A SHOT IN THE COLD: A NEW OPEN REGENERATIVE CYCLE FOR HEAT-PUMPS AND REFRIGERATORS

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ABSTRACT

Although Stirling heat-pumps and refrigerators offer the possibility of environmentally-friendly operation by using air as the refrigerant, the Coefficient of Performance (COP) of low-cost Stirling systems is only about the same as that of conventional vapour-compression technology. For this reason, Stirling machines have been perceived as lacking the significant advantages required to supplant vapour-compression systems in heat-pump and refrigerator applications. This paper discusses a new open regenerative cycle for heat-pumps and refrigerators that offers similar performance to practical Stirling systems. Additionally, it eliminates the need for a heat absorbing heat-exchanger (the most costly component of conventional vapour-compression heat-pumps), and has inherent frost-free operation (frosting significantly degrades the performance of conventional vapour-compression heat-pumps at low temperatures). An experimental prototype has been designed by modifying the MAG1 Regenerator Test Rig. *Sage* simulations of the experimental prototype suggest that an indicated Heating COP of at least 3.4 may be possible at the standard heat-pump rating temperatures of 20ºC (hot space) and 5ºC (cold space). Although the new open regenerative cycle has low Heating and Refrigeration Effect per unit volume, it appears to offer a number of potential advantages over both Stirling and conventional vapour-compression machines in the context of small systems.

1.0 INTRODUCTION

The Stirling Cycle Research Group at the University of Canterbury has worked with Whisper Tech Ltd on a variety of projects involving Stirling engines, refrigerators, and heat-pumps. Over the last five years, the Stirling Cycle Research Group has successfully produced both heat-pump and near-ambient temperature refrigerator prototypes that operate according to the Stirling Cycle. Although these machines offer genuinely environmentally-friendly operation (by virtue of using air as the working fluid), the COP achieved for low-cost design has been only slightly better than for conventional vapour-compression technology [1]. The prototypes have therefore been perceived as lacking the significant advantages required to supplant conventional systems in the heat-pump or refrigerator contexts, and this has necessitated the investigation of other, more innovative, approaches to the design of environmentally-friendly air-cycle machines.

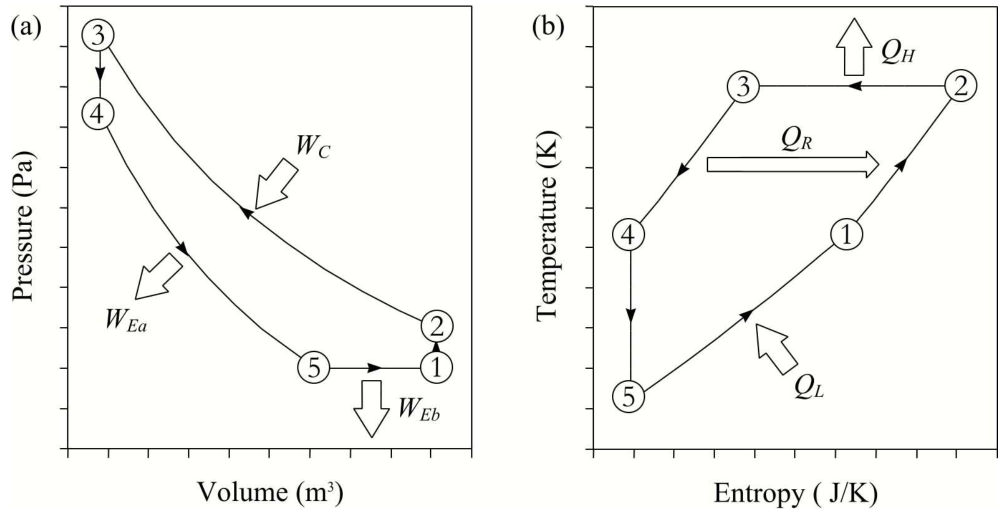

Figure 2.1: Thermodynamic diagrams (a) pressure volume and (b) temperature-entropy

