

Measurement and Prediction of Heat Transfer Losses on the XMv3 Rotary Engine

Tiago J. Costa
Universidade do Minho

Mark Nickerson and Daniele Littera
LiquidPiston Inc

Jorge Martins
Universidade do Minho

Alexander Shkolnik and Nikolay Shkolnik
LiquidPiston Inc

Francisco Brito
Universidade do Minho

ABSTRACT

This paper describes predictive models and validation experiments used to quantify the in-chamber heat transfer of LiquidPiston's rotary 70cc SI "XMv3" engine.

The XMv3 engine is air cooled, with separate cooling flow paths for the stationary parts and the rotor. The heat transfer rate to the stationary parts was measured by thermal energy balance of that circuit's cooling air. However, because the rotor's cooling air mixes internally with the engine's exhaust gas, a similar procedure was not practical for the rotor circuit. Instead, a CONVERGE CFD model was developed, and used together with GT-POWER to derive boundary conditions to estimate a ratio between rotor and stationary parts heat transfer, thus allowing estimation of rotor and total heat losses.

For both cases studied (5000 and 9000 rpm under full load), the rotor's heat loss was found to be ~60% that of the stationary parts, and overall heat losses were less than 35% of supplied fuel energy.

The significance of this work relates to the following facts:

- It represents the first time that heat transfer was quantified for the "X" engine architecture;
- Preliminary experimental and modelling results show reasonable correlation
- The predictive models developed will inform future engine cooling system optimization work, leading to higher power densities and thermal efficiencies. Results for these two metrics are already market competitive in the 3 horsepower engine size.

CITATION: Costa, T., Nickerson, M., Littera, D., Martins, J. et al., "Measurement and Prediction of Heat Transfer Losses on the XMv3 Rotary Engine," *SAE Int. J. Engines* 9(4):2016, doi:10.4271/2016-32-0033.

INTRODUCTION

Heat loss from the working fluid to the combustion chamber is an important driver for engine performance, durability, fuel economy, and exhaust emissions. For a given mass of fuel within the chamber, higher heat transfer to the walls lowers the average gas temperature and pressure, and therefore reduces the work produced per cycle. As engine size decreases, the energy fraction lost through heat transfer

tends to increase due to the associated increase in surface area to volume ratio. Engine architecture also plays a big rule on this ratio, with rotary engines having a higher ratio than conventional piston engines for the same chamber displacement [1], [2]. Overall, the amount of heat transferred to the engine walls is predominantly affected by the surface to volume ratio at Top Dead Center (TDC).



charge motion and flame quenching distance (both affecting the heat transfer coefficient), the difference between gas and wall temperature, and the time available for heat transfer to occur (engine speed).

LiquidPiston's "X" architecture is a rotary engine embodiment similar in some aspects to the Wankel. A significant benefit of the "X" engine architecture is the ability to more closely follow LiquidPiston's optimized 4-stroke cycle, dubbed the High Efficiency Hybrid Cycle (HEHC) [3], which incorporates both constant volume combustion and over-expansion [4], [5]. The surface area to volume ratio (SA/V) of the combustion chamber is lower for "X" architecture, compared to piston engines of similar displacement and performance. However, at the beginning of the expansion stroke, the combustion chamber opens into a larger area that has rather larger SA/V than piston engines. It is, therefore, important to quantify real-life heat transfer losses, and also to develop predictive models for this purpose.

The XMv3 (at 70cc total displacement) is LiquidPiston's smallest "X" engine prototype, and it is cooled by a crankshaft mounted centrifugal fan. The design of the cooling circuit separates airflow paths for 1) all stationary parts (main housing, side covers, etc.), and 2) the rotor (also the most challenging component to cool). Heat transfer through the stationary parts was measured experimentally by a standard thermal energy balance technique. However, a similar procedure was not possible for the rotor circuit, as the exiting cooling air is mixed with engine exhaust flow before those temperatures can be separately measured. Instead, a CONVERGE CFD model was used to derive a ratio between "rotor" and "stationary parts" heat losses, and the ratio then used to estimate rotor and total heat losses during the experiments.

Following a literature review, an overview of the tested XMv3 engine will be presented, focusing on its unique gas flow paths. LPI's experimental methods, GT-POWER 1D, and CONVERGE CFD models are next described. Finally, the calculation procedure used to derive net heat transfer losses is presented. The paper is concluded with a discussion of anticipated future work.

