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Early Development of a 100 Watt Low Temperature Difference Stirling Engine

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Abstract**—** Low temperature difference Stirling engines are investigated for power generation from low grade geothermal and waste heat. Findings from the early development of an engine targeting a shaft output of 100 Joules/cycle are presented here. A concept was drafted that is a beta engine with ~90 litres of internal volume, powering a bellcrank-style drive mechanism. Key features include exchangeable heat exchanger modules that allow volume variation, variable compression ratio by changing the power piston stroke, and a charge pressure of up to 10 atm. The purpose of this engine will be to provide experimental data for validation of a numerical model and ultimately prediction of the performance of larger scale machines. The detailed design process following the specifications outlined here is currently in progress.

Keywords – Stirling, engine, low temperature, scaling, layout, mechanism, concept, prototype, development, design

I. INTRODUCTION

There is a significant energy resource in the form of low temperature (LT) heat that is widely available but currently not being utilized. This refers to heat sources with temperatures of 150 °C or lower. Much of this energy potential exists from geothermal heat sources, which are available in great capacity within the province of Alberta [1]. In addition, LT heat is commonly discarded as a by product of energy intensive industries. These sources of renewable energy have so far not played a role in power supply because of their poor exergy (available energy) compared to higher temperature sources. This causes their utilization to be inefficient and not economically viable with conventional technology.

Two competing technologies for power generation from LT heat are the Organic Rankine Cycle (ORC) and reciprocating heat engines such as the Stirling engine. While the ORC uses turbomachinery and is viable for large scales in terms of power and size at a steady operating point, Stirling machines have the advantage of lower complexity and possibly higher efficiency [2] at small scales and can be adaptable to a variable source temperature by changing their compression ratio (CR). The Dynamic Thermal Energy Conversion (DTEC) Lab at the University of Alberta investigates Low Temperature Difference Stirling Engines (LTDSE) experimentally and numerically [3-8]. The goals are to model the energy flows and losses that are significant in the LT regime, and to assess the feasibility of LTDSE as part of the power supply infrastructure on a broad scale.

Research so far has focused on small scale lab engines (shaft power P_{shaft} < 10 W) of the gamma type which were used to gain a basic understanding of the parameters that govern LTDSE. To initiate this research, an existing high temperature engine design was modified to operate at lower temperatures [3,8]. Furthermore, a series of experiments was performed on a custom-built engine to find optimal working parameters for LTDSE, such as CR [4], and assess the potential of non-sinusoidal motion kinematics [5]. Alpha type engines were also investigated and tested but were found to have poor characteristics for LT engines [6]. Parallel to the experimental work, a 3rd order numerical model is being developed specifically to account for the physics of LTDSE [7].

Informed by the findings from this work, a new engine ('2021 engine') is currently in development that will advance the research from a small to an intermediate scale (100-1000 W). This machine will have variable components in order to study the effect of CR, charge pressure, and heat exchanger size and type on LTDSE performance. The goal is to collect data from this engine at a wide range of operating points that can be used as a benchmark for LTDSE. Using the data to verify the numerical model under development, it would become possible to make reliable predictions about the performance of a large scale commercial LTDSE. This would be a significant step towards addressing the overarching question regarding the viability of LTDSE as a power generating machine. This paper covers findings and decisions about size, layout and mechanism from the early design stage of the 2021 engine.

II. STIRLING ENGINE CYCLE, LAYOUTS & PERFORMANCE

An overview of the fundamental working principle, cycle, types and components of Stirling engines is given by West [9]. The typical Stirling indicator diagram is shown in Fig. 1.