The title of this paper alludes to Organ's 2001 ISEC presentations on a regenerative air-scavenged engine [2]. The Stirling Cycle Research Group at Canterbury has also been investigating open regenerative cycles, albeit for entirely different reasons and applications. Our work on open cycle systems has since 1999 been seeking a suitable configuration for heat-pumps and near-ambient temperature refrigerators. A prototype machine is now being constructed to investigate the open regenerative cycle proposed by Haywood [3]. The operation of this machine is thermodynamically similar to the Stirling Cycle, except that cold space heat exchange is accomplished via a scavenging stroke rather than a heat-exchanger. This has some significant advantages in comparison with conventional vapour-compression technology:

(a) It uses environmentally-friendly air as the working fluid (as per Stirling machines).
(b) It removes the need for a heat absorbing heat-exchanger (the most costly part of a conventional vapour-compression heat-pump) and blower, and also eliminates the temperature drop across the heat-exchanger (which is in itself an important cause of reduced performance) .
(c) It eliminates the frost problems associated with air heat-exchange systems, which (particularly in heat-pumps) significantly degrade performance at cold space temperatures less than approximately 6°C.

This paper describes the ideal thermodynamic operation of the new open regenerative cycle as well as the working principles of a practical embodiment, and the design and simulation results of an experimental prototype.

2.0 DESCRIPTION OF THE IDEAL THERMODYNAMIC CYCLE

As Organ [4] points out, an ideal "textbook" cycle based on a single homogeneous mass of gas cannot be achieved with a practical Stirling-type machine. However, it is still useful to consider the hypothetical ideal situation for comparison purposes with other ideal cycles, in particular, the Carnot Cycle. Thermodynamic diagrams for the new open regenerative cycle (operating on an ideal "textbook" single homogeneous mass of gas) are shown in Figure 2.1. It can be seen that five separate thermodynamic processes comprise the ideal cycle:

- (1→2) The single homogeneous mass of gas undergoes isochoric heating Q_R. Heat is transferred reversibly from corresponding isothermal temperature points in process 3→4.
- (2→3) Work W_C is performed on the single homogeneous mass of gas. This produces an isothermal compression process, and heat Q_H is transferred reversibly to a high temperature heat reservoir.
- (3→4) The single homogeneous mass of gas undergoes isochoric cooling Q_R. Heat is transferred reversibly to corresponding isothermal temperature points in process 1→2.
- (4→5) The single homogeneous mass of gas expands in an isentropic process, and produces work W_{Ea}.
- (5→1) The single homogeneous mass of gas expands in an isobaric process, and produces work W_{Eb}. Heat Q_L is transferred reversibly from a low temperature heat reservoir. The system has now returned to its original conditions.

The processes are very similar to the ideal "textbook" Stirling Cycle, except that the isothermal expansion has been replaced by the combination of an isentropic and an isobaric process, i.e. the ideal Stirling Cycle would follow the process path 1→2→3→4→1. Inspection of the thermodynamic diagrams reveals that the ideal Coefficients of Performance (i.e. Q_H/W and Q_L/W) will be significantly lower than for the ideal Stirling Cycle, however, the limitations of practical Stirling machines means that real-life performance of the two thermodynamic cycles may actually be very similar, as will be shown in Section 4.

3.0 DESCRIPTION OF THE PRACTICAL THERMODYNAMIC CYCLE

Thermodynamic diagrams for a practical Stirling machine are not easy to interpret (since the gas is not homogeneous, and different temperatures and pressures occur within the various components), and therefore the embodiment of the new open regenerative cycle is most clearly explained by the α-configuration Stirling-type machine shown in Figure 3.1. In operation, the cycle incorporates the six different steps shown in Figure 3.2; however, gas momentum effects and sinusoidal piston motion mean that a real machine will have a blended transition between the various steps of the cycle, rather than the strict delineation indicated by this simplified explanation:

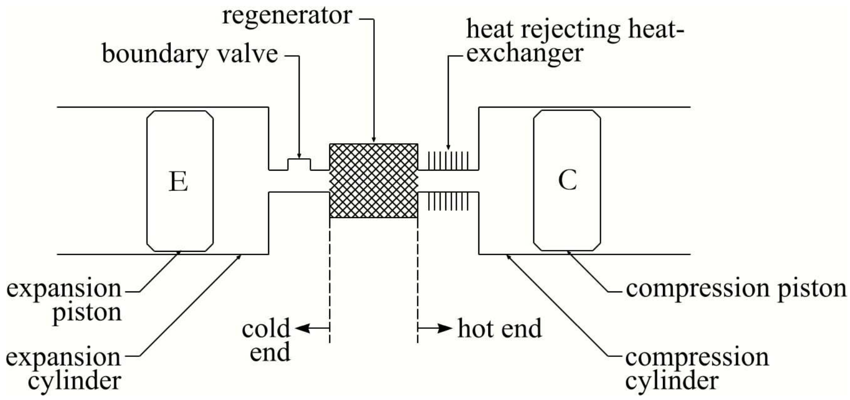

Figure 3.1: Simplified diagram showing a possible embodiment of a machine operating on the new open regenerative cycle.

(0→1) Isobaric inlet—a fresh charge of working gas (and the heat Q_L contained within it) is drawn into the system from the cold space at temperature T_L. When the system is fully charged the boundary valve closes.

(1→2) Isochoric heating—the expansion and compression piston move together to transfer the working gas isochorically through the regenerator to the hot end of the machine. Heat Q_R is delivered to the gas as it passes through the regenerator, thus raising the temperature of the gas to approximately that of the hot space. As the temperature rises, the gas pressure increases significantly.

(2→3) Polytropic compression—the compression piston does work W_C to the gas and compresses it polytropically, hence rejecting heat Q_H to the hot space (via the heat rejecting heat-exchanger).

(3→4) Isochoric cooling—the expansion and compression piston move together to transfer the working gas isochorically through the regenerator to the cold end of the machine. Heat Q_R is absorbed from the gas as it passes through the regenerator, thus lowering the temperature of the gas to approximately that of the cold space.

(4→5) Adiabatic expansion—the low-pressure working gas expands adiabatically, lowering the temperature of the working gas below that of the cold space, and doing work W_E to the expansion piston.

(5→0) Isobaric exhaust—the boundary valve opens and the very cold working gas at temperature T_{L-} is expelled to the cold space. The system has now returned to its original conditions.

Note that the inlet stroke (0→1) and exhaust stroke (5→0) superimpose to give the isobaric heat-absorption process (5→1) as described for the ideal cycle in Section 2. However, the isobaric expansion work W_{Eb} does not exist in this case, as the process occurs at zero gauge pressure, i.e. the pressure on both sides of the expansion piston is the same, and therefore no work is produced (in fact, in a practical machine, there will be a small pumping loop loss during this process).

4.0 PERFORMANCE OF THE NEW OPEN REGENERATIVE CYCLE

Practical Stirling machines differ from the ideal Stirling Cycle in a number of significant ways. Perhaps the largest contribution to the discrepancy between ideal cycle Co-efficient of Performance (COP) and the performance achieved by real machines relates to cylinder gas behaviour [5]. Haywood [3] has developed a set of simple equations which approximate the ideal performance limits for Stirling machines under the constraints that more realistically occur during actual operation. These equations make the following assumptions:

(a) that the compression and expansion cylinders are separated from the heat-exchangers
(b) that the machine operates so rapidly that the working gas in the compression and expansion cylinders exhibits adiabatic behaviour
(c) that the heat transfer is so good in the rest of the system (i.e. in the volume of the heat-exchangers and regenerator) that the working gas at any given point exhibits isothermal behaviour
(d) that the system conforms to the usual ideal cycle assumptions of frictionless flow, zero heat leakage losses, etc.

Although quite different in form, these equations demonstrate very close agreement to those developed by West [5] under a similar set of assumptions, and show that the upper limits for Heating COP C_{pH} and Refrigeration COP C_{pL} are determined by hot space temperature T_H and cold space temperature T_L in conjunction with two polytropic compression ratio numbers:

$$C_{pH} = \frac{N_{DMC} T_H}{N_{DMC} T_H - N_{JKR} T_L} \tag{4.1}$$

$$C_{pL} = \frac{N_{JKR} T_L}{N_{DMC} T_H - N_{JKR} T_L} \tag{4.2}$$

where the Clucas polytropic compression ratio number N_{DMC} and the Raine polytropic compression ratio number N_{JRK} can be calculated from the compression ratio r_c and the ratio of specific heats γ:

$$N_{DMC} = r_c^{\frac{\gamma-1}{\gamma}} - 1 \tag{4.3}$$

$$N_{JKR} = 1 - r_c^{\frac{1-\gamma}{\gamma}} \tag{4.4}$$

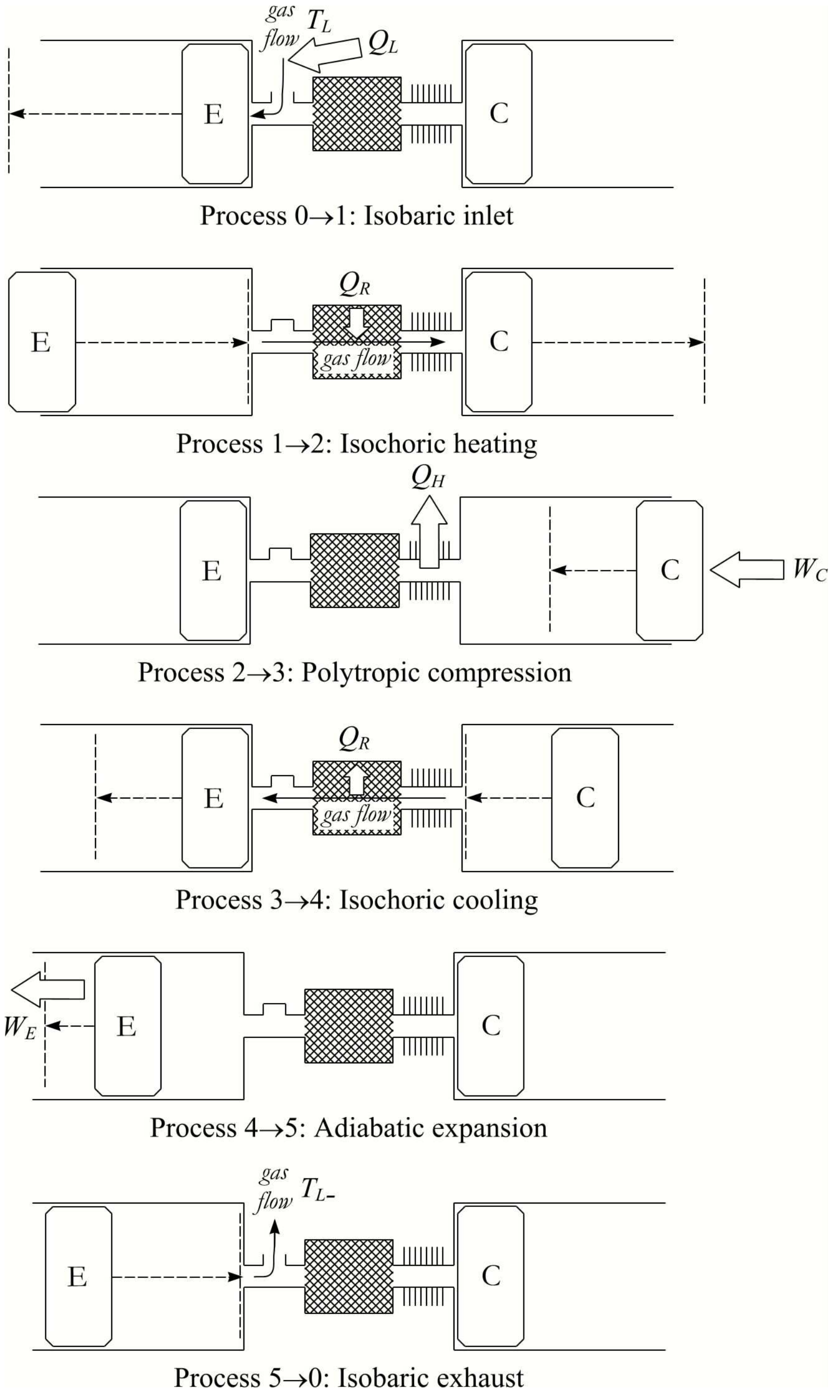

Figure 3.2: Simplified diagram showing the operation of the new open regenerative cycle in a possible machine embodiment.

