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Preliminary Development of a 30 kW Heavy Fueled Compression Ignition Rotary 'X' Engine with Target 45% Brake Thermal Efficiency

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Abstract

This paper presents initial progress in the development of LiquidPiston's 'X4', a 30 kW heavy-fueled rotary compression ignition engine prototype. The X4 is the newest version of the unique rotary 'X' engine architecture. This development is partially funded by the Defense Advanced Research Projects Agency (DARPA), with broad dual-use civilian / military applications including UAVs, generators, and for propulsion. The 30 kW size may be of significant interest in the automotive application as a range extender for Electric Vehicles. The final performance objectives of the program are aggressive: 45% brake thermal efficiency; > 1 hp / lb power-to-weight; and the engine is targeted to fit within a 10"x10"x10" box weighing <40lbs.

A first prototype engine "core" has been designed, analyzed, prototypes, and completed its initial testing. This stage of development has focused on verifying the structural integrity of the main engine components including the shaft, bearing, rotor and gears, as well as the function of the engine's gas seals, at the high pressures (in excess of 100 bar) expected of a CI engine.

Motoring results from initial testing show reasonable agreement to 1D performance modeling. The engine has exceeded the mechanical verification target, demonstrating stable operation with in-cylinder pressures of 150 bar. This paper will summarize the design and preliminary experimental results achieved in the first year of development, including:

- 1D thermodynamic Modeling
- Design considerations of the main engine components including the crank shaft, bearings, gears, combustion system
- Initial Testing to verify system integrity

With the X4 "core engine" structure and gas seals validated, future work is discussed, which will include performance optimization to achieve power and efficiency objectives.

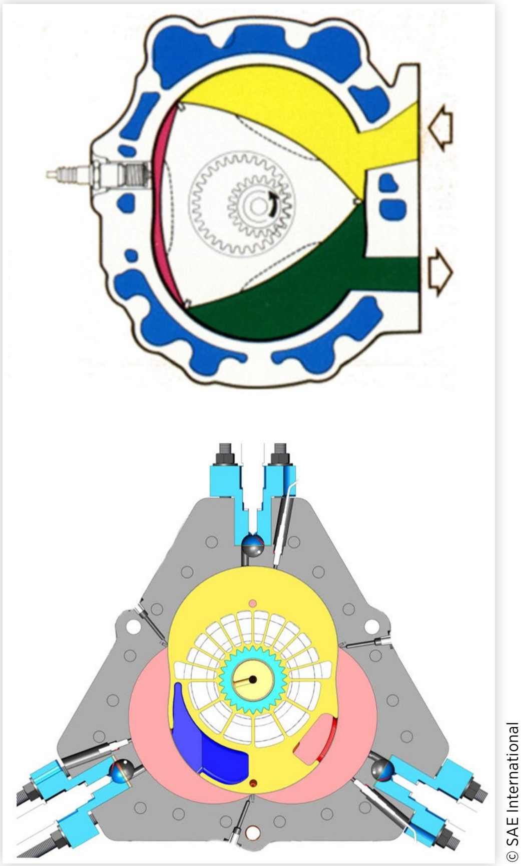
Introduction

The Wankel rotary engine [5] was developed in the 1960s as an alternative engine architecture. The engine demonstrated excellent power to weight characteristics, and exhibited low vibration even at high RPM. Despite these advantages, the Wankel was always plagued by poor fuel economy, emissions problems, and durability issues, especially in the apex / tip seals. These challenges are due to a number of inherent issues: 1) a narrow combustion chamber prevents adequate flame propagation, while also having high surface to volume ratio; 2) the engine is poorly sealed, leading to significant blowby; 3) the Wankel engine operates on the same conventional 4-stroke Otto cycle with Spark Ignition as a piston engine; however there are inherent challenges to operate >10:1 compression ratio, and this engine was forced to compete with piston engines that had over one hundred years of prior development; and 4) the tip seals, in addition to being difficult to seal, are also difficult to lubricate; oil must be injected into the charge, with the majority of the oil burned in order to lubricate the gas seals.

LiquidPiston has developed the 'X' engine, which essentially "inverts" the Wankel engine (see Figure 1). While a Wankel engine has a 3-sided rotor, and a 2-lobed housing, the X engine has a 2-lobed rotor in a 3-sided housing. The X engine captures the main advantages of the Wankel, including 1) high power-to-weight ratio [one rotor engine behaves like a 3-cylinder 4-stroke]; 2) simplicity - having only 2 moving parts - a rotor, and a shaft; and 3) like the Wankel - the X engine is inherently balanced, therefore having minimal vibration. Unlike the Wankel however, there are several key differentiators which address the bulk of the older Wankel's design deficiencies:

- The combustion chamber in the X engine is located in the stationary housing, with most of the gas displaced during compression into this stationary combustion chamber. This makes the X engine uniquely suitable for

FIGURE 1 Cross sections of (Top): Wankel Engine; (Bottom): X4 Engine (shown with IDI type combustion chambers)



high compression ratio operation with Direct Injection and Compression Ignition (which is not possible in the Wankel without boosting or a second compression rotor). Additionally, the combustion chamber can take any geometry, and can be optimized for surface to volume ratio, thereby improving combustion efficiency and reducing heat transfer.

- The apex seals of the X engine are located within the stationary housing, and do not move with the rotor. The seals do not experience centrifugal forces, and can be lubricated directly by metering small amounts of oil directly to the sealing surface, which means that oil consumption can be reduced to levels potentially comparable to that of a 4-stroke piston engine (essentially negligible).
- The unique sealing geometry of the X engine has 3-5 times less blowby than the Wankel rotary. This is mainly because 1) the Wankel requires clearance at the corners between its side/face seals and its apex seals, while the X engine does not; and 2) the Wankel seals traverse across holes that contain spark plug(s), whereas the X engine does not. The sealing strategy, seal modeling, and testing validation is described in detail in [9].