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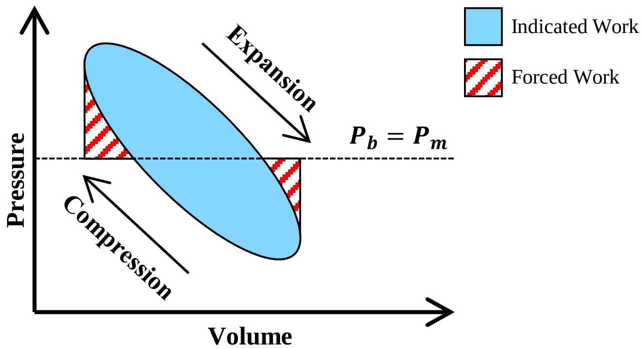

Figure 1. Indicator diagram of a real Stirling engine cycle. The area enclosed by the P-V-curve represents the indicated cycle work. Dashed line indicates engine buffer pressure, which is in this ase equal to the cycle mean pressure.

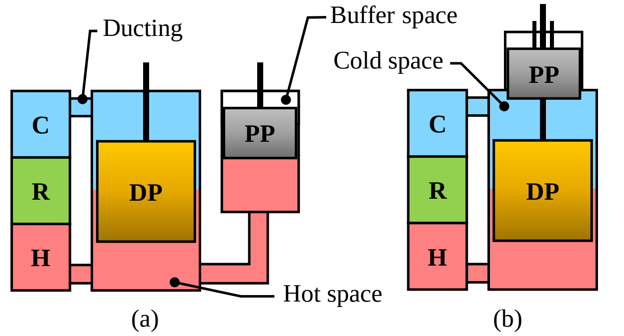

Figure 2. Examples of Stirling engine layouts: (a) Parallel gamma, (b) Beta. Meaning of the labels: C – Cold heat exchanger, R – Regenerator, H – Hot heat exchanger, DP – Displacer Piston, PP – Power Piston. Shown at maximum expansion – PP at top dead center, DP mid stroke. The thick black bars are piston connecting rods.

Essentially, a Stirling engine is a closed cycle reciprocating heat engine that draws energy from a temperature difference between a heat source and a heat sink (hot/cold heat exchangers). In the engine volume there is a constant mass of working gas (i.e. air) that is alternately heated and cooled by moving back and forth through a set of heat exchangers. The oscillation of the gas temperature causes a pressure swing in the engine. While the pressure is high, expansion of the engine volume occurs and during low pressure, compression takes place, similar to the cycle of an internal combustion engine. Thereby a net indicated work output is produced equal to the difference between expansion and compression work. This is represented by the enclosed area of the indicator diagram (Fig. 1, blue).

A. Stirling Engine Layouts

Stirling machines are divided into three types based on their layout and components [9], two of which are shown in Fig. 2:

*1) An **alpha** engine* has an expansion piston and a compression piston acting on its volume. The pistons take care of the gas displacement between hot and cold as well as the volume change. An inherent weakness of this layout is that a significant amount of energy is constantly transferred from expansion to compression piston through the mechanism connecting the two. This causes frictional losses that have proven to be detrimental to the performance of a LT alpha engine [6]. It is also difficult to vary CR and other engine parameters independently, which is a desired feature for an experimental engine. For these reasons the alpha layout is not further considered for the 2021 engine design.

*2) A **beta** engine* has a power piston for expansion and compression, and a double-acting displacer piston that moves the gas through the heat exchangers. The functions of volume change and displacement are fulfilled by separate components, which allows for uncomplicated adjustment of the engine parameters. By means of a mechanism, the pistons usually oscillate sinusoidally with a phase difference of 90°, resulting in the characteristic indicator curve shown in Fig. 1. In a beta layout, the power and displacer pistons operate on the same axis within a shared cylinder. The displacer connecting rod passes through the power piston. This layout is illustrated in Fig. 2b.

*3) A **gamma** engine* functions similarly to a beta, but has the displacer and power pistons arranged separately, for example in parallel cylinders as shown in Fig. 2a.

Only piston-displacer engines (beta and gamma types) are considered from hereon.