Interestingly, it can be shown that (under the same set of assumptions) the equations describing the limits of performance for the new open regenerative cycle are *exactly the same* as Equations 4.1, 4.2, 4.3, 4.4. The implications of this are that (in spite of the large differences in "textbook" ideal performance), the actual performance of the new open regenerative cycle may be very similar to that obtained from practical Stirling machines. For a more accurate prediction of real machine performance (when operating on the new open regenerative cycle) a computer simulation must be employed, and the results of such an analysis are given in Section 5.

4.0 INHERENT FROST-FREE OPERATION OF THE NEW OPEN REGENERATIVE CYCLE

At first glance, the possibility of ice formation in the regenerator of a machine operating on the new open regenerative cycle seems rather high. With sub-zero air at high relative humidity, the gas passing through the regenerator in the isochoric cooling (3➔4) and adiabatic expansion (4➔5) steps shown in Figure 3.2 may easily fall below dew-point temperature, and the resulting condensation could freeze as ice within the regenerator matrix voids. In actual fact, however, this will not occur—and preliminary testing has demonstrated that the regenerator will not ice-up even when liquid water is directly injected into the expansion cylinder.

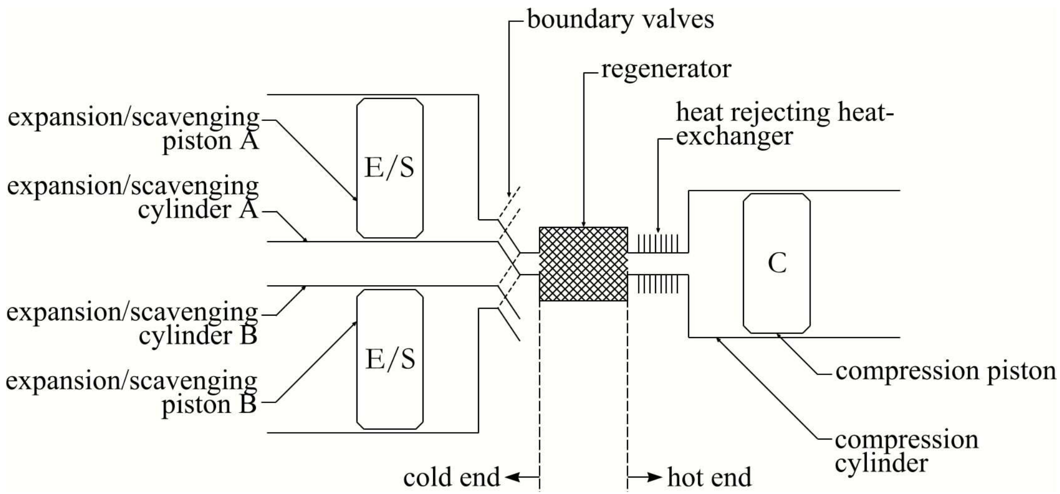

Figure 5.1: Simplified diagram showing the operational layout of the new open regenerative cycle on the MAG1 Regenerator Test Rig.

The reasons for this are not fully understood, however, it can easily be seen that the extreme situation (where the regenerator in a machine becomes completely blocked with ice) is impossible. If this were to happen then the cylinder spaces would become isolated from one another, and the compression ratio in each cylinder would consequently become extremely high (of the order of a Diesel engine!). The work input to the cylinder spaces would then be far greater than the heat-exchange capacity, and the gas temperature would rise rapidly in both cylinders until the ice melted and the obstruction cleared. In reality, the regenerator does not seem to ice-up at all, and it seems likely that several complex thermodynamic processes may contribute to this. An experimental prototype has been constructed to further investigate this phenomenon (along with several other important issues).

5.0 DESIGN AND SIMULATION OF AN EXPERIMENTAL PROTOTYPE

A number of different embodiment design possibilities exist for machines that can operate on the new open regenerative cycle. These include α and β-configuration Stirling-type two-cylinder systems with dwell mechanisms, blower-scavenged designs similar to two-stroke diesel engines, and various compound cylinder arrangements. For the initial experimental prototype, however, a layout has been chosen that can be easily adapted to operation in our existing MAG1 regenerator experimental rig, and a simplified diagram of this arrangement is shown in Figure 5.1.

