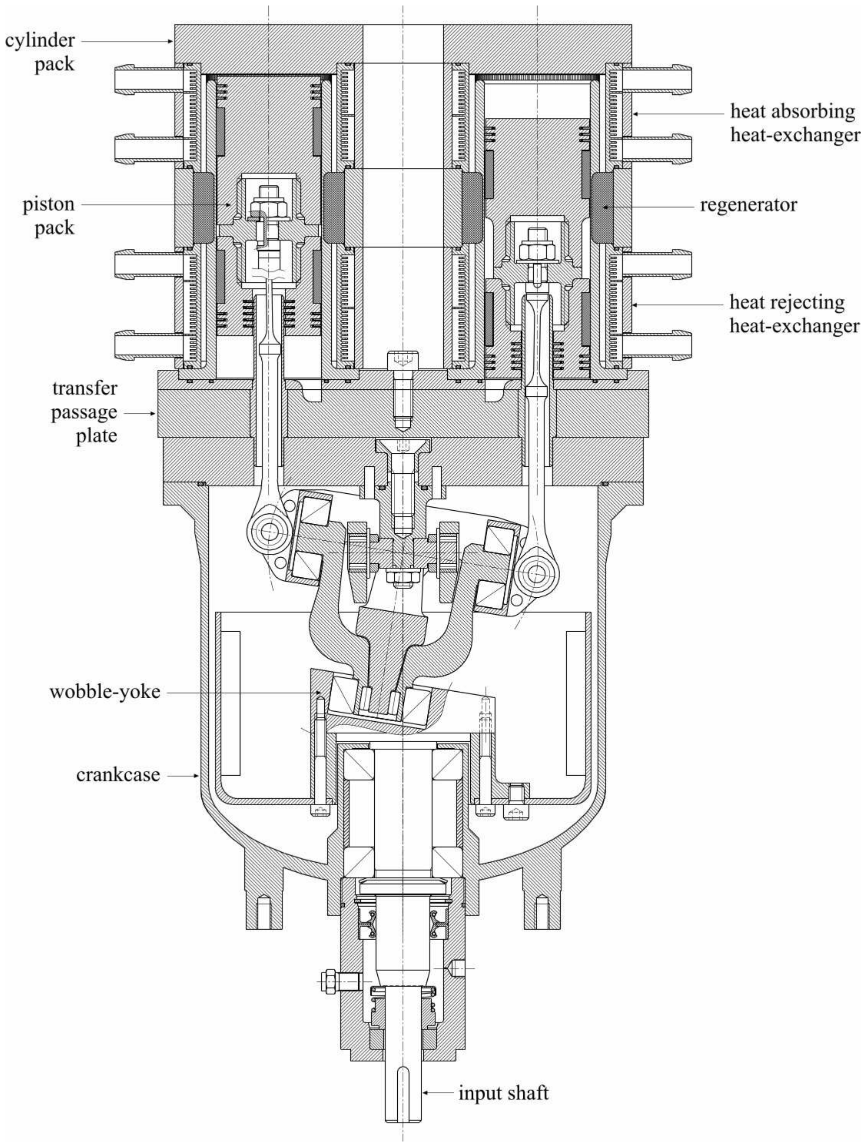

Figure 5. Cross-sectional general assembly drawing of DH1 heat-pump/refrigerator experimental rig in a 4-α double-acting configuration (some components removed for clarity).

At the heart of the DH1 heat-pump/refrigerator experimental rig is a wobble-yoke (a kinematic mechanism developed by D.M. Clucas (Clucas and Raine 1994 337-346) in 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 arrangement) as shown in Figure 5., but can easily be set-up in other configurations as well.

The MAG1 regenerator test rig is a single-cycle α-configuration machine based around two slider-crank mechanisms, as shown in Figure 6. Gas temperature and pressure are monitored within the regenerator matrix, and materials are tested by varying the package geometry and measuring regeneration efficiency.