B. Key Parameters for Power Output

In a strongly simplified sense, the power output of a Stirling engine behaves approximately proportional to the following parameters, as derived from the West number definition [9]:

$$P_{shaft} \propto P_m * f * dV * \frac{T_H - T_C}{T_H + T_C} \qquad (1)$$

Here, P_m is the engine cycle mean pressure or charge pressure, f is the frequency or engine speed, dV is the total volume change of the engine, T_H and T_C are the heat source and sink temperatures. Other major parameters that influence engine performance are:

- Heat exchanger performance, i.e. geometry, surface area, flow losses
- Dead volume, i.e. engine volume that is not swept by pistons (e.g. heat exchangers, ducts)
- Mechanical efficiency (seal & mechanism friction)

C. Forced Work, Buffer Pressure and Effectiveness

Out of all energy that passes through the mechanism from piston to flywheel or vice versa, a fraction E is transferred and the remaining fraction $(1 - E)$ is lost due to mechanism friction. E is called the mechanism effectiveness. This parameter is difficult to predict but can be determined experimentally by measuring the shaft work of an engine, and calculating the indicated and forced work from its indicator diagram. This concept of efficiency was developed by Senft [10].

The pressure acting on the outward side of the power piston, counteracting the internal engine pressure, is called the buffer pressure (P_b). It is critical for engine performance because it affects the amount of forced work, as can be seen in Fig. 1. Forced work is energy that must pass through the engine's mechanism twice in order to drive the power piston. This occurs when the engine pressure falls below P_b during expansion and when it rises above P_b during compression. The amount of forced work in the engine cycle must be minimized in order to minimize the mechanism losses. Depending on the effectiveness E, these losses can consume a significant share of the shaft power and in extreme cases prevent an engine from running. . It can be shown that the forced work amount is always minimal when P_b equals the mean pressure P_m of the engine cycle [10].

III. Design Specifications for the 2021 Engine

Based on the sum of findings from previous experiments and the research goals outlined in section I, the design specifications for the 2021 engine were laid out. The new prototype will be developed to meet these specifications, ensuring that it has the capabilities to provide all necessary data to tackle the research questions put forward. To determine the initial specifications, many comparisons were drawn with the most recent LTDSE built in the DTEC Lab, the EP-1 [4], since detailed experimental data from this engine is available to the authors.

A. Parameters to Scale Engine Volume

A number of parameters must be set before the engine volume size can be scaled using an isothermal model. These are:

1) Target Engine Power

The overall scale of the engine is defined by its targeted shaft power output. The main drivers to determine an appropriate scale are:

- Should be as large as possible for data to be scalable to large commercial scale and produce accurately measurable outputs (power, torque).
- Size limited by budget; lab space; lab power supply capacity for heating and cooling; difficulties in manufacturing, assembly, handling of large components.

A target power of 100 Joules/cycle at atmospheric charge pressure was set. That is about a 16-times increase compared to the EP-1, which produces about 6.2 Joules/cycle (7 W at 68 rpm [4]). Power is the product of cycle work and frequency. Cycle work was defined rather than power because it can be easily estimated for a given engine geometry by simple models. Engine frequency is difficult to predict in the design stage as it depends on heat exchanger performance and overall friction in the engine, which cannot be quantified without knowledge of the engine and heat exchanger geometry.

2) Heat Source & Sink Temperatures

As with the EP-1, liquid water will be used as a medium to carry energy in and out of the heat exchangers. This limits T_H to ~ 95 °C, which is appropriate as long as the temperature is sufficient for the engine to operate. A higher T_H would change the operating point, but not the conclusions from experiments. It would, however, require the use of oil, which would make handling and cleanup of the liquid more difficult, increase the risk of burns and limit the choice of materials for components that are hot or in contact with oil.

The heat sink could potentially be run with water of 5 °C that is cycled through a cold water bath like with the EP-1, but that would require significant cooling power at this engine scale. Instead, cold tap water can be used and discarded after passing through the heat exchanger, requiring no electric cooling power. Its temperature has been measured in the lab to 7 ± 1 °C, which will not noticeably change engine performance compared to 5 °C. Therefore, leaving room for fluctuations in the tap temperature, T_C = 10 °C is a reasonable assumption.