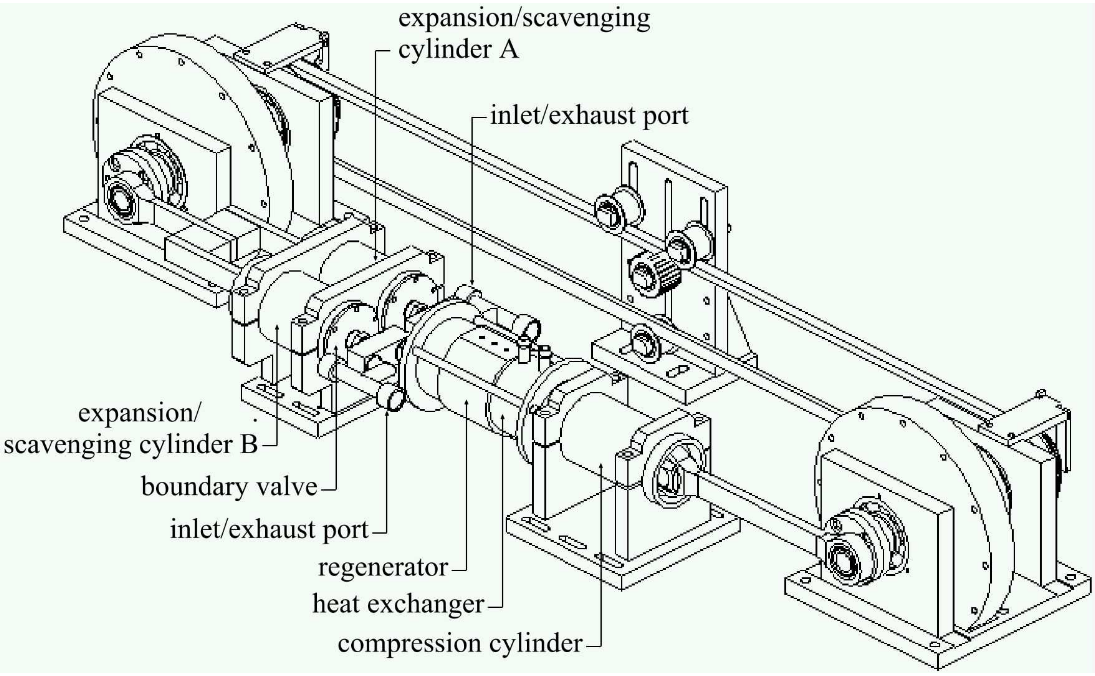

Figure 5.2: The MAG1 Regenerator Test Rig adapted for operation using the new open regenerative cycle.

The arrangement consists of three cylinders containing two expansion/scavenging pistons driven in parallel and a single compression piston at 90° phase spacing. The pistons move in near-sinusoidal motion and the boundary valve connects the compression cylinder to each expansion/scavenging cylinder in turn, i.e. as expansion/scavenging cylinder A is connected to the compression cylinder and executing steps 1→5 of Figure 3.2, then expansion/scavenging cylinder B is connected to the cold space and simultaneously executing steps 5→1, and vice versa. The MAG1 Regenerator Test Rig adapted for operation using the new open regenerative cycle is shown in Figure 5.2.

The thermodynamic design of the experimental prototype has been carried out using the *Sage* simulation software, and the critical parameters are listed in Table 5.1. The operating conditions are at the standard heat-pump rating temperatures of 20ºC (hot space) and 5ºC (cold space). It should be noted that the overall performance of the machine is limited by the heat-exchanger diameter, regenerator canister diameter, and stroke length available on the MAG1 test rig; without these constraints a heating effect of 250W can be achieved for the same cylinder bore with an indicated COP up to 3.4. It should also be noted that the *Sage* simulation has been conducted very conservatively, and it is likely that the indicated COP values given here are actually below that which will be achieved on the real machine.