The other key differentiator of the X engine over most other types of engines is the ability to run on the High Efficiency Hybrid Cycle. In an idealized sense, this thermodynamic cycle cherry picks the "best" features of a number of other cycles, combining them into an efficiency-optimized cycle. The features of HEHC (referring to Figure 2) include:

1-2. High compression ratio of air (similar to Diesel cycle).

2-3. Fuel is injected, mixed and burned under approximately constant volume conditions (similar to ideal Otto cycle).

3-4. The expansion ratio can be designed to be greater than the compression ratio, enabling the engine to run over-expanded (similar to Atkinson cycle).

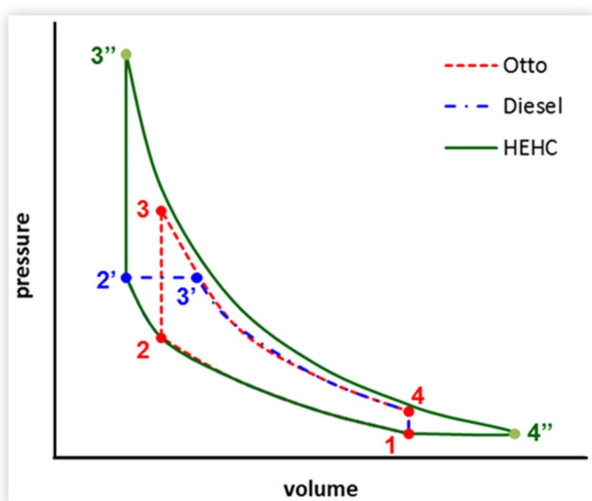
High compression is achieved by displacing and compressing the trapped air into a combustion chamber. The combustion chamber can have a small volume (thereby increasing compression ratio). This chamber can be thermally isolated from the rest of the engine and could potentially run hot to reduce heat transfer during combustion. Both SI and CI, as well as IDI and DI type combustion is possible.

The unique geometry of the engine causes a dwell near Top Dead Center (TDC), where the rotor is spinning, but the volume is not changing very much. This flatter volume profile gives the engine more time to combust more fully under near-"constant volume" conditions. The benefit of constant volume combustion is eloquently captured in a text by Gordon Blair [6, page 81]:

"The problem is simply that [a piston] engine cannot conduct combustion at constant volume, i.e., instantaneously at tdc, because a real burning process takes time, the piston keeps moving, and the cylinder volume changes. If this latter problem could be remedied by keeping the piston stationary at tdc while combustion took place and then moving it down on the power stroke when all is burned, ***the imep and power would increase by some 50%****. […] I would actually encourage the world's inventors to keep on trying to accomplish this ic-engine equivalent of the 'search for the Holy Grail'."*

The HEHC cycle has been analyzed and described previously [1]. Figure 2 (adapted from [1]) shows visually a comparison of ideal Otto, Diesel, and HEHC cycles. In that work, the HEHC is shown to have approximately 30% higher efficiency than comparable Otto / Diesel cycles. More complete thermodynamic modeling of the actual engine cycle

FIGURE 2 p-V diagram comparing ideal air-standard cycles



has also been completed using 1D simulation (GT-Power), and the results summarized in the Modeling section below, suggests that 45% brake thermal efficiency is possible in the 30 kW size X engine.

Additional details of the cycle, X engine architecture, and development of a small SI 70 cc gasoline X engine is described in [1-3]. The interested reader is also referred to view an animation of the 'X' engine here [4].

The optimal efficiency benefits described in [1] cannot be fully realized in the small gasoline-fueled spark ignition (SI) engine which had been described previously in [2, 3]. Like other engines, the HEHC cycle requires a high compression ratio for optimal performance. This is best achieved with compression ignition (CI) and heavy fuel. In this work, we take a step toward developing an engine which runs on Diesel / JP-8, achieves state-of-the-art (or better) CI fuel efficiencies, and offers a significant weight advantage comparable to current engines of similar performance.

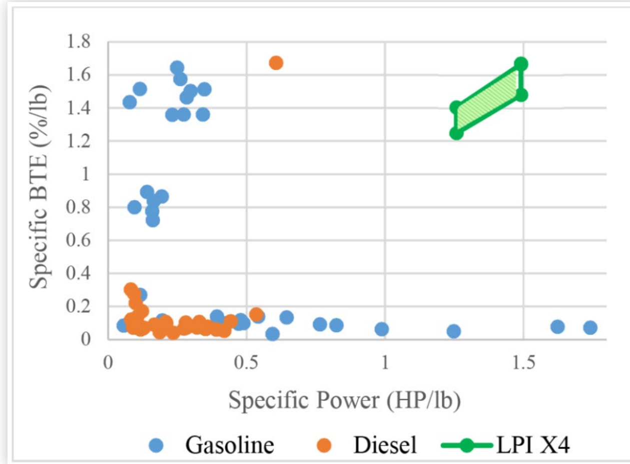
This paper summarizes the results of Phase I of a program to develop the .8 L 30 kW X4 rotary engine. The objectives of this Phase were to concept, design, prototype, and test the system integrity of the 30 kW engine core. The engine "core" (akin to a "long block") includes the main housing, side plates, shaft, gears, bearings, rotor, and gas seals; this does not include ancillary components such as the fuel system, oil pump, etc. While we had prior experience operating smaller SI 'X' engines, achieving high compression and constant volume combustion demands the engine to operate at much higher pressures, which had not previously been demonstrated. Therefore, the focus of this initial phase of the program was to operate the X4 at >100 bar of cylinder pressure, and show that the bearings, gears, shaft and rotor survive these high pressures, and further that the blowby / gas sealing performance is adequate to maintain such pressures without too much blowby (as determined by a 1D thermodynamic model). In this first stage, the engine was built without cooling, with the intent of demonstrating a few minutes of operation under load. The next stage of development (not covered in this paper) will include: 1) integration of cooling system into the engine; and 2) optimization of power and efficiency.

The paper is organized as follows: Motivation, Geometry Specification, Performance Modeling, Detailed Design and Analysis, Testing, and Future work.