3) Hot & Cold Side Engine Air Temperatures

Air temperatures in the real engine undergo continuous change throughout the cycle as the engine volume and flow rate through the heat exchangers are oscillating. The isothermal model simplifies this strongly by assuming the temperatures in the hot and cold spaces of the engine volume to be constant throughout the cycle. The EP-1 achieved hot and cold space temperatures of T_h = 67 °C and T_c = 21 °C [4] with similar source and sink temperatures to those specified in the previous section. Since the heat exchanger performance in the 2021 engine will be improved from the EP1's, temperatures are expected in the ranges of T_h = [75, 85] °C and T_c = [15, 25] °C.

4) Range of Compression Ratio

The compression ratio (CR) is the ratio of maximum to minimum engine volume. It is a central design parameter that interrelates the engine volume proportions to the engine pressure swing and performance. CR will be adjustable so that it can be optimized for a given engine working point. The analysis by Stumpf [4] on the EP-1 and other engines suggests a range for the optimal CR based on source and sink temperatures. Combined with the author's results from isothermal modeling, a target range of CR = [1.2, 1.4] is reasonable for optimization as well as for manufacturability of a mechanism with that range.

B. Engine Volume Scaling

The nature of Stirling engine design is that in order to specify a design, parameters need to be estimated because their actual values depend on the design that is so far unknown. Thus, it is helpful to create a range of parameters that the real engine will fall within. Additional parameters that must be estimated are:

- Volume ratios V_d/V_{swd} and V_{HX}/V_{swd}
- Mechanism Effectiveness E

where V_d is dead volume excluding heat exchangers, V_{swd} is the displacer swept volume, and V_{HX} is the volume of the heat exchangers. All parameters were estimated based on section III.A and data from the EP-1 to create three different scenarios as shown in Table 1. Scenario 1 represents an engine that falls short of its expected performance, while scenario 3 is an engine that outperforms expectations. Between them is the performance

TABLE I. PARAMETERS AND RESULTS OF 2021 ENGINE SCALING TO 100 JOULES CYCLE WORK USING ISOTHERMAL MODEL

	Parameters					**Results**					
Scenarios	V_d/V_{swd}	V_{HX}/V_{swd}	T_h[°C]	T_c[°C]	E	**Optimal** CR	V_{swd}[L]	V_{swp}[L]	V_{HX}[L]	V_d[L]	V_{total}[L]
1 - Pessimistic	0.4	0.4	75	25	0.75	1.209	76	28.6	30.4	30.4	165.4
2 - Realistic	**0.3**	**0.35**	**80**	**20**	**0.8**	**1.335**	**40.5**	**22.4**	**14.2**	**12.2**	**89.3**
3 - Optimistic	0.2	0.3	85	15	0.85	1.549	22.5	18.5	6.75	4.5	52.3

range that the real 2021 engine will most likely fall within. Scenario 2 is a realistic middle ground and will be used for the initial engine size estimate.

The parameter sets listed in Table 1 (left half) serve as the inputs to the isothermal (Schmidt) model as described by Senft [10]. This model heavily simplifies the physics of a real Stirling engine by dividing the engine volume into a hot space and a cold space, which are assumed to remain at constant temperatures T_h and T_c, respectively. The hot and cold space change in size due to the motion of the displacer, which causes the pressure swing. This approach allows for an estimate of cycle work without dealing with flow friction, heat exchange performance, engine frequency or an actual physical engine layout.

For each scenario an optimal CR is found by optimizing the power piston swept volume (V_{swp}) for maximum cycle work at a fixed V_{swd}. Now all of the engine volumes are defined in terms of V_{swd}, and the engine configuration for each scenario is scaled by volume to match the target shaft work output of 100 Joules/cycle . The results are listed in Table 1 (right half). $V_{swp}V_{total}$ is the total engine volume at maximum expansion.