Table 5.1: Critical parameters of the experimental open regenerative cycle machine

General	Working gas	air	**Regenerator**	Type	Wound mesh
	Operating speed (rev/min)	960			
	Hot space temperature, T_H (ºC)	20		Material	SS304
	Cold space temperature, T_L (ºC)	5		Wire diameter (mm)	0.08
	Mean operating pressure (kPa)	151.5		Porosity (void/total)	0.66
	Minimum cycle pressure (kPa)	101.3		Canister length (mm)	28.6
	Maximum cycle pressure (kPa)	201.7		Canister diameter (mm)	91
	Minimum cycle volume (cm^3)	207	**Boundary valve**	Passage length (mm)	75
	Maximum cycle volume (cm^3)	362		Passage diameter (mm)	19
	Compression ratio, r_c	1.75		Passage length (mm)	75
Cylinders	Bore (mm)	71	***Sage* predicted performance**	Indicated Refrigeration Effect (W)	67
	Stroke (mm)	27.8		Indicated Heating Effect (W)	100
	Clearance on stroke (mm)	1		Indicated Refrigeration Co-efficient of Performance C_{pL}	2.01
	Phase difference	90º			
Heat exchanger	Channel length (mm)	51	(constrained by MAG1 test rig dimensions)	Indicated Heating Co-efficient of Performance C_{pH}	3.01
	Channel height (mm)	3			
	Channel width (mm)	0.8			
	Number of channels	198			

On the negative side, the *Sage* predicted results for Refrigeration and Heating Effect demonstrate the consequences of operating at atmospheric pressure, and highlight the main drawback of the new open regenerative cycle, i.e. the very low heat and coolth production per unit volume. For this reason, the cycle would not be suitable for large heating or cooling load situations, however, it would still be applicable in smaller applications such as domestic refrigerators and heat-pumps. For the comparatively small loads required in these contexts, a machine operating on the new open regenerative cycle could quite easily fit within the standard volume envelope of conventional systems.

6.0 CONCLUSIONS

A new open regenerative cycle for refrigerators and heat-pumps has been developed that is similar to the Stirling Cycle except that cold space heat exchange is accomplished via a scavenging stroke rather than via a heat-exchanger. Some important features of this new cycle are:

- It uses environmentally-friendly air as the refrigerant.
- It does not require a heat absorbing heat-exchanger (the most costly component of a conventional vapour-compression heat-pump).
- It appears to have a similar Co-efficient of Performance (COP) to Stirling machines.
- It has inherent frost-free operation (frosting degrades low-temperature performance in conventional vapour-compression machines, particularly heat-pumps).

An experimental prototype has been designed to investigate the real-life performance of the cycle. This machine is based on the MAG1 Regenerator Test Rig. Some important features of this experimental prototype are:

- *Sage* thermodynamic simulation of the machine predicts an indicated Heating COP of at least 3.01 with a Heating Effect of 100W at the standard heat-pump rating temperatures of 20ºC (hot space) and 5ºC (cold space).
- Performance is limited by the geometric constraints of the regenerator, heat-exchanger, and stroke of the MAG1 test rig. For the same cylinder bore an indicated Heating COP of at least 3.4 is possible, and a Heating Effect of 250W.
- Operation at atmospheric pressure means that the machine has a low Heating and Refrigeration Effect per unit volume in comparison with both Stirling and conventional vapour-compression machines.

Despite the low Heating and Refrigeration Effect per unit volume, the new open regenerative cycle appears to offer a number of potential advantages over both Stirling and conventional vapour-compression machines in the context of small systems.

7.0 REFERENCES

[1] Haywood, D., Raine, J.K., Gschwendtner M.A. (2002) Stirling-cycle Heat-pumps and Refrigerators—a Realistic Alternative? *Proceedings of the IRHACE Technical Conference* (Christchurch), 26th April, 111-118.

[2] Organ, A.J., Larque, I.J., Roberts, D.A. (2001) A Shot in the Dark. *Proceedings of the 10th International Stirling Engine Conference* (Osnabrück), 24-28th September, 47-54.

[3] Haywood, D. (Forthcoming) *Investigation and Development of Stirling-type Heat-pump and Refrigerator Systems Using Air as the Refrigerant.* Ph.D. thesis, University of Canterbury.

[4] Organ, A.J. (1992) *Thermodynamics and Gas Dynamics of the Stirling Cycle Machine*. Cambridge University Press, Cambridge, p.53.

[5] West, C.D. (1986) *Principles and applications of Stirling engines*. Van Nostrand Reinhold Co., New York, pp.33-42.