Motivation

Diesel Compression Ignition engines, which operate at high pressure, tend to be more fuel efficient than gasoline engines, but are also large and heavy. For logistical purposes, military applications prefer to operate on JP8 fuel (a heavy fuel, similar to kerosene), and increasingly demand higher efficiency and improved power-to-weight. Non-military applications also benefit from similar improvements. One area of particular interest may be as a range extender for electric vehicles. In this approach, a vehicle may be powered electrically with a small battery pack which is intermittently charged by an efficient and compact Diesel power generator based on the 30 kW X4 engine. This approach could eliminate the bulk of an electric vehicle battery pack, reducing vehicle cost and weight, while improving efficiency (on a well to wheel basis), and also reducing logistical burden by allowing the vehicle operator to rapidly refuel at any existing fueling station.

FIGURE 3 X4 vs. Benchmark Data



The proposed program targets are very aggressive, as could be seen in comparison of the X4 to published gasoline and diesel specific power and specific efficiency data, which is shown in Figure 3.

The green box shows the bounds of the X4 program targets when development is completed.

Geometry Specification

The X4 program began with evaluating the potential of several X engine concepts to meet the stated objectives. The most significant parameters defining the geometry of an X engine are the values for E, R, Rr, and engine housing thickness. E is the eccentricity, R is the radius from the centerline of the crankshaft to the center of the apex seal radius, Rr is the apex seal roller radius, and the housing thickness is the width of chosen geometry parallel to the crankshaft center line. These parameters are described in Figure 3 below.

FIGURE 4 X Engine Generating Parameters

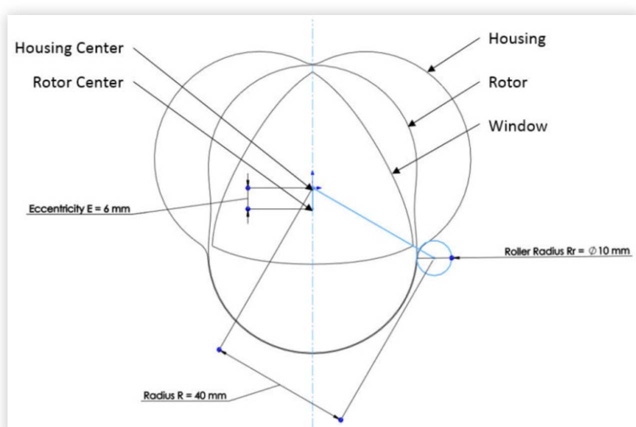


Figure 5 and Figure 6 show an effect of changing R, the distance from the engine center and the apex seals, and E, the crankshaft eccentricity, on the surface area to volume ratio of the working chamber. A low surface to volume ratio is desirable for reducing heat transfer losses.

With only this parameter in mind, a large R and E value would be chosen. However, as E increases, so does the diameter (size) of the engine; and as the ratio of R to E changes, the "Window Area" changes, which affects the ability of the X engine to intake and exhaust gas efficiently. The latter is shown graphically in Figure 7. To create this metric, available window area was calculated for several different E/R ratios at each crank angle and the integral of was this was taken over 720 degrees. This gives an indication of the trade-off made in window area for a selected Geometry.

FIGURE 5 Surface Area vs. CA for Varying R

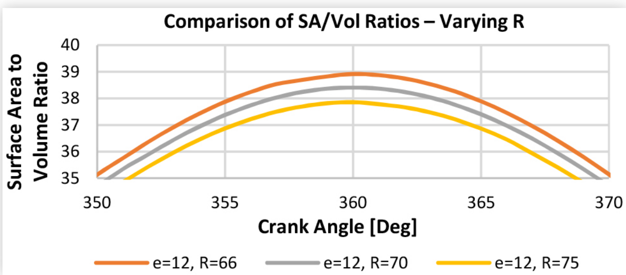


FIGURE 6 Surface Area vs. CA for Varying E

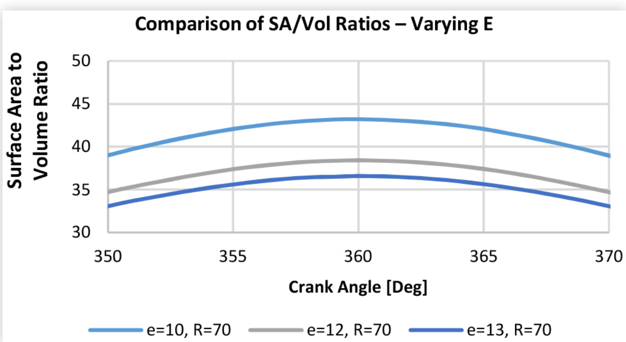


FIGURE 7 Window Area vs. R/E Ratio

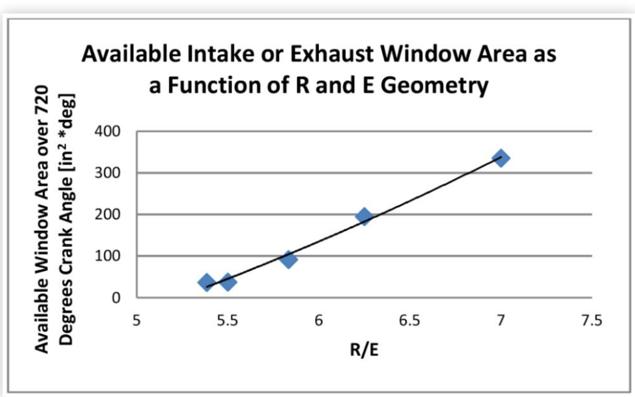
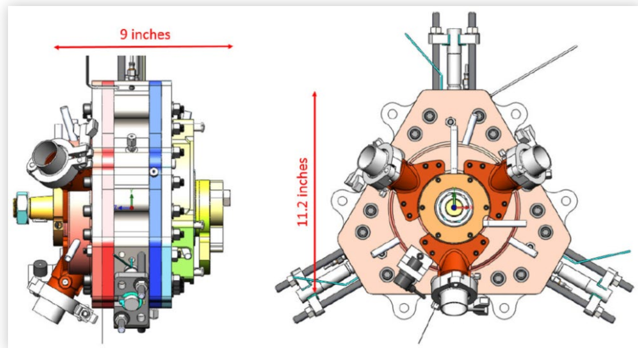


TABLE 1 X4 Design Parameters

E	13 mm
R	83 mm
Rr	1.46 mm
Housing Thickness	45.4 mm
Displacement (per chamber)	250 cc
Displacement (total)	750 cc

FIGURE 8 X4 Size



After optimizing all the conflicting restraints, and considering the performance modeling described in the next section, the concept phase concluded with the selection in Table 1.