It is evident that the overall engine size can vary wildly depending on how conservatively or optimistically the parameter values are estimated. According to scenario 2, the design should aim to incorporate displacer and power pistons with ~40 litres and ~22 litres of swept volume, respectively, to achieve the targeted cycle work. That is about a 7-fold increase in volumetric size compared to the EP-1 (V_{swd} ~6 L). It would be expected from the basic relation of engine power (1) that this increase is on the same order as the 16-fold increase in cycle work. However, the real performance of the EP-1 is being compared to the idealized, overestimated performance of an engine modeled as isothermal. The real 2021 engine will likely not reach the target cycle work at atmospheric pressure. The objective of this initial scaling is only to determine an approximate size scale to design for.

C. Mechanical & Thermal Efficiency

The EP-1 reference engine achieves around 2 % overall thermal efficiency (shaft power / heat input) [4]. The in-house thermodynamic model [7] suggests that there is significant potential to improve this number by investigating and optimizing heat exchanger size and performance, by reducing conduction losses through separation of hot and cold spaces and use of insulating materials, and the use of regenerator material to recover energy between the heat exchangers. Mechanical friction in seals and the mechanism also needs to be minimized. These are objectives for the 2021 engine design.

D. Modular Heat Exchanger Design

A main research question to be tackled is the effect of different heat exchanger types, internal volumes, surface areas and flow structures on LTDSE performance. To accommodate for the variation of these parameters, the heat exchangers cannot be integrated into the engine volume or limited in size and shape by the engine structure, but must be designed as a separate module. These modules will connect to the engine volume through plenums and can be exchanged for different heat exchangers without disassembly of the engine body. Within a module, the active heat exchanger volume can be varied by blocking off or adding/removing flow channels.

E. Elevated Charge Pressure

Equation (1) shows that the power of an engine scales with four main characteristics. These are charge pressure, frequency, volume (size) and temperature difference. The latter one is fixed in this case, and the frequency is a product of the real engine performance rather than an adjustable parameter. The scaling of the 2021 engine power by size was shown in section III.B. The effect of an above-atmospheric charge pressure on performance will also be explored with this design. A maximum charge pressure of 10 atm has been defined as a requirement. The amplitude of the pressure swing about this mean pressure is estimated to reach 4 atm. Consequently, the maximum differential pressure between the engine volume and the environment (1 atm) will be 13 atm. This results in substantial mechanical loads on the engine body structure that will dictate the structural design.

F. Buffer Pressure and Crankcase

Section II.C describes the need of equating the buffer pressure with the engine mean or charge pressure. Since the latter can go up to 10 atm as seen in the previous section, it needs to be possible to pressurize the buffer space to the same level. The buffer space (see Fig. 2) is located at the outside face of the power piston, where it connects to the mechanism that transfers the piston work to a crankshaft. The mechanism must be enclosed into a crankcase which is the engine's buffer space. During operation the crankcase will be sealed and pressurized to match the engine charge pressure, but it must be easily accessible during maintenance to change the CR setting on the mechanism. As the crankcase will have to withstand a significant pressure load, a compact mechanism is desired that fits into a small, structurally strong enclosure.

G. Dual Coupled Engines Setup

Apart from adjusting the buffer pressure, there is an alternative method to minimize the forced work passing through the mechanism. It requires two similar engines which have their power pistons mechanically coupled so that the mean pressure forces acting on both pistons counteract each other and cancel out. This has the same effect as equaling the buffer and charge pressures (see previous section), where the mean engine pressure is counteracted by the buffer pressure. The engines would be operating 180° phase-shifted, expansion in one engine coinciding with compression in the other. A pressurized crankcase is not necessary in this setup, which reduces complexity. It would also be an advantage for research purposes to have a second engine body in case one of them fails. To be able to test these benefits, the 2021 engine will be designed so that it can be extended into a dual engine in the future.