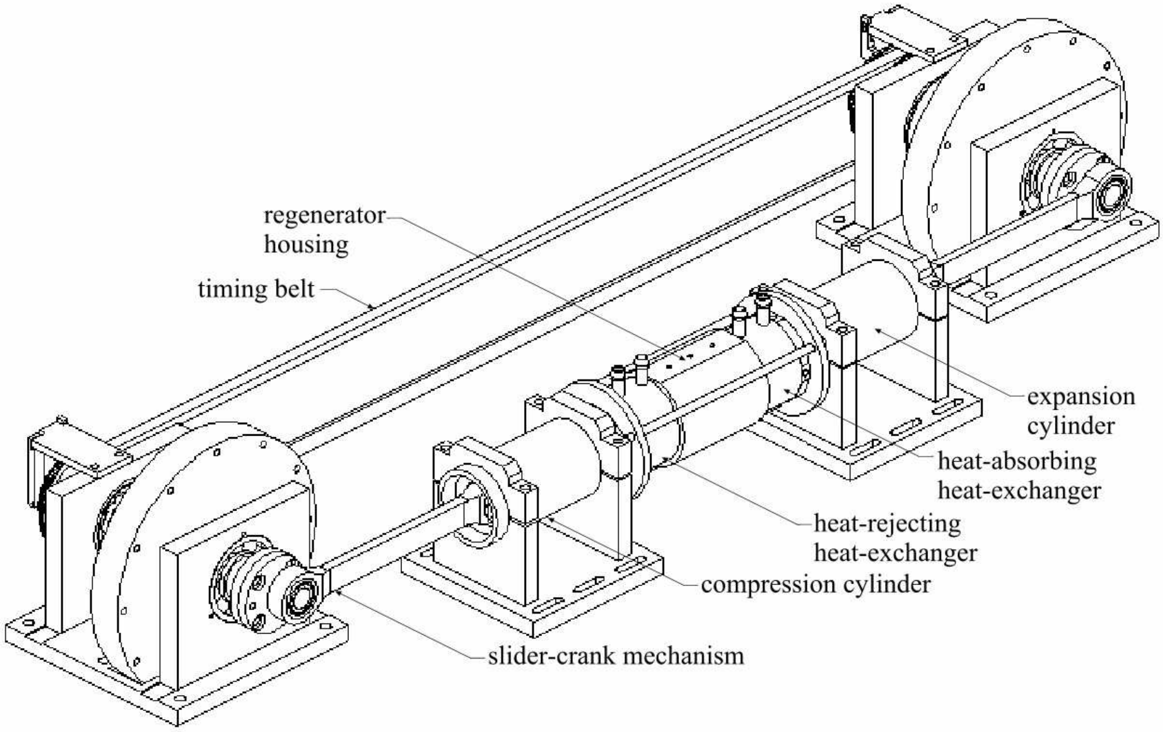

Figure 6. General assembly drawing of MAG1 regenerator test rig.

In both experimental rigs, overall system performance is evaluated by circulating a temperature-stabilized coolant and heatant liquid through the appropriate heat-exchangers, causing the Stirling-cycle 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.

Manufacturing cost is a major barrier to the introduction of Stirling-cycle heat-pumps and refrigerators, particularly when competing against the very large-scale production of vapour-compression technology. The Stirling-Cycle Research Group are therefore investigating potential niche applications in small markets where Stirling-cycle machines offer improved performance at similar cost to vapour-compression systems, e.g. low mean-outdoor-temperature heat-pump applications, very low temperate freezer (below -35°C) applications, CO_2 storage and transportation equipment, and high-speed bottle/can chillers.

5) SOME RESULTS FROM THE HEAT-PUMP DEVELOPMENT PROGRAMME

Preliminary work from the Stirling-cycle heat-pump development programme indicated superior performance (compared to vapour-compression systems) in certain situations. This is mainly associated with a lower performance droop as the hot and/or cold heat-exchangers varied from their design temperatures. Some results from this work are presented in Figure 7.