The overall size of the X4 test rig is shown below in Figure 8. The size is larger than the final intent in order to make space for combustion chamber variations, and to accommodate standard diesel fuel injectors. These dimensions will be reduced in the following phases of the program.

Performance Modeling

In order to decide on the displacement of the engine, analyze the feasibility of meeting the 45% brake efficiency target, and provide gas load curve for component analysis, a GT Power

FIGURE 9 X4 GT Power Model

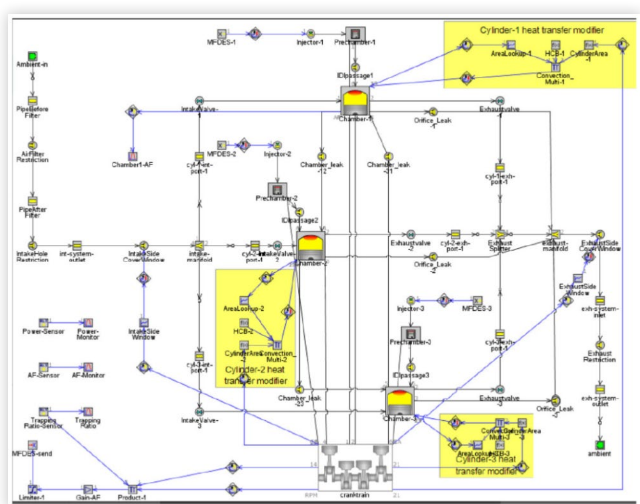
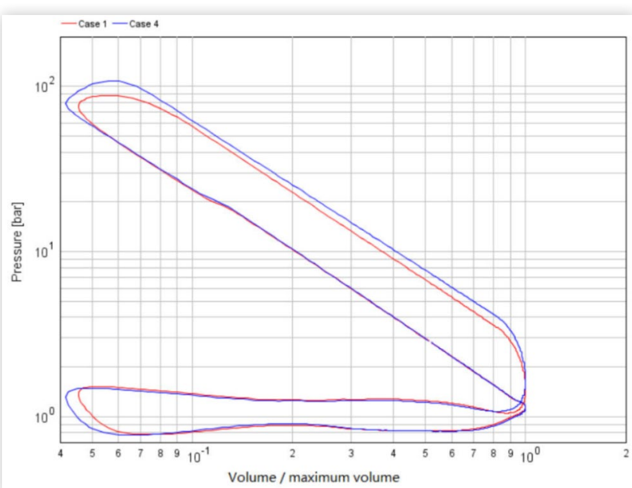


FIGURE 10 X4 GT Power Inputs/Outputs

Case No.	1	4
Burn Duration	66°	44°
Start Of Combustion timing	-30°	-16.5°
Leak areas (mm^2)	2.5	1.5
Friction	8.3% of fuel	7.8% of fuel
Heat transfer	27.9% of fuel	16.5% of fuel
Pumping loss	1.80% of fuel	1.66% of fuel
Geometric Compression Ratio	22	24
720° Indicated Thermal Efficiency	38.10%	45.50%
Brake power (kW)	24.5	32.8
PCP (bar, absolute)	88	108
Average cylinder P (bar)	8.52	9.46
Engine Speed (RPM)	7000	7000

FIGURE 11 PV Diagram for GT Cycles



model was created. GT Power is a 1D thermodynamics and flow simulation software by Gamma Technologies which is commonly used to simulate piston engines. In order to simulate the rotary X engine, some customizations were made to account for deviation such as non-standard crank-slider mechanism, volume and surface area profiles. The model is shown in Figure 8.

A summary of important input assumptions and output results are shown in Figure 10. Case 4 shows a pathway towards the Objective (aggressive) target, and Case 1 the Threshold (conservative) target. The "Leak Area" is an equivalent blow-by orifice which represents gas leakage through the combustion seals. This value has been measured at 0.5 to 1.5 mm^2 based on motoring compression data. The friction and heat transfer values are aggressive assumptions based on prior work on the 3 horsepower X engines. PCP stands for Peak Cylinder Pressure, and the value of 108 bar, predicted from the GT model, was used for component design. The corresponding Pressure vs. Volume diagram for Case 1 (red) and Case 4 (blue) are shown in Figure 11.

Detailed Design and Analysis

As described in the introduction, one challenge with CI engines is to build a lightweight structure capable of supporting the gas loads. GT-Power predicts up to 108 bar of gas loads in the naturally aspirated X4. This section will describe the design process used for the "bottom end" of the engine, and touch on the preliminary combustion system.

Crankshaft/Bearings

Figure 12 below shows possible gas exchange mechanisms. In the prior generation XMv3 engine, a hollow shaft was used for intake. In the X4 we chose to use that space for oil for bearings and rotor cooling. With the "Hollow Shaft" ruled out, the remaining options were "Window in Cover" and "Poppet Valves", and "Window in Cover" was chosen in order to retain the advantages associated with simplicity.

For the X4 engine, the loads and speeds were out of range for roller bearings, especially when considering the packaging constraints. Therefore, the crankshaft was designed to be solid construction and to make use of journal bearings.

The final crankshaft design is shown in Figure 13.

