# THE NEW RIPL ROTARY ENGINE

**The Ripl Rotary internal combustion engine is not an improved Wankel iterant (i.e. a Reuleaux triangular rotor rotating inside an oval-like housing). It is an entirely novel solution for a rotary ICE.**

It follows the Atkinson cycle and is fully configurable over the intake volume, the compression ratio, and the final expansion volume resulting in optimised extraction of energy from the fuel.

It can compress the fuel air mix to a chosen volume before ignition and maintain that volume during burn before starting that power stroke, without interrupting engine rotation.

The Ripl Rotary makes pops, rather than a stream of fast-moving exhaust air, meaning effective noise dampening can be achieved.

The engine design is entirely scalable, with only 4 bearing-mounted moving parts and 1 seal, working on both sides of the piston head at all times.

Results from an independent thermodynamic study of the Ripl Rotary ICE suggest a step-change capability:

- Relative efficiency improvement over piston engine of 40.1%
- 4x power to weight over piston engine
- 1kg of engine weight will return 9.8kw of power generation @3,000rpm

The full report (IP redacted) follows ⟶

Further information: invent@ripl.ltd www.ripl.ltd

ripl

# Technical Report:

## Simplified Thermodynamic Study and Analysis of the RIPL Rotary Internal Combustion Engine

Company:

**RIPL Ltd**

Date:

**13th December 2022**

Version:

**01**

Telephone: +44 (0) 1872 260005
Email: info@riventa.com
WWW.RIVENTA.COM

Unit B1 Walker Business Park
Threemilestone Industrial Estate, Threemilestone,
Truro, Cornwall TR4 9FB United Kingdom

# Document Properties

| | | |
|---|---|---|
| Report by | **Owain Hughes** B.Eng (Hons) | |
| | Riventa Ltd :: 21 November 2022 | |
| Approved by | **Thomas Clifford** B.Eng (Hons), ACSM, PhD, CEng MIMMM | T. C. Clifford |
| | Riventa Ltd :: 28 November 2022 | |

# Version Control

| Version | Issued | Comment |
|---|---|---|
| Version 01 | 13/12/2022 | First Issue |
| | | |




Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

# Contents

# 1 Executive Summary

- A digital model has been created to simulate the performance of a conventional piston internal combustion engine and contrast this, with comparable assumptions, to a novel design of rotary internal combustion engine proposed by RIPL Ltd.
- The key novel elements are that the;
  - Induction, compression, and expansion process are not the same volume, and these volume ratios can be defined and optimised
  - Combustion process happens at a constant volume where the duration can be defined and optimised
- The following engine performance data was calculated;
  - RIPL rotary engine efficiency (v1.2) **51.0%**
  - Conventional piston engine efficiency **36.4%**
  - Relative improvement **40.1%**
- Due to the there being [redacted] power strokes per revolution and an increase in efficiency of the RIPL engine, there is a **four-fold** increase in the power to weight ratio over the contemporary technology (Ford Ecoboost V6 3.5L engine).

## 2 Engineering Summary

- Three separate design versions of the model were simulated with each iteration designed to improve the limitations discovered in the previous iteration
  - Initial RIPL Design v1.0
  - RIPL Design with Variable Spark Ignition v1.1 see 7.2.
  - The inclusion of a combustion chamber at constant volume v1.2
- Increasing the volume of the expansion stage beyond the standard 1:1 Intake Volume: Expansion Volume, yields improvements in thermal efficiency of the RIPL engine under the given conditions.
  - For v1.2, depending on the compression ratio, and optimising for expansion ratio and spark angle yields an improvement in efficiency of:
    - 11.7 – 15.5% from the initial 1:1 expansion ratio, with the improvement in efficiency reducing with higher compression ratios.
    - 20.3 – 24.2% from a standard piston engine, with the improvement in efficiency reducing with higher compression ratios.
- The RIPL engine (v1.0-1.1) at the standard 1:1 expansion ratio has a worse performance than a standard piston engine with a shortfall in efficiency of 9.2% (v1.0) and 2.0% (v1.1). This indicates a potential area for improvement in the engine design by modifying the expansion stage of the engine to maximise combustion in periods of higher pressure or the installation of a combustion chamber seen in v1.2.
  - This improvement would also reduce the effects of slow/delayed ignition on the overall engine efficiency.
- Power density for a RIPL engine is significantly higher than that of a piston engine, with [redacted] power strokes possible per revolution, compared to 0.5 for a 4-stroke engine. Thus, for an equivalent power output multiple pistons would be needed.
  - Under the conditions set out in 7.1. the indicated power and break power for the engines per cylinder is:
    - RIPL Engine v1.0: 3.17$kW_i$, 2,27$kW_B$
    - RIPL Engine v1.1: 4.07$kW_i$, 2.93$kW_B$
    - RIPL Engine v1.2: 5.03$kW_i$, 3.61$kW_B$
    - Piston Engine (4-Stroke): 0.60$kW_i$, 0.43$kW_B$
  - The fuel consumption of the RIPL engine will be treble that of typical piston of the same initial volume because of having [redacted] power strokes per revolution.
  - With the increase in power strokes per revolution, and the increase in efficiency, for the same swept volume and compared to a Ford Ecoboost V6 3.5L engine, the RIPL engine has the following improvements in output power per cylinder of equal initial volume and power to weight ratios
    - RIPL v1.0: 5.2x Power Output, 3.2x Power to Weight Ratio

- RIPL v1.1: 6.8x Power Output, 3.6x Power to Weight Ratio
- RIPL v1.2: 8.4x Power Output, 4.3x Power to Weight Ratio

- Approximating mechanical and frictional losses within the engines under the conditions set out in 5.1 yields an efficiency of:
    - RIPL Engine v1.0: 32.1%
    - RIPL Engine v1.1: 41.5%
    - RIPL Engine v1.2: 51.0%
    - Piston Engine: 36.4%
    - A net improvement of -12%, +14% and +40% respectively (relative to the piston engine)
- Several approximations and assumptions have been used to perform this analysis, it is recommended to conduct more complex analysis through further modelling, CFD and real-world testing to improve the overall accuracy of the model.

# 3 Project Scope

The specified project scope for this study is as follows.

- Simplified Thermodynamic analysis of engine to include:
    - Pressure-Volume & Temperature-Entropy graphs for a complete cycle
    - Pressure, Temperature and Volume relationships to Stroke/rotation position
    - Comparison to an equivalent piston engine
    - Comparison to ideal Otto and Atkinson Cycles
    - The relationship between Fuel Fraction Burned and Combustion Time
    - The relationship between Ignition Advance and Net Work
        - Identify optimal ignition angle for given inputs
        - Predicted Power Output
- Using the above thermodynamic analysis identify areas of potential improvement of the model. And model these thermodynamically.
- Approximation of thermal losses through cycle and an estimation of mechanical efficiency from the literature, unless provided by RIPL.
- Produce a theoretical model that will enable RIPL to edit the input parameters to produce the above under different design scenarios.
- Produce an output report from the model detailing all the above, highlighting the efficiencies of the different steps in the process.


Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

# 4 Theoretical Optimum Efficiency

## 4.1 Otto Cycle

An Otto cycle is an idealized thermodynamic cycle that describes the functioning of a typical spark ignition piston engine.

The Otto cycle is a description of what happens to a gas as it is subjected to changes of pressure, temperature, volume, addition of heat, and removal of heat. The gas that is subjected to those changes is called the system. The system, in this case, is defined to be the fluid (gas) within the cylinder. By describing the changes that take place within the system, it will also describe in inverse, the system's effect on the environment.

The Otto cycle is constructed from:

- Top and bottom of the loop: a pair of quasi-parallel and isentropic processes (frictionless, adiabatic reversible).
- Left and right sides of the loop: a pair of parallel isochoric processes (constant volume).

The isentropic process of compression or expansion implies that there will be no inefficiency (loss of mechanical energy), and there be no transfer of heat into or out of the system during that process. The cylinder and piston are assumed to be impermeable to heat during that time. Work is performed on the system during the lower isentropic compression process. Heat flows into the Otto cycle through the left pressurizing process and some of it flows back out through the right depressurizing process. The summation of the work added to the system plus the heat added minus the heat removed yields the net mechanical work generated by the system [18].

Typical piston ICE’s operate to the same principles as the Otto cycle, the stages of which can be seen outlined below.

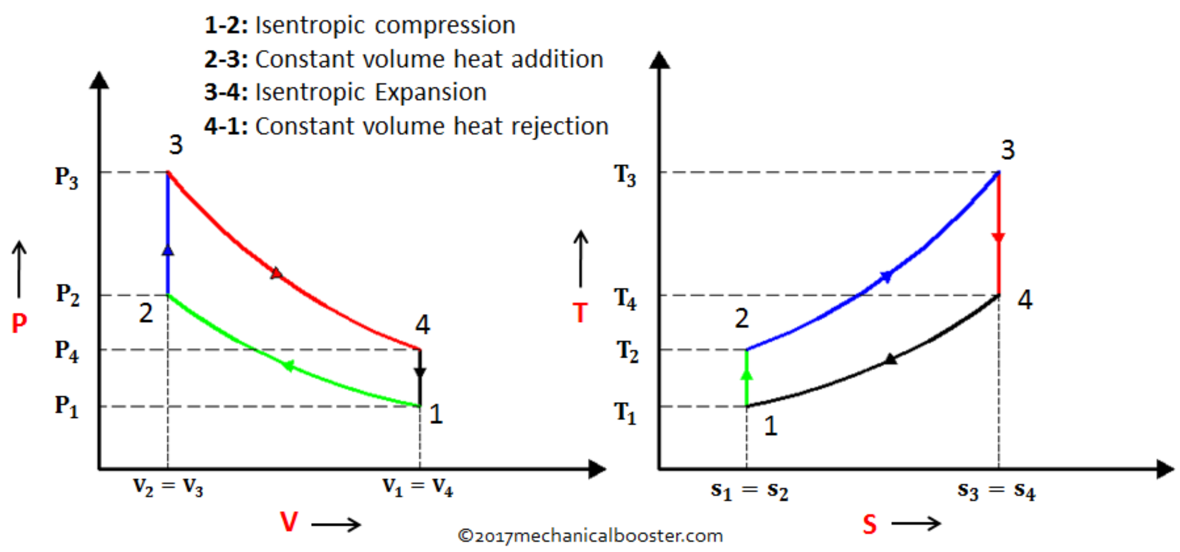



**Figure 1 Pressure-Volume and Temperature-Entropy diagrams of the Otto Cycle [1]**

The theoretical efficiency of an Otto cycle can be calculated using the formula below.

$$\eta_{Otto} = 1 - \left(\frac{V_2}{V_1}\right)^{\gamma-1} = 1 - \frac{1}{CR^{\gamma-1}}$$

**Equation 1 Otto Efficiency Equation**

Where;

$\gamma$ = Ratio of specific heat capacities (specific heat capacity at constant pressure / specific heat capacity at constant volume)

$CR$ = Compression ratio

$V$ = Volume ($m^3$)

This provides an optimal benchmark with which to compare the performance of ICE designs. In practice the Otto efficiency is not achievable as instantaneous combustion is not possible and thermal and other losses are present in an engine.

## 4.2 Atkinson Cycle

The design of RIPL's proposed ICE differs in that the initial inducted volume is not required to equate to the final expanded, exhausted volume as per typical ICE designs. This theoretically enables more energy to be extract from the fluid during the expansion phase of the process as compared to contemporary designs. Thus, a thermodynamic cycle that is more comparable to RIPL design is the Atkinson Cycle.

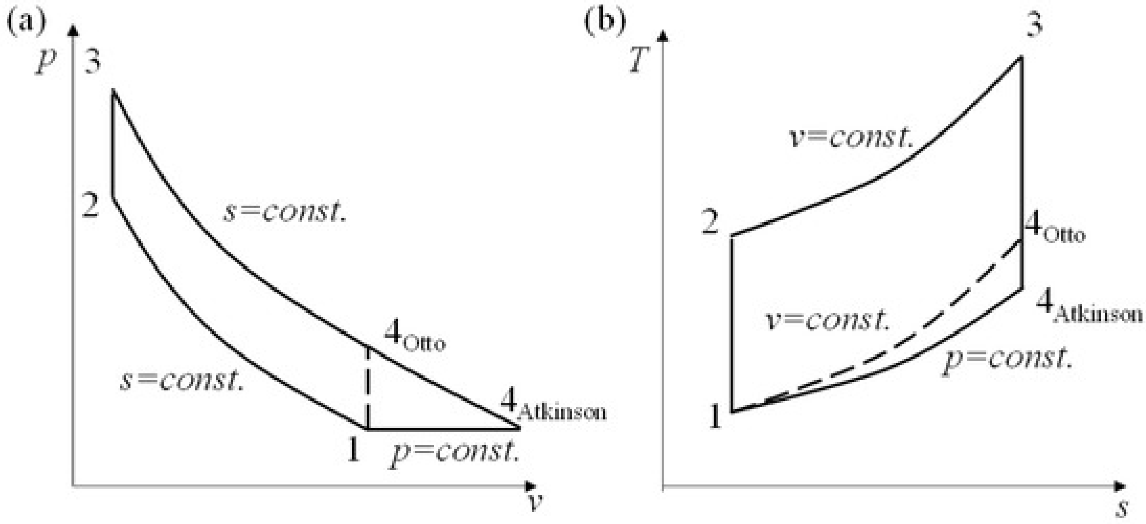



**Figure 2 Atkinson vs Otto Cycle [19]**

The fundamental difference between Atkinson and Otto Cycles is that the isentropic expansion stage is able to expand beyond the initial volume at the beginning of the cycle, before returning to the initial volume during the exhaust phase.