Piston/Wankel Engine Heat Transfer Modeling Methods

For piston engines, empirical correlations have been formulated to predict in-chamber heat transfer losses as functions of engine operating conditions [6]. These models typically calculate instantaneous convective heat transfer coefficients through simplified empirical correlations for flow over flat plates or in pipes. The most widely used piston engine correlation was published by Woschni in 1967 [7]. A heat transfer correlation based on turbulent Couette flow was developed to better predict heat transfer losses in Wankel rotary engines [8]. This model was able to differentiate between the rotor and housing heat transfer coefficients. However, in general, these models are not used to describe the details of local or unsteady heat transfer effects, and must be calibrated using experimental results for different engine types. Once all coefficients are tuned, a completed "0D/1D" model can be used to explain or roughly predict overall engine performance such as power output, fuel consumption, total

heat loss, exhaust temperature, etc. These types of models are most often used in the early stages of engine design, due to their simplicity, albeit at limited accuracy.

Heat transfer in real combustion engines is known to be an unsteady and three-dimensional phenomenon. More recently, thanks to advances in computing, CFD numerical codes have become increasingly valuable tools for performing accurate heat transfer simulations [6]. These models have the advantage of being able to provide more detailed information about the local in-cylinder flow pattern and behavior, at the expense of added complexity and time. A CFD model requires a CAD geometry and an accurate understanding of boundary conditions for fuel, intake air, exhaust, etc., so they are typically used later in the engine design process.

Another challenge in CFD modeling is choosing appropriate sub-models, for example, one to describe near wall heat transfer. An extensive CFD code investigation was performed by Rakopoulos [9], where different wall heat transfer models were compared with experimental heat flux measurements from spark ignition and diesel engines. For the calculation of the wall heat fluxes, an in-house CFD model was validated against numerical results obtained using wall-function formulations, such as those of Han [10], and Huh. Han's model, although having a very simple formulation, consistently captured the magnitude of peak heat flux and its trend during the compression and expansion strokes.

It is of the utmost importance to experimentally measure the heat transfer to the engine components in order to validate the empirical or computational models. This measurement is typically performed by an energy balance analysis as the engine is run under controlled conditions. It is also possible to directly measure heat flux to the combustion chamber walls, but this technique requires the adoption of complex, expensive, and intrusive sensors [11], which in some cases cannot be easily used, for example, to measure the heat flux on moving parts such as a piston or the rotor of a Wankel engine [12].

Wankel Experimental Heat Transfer Results

Due to the similarities between the "X" and the Wankel engine architectures, it is helpful to review some research on Wankel engine cooling losses, with focus on experimental data.

One of the most complete sets of experimental thermal balance data was published by Badgley [13]. A stratified JP-8 charge, 160 horsepower, turbocharged Wankel engine was tested at 8000 RPM. Extensive energy related measurements were taken, allowing the calculation of heat losses with two different methods. The final error between methods was 3% of the fuel energy. These results are relevant because the Wankel engine, has a relatively larger SA/V ratio than a conventional piston engine, which is also true during expansion stroke of "X" engines, so intuitively could suffer from higher heat transfer losses. This was not the case, however, with just 12% of fuel energy in this experiment being rejected to the cooling and oil systems. A typical value for a piston engine would be 20-36% [14]. The reasons for such low heat loss were not discussed in the

work, but there are three likely hypotheses. These are: the unusually low compression ratio (7.5:1), the stratified charge fueling strategy, and relatively high engine speed.

Another report from NASA [15], showed test results of a Mazda 12B spark ignition engine (2 rotor, carbureted, with a 9.4:1 compression ratio, and a total displacement per rotor of 573cc). The results showed a maximum cooling loss of 20% at 4500 rpm.

An important takeaway from these investigations is that rotary engines are capable, under the right circumstances, of quite low heat transfer losses. In fact, no engineering literature was found to report the intuitively expected high Wankel heat losses. With this in mind, it is pertinent to perform a comparison between the “X” and the Wankel engines, in order to understand if similar results might be expected for LiquidPiston’s engine.

ENGINE OVERVIEW

The prototype engine investigated in this work, XMv3, is a spark ignition (SI), four-stroke variant of the “X” architecture, and its development is ongoing. It has a single rotor, three combustion chambers, an overall displacement of 70cc (23 cc per chamber), and produces 3hp at 10,000 rpm. A detailed description of prior XMv3 engine developments can be found in papers [16], [17].

For an animation describing the operation of the XMv3 engine, please visit: <http://liquidpiston.com/technology/x-mini-gasoline/>.