To assist in the design of journal bearings, another Gamma Technologies product, GT Suite, was used to predict

FIGURE 12 Breathing Mechanisms

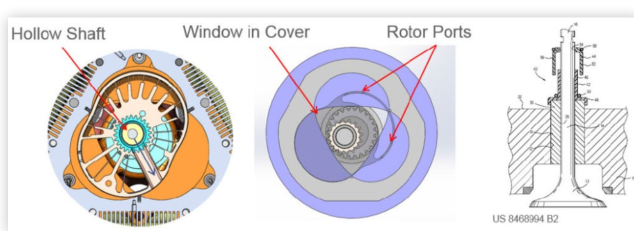
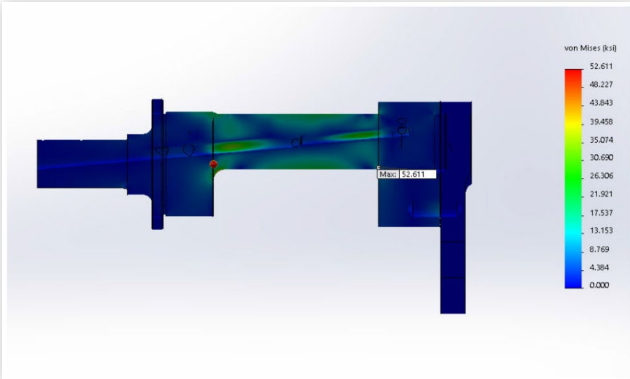


FIGURE 13 X4 Crankshaft (showing FEA result of stress from gas loading)



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performance parameters such as minimum oil film thickness, oil exit temperature, and peak film pressure. The GT Suite bottom end model is shown in Figure 14.

Key bearing performance parameters are shown in Figure 15.

The calculated oil film pressure, thickness and temperature are acceptable, but the friction is higher than desirable at over 1 hp per bearing. These values were acceptable for initial development stages, and will be reduced as development progresses.

Gears

The GT Suite bottom end model was also used to predict loading of the phasing gear. The phasing gear is used to prevent collision of the rotor with the housing as the crankshaft rotates. Due to high loads on the rotor, X4 requires a ring and pinion set on each side of the rotor. This configuration is shown in Figure 16.

Because of the dual gears, there was concern about misalignment, and the gears “fighting each other”. This would be a high risk in a rigid system with no clearances. In order to allow for expected deviations from an ideal case (due to manufacturing accuracy, thermal expansion, mechanical deflection), a crowned tooth profile was designed for the ring gears such that they initially contact the pinion on a point rather than along a line. A FEA prediction done via Solidworks Simulation was used to estimate contact stress for various levels of crowning using GT-Crank predicted loading. An example is shown in Figure 17.

Extensive work was done to study the optimal amount of crowning for the expected misalignment. For every level of misalignment there is an optimum crowning value which minimizes contact and bending stress by moving the contact point far enough away from the edge of the gear without reducing the deformed contact patch area too much.

Another factor which enabled the use of dual gears is the journal bearing design, allowing more compliance than a roller bearing design. Any deviation of the rotor/pinion shaft from

FIGURE 14 GT Suite Bottom End Model

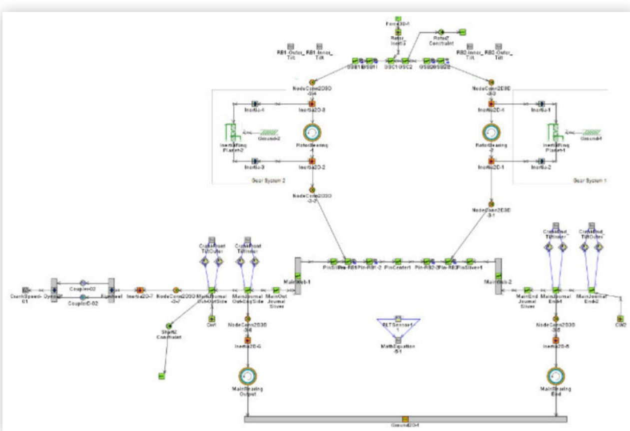

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FIGURE 15 X4 Bearing Predictions

Bearing Operating Conditions - 7000 RPM Full Load	Units	Limit	Rotor Bearing 1	Rotor Bearing 2	Main Bearing (Output)	Main Bearing (Front End)
Min oil film thickness	[micron]	1	1.38	1.69	4.32	4.9
Peak Oil Film Pressure	[Mpa]	150-250	166	159	109	87.1
Friction	[hp]	1	0.723	0.713	1.04	1.48
Oil Outlet Temperature	[C]	140	130	128	124	134

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FIGURE 16 X4 Cross Section with Ring Gears Shown

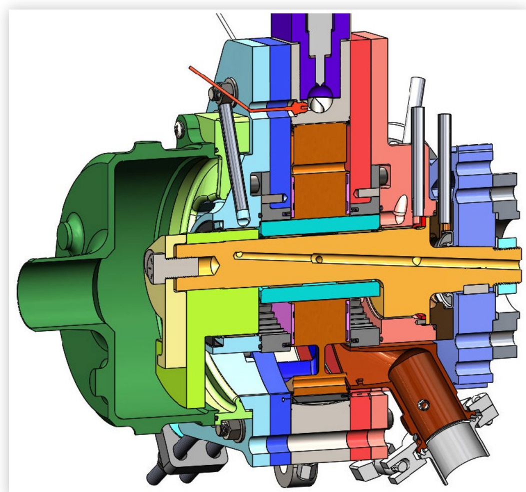

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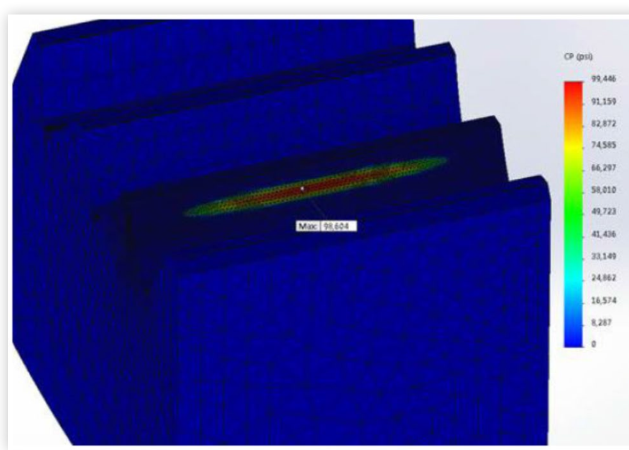
FIGURE 17 Crowned Tooth Stress Distribution



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its ideal position (in any degree of freedom) can be taken up by a combination of backlash, crowning, and bearing clearance to allow load sharing between gears.