The theoretical efficiency of an Otto cycle can be calculated using the formula below.

$$\eta_{Atk} = 1 - \gamma \frac{CRER - CR}{(CRER)^{\gamma} - CR^{\gamma}}$$

**Equation 2 Atkinson Efficiency Equation**

Where;

$\gamma$ = Ratio of specific heat capacities (specific heat capacity at constant pressure / specific heat capacity at constant volume)

$CR$ = Compression ratio

$ER$ = Expansion Ratio, (Initial Volume: Expanded Volume)

Theoretically the Atkinson cycle can operate at much higher efficiency than that of an Otto cycle given the same compression ratio. For instance, at a compression ratio of 14, an expansion ratio of 4 and a ratio of specific heats of 1.4.

- Otto Cycle Efficiency: 65.4%
- Atkinson Cycle Efficiency: 75.5%
- Proportional improvement: 21.4%

Therefore, the prosed RIPL design will have an intrinsic advantage over the contemporary piston technology.

Ultimately the limit on how much energy can be extracted is limited by the pressure, such that pressure in the expansion chamber does not drop below that of the atmospheric pressure at the exhaust point.

Studies and analysis suggest that extending the expansion process has benefits on both fuel efficiency and thermal efficiency. Andre et al, modelled a toroidal-shaped piston rotary engine and saw a noticeable improvement in efficiencies when varying the expansion ratio [20].

# 5 Engine Design

## 5.1 Initial Design

The proposed RIPL internal combustion engine (ICE) design is centred on a novel approach to rotary engine principles. It is constructed primarily of three segments, that act as separate zones for each step in the thermodynamic cycle.

1. Air Intake and Fuel Injection
2. Compression and Ignition
3. Combustion and Expansion

REMOVED

**Figure 3 Vertical Cross Section of RIPL ICE**

REMOVED

**Figure 4 Horizontal Cross Section of RIPL ICE**

As the rotor advances the engine goes through the following stages.

[redacted]

The optimum ratio of chamber depths (i.e. [redacted]) was unknown and thus the optimum was computed. The remaining dimensions of the engine have been maintained from the issued drawing files. The volume profile used in the calculation has been simplified to remove the size of the [redacted] to simplify the volume profile model, as such chamber volume is taken from port to port.

REMOVED

**Table 1 Fixed Dimensions of RIPL ICE**

Due to the design principles of the proposed RIPL ICE, the engine will be capable of [redacted] power strokes per revolution, compared to three for the conventional 'state-of-the-art' rotary engine: the Wankel. Traditional piston engines usually produce 1 or 0.5 power strokes per revolution depending on if it's a two stroke or four stroke design. This provides an initial indication as to the significant power to weight ratio improvements.


Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

## 5.2 Design Proposals

Initial thermodynamic analysis highlighted two key areas for improvement from the initial design:

- Improvement in engine design to allow spark angle to be optimised is estimated to yield significant efficiency improvements. The model should be reviewed to enable advanced spark ignition.
- Inclusion of a [redacted] at constant volume or a review of the expansion chamber to enable combustion to occur over a less varying volume.

From these suggestions 3 thermodynamic models were made:

- Initial RIPL Design v1.0
- RIPL Design with Variable Spark Ignition v1.1 see 7.2.
- The inclusion of a combustion chamber at constant volume v1.2

The inclusion of a [redacted] is modelled as a duplicated [redacted] to maintain constant volume for the combustion process to occur.

Constant volume combustion can significantly increase efficiency by maximising pressure at the point of minimum volume and ensuring maximum work can be extracted [21]. An added benefit of this means that the engine is less susceptible to limitations in the spark angle as it can combust over a longer time frame, see 7.2.

# 6 Combustion Cycle Theory

The Work Done by an engine is defined by the area enclosed within a Pressure-Volume model, calculated from the following equation;

$$W = \sum_{0}^{2\pi} \frac{dP}{d\theta} V$$

**Equation 3: Work Done for an ICE**

Where;

$W$ = Work Done (J)

$P$ = Pressure ($Nm^{-2}$)

$\theta$ = Rotational position (Radians)

$\pi$ = Pi (~3.142)

Equation 3 enables determination of the Work Done per rotation and given the engine speed, the power of the engine can then be computed.

The pressure profile across the compression and expansion phases of the cycle are modelled according to the laws of isentropic expansion and compression:

$$P_2 = \frac{V_1^{\gamma} P_1}{V_2^{\gamma}}$$

**Equation 4 Isentropic Pressure Volume relationship**

Where;

$\gamma$ = Ratio of specific heat capacities (specific heat capacity at constant pressure / specific heat capacity at constant volume)

During the combustion process, the pressure profile is defined as [2,3,4];

$$dP = \frac{Q_{CV}.m_f.dX_b.(\gamma - 1) - \gamma.P_1 dV}{(V_1 + V_2)/2}$$

**Equation 5**

Where;

$Q_{CV}$ = Fuel calorific value (J/kg)
$m_f$ = Fuel mass (kg)
$dX_b$ = Change in mass fraction burned

The mass fraction burned is defined by [2,4-9];

$$X_b = 1 - e^{\ln(1-\eta_c).(\frac{\theta-\theta_s}{\theta_c})^{(f+1)}}$$

**Equation 6 Mass Fraction Burned**

Where;

$\eta_c$ = Combustion efficiency approximated from the fuel-air equivalence factor [10]

$\Theta s$ = Spark angle from 0° (Radians)

$\Theta c$ = Combustion angle (Radians) approximated from engine speed, spark angle, compression ratio and combustion efficiency [11-12]

$f$ = Form factor of the fuel air mix when ignited

Thermal losses were calculated using the convective heat transfer formula:

$$\dot{Q} = h.A.\Delta T$$

**Equation 7 Convective heat transfer**

Where;

$\dot{Q}$ = Heat transfer from the working fluid to the engine block (J/s)

$h$ = Convective heat transfer coefficient (any units?) approximated from air properties and working fluid velocity [13]

$A$ = Contact area between the engine and the working fluid ($m^2$)

$\Delta T$ = Temperature differential between the working fluid and the engine block (°K)

The effect of thermal energy losses is calculated using the ideal gas laws to determine the change in pressure and temperature.

# 7 Results

The theory outlined in section 6 has been populated for both the novel RIPL rotary engine design and also, by way of a relative benchmark, a contemporary piston engine design. The same theoretical generalisations are therefore made to each engine which provides an equal support comparison and a reliable relative contrast in performance between the two designs of engine appraising efficiency improvement.

## 7.1 Thermodynamic Analysis

### 7.1.1 Assumptions

Several assumptions and parameters where fixed to constrain the extents of the models. Assumed variables which can be edited in the supplied digital model for re-simulations are superscripted with an *.

[redacted]

- Model size was inputted as per the dimensions provided. A scalar is included to enable to model to be scaled up if needed (Dimension Scalar)*
- Fuel Calorific Value: 44MJ/kg * (Petrol was used)
- Air Fuel Ratio: 15.4 * (This is the empirical optimum for efficiency for petrol) [10]
- Air Density at Intake = 1.243kg/m3 *
- Air Fuel Mixture was deemed to have the same ratio of specific heats as air under the given conditions. Research suggests fuel mixture has minimal difference from air. [14]
- Engine Design Speed: 1000rpm *
- Form Factor: 3 * (This is the forms part of the model for combustion dynamics, see 7 for further suggestions)
- Compression Ratio: 14 *
- Expansion Ratio: Optimised to nearest 0.5x ratio *
- Air Inlet Temp: 20C *
- Engine Block Operating Temperature: 200C *
- Convective Heat Transfer Coefficient (Heat Losses to Engine Block): Convective heat transfer assumed peak air-fuel velocity for the approximation.
- Spark Angle: Optimised to nearest degree *
- Piston Initial Volume is equal to that of the supplied RIPL engine design.
- Piston depth is deemed to be 1.5x that of the diameter.
- Drive Train Efficiency: From Empirical Data [15] *
- RIPL and Piston Engine Frictional/Mechanical Losses: From Empirical Data [16-17] *

### 7.1.2 Outputs

The following outputs can be generated using the spreadsheet provided, with any of the variables marked with a * changeable to produce different results. The results listed here are for RIPL Engine v1.2. For previous version see 11.2.

# RIPL Engine Thermodynamic Performance Analysis

Comparitive to a Piston Engine, Otto Cycle and Atkinson Cycle

RIVENTA

## 1. Inputs

| | | | | | |
|---|---|---|---|---|---|
| Fuel Type* | Petrol | | Compression Ratio* | 14 | |
| Engine Design Speed* | 1000 | rpm | Include Thermal Leakage* | Y | |
| Intake Air Temperature* | 20 | °C | Intake Air Density* | 1.243 | kg/m³ |
| RIPL Spark Angle | 120 | ° | Piston Engine Spark Angle | 111 | ° |
| Engine Block Operating Temp* | 200 | °C | Intake Air Pressure* | 101.325 | kPa |
| Form Factor* | 3 | | Dimension Scalar* | 1 | |
| RIPL Expansion Ratio | 4 | | Key: applies to both engines | * | |

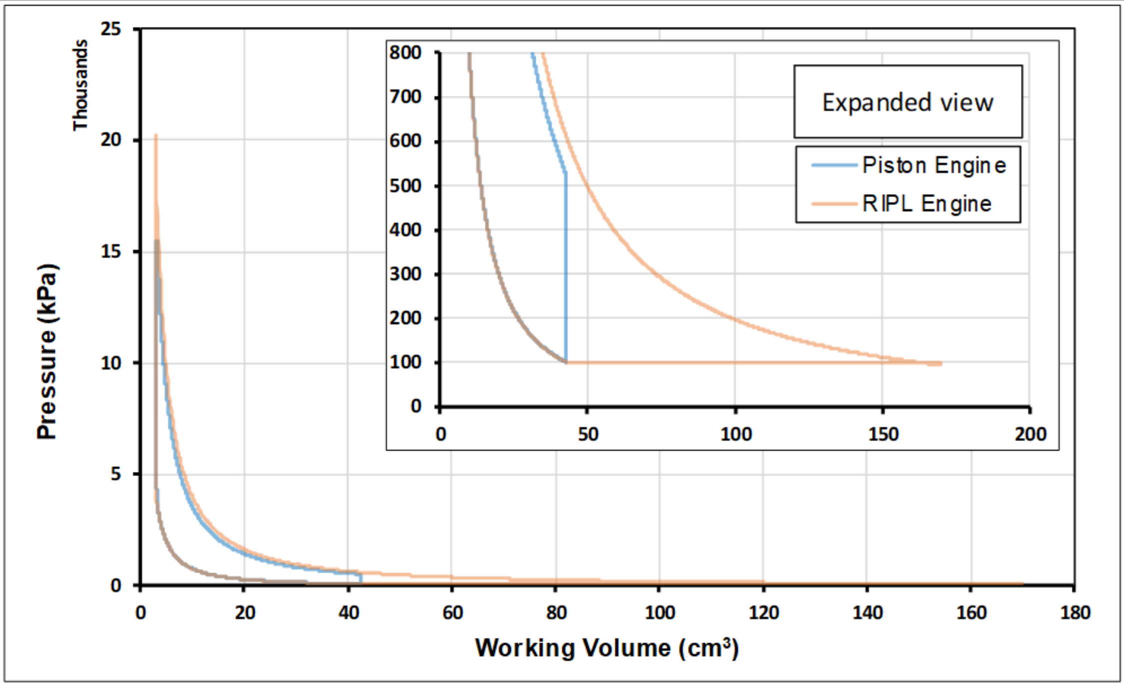
## 2. Thermodynamic Summary Data (does not include mechanical losses)

| | | | | | |
|---|---|---|---|---|---|
| RIPL Engine Efficiency | 70.9% | % | Theoretical Otto Efficiency* | 65.4% | % |
| Piston Engine Effiency | 50.6% | % | RIPL shortfall from Otto Efficiency | -8.4% | % |
| Efficiency Improvement | 40.0% | % | Theoretical Atkinson Efficiency* | 75.5% | % |
| RIPL Thermal Power Output | 15.0 | kW | 2 Stroke Piston Thermal Power Output | 1.19 | kW |
| RIPL Thermal Power Output | 20.2 | HP | 4 Stroke Piston Thermal Power Output | 0.60 | kW |
| Chemical Energy In per Power Stroke* | 141.4 | J/power stroke | 2 Stroke Piston Thermal Power Output | 1.60 | HP |
| RIPL Net Work | 100.3 | J/power stroke | 4 Stroke Piston Thermal Power Output | 0.80 | HP |
| Piston Net Work | 71.6 | J/power stroke | Engine Cc per Cylinder/Chamber | 42.41 | cm³ |

## 3. Engine Dimensions

| | | | | | |
|---|---|---|---|---|---|
| RIPL Stage 1 Depth | 20.0 | mm | RIPL Stage 2 Depth | 2.1 | mm |
| RIPL Stage 1 Inner Radius | 60.0 | mm | RIPL Stage 2 Inner Radius | 65.0 | mm |
| RIPL Stage 1 Outer Radius | 75.0 | mm | RIPL Stage 2 Outer Radius | 75.0 | mm |
| RIPL Stage 3 Depth | 80.0 | mm | | | |
| RIPL Stage 3 Inner Radius | 60.0 | mm | Piston Cylinder Diameter | 33.0 | mm |
| RIPL Stage 3 Outer Radius | 75.0 | mm | Piston Stroke Length | 49.5 | mm |