IV. Engine Layout & Mechanism Choice

With the most important design specifications for the 2021 engine being established, the next step is to arrange the engine components in a layout that is best suited to fulfill the set requirements. This gives the engine a geometry that will be the foundation for the detailed design to work with at a later stage. Closely tied to the layout is the choice of mechanism that will provide the kinematic linkage between the moving components. Thus, layout and mechanism are chosen simultaneously as a combination.

A. Displacer and Power Piston Layout

The layouts considered differ only in the arrangement of the displacer and power pistons:

1. Coaxial beta, as seen in Fig. 2b
2. Parallel gamma, as seen in Fig. 2a
3. Stacked gamma – Similar to beta but not coaxial
4. 90° gamma – Piston axes orthogonal to each other and in one plane (L-shape)

Based on the compatible mechanisms for each of these layouts, it was determined that only the first two options, which are shown in Fig. 2, can be chosen to satisfy the specifications.

B. Choice of Mechanism

A wide range of mechanism options were considered, including slider crank, scotch yoke, Ross yoke, rhombic drive, several bellcrank linkages, camshafts, rocker arms, and several axial motion generators such as swashplate, barrel cam and leadscrew. To evaluate the degree to which each of these would suit the specifications, the following criteria were used:

- Compactness (size of bounding box)
- Complexity of parts and assembly
- Bearing friction: Number of load-carrying rotating or sliding (e.g. scotch yoke) bearings
- Ease of varying power piston stroke for changing CR while maintaining top dead center position
- Allows mechanical coupling of dual engine
- Minimal side load angle on piston connecting rod

The side load angle is the angle between the piston axis and the connecting rod that connects piston and mechanism. This angle varies throughout the engine cycle. Its maximum is much larger in simple mechanisms (e.g. slider crank) than in others which apply only little rotation to the connecting rod (e.g. bellcrank). The greater the side load angle, the stronger is the radial force (side load) applied to the piston by the mechanism. This side load must be absorbed by linear bearings or a piston ring guiding the piston, which causes friction and results in a loss of shaft power. Therefore, a mechanism should be designed to ensure a small side load angle.

The axial motion options were discarded because of concerns with friction and lubrication. The ability of cam-driven mechanisms to change motion profiles away from sinusoidal was not considered an advantage because motion variation is not an objective of this research. Promising mechanism candidates were modeled as interactive sketches or solid models in SOLIDWORKS® to determine their geometrical relations.

Two engine concepts, consisting of layout and mechanism, were developed that match the criteria best. The first is a beta engine (see Fig. 2b) driving a bellcrank-style mechanism, which was developed from a mechanism concept presented by Organ [11]. Two more members were added to Organ's linkage and the bellcrank pivot points were changed to allow the displacer and power piston strokes to be controlled independently. Fig. 3 shows the geometry and parts of this mechanism in detail. The pivot point of the power piston bellcrank can be set to different positions via the hole pattern to adjust CR. The second concept is a gamma engine with parallel cylinders (see Fig. 2a). Its pistons are driven by an inverted Ross yoke mechanism, shown in Fig. 4. The power piston connecting rod can be linked to the yoke at different positions to change CR.

Both mechanisms fit into a compact crankcase and are made from simple parts and bearings. They have no sliding features and allow adjustment of CR by moving the position of one fixed pivot point. They also allow the attachment of two opposed engines to the same linkage and can equally be designed for a small side load angle. The two described engine concepts are both well suited to become the basis for the development of the 2021 engine. It was concluded that the beta variant should be pursued by considering the following small advantages of this layout over the gamma:

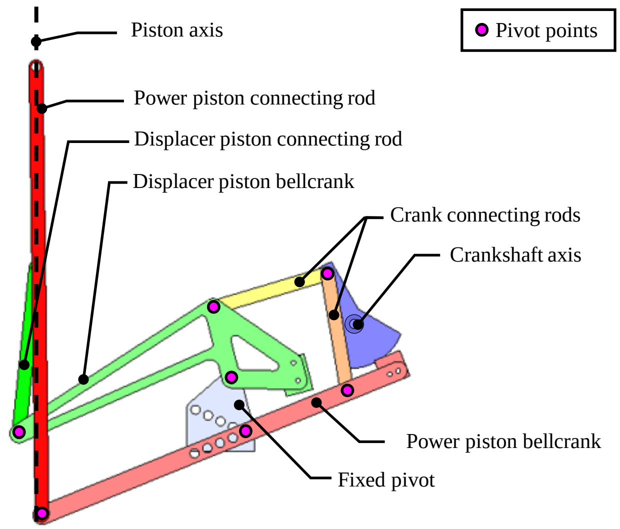

Figure 3. Schematic of the beta engine bellcrank mechanism, rendered in SOLIDWORKS®. The crank (purple) is linked to the two bellcranks independently, which produce oscillating motion parallel to the piston axis. The side load angle between connecting rods and the piston axis is very small.

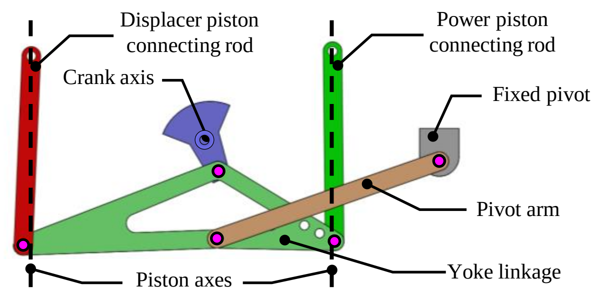

Figure 4. Schematic of the inverted Ross yoke mechanism, rendered in SOLIDWORKS®. The yoke linkage follows a rocking motion about the pivot arm and a translation along the piston axes. The resulting motion has both pistons move with a fixed phase difference while keeping side load angles very small.

The beta setup contains both pistons in a shared engine body and the piston working spaces are directly connected. A single pressure vessel (cylinder) can house the engine working volume, reducing the number of pressure-bearing structures. The gamma setup requires air ducts between the parallel cylinders, which adds dead volume and flow friction.

A unique beta feature is that the stroke of the pistons may overlap during part of the cycle, which is possible with the shared working space. This allows part of the space to be swept by both pistons, reducing the overall engine volume and theoretically increasing the pressure swing without shrinking the active working spaces. Volume savings are significant: In the given design, about 35 % of V_{swd} can be overlapped.

Making use of stroke overlap requires the power piston to be attached at the 'cold' side of the displacer space (as shown in Fig. 2b) due to the phase shift between the piston motions. The in-house model [7] suggests this should have no notable effect on power compared to attaching the power piston to the 'hot' side. The piston seal would operate at a low temperature, which is beneficial for elastomer or polymer materials.

For these reasons the beta layout with a bellcrank mechanism has been determined as 'Plan A' and is being developed in detail at this time. The parallel gamma with Ross yoke is 'Plan B' in case an insurmountable problem is encountered with the current design choice.

V. Conclusion

An LTDSE is being developed with a target shaft work output of 100 Joules/cycle. This research goal for this machine is to study the effect of various operating parameters on LTDSE performance, using the experimental data to verify a numerical model, providing a foundation to predict the performance of a large scale commercial LTDSE and assess the viability of LTDSE for power generation.

Design requirements were defined and the concept development stage was concluded, resulting in the specification of an engine scale, component layout and drive mechanism. The concept consists of a beta-style engine with ~90 litres of internal volume, modular heat exchangers, a custom bellcrank-style mechanism with adjustable CR, designed to operate at up to 10 atm of charge pressure. Detailed design of components and concurrent thermodynamic modeling are currently ongoing at an advanced stage, but would go beyond the scope of this paper which is limited to the decisions of the early design phase.

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