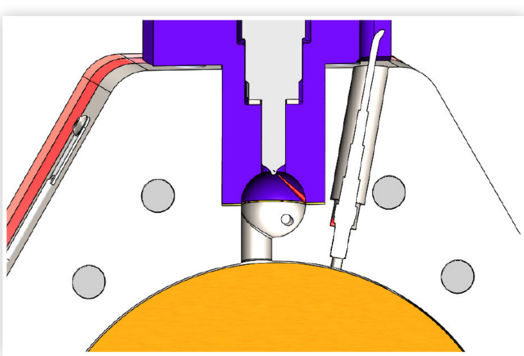
Predicted contact and bending stresses were low enough to use conventional gear materials and heat treatment processes with reasonable face widths for each ring gear.

Combustion System

As the X4 engine is intended to run at up to 7000 RPM, one of the keys to enabling high performance is combustion speed. It is well known that Indirect Injection (IDI) type combustion is typically capable of achieving higher engine speeds when compared to a direct injection (DI) CI counterpart [6]. With IDI, fuel is injected into a pre-chamber, where charge motion is high due to the inrush of compressing air from the main chamber. Since the velocity of the air in the chamber increases with engine speed, IDI typically allows faster engine operating speeds [6]. As a trade-off, typically there is a 10 to 17% efficiency penalty partly due to increased heat transfer and the slower burn characteristic typical of IDI [8] [6]. Because the initial program objective was only intended to show the mechanical integrity of the core engine components, IDI type chambers were selected for the first prototype. DI type chambers have also been used, and also showed good combustion without requiring a glow plug for ignition. The IDI chamber design is shown in cross section in Figure 18.

The purple half of the round chamber is an insert that can be spaced up by inserting a spacer between it and the main housing (white), which allows different compression ratios to be evaluated. Additionally, the diameter of the throat connecting the main chamber (orange) to the IDI chamber started with the smallest estimate (6 mm), and was progressively drilled out during testing. The fuel injector and pump used are standard components for automotive use and will eventually be replaced by custom, smaller components. Not shown is a glow-plug (which was not required for startup at room temperatures), and on the right is a pressure transducer.

FIGURE 18 IDI Chamber Design



Testing

Some images of engine parts and assembly are shown in figures 19-21. LiquidPiston's AC dyno test cell with the X4 engine mounted is shown in Figure 22. The engine was equipped with in-cylinder pressure transducers, torque sensor, various thermocouples, and was controlled by a National Instruments / Drivven data acquisition / control system.

Preliminary testing focused on achieving >108 bar of chamber pressure, as the X geometry had never been subjected to such loads. The goal of this phase of development was to show that the bearings, seals, and gears were adequate, and that this "Test Rig" was ready for performance optimization. This Test Rig has no cooling system, so the engine was run only briefly (~ 1 minute at a time) to get an initial look at sealing and component wear.

Before firing the engine, motoring pressure traces were used to check the expected seal function.