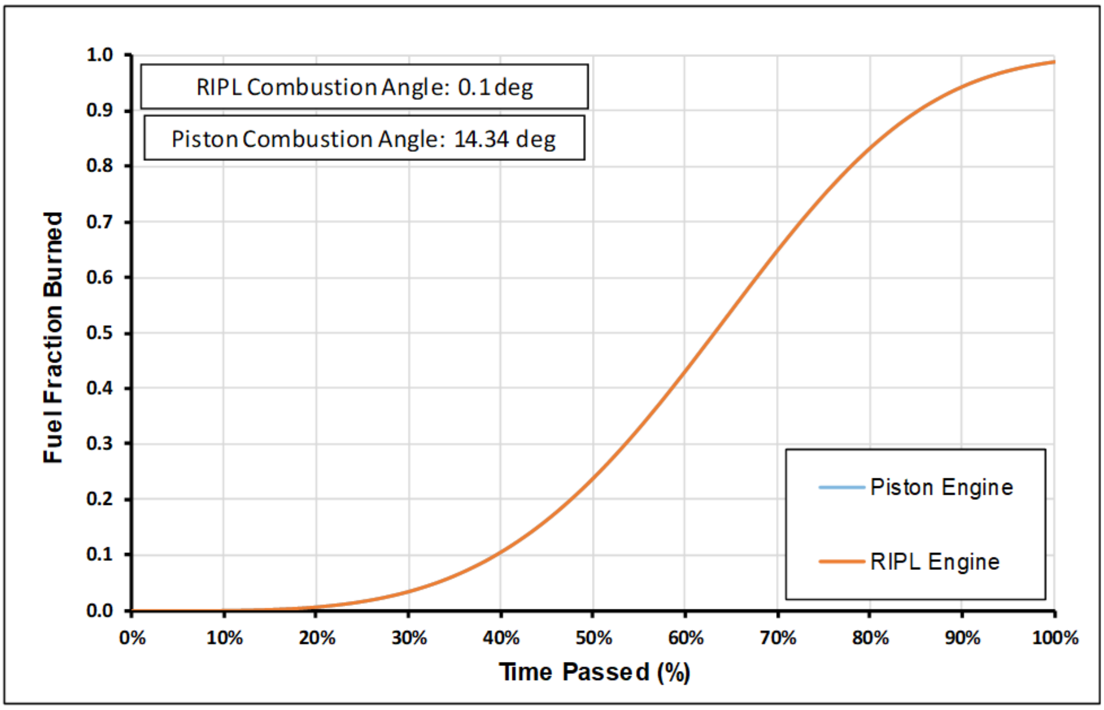
## 4. P-V Graph



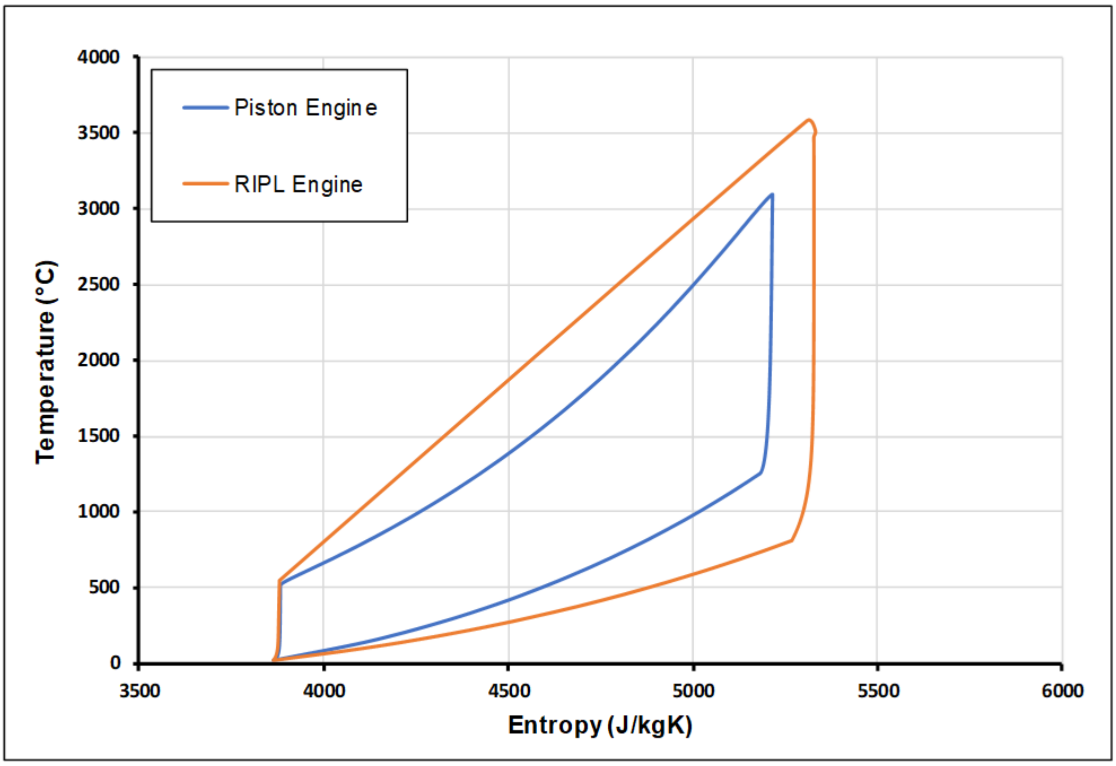



Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

5. Fuel Fraction Burned vs Time Passed During Combustion




6. T-S Graph





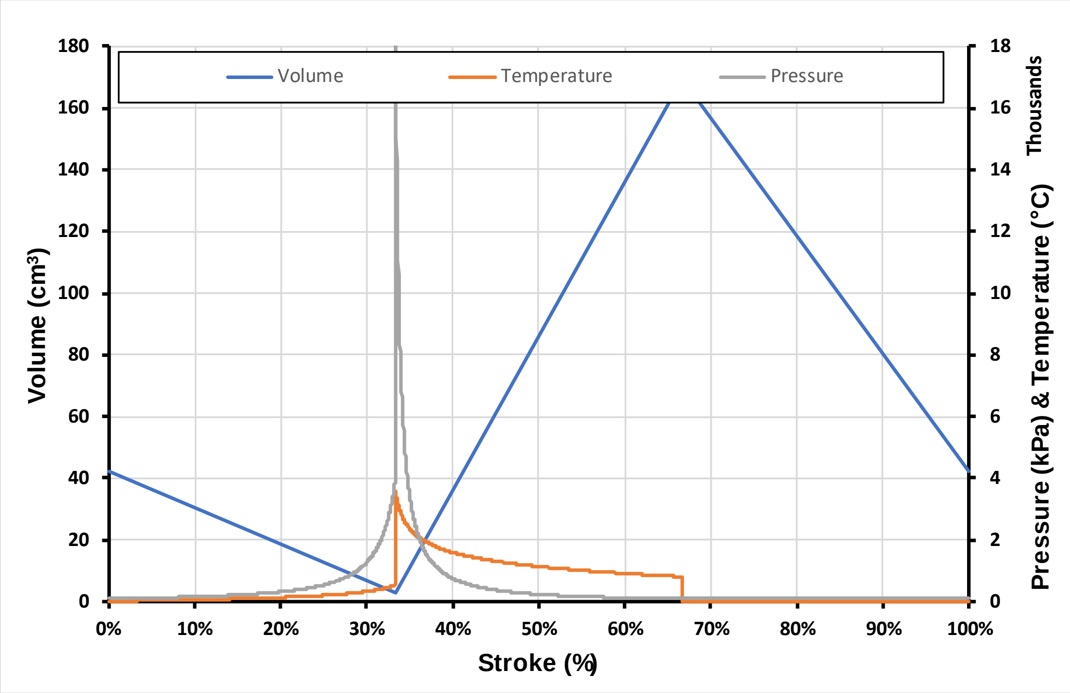
Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

**RIVENTA Ltd**
www.riventa.com

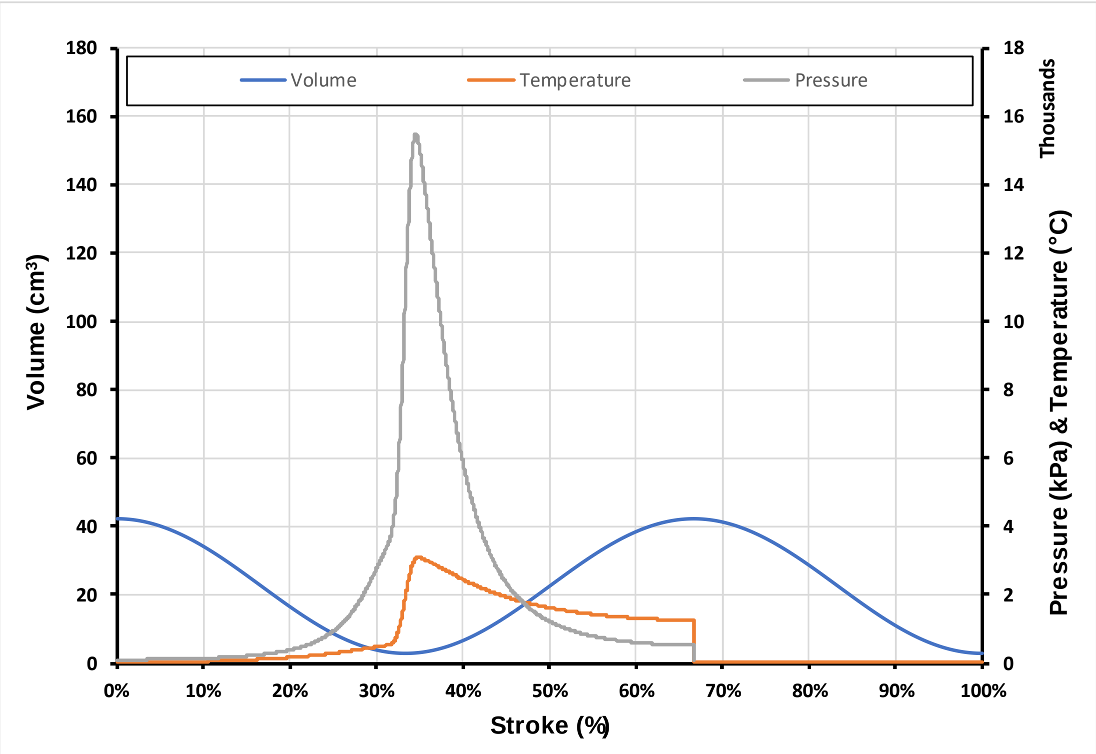
RIVENTA

Tel: +44 (0)1872 260005
e: info@riventa.com

**7. Pressure-Volume-Temperature Vs Stroke - RIPL Engine**




**7. Pressure-Volume-Temperature Vs Stroke - Piston Engine**

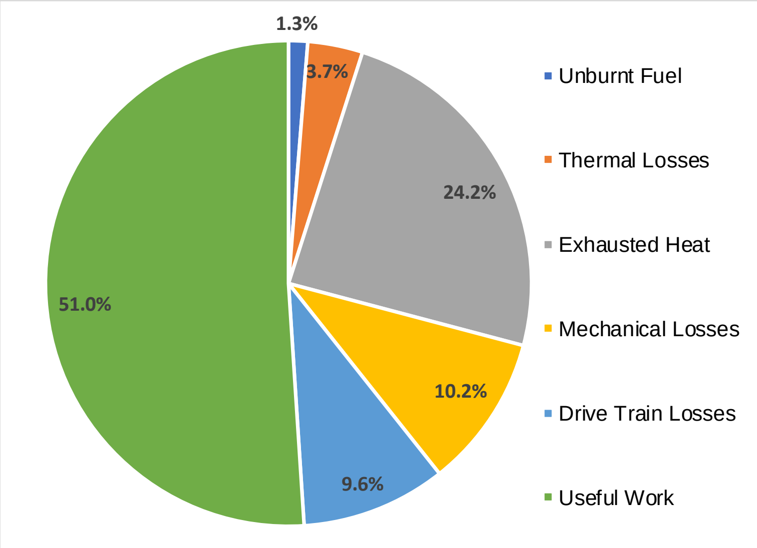



Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

RIVENTA Ltd
www.riventa.com

Tel: +44 (0)1872 260005
e: info@riventa.com

## 9a. RIPL Engine Overall Efficiency Analysis



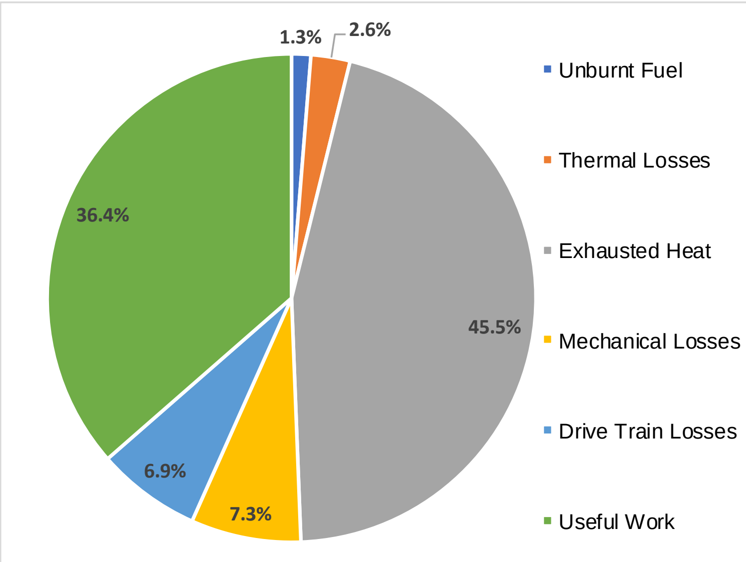

| Inputs | |
|---|---|
| RIPL Engine Power to Shaft Efficiency (%) | 85.6% |
| Drive Train Efficiency | 84.1% |

| Results | |
|---|---|
| Overall efficiency (%) | 51.0% |
| Overall efficiency improvement over piston (%) | 40.0% |
| Increase in power output for the same swept volume | 8.4 times |

## 9b. RIPL Engine Per Revolution Power Breakdown

| | | | | | |
|---|---|---|---|---|---|
| **Chemical Power in** | **21.22** | **kW** | Exhausted heat | 5.13 | kW |
| **Useful Work** | **10.83** | **kW** | Frictional Losses | 2.17 | kW |
| Unburnt Fuel | 0.27 | kW | Drive Train Losses | 2.05 | kW |
| Thermal Losses | 0.78 | kW | | | |

## 10a. Piston Engine Overall Efficiency Analysis - 2 Stroke




| Inputs | |
|---|---|
| Piston Engine Power to Shaft Efficiency (%) | 85.6% |
| Drive Train Efficiency | 84.1% |

| Results | |
|---|---|
| Overall efficiency (%) | 36.4% |
| --- | --- |
| --- | --- |

## 10b. Piston Engine Per Revolution Power Breakdown

| | | | | | |
|---|---|---|---|---|---|
| **Chemical Power in** | **1.18** | **kW** | Exhausted heat | 0.54 | kW |
| **Useful Work** | **0.43** | **kW** | Frictional Losses | 0.09 | kW |
| Unburnt Fuel | 0.01 | kW | Drive Train Losses | 0.08 | kW |
| Thermal Losses | 0.03 | kW | | | |


Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

**RIVENTA Ltd**
www.riventa.com

RIVENTA

Tel: +44 (0)1872 260005
e: info@riventa.com

## 7.2 Thermodynamic Comparative Analysis

Thermodynamic analysis comparing the piston and RIPL engine to the Otto & Atkinson Standards was conducted at varying different inputs.


Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

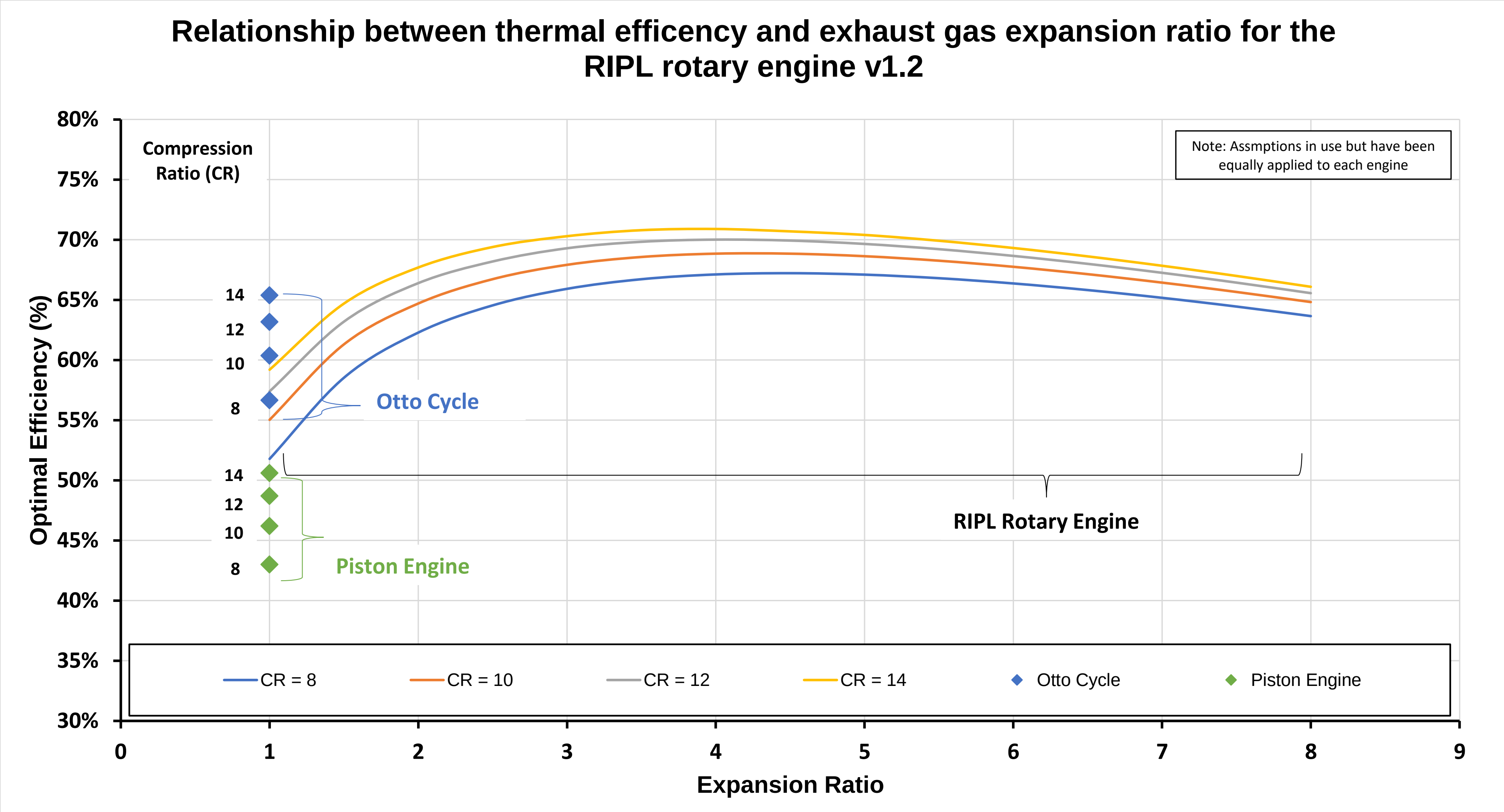



**Figure 5 Changing Expansion Ratio at Varying Compression Ratios**

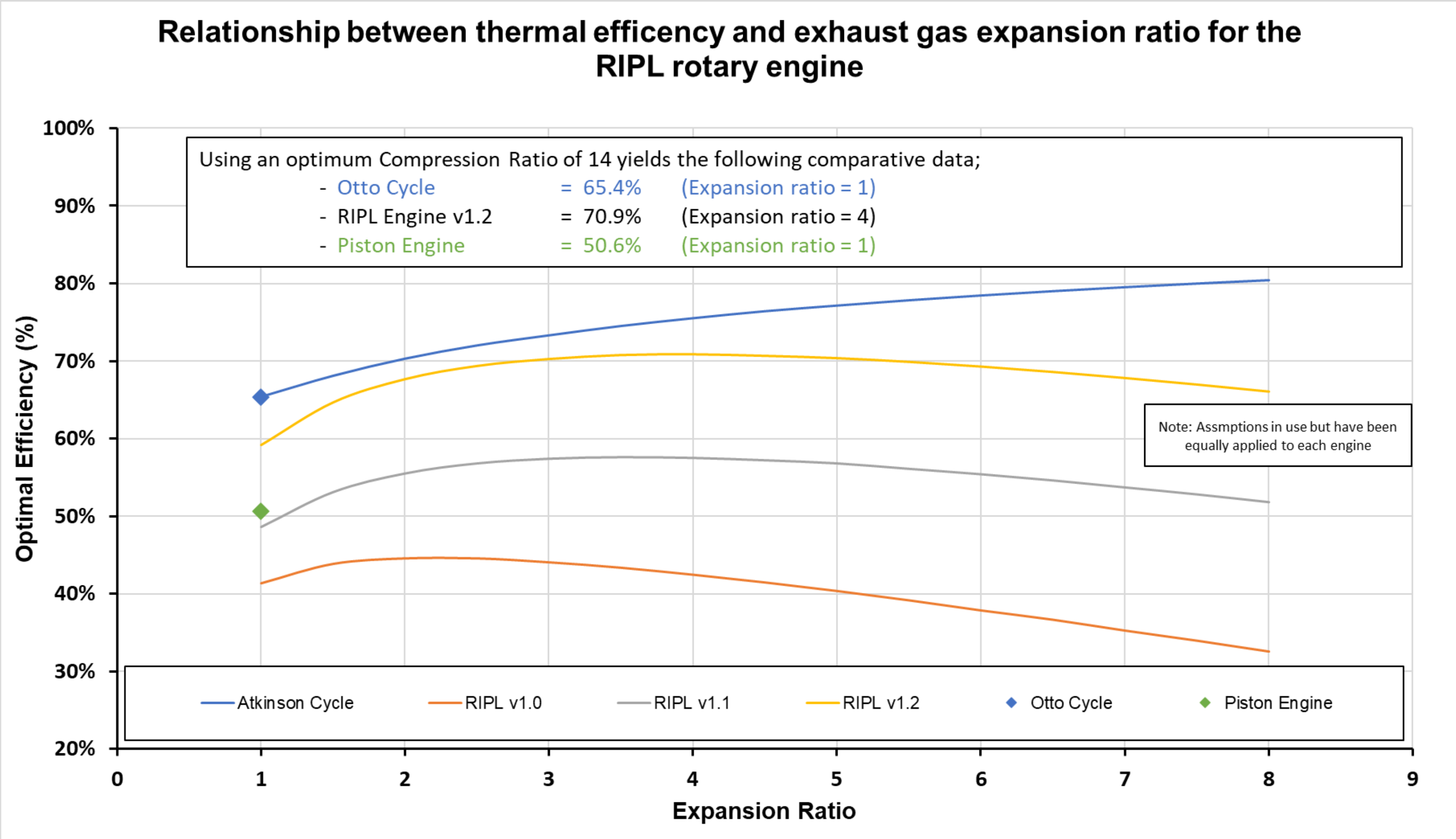



**Figure 6 Comparing Differing Iterations of the RIPL Engine at CR = 14**



Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

- The optimum expansion ratio for the RIPL engine (v1.2) is ~4 (depending on final selection of compression ratio) and the inputs used.
- At a 1:1 expansion ratio. The standard piston engine outperforms the RIPL engine v1.0-1.1, this is due to the volume profile of the RIPL engine. See 7.
- Increasing the expansion ratio, increases the efficiency of the RIPL engine. At the point where the final pressure is equal to atmospheric, we see no further benefit of increasing the expansion ratio. And efficiency starts to drop off.
    - In increasing the expansion ratio, the performance of the RIPL engine can be made to exceed that of the standard piston (v1.1).
- As compression ratio increases there is a reduced benefit from increasing expansion ratio.
- By operating outside the standard Otto cycle limits, the optimal efficiency for v1.2 can exceed the otto efficiency in a idealised thermodynamic cycle.

