On designing low pressure loss working spaces for a planar Stirling micromachine

M.-A. Hachey¹, É. Léveillé¹, L.G. Fréchette¹, F. Formosa²
¹Institut interdisciplinaire d’innovation technologique (3IT) Labo. Nanotech. Nanosystèmes (LN2) CNRS UMI 3463, Université de Sherbrooke, Sherbrooke, Canada
²Université Savoie Mont Blanc, Annecy, France

Luc.Fréchette@usherbrooke.ca

Abstract. In this paper, research was undertaken with the objective to design low pressure loss working spaces for a Stirling cycle micro heat engine operating from low temperature waste heat. This planar free-piston heat engine is anticipated to operate at the kHz level with mm³ displacement. Given the resonant nature of the free-piston configuration, the complexity of its working gas flow geometry and its projected high operating frequency, flow analysis is relatively complex. Design considerations were thus based on fast prototyping and experimentation. Results show that geometrical features, such as a sharp 90° corner between the regenerator and working spaces, are strong contributors to pressure losses. This research culminated into a promising revised working space configuration for engine start-up, as it considerably reduced total pressure losses, more than 80% at Re = 700, from the original design.

1. Introduction

Ongoing research is being undertaken by an international consortium (France-Canada) to produce a novel Stirling micro heat engine, the MISTIC (micro Stirling cluster), which will use low temperature waste heat (150-200° C) as a primary energy source for cogeneration. Three double-acting piston membranes (the double-acting idiom designating that the cold and hot working spaces are simultaneously acting on their respective faces) are set up in a Siemens (daisy chained) configuration [1]. These specially-designed double-acting piston membranes forgo the typical reciprocating pistons, piston rings and mechanical linkages found in Stirling engines for an airtight layered construction of the working spaces. This makes way for enabling the use of both MEMS fabrication and conventional bulk fabrication processes.

Each of the heat engine’s three modules (a cold and hot piston-membrane face, a cold and hot working space, and the corresponding regenerator) (figure 1) exploit the inertia of the suspended structures to act on the working gas. Thus, cyclic pressure applied between the modules cause the piston-membranes of each module to phase by 120°; i.e., the piston-membranes act as both a piston (produce work with the working gas) and displacer (displace the working gas) over one cycle.

This paper discusses how the MISTIC’s working spaces were designed to reduce pressure losses, also called pumping losses, as the working gas is displaced in and out of them. This entails having comprehensive knowledge of the flow characteristics, which is hard to achieve through analytical means because of the scale, oscillating flow and non-traditional geometry [2]. Hence, an experimental approach...
was favored where the characterization of the pressure losses are first investigated in steady-state unidirectional flow to identify promising configurations. Rapid prototyping, where a set of non-dimensional numbers focusing on certain geometric features, helped iterate working spaces [3]. In this article, working space geometries were trialed in both directions. One of them was found most satisfactory for engine start-up with a given set of mechanical and thermal parameters.

![Diagram of MISTIC’s three modules](image)

**Figure 1.** Depiction of a cross-section plane passing through the MISTIC’S three modules

The start-up criterion is currently determined through the use of a specially-developed electrical circuit equivalent (ECE) model elaborated by the Université Savoie Mont-Blanc to analytically characterize free-piston Stirling engines [1]. The model also helped determine the operating frequency of the structure and, consequently, the parameters of this study; given that frequency scales inversely to length, operation is expected to reach at least 2000 Hz with mm³ displacement. As most structures have a few hundred micrometer diameters, this implicates high flow velocities (some possibly reaching Ma = 0.2 in the original configuration).

2. Designing the MISTIC’s working spaces

Each of the MISTIC’s three modules comprises cold and hot working spaces. One of the plane walls of these open spaces is in contact with a thermal reservoir. The other plane is occupied by the piston membrane which compresses and or displaces the working fluid (gas). Given that the exchanger wall and the membrane piston share the same working space, its simplest (and most compact) form is found when most of the shape corresponds to the membrane’s perimeter, occupying a 5mm diameter. Moreover, because these piston membranes are anticipated to have a short stroke, the working spaces are required to be thin and the current design expects to have a gap that’s approximately 100µm high.

Joined to the working space is the regenerator cartridge that goes through the insulated core that also reaches the opposite working space. The volume contained in all three is the total gas volume contained in one module. Given that the regenerator is at 90° with the working space, local flow phenomena can be problematic in this region and transition effects should be mitigated.

2.1. Design approach

2.1.1. Geometry breakdown. To give some direction to the design process, the working spaces were deconstructed into easily identifiable geometrical elements and non-dimensional groups (NDGs). This enabled us to delineate the changes to the working spaces that were deemed more likely to reduce pressure losses incurred from the gas’ displacement. Three NDGs were identified (refer to figure 3):

- NDG₁: ratio of regenerator (1) hydraulic diameter to that of exchanger channel (2)
- NDG₂: ratio of exchanger outlet (3) width to working space diameter (4)
- NDG₃: ratio of exchanger volume (2’, 3, 5) over maximum swept volume (2, 4)
Given these sets of NDGs, one can surmise that the pressure losses reach a minimum when NDG\(_i\) and NDG\(_j\) are at unity; this would implicate the elimination of the minor losses at zone 3 and the non-recoverable losses from the steep velocity change at the 90° corner. However, increasing NDG\(_j\) increases NDG\(_3\) and subsequently, dead space or volume on which no work is exerted. It should be noted that as trials were being conducted, the regenerator cartridge was found to be more efficiently manufactured by MEMS processes. As such, the regenerator’s shape was changed from a cylinder to a rectangular prism, immediately affecting NDG\(_i\) and the working space’s shape in subsequent revisions. This resulted in Revision A exploiting three of the four faces of the regenerator interface to increase NDG\(_i\) while slightly increasing NDG\(_2\). Revision B was proposed to maximize the regenerator interface’s hydraulic diameter (and NDG\(_i\)) by exploiting all four faces of the cartridge at the corner by shaping it similarly as a golden (or logarithmic) spiral. Revision B also maximized NDG\(_2\) as a means to eliminate possible flow separation at the wall at the cost of an increased NDG\(_3\).