FIGURE 19 Rotor Assembly

FIGURE 20 Side Housing with Windows

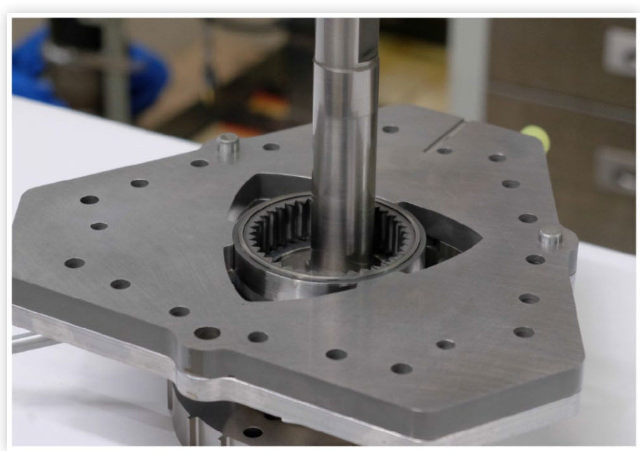


FIGURE 21 X4 assembly, view from Flywheel Side

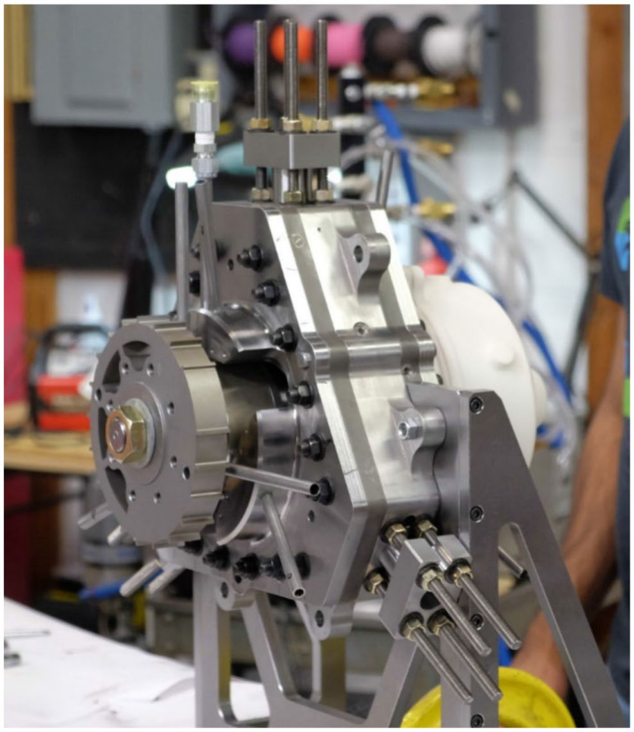


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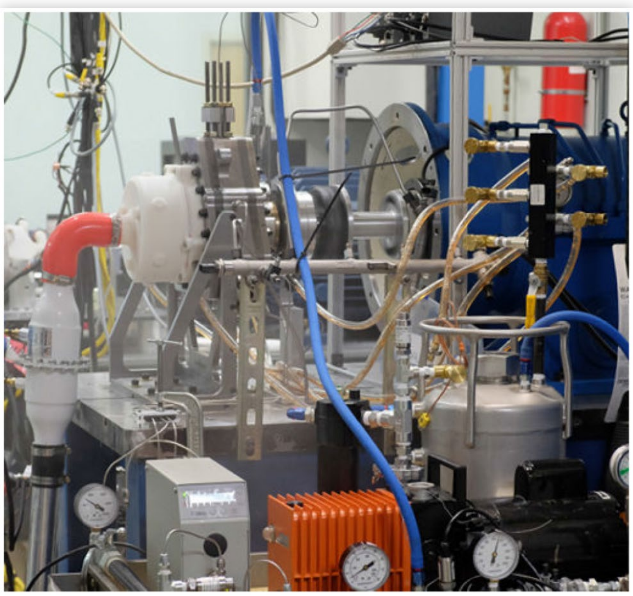
FIGURE 22 X4 Test Setup



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As seen in Figure 23, the pre-chamber pressure lags the main chamber pressure slightly and reaches a lower peak value. Figure 24 below shows good agreement between GT Power and experimental data, indicating a reasonable understanding of the blow-by orifice size. This motoring data shown below was at a low compression ratio (16:1). The compression ratio was adjustable up to 26:1 with the combustion chamber inserts.

Figure 25 shows that during firing cycles (fuel injected), the peak pressure in the pre-chamber is higher than in the main chamber due to the restriction of the throat (as expected).

FIGURE 23 Motoring Pressure Traces for 3 Speeds; blue curves are the pressures in the main chamber; red curves are in the IDI chamber.

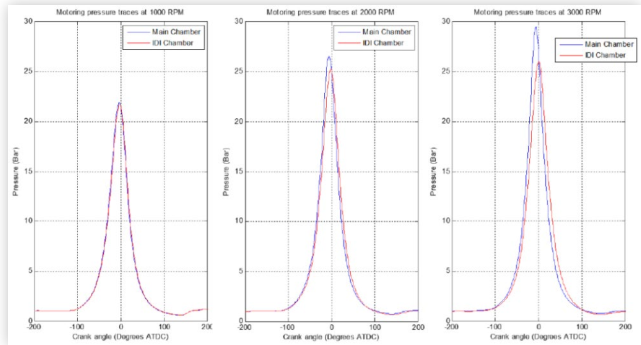

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FIGURE 24 Motoring Pressure (model vs. experiment)

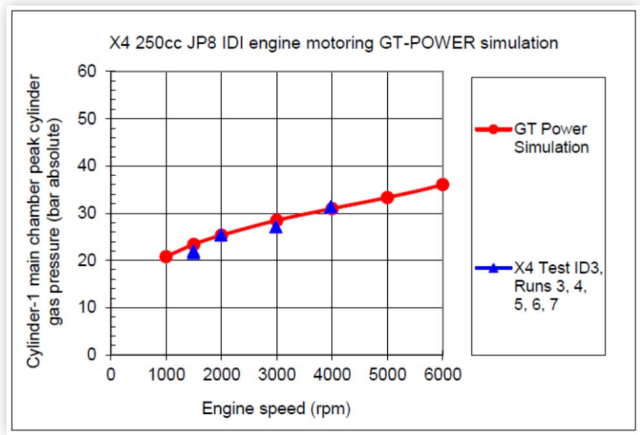

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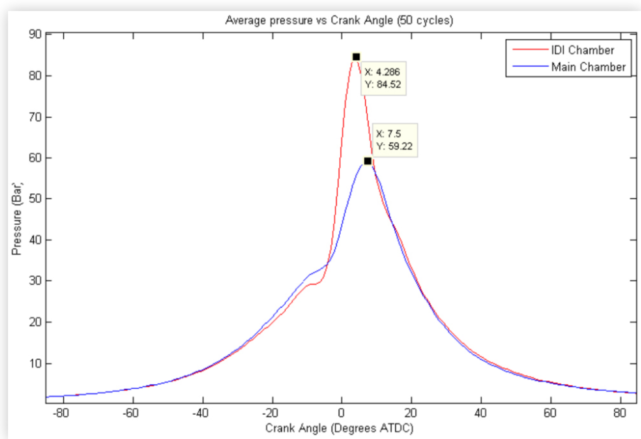
FIGURE 25 Firing Pressure Traces, 3000 RPM



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Performance of the engine improved with larger and larger throat sizes, so the next iteration of hardware will move back to a direct injection (DI) design. Combustion speed will be optimized through the use of pre-injections, chamber geometry to control charge motion, and compression ratio to control charge temperature.

As for peak pressures recorded, the highest measured was almost 150 bar, exceeding the mechanical verification objective pressure of 108 bar (Figure 26).