Table 1 lists basic XMv3 engine parameters. Worthy of note is the compression ratio vs. expansion ratio, the difference intentionally designed to provide an adjustable degree of over-expansion by asymmetrical location of the intake and exhaust ports in the rotor. This asymmetry is also seen in the port timing numbers. While the XMv3 configuration as tested had a limited over-expansion ratio of 1.12, some of LiquidPiston’s other prototype engines, designed for higher efficiency operation on diesel, have significantly higher over-expansion ratios in excess of 1.5.

Table 1. Tested XMv3 engine specifications.

Number of rotors	1
Overall displacement (cc)	69
Displacement per chamber (cc)	23
Nominal compression ratio (CR)	9.26:1
Effective CR	8.07:1
Effective expansion ratio	9.05:1
Over expansion ratio (Effective expansion ratio/ Effective CR)	1.12
Intake valve closing (°CA BTDC)	135
Exhaust valve opening (°CA ATDC)	161

Comparison to Wankel

In many ways, the “X” engine can be thought of as an inverted Wankel. The rotor in both engine types is mounted on an eccentric shaft, so that its center moves on a circle, with the rotor being turned by a gear. In the case of the Wankel, however, the small gear is fixed to the stationary housing, while in the “X” engine the small gear is fixed to the moving rotor. As can be seen in Figure 1, the “X” engine rotor has the same epitrochoid shape as the Wankel housing. These parts are both divided into two symmetric lobes, one seeing mainly intake/exhaust strokes, and the other compression and expansion strokes (the “hot side”). The Wankel has a three lobed rotor, with each of the lobes defining a combustion chamber, similar to the housing of the “X” engine.

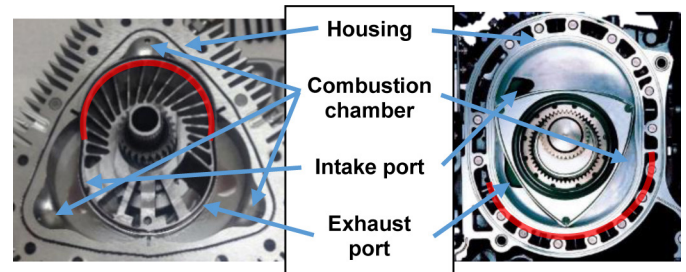


Figure 1. Comparison between the Wankel engine (right), and the “X” engine (left). The hot region of each engine is marked with a red arc.

The “X” engine has three chambers, delineated by the rotor, housing and two cover plates, and sealed by the rotor face seals and three stationary apex seals. The Wankel’s three moving chambers are sealed to the side covers similarly by rotor face seals, and separated by rotor mounted (moving) apex seals. The Wankel “hot part” is the housing section that only sees combustion/expansion and exhaust strokes. Whereas the Wankel engine executes the 4-stroke cycle while moving the charge over different parts of the engine, the X engine executes the three 4-stroke cycles simultaneously in three independent combustion chambers.

The Wankel engine intake and exhaust ports are either located “peripherally” in the main housing, or “side mounted” in the side plates. The “X” engine ports, when configured as a 4-stroke, are located peripherally in the rotor.

As a result of the aforementioned porting and architecture differences, there is a significant difference between the gas flow in the combustion chamber of each engine, as shown in Figure 2. In the Wankel, gas travels with the rotor as it sweeps across the stationary housing surface, similarly to Couette flow, described as a parallel-plate flow developed by fixing one plate and moving the other one with constant velocity [18]. The “X” engine flow, on the other hand, is more similar to that of a piston engine, with the rotor rotating (relatively slowly) at the same time as translating (relatively quickly) towards or away from the housing profile, similarly to a piston moving relative to a cylinder head. The Wankel combustion chamber geometry is defined by the rotor surface geometry (like bowl-in-piston), whereas the “X” engine combustion chambers are defined by the housing, thus allowing more freedom for the X engine chamber design.

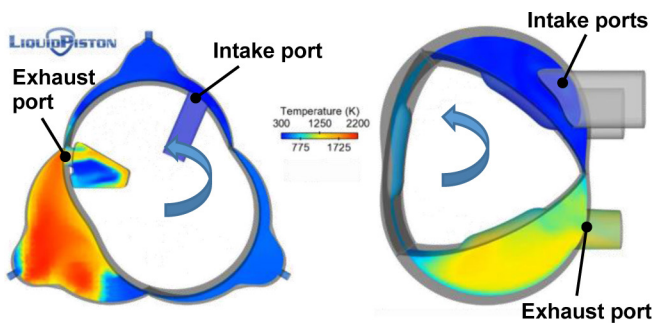


Figure 2. Fluid motion comparison between the “X” engine (left) and the Wankel engine (right).