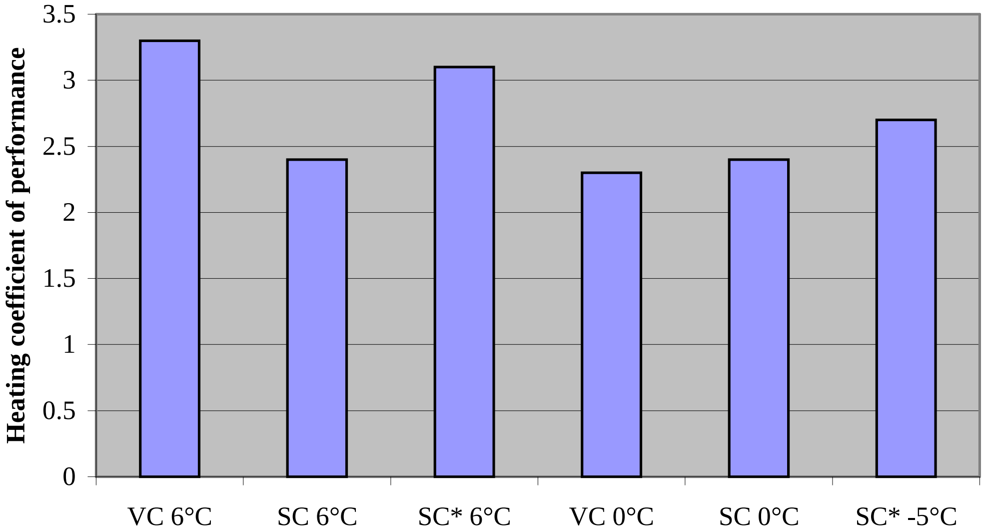

Figure 7. Some results from the Stirling-cycle heat-pump development programme. VC = typical vapour compression machine, SC = DH1 prior to seal development programme, SC* = DH1 after seal development programme. The temperature stated is the outdoor temperature. The indoor temperature in all cases is 20°C.

It can be seen from the results obtained that (following the seal development programme) the heating COP of the Stirling-cycle machine at 6°C outdoor temperature is only very slightly less than that of a typical vapour-compression system. However, at 0°C outdoor temperature the vapour-compression machine has considerable poorer COP than the Stirling-cycle system, even though the Stirling-cycle machine is operating at an even lower outdoor temperature of -5°C. For situations where there are large diurnal temperature swings during winter this could imply a greater net COP for Stirling-cycle systems. There are also other advantages for Stirling-cycle machines in such contexts.

The results prior to the seal development programme are included in order to demonstrate the importance of seal design in Stirling-cycle machinery. Further information on the seal development programme can be found in the proceedings of the International Stirling Engine Conference 2001 (Haywood, Raine, and Gschwendtner 2001 181-188).

6) CONCLUSIONS

As well as their role as high-efficiency low-noise engines, Stirling-cycle machines offer an environmentally-friendly alternative to vapour-compression systems, by virtue of their ability to use air as a refrigerant (the ultimate environmentally-safe chemical).

In comparison to other thermodynamic cycles, the Stirling Cycle is extremely difficult to optimize for a practical machine. When optimized, however, Stirling-cycle machines have similar performance to traditional vapour-compression systems; although there are certain contexts where they can, in fact, provide significantly higher performance. It can therefore be said that Stirling-cycle machines provide a technically realistic alternative to current vapour-compression technology. The main challenge will lie in achieving a competitive price through a low-cost mechanical design configuration, and manufacturing in sufficiently large volumes.

The Stirling-Cycle Research Group at Canterbury University are working on the development of low-cost high-performance machines. Initially these systems will probably be introduced in niche applications for small markets, where they can offer improved performance at similar cost to vapour-compression systems.

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