FIGURE 26 Highest Pressure Cycles

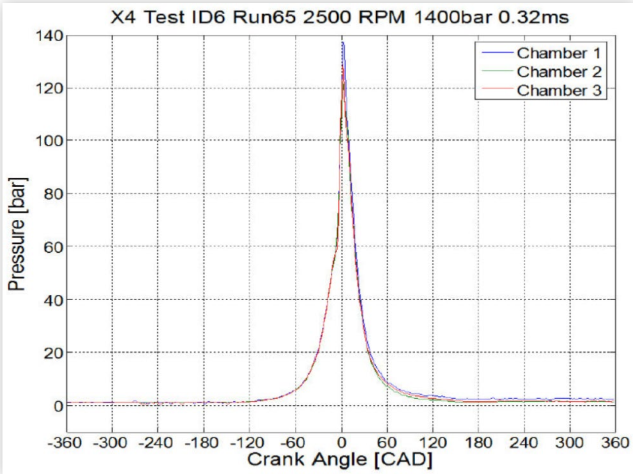
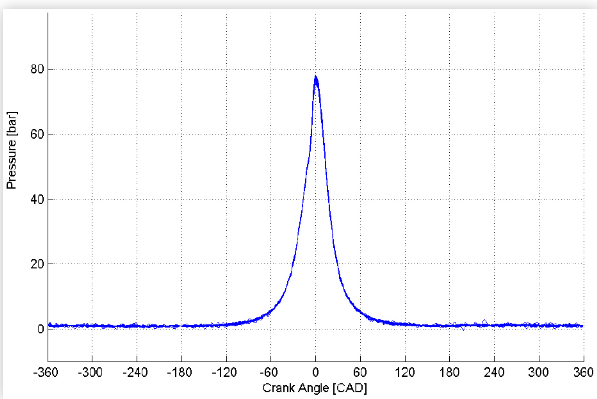


FIGURE 27 Example of Pressure (bar) vs. Crank Angle (deg.) of 50 consecutive firing cycles overlaid, showing COV IMEP of 0.4%. This data had early fuel injection with objective of increased cylinder pressure rather than increased IMEP.



The engine was fired on all three chambers for up to one minute. One example is shown in Figure 27 which shows 50 back-to-back cycles overlaid. The Coefficient of Variation (COV) of IMEP during firing was very low, less than .5%.

Disassembly of the engine after several hours of intermittent testing under these conditions showed no damaged hardware, with light polishing of the gears, crankshaft, and seals. No wear was measurable beyond what is considered normal break-in.

Future Work

The work presented in this paper is just the early stages (first year) of development of the X4. Future work on the X4 program will focus on achieving high efficiency targets (45%), and high power (>30 kW). A cooling system is currently being developed for the engine which includes a water jacket for the housing as well as an oil cooling supply for the rotor. The next version of the engine, which is expected to run cooled / steady state, can then be optimized for indicated efficiency and power, while still using many test cell support systems. Additional effort will then focus on optimization of the engine for brake power / efficiency (friction and ancillary component power reduction), improving durability, characterizing noise and vibration (NVH), and characterizing and reducing emissions as necessary. While the emissions have yet to be assessed, there is some concern that constant volume combustion could produce higher peak cylinder pressures and therefore increased NOx. However, the GT modeling shows that cylinder pressures and temperatures of the naturally aspirated X4 are not excessive in comparison to turbocharged diesels, which indicates that NOx levels may be similar to that of a piston engine, and similar emissions mitigation techniques used on piston engines may be applicable.

Finally, the balance of plant (oil pump, fuel pump, injectors, etc.) will be customized and packaged to the X4 engine.

Conclusions

The first year development for LiquidPiston's 'X4', a 30 kW heavy-fueled rotary compression ignition engine prototype was summarized. This work represents important progress in development, and focused on selecting appropriate displacement, basic engine layout / influence of geometric variables, and creating baseline performance models. The engine successfully completed its objectives of mechanical validation, surviving short operation at nearly 150 bar of peak cylinder firing pressure. A GT model was calibrated based on available engine data, and showed that seals performed adequately and that blowby was significantly reduced compared to Wankel type rotary engines.

The work continues, and if successful, will result in a JP-8 fueled, rotary, CI engine with specific power and specific density significantly improving upon today's state of the art. As a high-speed diesel, the engine will demonstrate its advantages particularly in applications that need power more than torque, and thus the engine couples naturally with high speed electric generator. A further advantage is that electric machine power density and efficiency tends to improve with increased speed as well. Such a compact and efficient power system could be used in hybrid electric applications, for mobile power, or for primary propulsion for ground, sea and air vehicles.

References

1. Shkolnik, N. and Shkolnik, A., "Rotary High Efficiency Hybrid Cycle Engine," SAE Technical Paper 2008-01-2448, 2008, doi:10.4271/2008-01-2448.
2. Shkolnik, A., Littera, D., Nickerson, M., Shkolnik, N. et al., "Development of a Small Rotary SI/CI Combustion Engine," SAE Technical Paper 2014-32-0104, 2014, doi:10.4271/2014-32-0104.
3. Littera, D., Kopache, A., Machamada Sun, C. et al., "Development of the XMv3 High Efficiency Cycloidal Engine," SAE Technical Paper 2015-32-0719.

4. "How It Works." LiquidPiston. Accessed Jan 22, 2018. http://liquidpiston.com/technology/how-it-works/.
5. Wankel, Felix. Rotary internal combustion engine. US Patent US2988065 A, filed Nov 17, 1958, and issued Jun 13, 1961.
6. Blair, G.P., "*Design and Simulation of Four-Stroke Engines*," (Warrandale, PA, SAE International, 1999).
7. Heywood, J.B., "*Internal Combustion Engine Fundamentals*," (New York, McGraw-Hill, 1988).
8. Atkins, R.D., "*An Introduction to Engine Testing and Development*," (Warrandale, PA, SAE International, 2009).
9. Leboeuf, M., Picard, M., Shkolnik, A., Shkolnik, N., et al., "Performance of a Low Blowby Sealing System for a High Efficiency Rotary Engine" SAE Technical Paper 2018-01-0372. Manuscript submitted for publication (2018 SAE World Congress), and is under revision.

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Definitions/Abbreviations

CI - compression ignition

SI - spark ignition

HEHC - High Efficiency Hybrid Cycle

DOD - Department of Defense

E - eccentricity

Rr - roller radius

R - radius

PCP - peak cylinder pressure

IDI - indirect injection

DI - direct injection

FEA - finite element analysis

TDC - top dead center

DARPA - Defense Advanced Research Projects Agency

COV - coefficient of variation

IMEP - indicated mean effective pressure

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