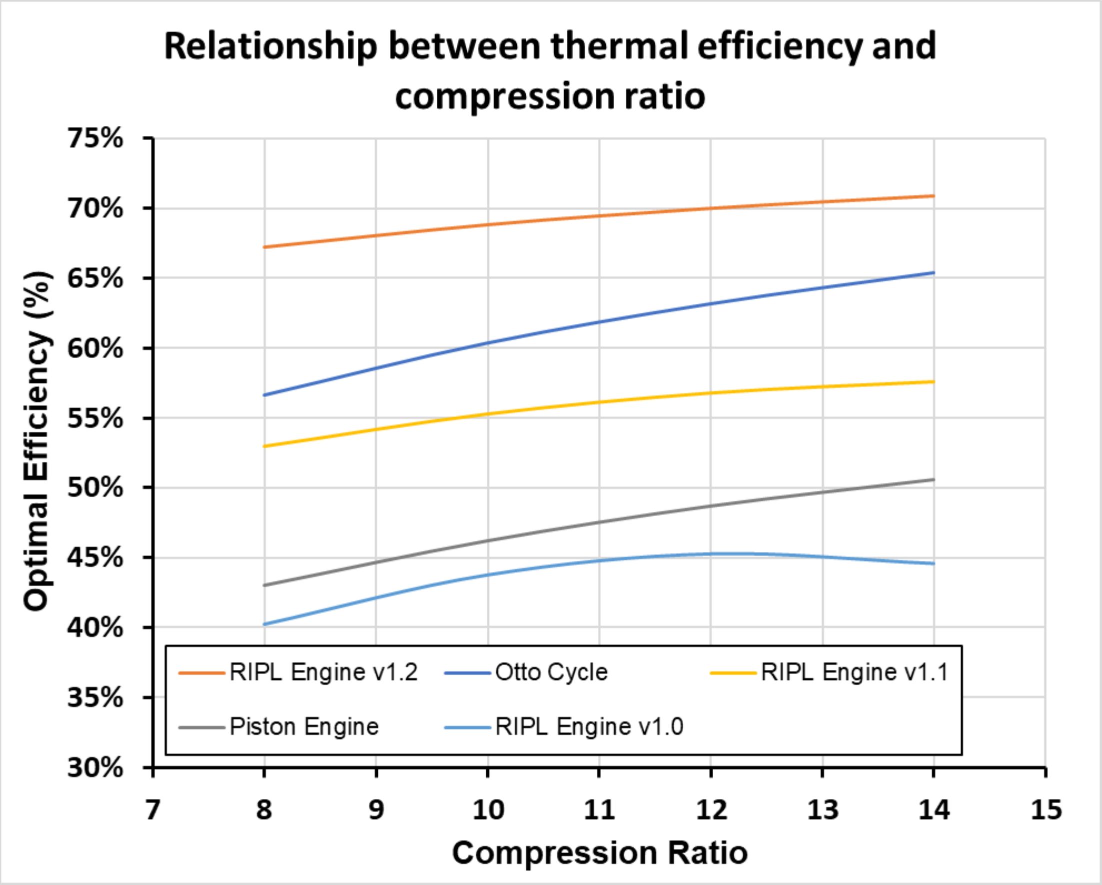



Figure 7 Optimal Efficiency Under the Given Compression Ratio

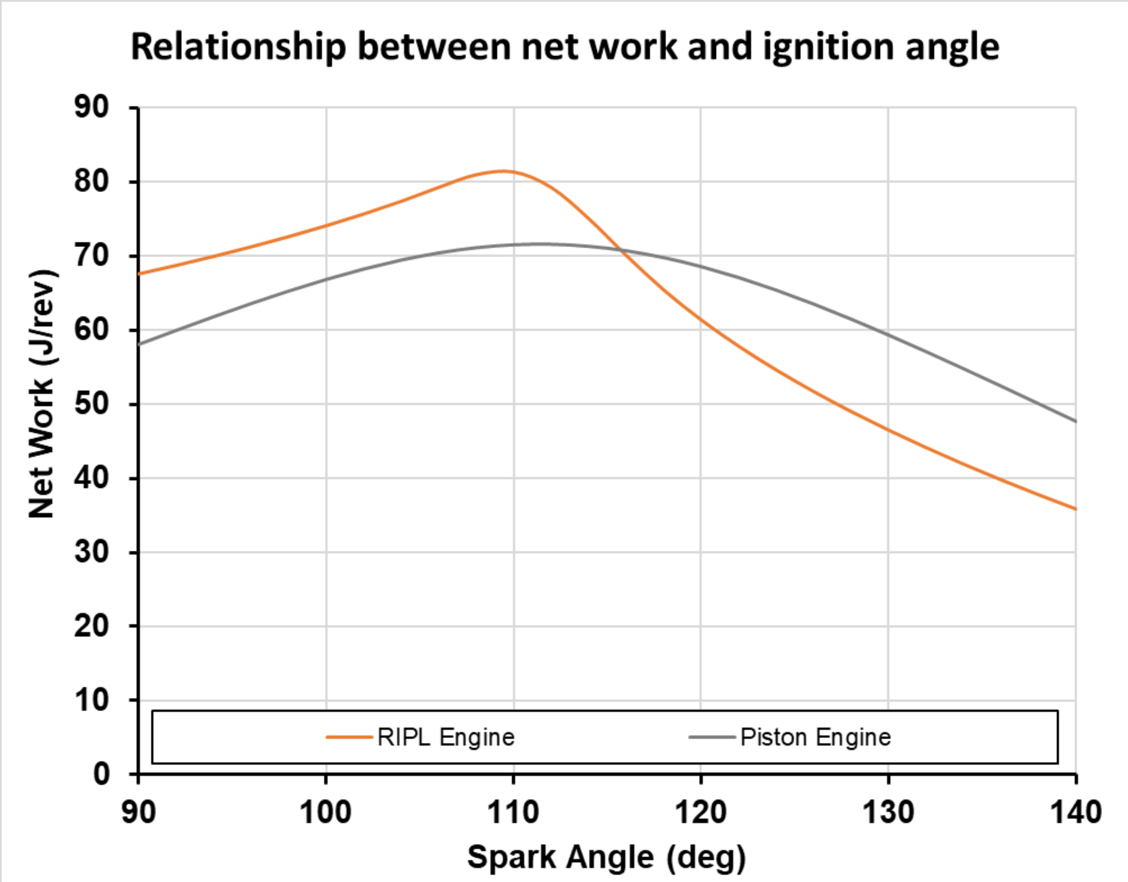



Figure 8 Net Work vs Spark Angle

Under the conditions set out in 5.1.2, the effect of ignition timing can be seen in Figure 6 for v1.1.

- The optimal spark angle for the RIPL and Piston engines for the given conditions is 110$^o$ and 112$^o$ respectively.
- The design of the RIPL engine results in a sharper drop off in efficiency if ignition is delayed. This has the following implications:
    - Engine performance is more affected by ignition timing, thus more susceptible to any delay
    - The higher advance also increases the probability of knocking occurring, which cannot be easily countered by delaying ignition without significantly affecting efficiency.
    - Improved engine design would reduce this limitation see 7.
- Before the optimal spark angle, the performance difference between the RIPL and piston engines remains approximately consistent when reducing the spark angle.


Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

## 8 Limitations

This study focuses on the idealised thermodynamic cycle that has been moderated for losses, as such no computational fluid dynamics or combustion analysis has occurred. Several assumptions and approximations were made in the model to generate comparable outputs for both the piston engine and RIPL's engine design. In practice it's likely that these approximations will not be accurate to the true combustion and fluid mechanics in the RIPL engine due to differing design. Where possible both models were optimised according to empirical data or calculated inputs to enable a fair comparison. Losses due to seal leakage, knocking and frictional losses in the fluid due to valves and pipework have been excluded from both models, as there is no information available modelling these losses for the RIPL engine without complex CFD analysis and real-world testing.

The ratio of specific heats is a thermodynamic property of fluids; this is determinable by the fluid and its pressure and temperature. This variable requires both the initial and final properties to be known to accurately lookup this value for calculations. This can be done through an iterative process; however, the computational time was too excessive. So, the initial value was used at small increments in the crank angle ($0.1^o$), to minimise the potential error. It was found that under combustion conditions where we see a sharp change in thermodynamic properties, the error in the ideal gas law, within a $0.1^o$ period peaked at ~0.2%. This error was then corrected to ensure the ideal gas law was maintained for a sealed system.

There is limited empirical data for the change in ratio of specific heats for air-fuel mixture, thus it was presumed that the ratio between the specific heats for air and the fuel mixture was maintained. And that it varied proportionally with air as the conditions changed.

Form factor models the shape of the combustion profile within the engine. Accurate CFD modelling will be required to determine this. In this study it was presumed that form factor is consistent across both engines.

Combustion efficiency was approximated from an empirical formula determined by the air fuel ratio, and the fuels equivalence factor. It is presumed that the RIPL engine will perform similarly.

Combustion angle was approximated using an empirical formula based on spark angle, combustion efficiency, engine speed and compression ratio. This formula is based on typical piston engines. In this study it was presumed this translates across to the RIPL engine.

The combustion approximation used had an error from ideal gas law of ~0.005% per $0.1^o$, similarly to the error incurred by ratio of specific heats this was corrected out at each step to reduce an accumulation in error.

Thermal losses from the working fluid were calculated by using the convection heat loss formula. Where the heat convective coefficient for air was approximated given the peak velocity of the fluid. Theoretically, the instantaneous velocity relative between object surface and fluid should be used. However, the impact of turbulent flow is not known, so peak velocity is used to account for any variation, this is duplicated across both the RIPL and the Piston engine. Additionally for the model a consistent uniform engine block temperature was used to calculate heat losses. The effect of water/oil cooling, may contribute to a non-uniform temperature distribution, resulting in differing levels of thermal losses in the model from reality.

The volume profile was simplified from reality for the RIPL engine, to minimise time spent interrogating the model as no volume equation was provided. The assumption is that this will have minimal impact on the results as the fundamental shape of the volume profile will differ minutely.

It was assumed that drive train efficiency would be equal for both engine types. An approximation was entered based on the efficiencies of each constituent part that makes up a standard vehicle drive train.

Frictional and conversion losses within the engine and to the drive shaft were determined by empirical data for the piston engine. Limited data was available for rotary machines, the literature that was available indicated that the largest portion of mechanical losses centred in the frictional losses inside the engine. Thus, it was estimated that any improvement in power transmission to the drive shaft was negated by the increase in frictional losses inside the engine.

# 9 Future Recommendations

As highlighted in this report; two steps in improving the engine to maximise efficiency are as follows:

- Improvement in engine design to allow spark angle to be optimised is estimated to yield significant efficiency improvements. This should be reviewed to enable advanced spark ignition.
- Inclusion of a [redacted] at constant volume.

Both design suggestions should be modelled to determine the possibility of including these changes into the existing design.

As indicated in 5.2, the Piston engine outperforms the RIPL engine at a standard 1:1 expansion ratio. The reason for this is due to the shape of the expansion stage. If this was designed to taper out into a large volume more slowly or the addition [redacted], thereby enabling a larger proportion of the combustion to occur at a higher pressure. We would expect to see an improvement in performance. An added benefit of this, would be less susceptibility in poor ignition timing hampering efficiency.

It is recommended to conduct some advanced simulation computational fluid dynamics (CFD) on the model to determine the proportion of losses that will be made in valving the working mixture between stages and the effect this will have on efficiency. This should also give some indication as to the strength of the seals required to ensure no leakage occurs between stages. In conjunction with this some combustion analysis should be undertaken to ascertain the effects of exhaust gases, engine block temperature and combustion profile.

Given the strength required for the seals, a better approximation of frictional losses inside the engine can be made. And through some advance modelling and/or real-world testing, a more accurate modelling of the mechanical losses can be made to ensure a fair comparison to the empirical modelling of the piston engine.

## 10 References

[1]: Mishra, Pankaj. “What is Otto Cycle – P-V and T-S Diagram Easiest Explanation?”. https://www.mechanicalbooster.com/2017/10/otto-cycle.html

[2]: Espinosa, Luis F. and Lappas, Petros. "Mathematical Modelling Comparison of a Reciprocating, a Szorenyi Rotary, and a Wankel Rotary Engine" Nonlinear Engineering, vol. 8, no. 1, 2019, pp. 389-396. https://doi.org/10.1515/nleng-2017-0082

[3]: G. H. Abd Alla, “Computer simulation of a four-stroke spark ignition engine,” Energy Conversion and Management, 2002, 43, 1044-1047.10.1016/S0196-8904(01)00092-9

[4]: S. Klein and G. Nellis. (2011). Thermodynamics. Available: http://RMIT.eblib.com.au/patron/FullRecord.aspx?p=80713410.1017/CBO9780511994883

[5]: T. J. Norman, “A performance model of a spark ignition Wankel engine: including the effects of crevice volumes, gas leakage, and heat transfer,” Master´s Degree, Dept. of Mechanical Engineering, Massachusetts Institute of Technology, 1983.

[6]: J. B. Heywood, Internal combustion engine fundamentals vol. 930: Mcgraw-hill New York, 1988.

[7]: J. I. Ghojel, “Review of the development and applications of the wiebe function: A tribute to the contribution of ivan wiebe to engine research,” International Journal of Engine Research, 2010, 11, 297-312.10.1243/14680874JER06510

[8]: G. Abbaszadehmosayebi and L. Ganippa, “Characterising Wiebe Equation for Heat Release Analysis based on Combustion Burn Factor (Ci),” Fuel, 3/1/ 2014, 119, 30110.1016/j.fuel.2013.11.006

[9]: G. Abbaszadehmosayebi and L. Ganippa, “Determination of specific heat ratio and error analysis for engine heat release calculations,” Applied Energy, 6/1/ 2014, 122, 144.10.1016/j.apenergy.2014.01.028

[10]: x-engineer. “Air Fuel Ratio”. https://x-engineer.org/air-fuel-ratio/

[11]: Farrag, A. & Gad, Mohammed. (2005). Investigations in Combustion of Hydrogen Enriched Natural Gas.

[12]: Bayraktar H., Durgun O."Development of an empirical correlation for Combustion Duration in Spark Ignition Engines". Energy Conversion and Management Journal, Vol.45, PP.1419-1431, 2004.

[13]: Engineering ToolBox, (2003). Convective Heat Transfer. [online] Available at: https://www.engineeringtoolbox.com/convective-heat-transfer-d_430.html [Accessed 17.11.22].

[14]: Ceviz, Mehmet & Kaymaz, Irfan. (2005). Temperature and air-fuel ratio dependent specific heat ratio functions for lean burned and unburned mixture. Energy Conversion and Management. 46. 2387-2404. 10.1016/j.enconman.2004.12.009.

[15]: x-engineer. “Drivetrain losses (efficiency)” https://x-engineer.org/drivetrain-losses-efficiency/

[16] Boru Jia, Rikard Mikalsen, Andrew Smallbone, Anthony Paul Roskilly, A study and comparison of frictional losses in free-piston engine and crankshaft engines, Applied

Thermal Engineering, Volume 140, 2018, Pages 217-224, ISSN 1359-4311, https://doi.org/10.1016/j.applthermaleng.2018.05.018.

[17] Eastern-European Journal of Enterprise Technologies, 3 (8(117)),32–38. doi: https://doi.org/10.15587/1729-4061.2022.256115

[18] Wikipedia, Otto Cycle, accessed online 22/11/2022, https://en.wikipedia.org/wiki/Otto_cycle

[19] Chen, K.-H.; Chao, Y.-C. Characterization of Performance of Short Stroke Engines with Valve Timing for Blended Bioethanol Internal Combustion. Energies 2019, 12, 759. https://doi.org/10.3390/en12040759

[20] Denis Allemant Andre, Matthew James Jensen, Gerald Micklow, James Brenner & Helgevon Helldorff | Duc Pham (Reviewing Editor) (2017) Simulation and thermodynamic analysis of extended expansion on a concept rotary engine including its effects on fuel efficiency, Cogent Engineering, 4:1, DOI: 10.1080/23311916.2017.1418131

[21] Dorić, Jovan & Klinar, Ivan & Dorić, Marko. (2011). Constant Volume Combustion Cycle for IC Engines. FME Transactions. 39. 97-104.

# 11 Appendices

## 11.1 Appendix 1: Approximations

### 11.1.1 Combustion Efficiency

Combustion efficiency is how much of the fuel is successfully combusted in the combustion phases, and thus how much fuel is remaining in the exhaust gases.

It can be approximated using the following:

$$\eta_c = -1.6082 + 4.6509\lambda - 2.0746\lambda^2$$

**Equation 8 Combustion Efficiency [10]**

Where λ is defined as the air fuel equivalence ratio:

$$\lambda = \frac{AFR}{AFR_S}$$

**Equation 9 Air Fuel Equivalence Ratio**

AFR is the air fuel ratio, $AFR_S$ is stoichiometric air fuel ratio, the atomic minimum ratio required for complete combustion to occur.

### 11.1.2 Combustion Angle

Combustion angle can be calculated using the equations below, from engine speed, combustion efficiency, spark angle and compression ratio.

$$\begin{aligned}
&F_1(N) = 0.1222 + 0.9717*(N/N_1) - 0.05051*(N/N_1)^2 \\
&F_2(\Phi) = 4.3111-5.6383*(\Phi/\Phi_1)+20304*(\Phi/\Phi_1)^2 \\
&F_3(\theta_s) = 1.0685-0.2902*(\theta_s/\theta_{s1}) +0.2545*(\theta_s/\theta_{s1})^2 \\
&F_4(r) = 3.2989-3.3612*(r/r_1) +1.08*(r/r_1)^2 \\
&\Delta\theta/\Delta\theta_1 = F_1*F_2*F_3*F_4
\end{aligned}$$

**Equation 10 Combustion Angle Approximation [11-12]**

Given some base reference values or $r_1 = 7.5$, $N_1 = 1000$rpm, $\Phi = 1$, $\theta_{s1} = -30$, $\Delta\theta_1 = 24$, from empirical data sets.