Fast prototyping: The experimental trials consisted of reproducing the different working space shapes into a laminate chip. Thus, an LPKF Protolaser U3 was used to cut the working space’s shape into a 130\(\mu\)m thick Kapton 500FN131 and mill the two glass caps in which the Kapton was laminated with a hot plate. The Kapton layer included five pressure taps (numbered 1 through 5 in figure 4) to measure the static pressure at the perimeter of the reproduced working space. Given that these pressure taps were set at dominant geometrical features, they would help determine the pressure loss in these areas. In addition, radial relief structures were added in the Kapton’s pattern to account for degassing and warping due to thermal dilatation during the device assembly.

The resulting structure, a 15x15 mm glass-Kapton-glass laminate, could then be easily socketed into a Plexiglas/aluminium packaging. The packaging granted the user access to the chips’ pressure taps through conventional fittings and tubing. Lastly, because the flow was to be unidirectional, the regenerator interface and the working space median were exploited as inlets or outlets to the chip, that is, depending on the gas flow’s direction (figure 5).
Figure 5. Packaging used in experiments pictured with a chip exposed (left) and a CAD drawing of the packaging’s ability to accept the original and Revision A versions of the working space (right).

2.1.2. Experimental setup. With the packaging sealed by O-rings and the tubes secured and tested for integrity, each chip was tested in an incrementing unidirectional flow rate. Static pressure at the taps were recorded to be later processed for determining the pressure losses that correspond to geometrical features and flow rate. The inlet flow rate, controlled (and corroborated) digitally in a range of 100 to 1500 cm³/min, represented the transient range of a pulsating flow, given the MISTIC’s operating parameters and displacement. Meanwhile, the packaging permitted the measurement of the static pressure at taps 1 through 5 by an Omega PX4202 pressure sensor (0-30psig resolution). An additional junction, tap 0, was located in a relatively large inlet manifold; thus, its static pressure was deemed to be the stagnant (total) pressure of the inlet. It is worth of note that all values were recorded relative to the transducer’s readout of local atmospheric pressure and inlet flow resistance, i.e. the resistance between tap 1 (or 5 if in reverse flow) and tap 0, was deducted.

3. Results and discussion

Post processing of the static pressures in the general equation for conservation of energy was required to determine pressure losses at each zone (Table 1). These zones were length-wise portions of the working space between the pressure taps. Any non-recoverable losses between two taps were to be found in the general equation by substituting stagnation pressure from one zone to another. This means that kinetic (average velocity) and semi-compressible (local change of density) effects were considered. Pressure at tap 0, being stagnate, corroborated the sum of the calculated stagnation pressure losses.

| Zones | Δp_{loss} (Pa) |
|-------|----------------|
|       | Original | Revision A | Revision B |
| 0→1   | 394     | 1473       | 3563       |
| 1→2   | 3781    | 1947       | -395⁵⁺    |
| 2→3   | 1241    | 1760       | 129        |
| 3→4   | 2187    | 57         | 925        |
| 4→5   | 2003    | 126        | -307⁵⁺    |
| total | 9605    | 5361       | 3915       |

| p_{tap} deviation | 9515 (<1%) | 5171 (3.7%) | 4137 (5.4%) |

⁵⁺ Negative values are within the range of 0.25% measurement uncertainty (+/-500Pa)
Results show that in a regenerator-to-working-space direction, pressure losses occurred mostly in the 0-1 zone, which corresponds to the 90° angle between the regenerator and the exchanger portion of the working space. The effect of increasing NDG\textsubscript{1} is prominently featured in the left portion of figure 6 where 84% of the pressure drop at Re = 700 was reduced between the original configuration and Revision A. This was likely due from increasing the hydraulic diameter at tap 1 from 200 to 260 µm. Improvement between Revisions A and B was marginal, even if the hydraulic diameter was maximized from 260 µm to 320 µm by way of a spiral relief. The wall’s spiral relief proved detrimental to reverse flow, as overall resistance increased for this case in Revision B. Thus, the working space in Revision A was retained for having more favourable (and symmetrical) flow pressure loss in steady state flow.

![Figure 6. Total pressure loss of the original, Revision A and Revision B configurations in a regenerator-to-working-space flowrate (left) and total pressure loss between Revision A and Revision B according to flowrate direction (right), i.e. regenerator-to-working-space (F) and working-space-to-regenerator (R). Data was processed in relation to tap 1’s Re number.](image)

4. Conclusion

This research’s objective was to design a working space which would reduce geometry-based pressure losses in the MISTIC’s engine modules as they operated at kHz levels. Practical considerations and the use of non-dimensional groups helped to design optimized working spaces for the projected operating conditions. By breaking down the working space geometry into a given set of zones, it was possible to determine which geometrical features were likely to affect the working gas total pressure loss. These findings were further corroborated by the retrofitting of the data into an ECE-based global model of the engine that showed favorable start-up conditions with the configuration that was retained. However, further study is necessary in determining the losses in oscillatory flow and the efficiency of the working space’s heat transfer.

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