Another important factor to compare between the two engines is the relative rotational speed of the rotor and crankshaft. The Wankel rotor runs at 1/3 shaft RPM, whereas the “X” engine ratio is 1/2. By having relatively higher rotational speed for a given airflow rate and engine size, the “X” rotor will have higher tangential velocities, possibly leading to higher fluid shear stress near the rotor walls. This could tend to increase local heat transfer coefficients if comparing at the same shaft RPM.

In conclusion, while both engines belong to the same geometrical family of rotary engines, there are significant differences in internal gas flow. These differences will ultimately affect the amount of energy that each architecture loses to heat transfer. Therefore, experimental measurement and development of a new model is necessary to best inform future work, rather than attempting to adapt existing design procedures from Wankel or piston engines to the “X” architecture.

XMv3 Gas Flow Paths

The XMv3 intake, exhaust, and cooling paths are significantly different from those of conventional piston/Wankel engines. Figure 3 shows section and front views of the XMv3, with arrows indicating the direction and position of the different flows.

Intake air (green) enters the engine through a bellmouth, where a fuel injector (not shown) is placed along the same axis as the crankshaft. The premixed charge then proceeds through a hollow crankshaft and turns 90 degrees, passing through holes drilled radially through the shaft. These holes open to a channel in the rotor, and then to the intake port, where the charge enters the working chamber.

During the exhaust stroke, combustion products (marked with red arrows) exit the active chamber via the rotor exhaust port, entering the interior volume of the rotor, therefore mixing with the rotor cooling air (blue arrows). Fan forced rotor cooling air continually passes through the engine via three matching triangular ports (red outlines) in each of the engine’s side plates.

The engine’s stationary elements are also air cooled, this circuit defined by plastic shrouds covering the exterior of the engine (Figure 4). The shroud’s insides are fitted with air distribution vanes. Note that for these heat transfer experiments, the shaft mounted cooling fan was replaced with an external blower, allowing for more convenient measurement and control of each flow rate and temperature.

The exiting cooling air, and in the rotor circuit case, also the mixed exhaust gases, leave the engine shrouds through three segregated openings each, shown again in Figure 4

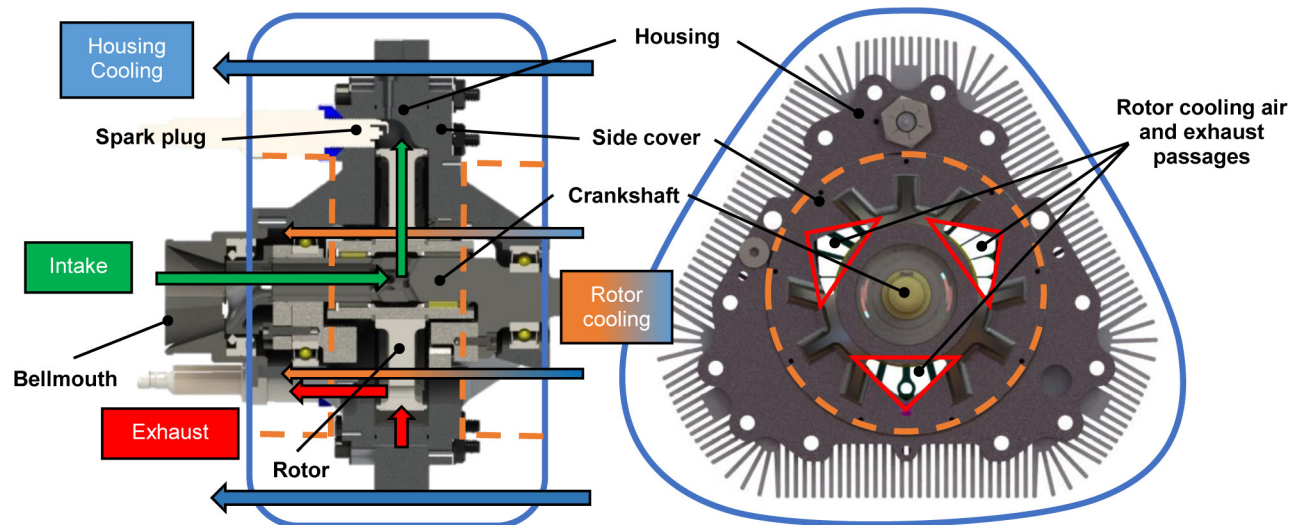


Figure 3. XMv3 engine gas path flows. Green is the intake charge, red is the exhaust, and blue is the cooling flow. The blue outlines represent the outer engine shroud, which separates the “stationary part” and “rotor cooling” flow paths at the orange dashed lines. Shown in red, three triangular openings on each side of the engine allow the passage of rotor cooling air and exhaust flows.