### 11.1.3 Convective Heat Transfer Coefficient

Convective heat transfer for air can be approximated given fluid velocity using the following equation:

$$h = 12.12 - 1.16v + 11.6\sqrt{v}$$

**Equation 11 Convective Heat Transfer Coefficient Approximation**

## 11.2 Further Modelling Outputs

### 11.2.1 No Optimised Spark Angle (v1.0)

# RIPL Engine Thermodynamic Performance Analysis

Comparitive to a Piston Engine, Otto Cycle and Atkinson Cycle

RIVENTA

**1. Inputs**

| | | | | | |
|---|---|---|---|---|---|
| Fuel Type* | Petrol | | Compression Ratio* | 14 | |
| Engine Design Speed* | 1000 | rpm | Include Thermal Leakage* | Y | |
| Intake Air Temperature* | 20 | °C | Intake Air Density* | 1.243 | kg/m³ |
| RIPL Spark Angle | 120 | ° | Piston Engine Spark Angle | 111 | ° |
| Engine Block Operating Temp* | 200 | °C | Intake Air Pressure* | 101.325 | kPa |
| Form Factor* | 3 | | Dimension Scalar* | 1 | |
| RIPL Expansion Ratio | 2 | | Key: applies to both engines | * | |

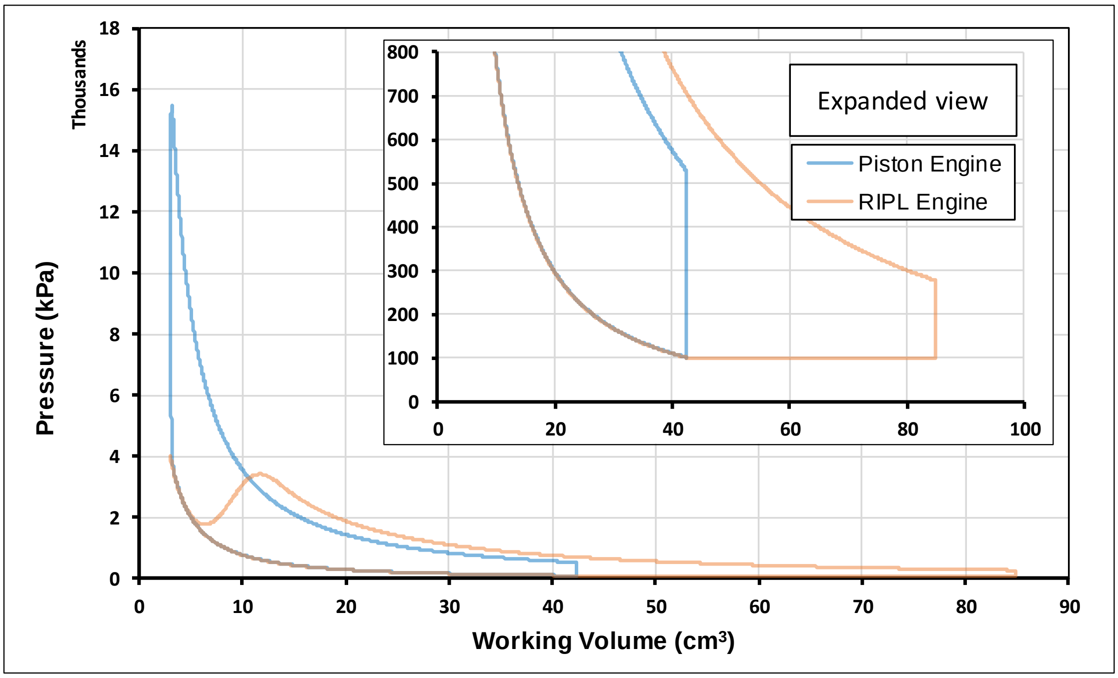
**2. Thermodynamic Summary Data (does not include mechanical losses)**

| | | | | | |
|---|---|---|---|---|---|
| RIPL Engine Efficiency | 44.6% | % | Theoretical Otto Efficiency* | 65.4% | % |
| Piston Engine Effiency | 50.6% | % | RIPL shortfall from Otto Efficiency | 31.8% | % |
| Efficiency Improvement | -11.8% | % | Theoretical Atkinson Efficiency* | 70.3% | % |
| RIPL Thermal Power Output | 9.5 | kW | 2 Stroke Piston Thermal Power Output | 1.19 | kW |
| RIPL Thermal Power Output | 12.7 | HP | 4 Stroke Piston Thermal Power Output | 0.60 | kW |
| Chemical Energy In per Power Stroke* | 141.4 | J/power stroke | 2 Stroke Piston Thermal Power Output | 1.60 | HP |
| RIPL Net Work | 63.1 | J/power stroke | 4 Stroke Piston Thermal Power Output | 0.80 | HP |
| Piston Net Work | 71.6 | J/power stroke | Engine Cc per Cylinder/Chamber | 42.41 | cm³ |

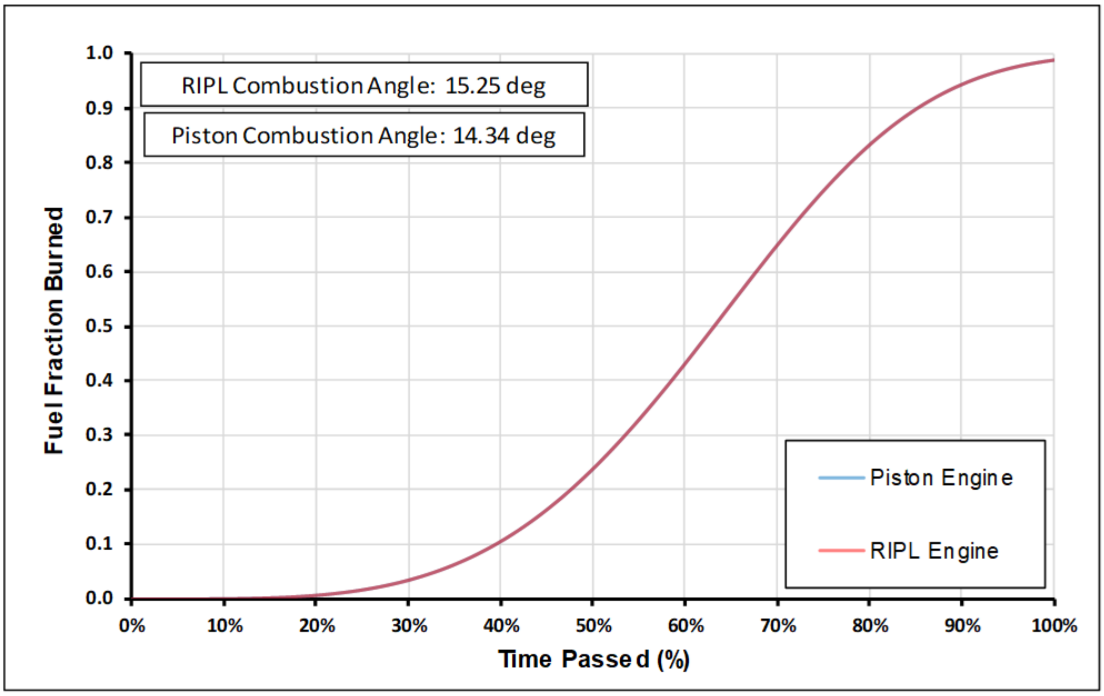
**3. Engine Dimensions**

| | | | | | |
|---|---|---|---|---|---|
| RIPL Stage 1 Depth | 20.0 | mm | RIPL Stage 2 Depth | 2.1 | mm |
| RIPL Stage 1 Inner Radius | 60.0 | mm | RIPL Stage 2 Inner Radius | 65.0 | mm |
| RIPL Stage 1 Outer Radius | 75.0 | mm | RIPL Stage 2 Outer Radius | 75.0 | mm |
| RIPL Stage 3 Depth | 40.0 | mm | | | |
| RIPL Stage 3 Inner Radius | 60.0 | mm | Piston Cylinder Diameter | 33.0 | mm |
| RIPL Stage 3 Outer Radius | 75.0 | mm | Piston Stroke Length | 49.5 | mm |

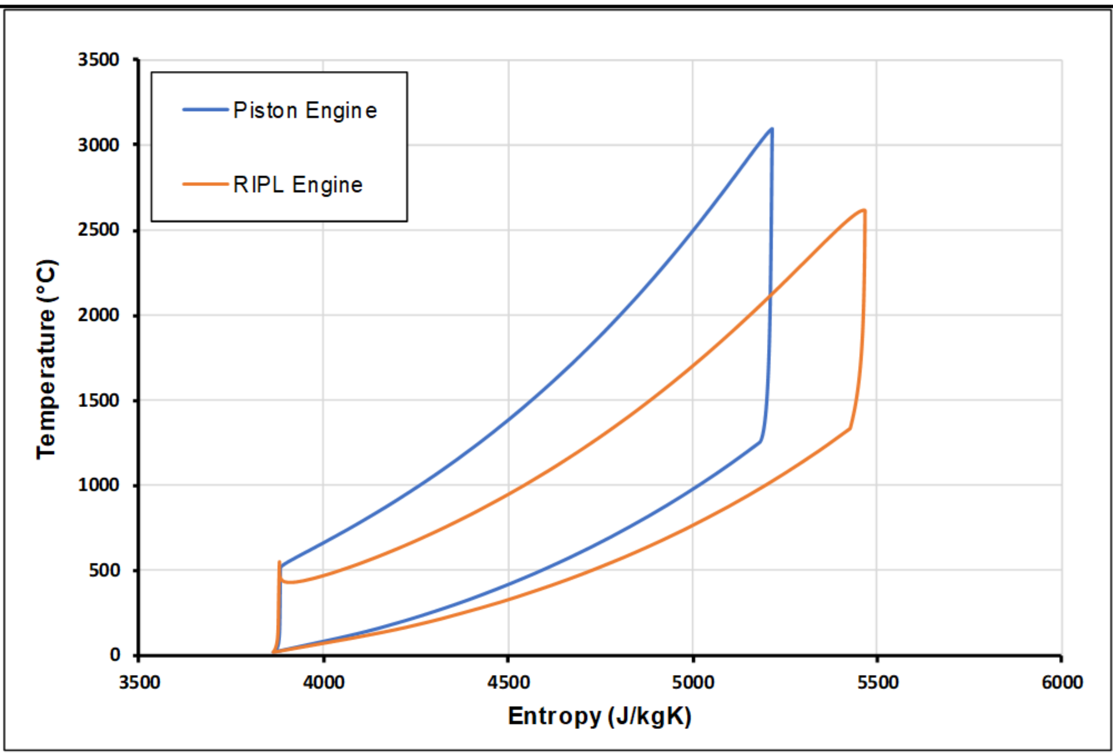
**4. P-V Graph**



5. Fuel Fraction Burned vs Time Passed During Combustion




6. T-S Graph



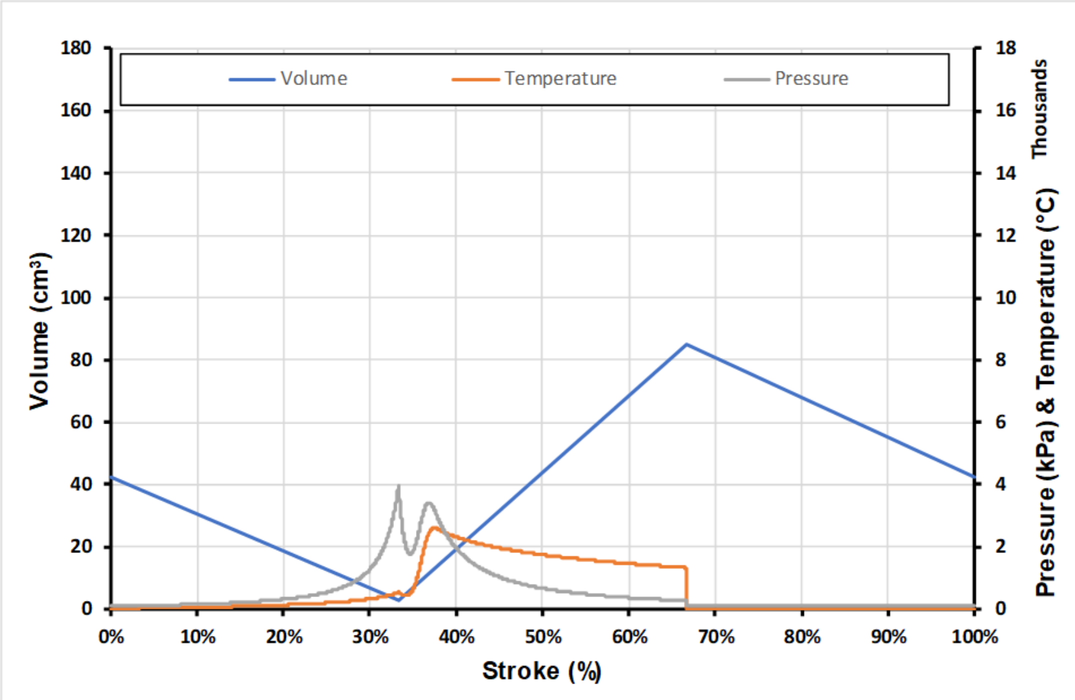


Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

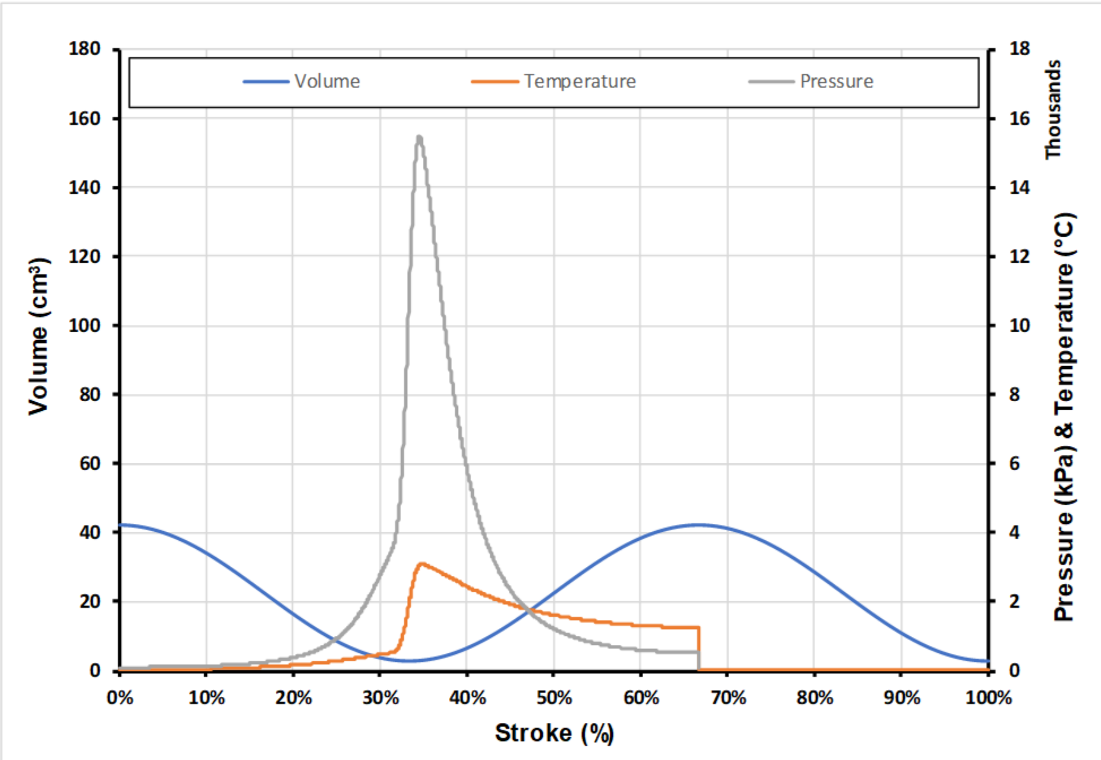
RIVENTA Ltd
www.riventa.com

Tel: +44 (0)1872 260005
e: info@riventa.com

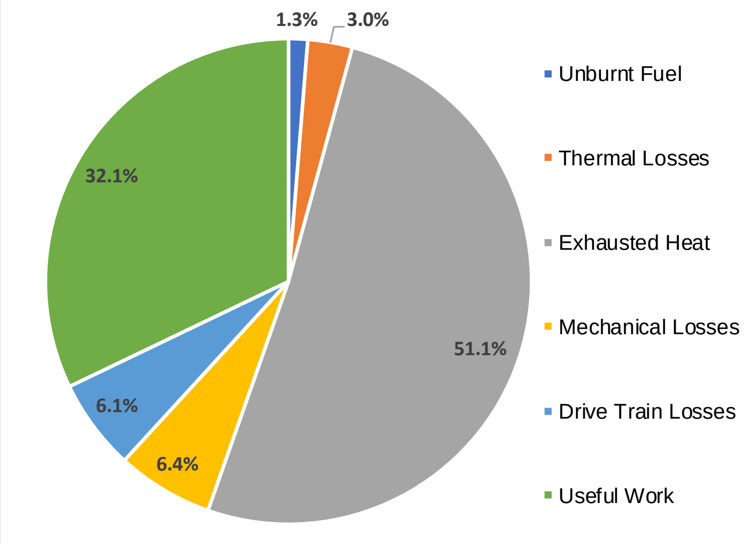
7. Pressure-Volume-Temperature Vs Stroke - RIPL Engine




7. Pressure-Volume-Temperature Vs Stroke - Piston Engine



### 9a. RIPL Engine Overall Efficiency Analysis



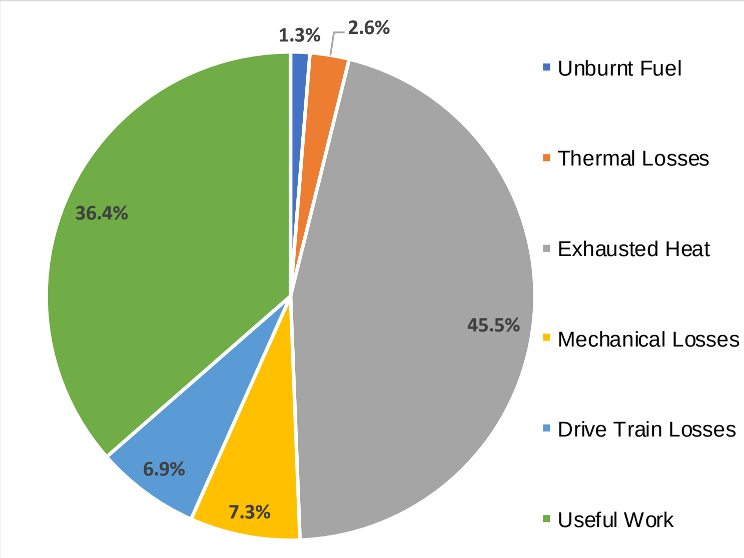

| Inputs | |
|---|---|
| RIPL Engine Power to Shaft Efficiency (%) | 85.6% |
| Drive Train Efficiency | 84.1% |

| Results | |
|---|---|
| Overall efficiency (%) | 32.1% |
| Overall efficiency improvement over piston (%) | -11.8% |
| Increase in power output for the same swept volume | 5.3 times |

### 9b. RIPL Engine Per Revolution Power Breakdown

| | | | | | |
|---|---|---|---|---|---|
| **Chemical Power in** | **21.22** | **kW** | Exhausted heat | 10.85 | kW |
| **Useful Work** | **6.82** | **kW** | Frictional Losses | 1.36 | kW |
| Unburnt Fuel | 0.27 | kW | Drive Train Losses | 1.29 | kW |
| Thermal Losses | 0.63 | kW | | | |

### 10a. Piston Engine Overall Efficiency Analysis - 2 Stroke




| Inputs | |
|---|---|
| Piston Engine Power to Shaft Efficiency (%) | 85.6% |
| Drive Train Efficiency | 84.1% |

| Results | |
|---|---|
| Overall efficiency (%) | 36.4% |
| --- | --- |
| --- | --- |

### 10b. Piston Engine Per Revolution Power Breakdown

| | | | | | |
|---|---|---|---|---|---|
| **Chemical Power in** | **1.18** | **kW** | Exhausted heat | 0.54 | kW |
| **Useful Work** | **0.43** | **kW** | Frictional Losses | 0.09 | kW |
| Unburnt Fuel | 0.01 | kW | Drive Train Losses | 0.08 | kW |
| Thermal Losses | 0.03 | kW | | | |


Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

**RIVENTA Ltd**
www.riventa.com

RIVENTA

Tel: +44 (0)1872 260005
e: info@riventa.com

### 11.2.2 Optimisable Spark Angle (v1.1)

# RIPL Engine Thermodynamic Performance Analysis

Comparitive to a Piston Engine, Otto Cycle and Atkinson Cycle

RIVENTA

**1. Inputs**

| | | | | | |
|---|---|---|---|---|---|
| Fuel Type* | Petrol | | Compression Ratio* | 14 | |
| Engine Design Speed* | 1000 | rpm | Include Thermal Leakage* | Y | |
| Intake Air Temperature* | 20 | °C | Intake Air Density* | 1.243 | $kg/m^3$ |
| RIPL Spark Angle | 109 | ° | Piston Engine Spark Angle | 111 | ° |
| Engine Block Operating Temp* | 200 | °C | Intake Air Pressure* | 101.325 | kPa |
| Form Factor* | 3 | | Dimension Scalar* | 1 | |
| RIPL Expansion Ratio | 3.5 | | Key: applies to both engines | * | |

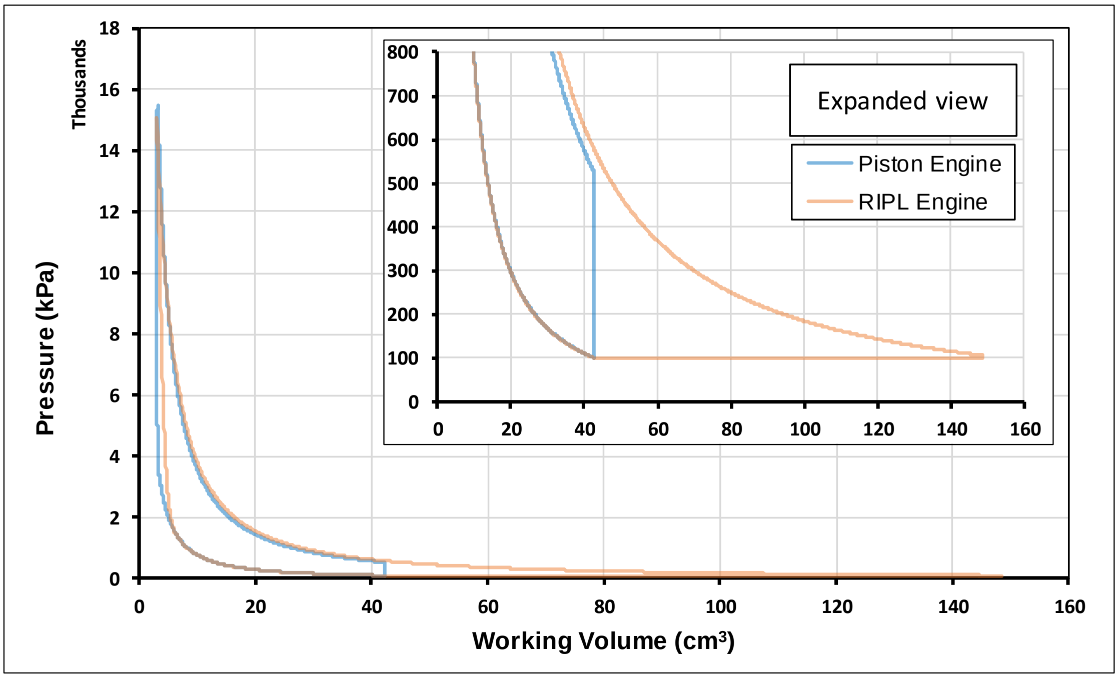
**2. Thermodynamic Summary Data (does not include mechanical losses)**

| | | | | | |
|---|---|---|---|---|---|
| RIPL Engine Efficiency | 57.6% | % | Theoretical Otto Efficiency* | 65.4% | % |
| Piston Engine Effiency | 50.6% | % | RIPL shortfall from Otto Efficiency | 11.9% | % |
| Efficiency Improvement | 13.8% | % | Theoretical Atkinson Efficiency* | 74.5% | % |
| RIPL Thermal Power Output | 12.2 | kW | 2 Stroke Piston Thermal Power Output | 1.19 | kW |
| RIPL Thermal Power Output | 16.4 | HP | 4 Stroke Piston Thermal Power Output | 0.60 | kW |
| Chemical Energy In per Power Stroke* | 141.4 | J/power stroke | 2 Stroke Piston Thermal Power Output | 1.60 | HP |
| RIPL Net Work | 81.5 | J/power stroke | 4 Stroke Piston Thermal Power Output | 0.80 | HP |
| Piston Net Work | 71.6 | J/power stroke | Engine Cc per Cylinder/Chamber | 42.41 | $cm^3$ |

**3. Engine Dimensions**

| | | | | | |
|---|---|---|---|---|---|
| RIPL Stage 1 Depth | 20.0 | mm | RIPL Stage 2 Depth | 2.1 | mm |
| RIPL Stage 1 Inner Radius | 60.0 | mm | RIPL Stage 2 Inner Radius | 65.0 | mm |
| RIPL Stage 1 Outer Radius | 75.0 | mm | RIPL Stage 2 Outer Radius | 75.0 | mm |
| RIPL Stage 3 Depth | 70.0 | mm | | | |
| RIPL Stage 3 Inner Radius | 60.0 | mm | Piston Cylinder Diameter | 33.0 | mm |
| RIPL Stage 3 Outer Radius | 75.0 | mm | Piston Stroke Length | 49.5 | mm |

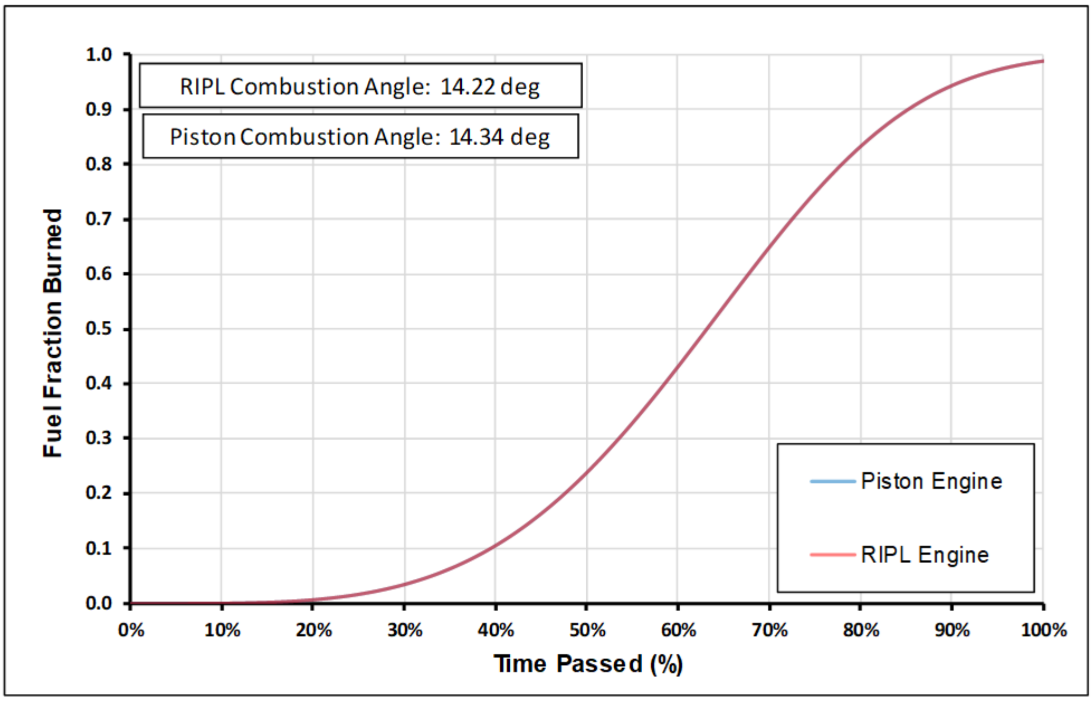
**4. P-V Graph**



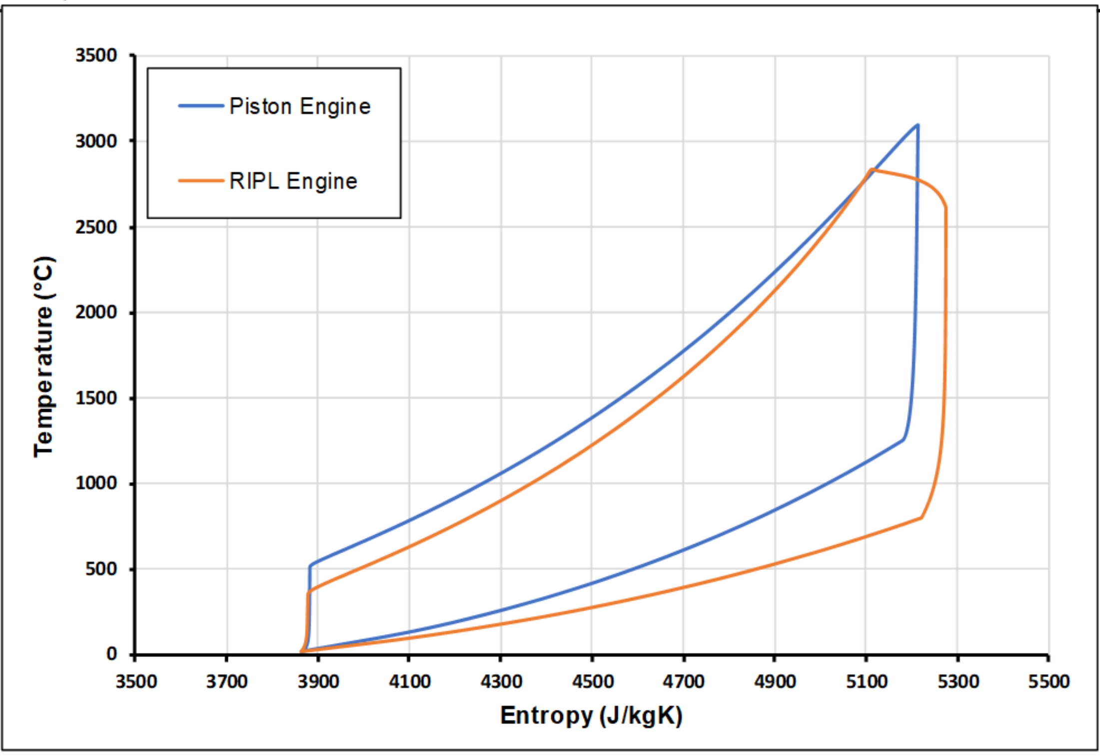



Unit B1, Walker Business Park, Truro, UK, TR4 9FB.
Tel: 01872 260005. www.riventa.com

5. Fuel Fraction Burned vs Time Passed During Combustion




6. T-S Graph





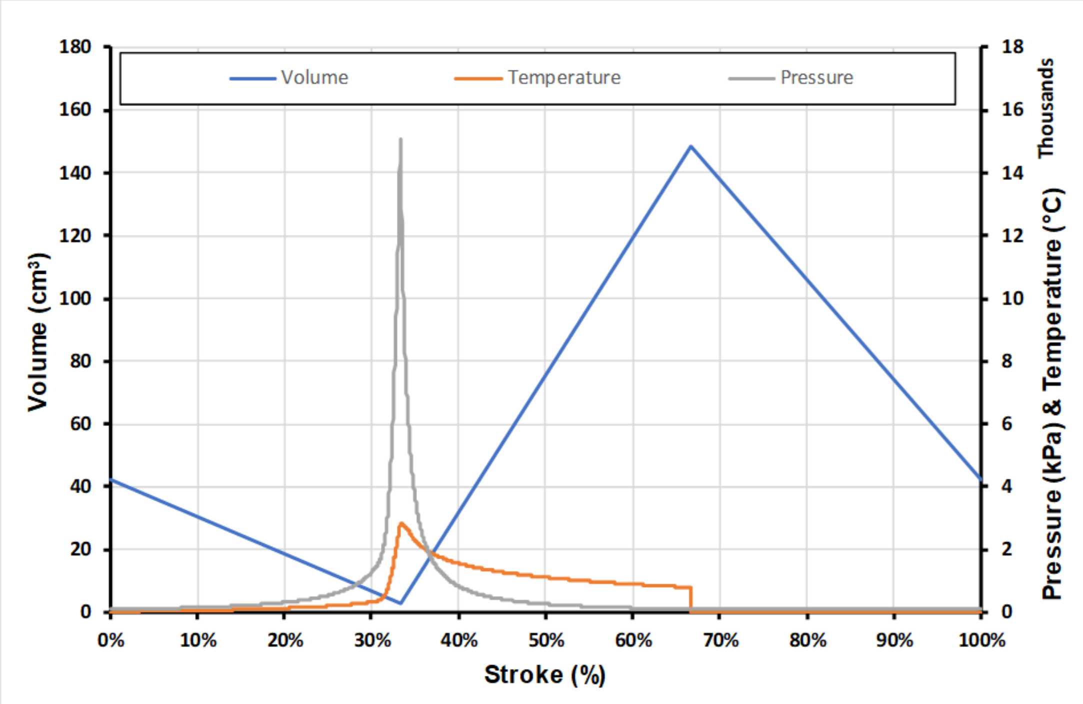
Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

RIVENTA Ltd
www.riventa.com

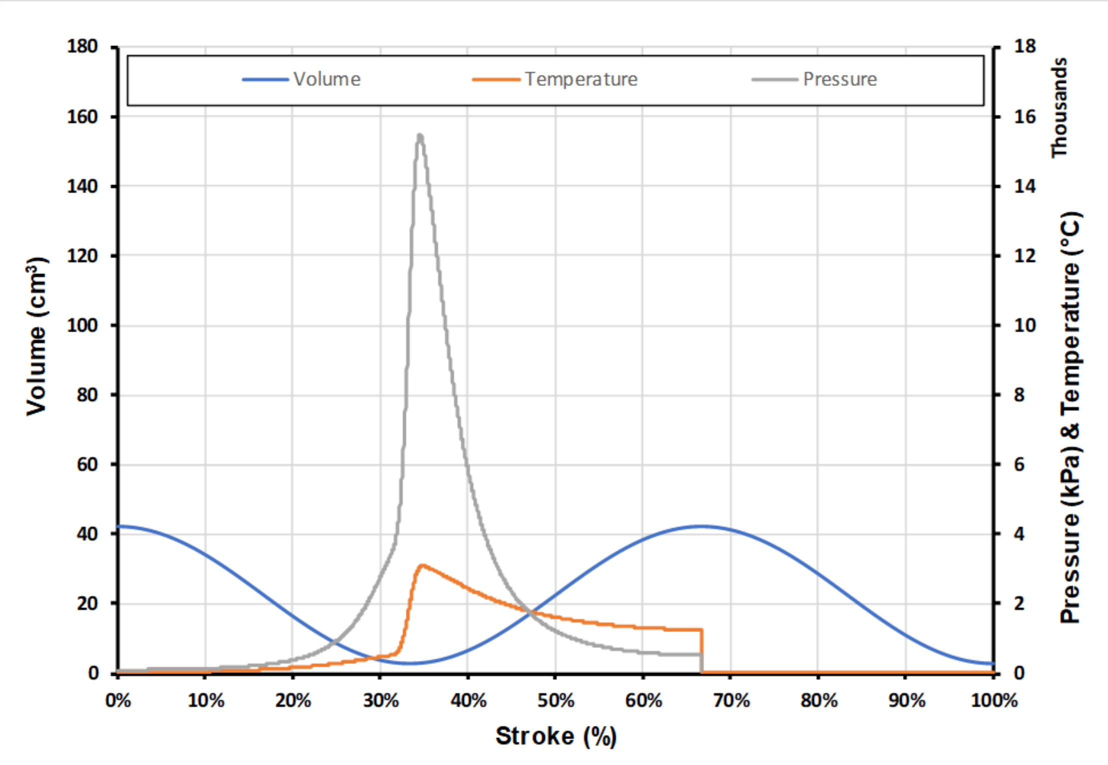
RIVENTA

Tel: +44 (0)1872 260005
e: info@riventa.com

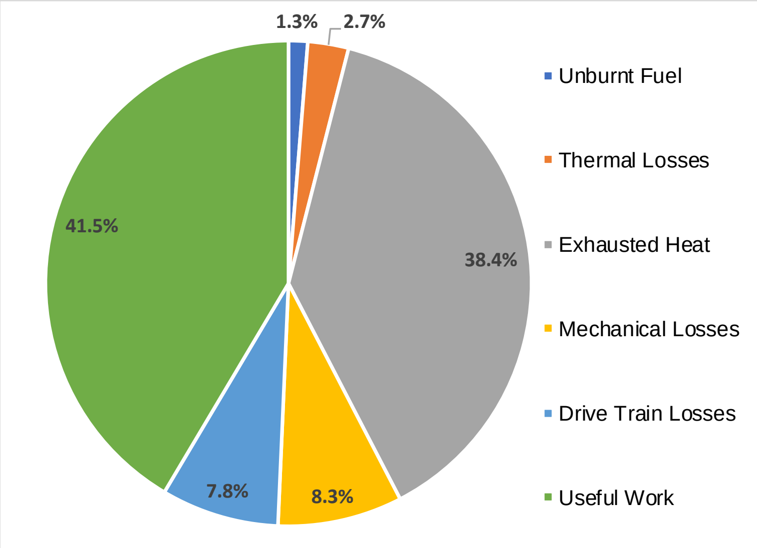
7. Pressure-Volume-Temperature Vs Stroke - RIPL Engine




7. Pressure-Volume-Temperature Vs Stroke - Piston Engine



## 9a. RIPL Engine Overall Efficiency Analysis



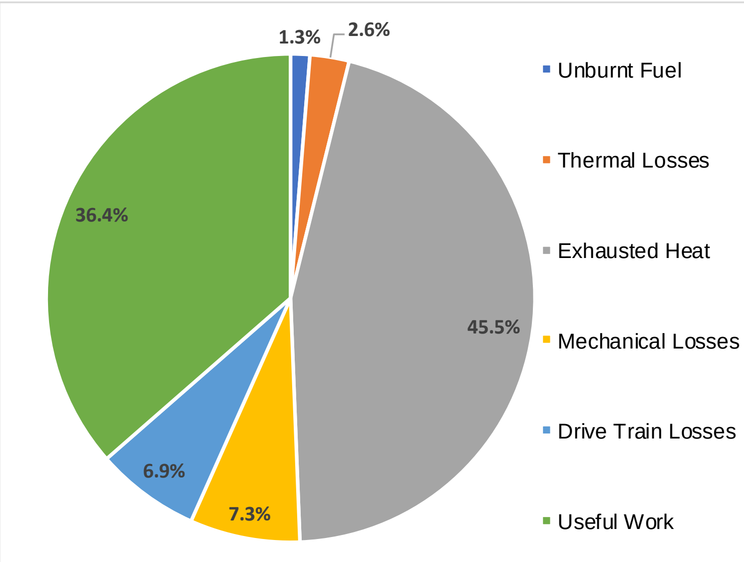

| Inputs | |
|---|---|
| RIPL Engine Power to Shaft Efficiency (%) | 85.6% |
| Drive Train Efficiency | 84.1% |

| Results | |
|---|---|
| Overall efficiency (%) | 41.5% |
| Overall efficiency improvement over piston (%) | 13.8% |
| Increase in power output for the same swept volume | 6.8 times |

## 9b. RIPL Engine Per Revolution Power Breakdown

| | | | | | |
|---|---|---|---|---|---|
| **Chemical Power in** | **21.22** | **kW** | Exhausted heat | 8.15 | kW |
| **Useful Work** | **8.80** | **kW** | Frictional Losses | 1.76 | kW |
| Unburnt Fuel | 0.27 | kW | Drive Train Losses | 1.66 | kW |
| Thermal Losses | 0.58 | kW | | | |

## 10a. Piston Engine Overall Efficiency Analysis - 2 Stroke




| Inputs | |
|---|---|
| Piston Engine Power to Shaft Efficiency (%) | 85.6% |
| Drive Train Efficiency | 84.1% |

| Results | |
|---|---|
| Overall efficiency (%) | 36.4% |
| --- | --- |
| --- | --- |

## 10b. Piston Engine Per Revolution Power Breakdown

| | | | | | |
|---|---|---|---|---|---|
| **Chemical Power in** | **1.18** | **kW** | Exhausted heat | 0.54 | kW |
| **Useful Work** | **0.43** | **kW** | Frictional Losses | 0.09 | kW |
| Unburnt Fuel | 0.01 | kW | Drive Train Losses | 0.08 | kW |
| Thermal Losses | 0.03 | kW | | | |

Unit B1, Walker Business Park
Threemilestone, TRURO
Cornwall, TR4 9FB, UK

**RIVENTA Ltd**
www.riventa.com

RIVENTA

Tel: +44 (0)1872 260005
e: info@riventa.com