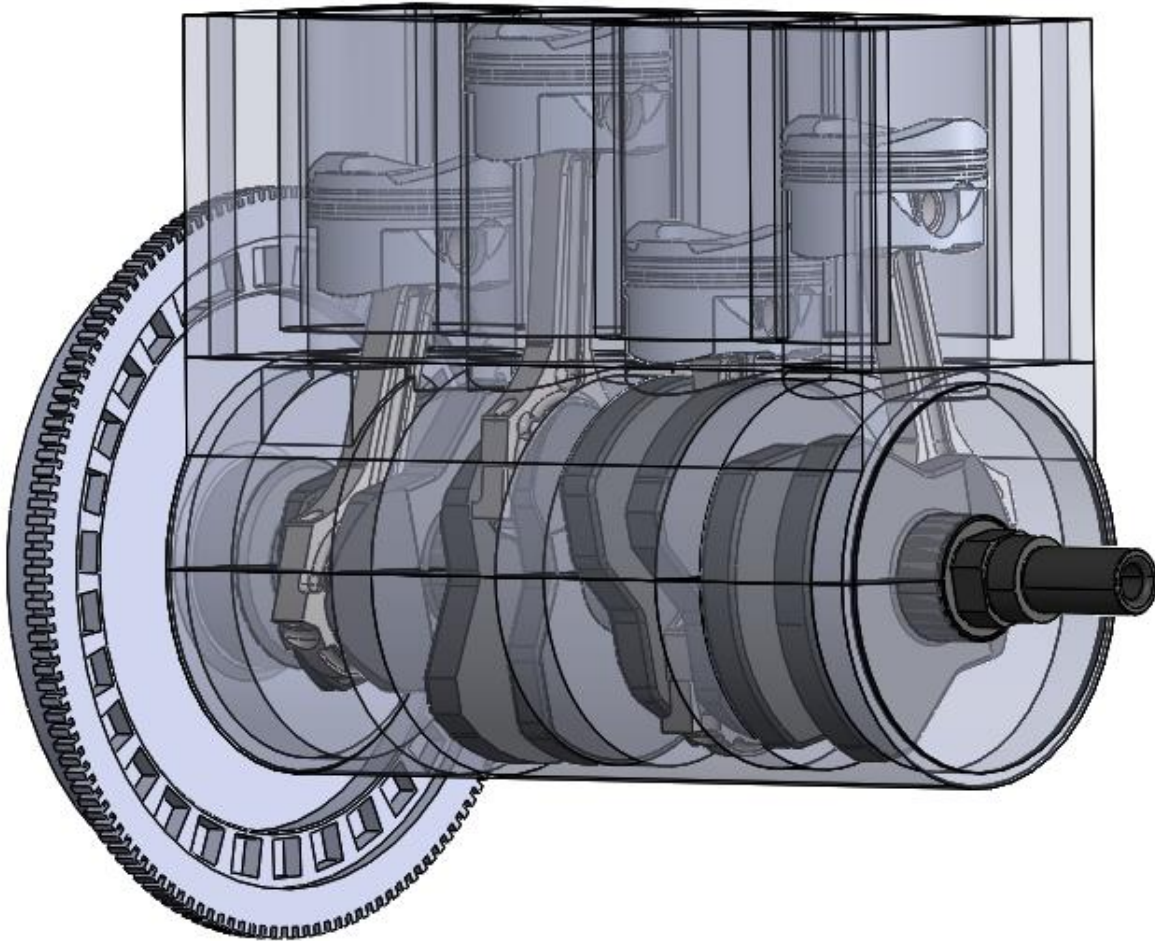




CRANKSHAFT BALANCING AND THE YAMAHA YZF-R1M CROSSPLANE

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Abstract

With the current increase in more efficient engines pushing the automotive industry, knowledge and understanding of how this can be carried out is essential. The crankshaft is at the heart of any internal combustion engine, therefore the following project investigation into 3 areas of this component. Examining the fundamentals of crankshaft balancing in several circumstances, followed by simulation of crank balancing in a single cylinder engine and finally the production of crankshafts using computer aided design.

Furthermore, the project is based around the Yamaha YZF-R1 crossplane engine. The report follows an investigation into allegations made by Yamaha in a promotional video where claims were made concerning the advantages to a crossplane crankshaft in comparison to a single plane.

Acknowledgements

I would like to thank Owen Williams and Malcolm McDonald for their help throughout this project and the contribution of knowledge that has been considered throughout all three investigations.

I would also like to thank my fellow students who helped me throughout the degree from start to finish, without whom I would not have gained the knowledge and understanding of engineering that I have today.

Declaration of Authority

This thesis is the result of the authors original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

Signed: Joseph Ieuan Woodhouse-Thomas

Date: 21/04/2020

1. Contents

Abstract	1
Acknowledgements	1
Declaration of Authority	1
1. Contents	2
2. Table of Figures	3
3. Table of Tables	5
4. Table of Equations	5
5. Introduction	6
6. Aims objectives and time scale	7
7. Literature Review	8
Matlab	8
Crankshafts	9
Design elements and criteria	9
Crankshaft balancing	10
Balance shafts	13
Cross plane crankshaft	14
Connecting rod theory	16
Balancing machine	18
Methods of balancing	19
External Balancing	19
Overbalancing	23
2 nd order balancing	25
2 nd order balance shaft	26
Lubrication	27
Conclusion to literature review	28
8. Computer Aided Design (CAD) simulation	30
Introduction to simulation experiment	30
Methodology	30
Production of CAD parts	31
Assembly	33
Material allocation	34
Motion simulation	35
Test conditions	35

Contrast test.....	37
Bob weight testing.....	38
Results.....	40
Comparative test.....	40
Conclusion to motion simulation	41
9. Matlab investigation.....	41
Introduction to the Matlab investigation	41
Methodology.....	41
Results.....	43
Inertia torque	44
Gas torque.....	45
Composite torque.....	46
Single plane	46
Crossplane.....	46
Centrifugal forces	48
Conclusion to Matlab experiment.....	48
10. Model production	49
Conclusion to model production	50
11. Conclusion	51
Limitations due to current situation.....	51
12. References.....	52
13. Appendices	55

2. Table of Figures

Figure 1 - Gannt chart of overall project	7
Figure 2- Schematic of forward and reverse vectors (Static balancing)	10
Figure 3 – Single cylinder crankshaft.....	11
Figure 4 - Weight added to single cylinder crankshaft	11
Figure 5- 1st order crankshaft balancing	12
Figure 6 – 90-degree V-Twin.....	13
Figure 7 - 70-degree V-Twin.....	14
Figure 8 - Crossplane crankshaft (Solidworks, 2020)	15
Figure 9 – Flat-plane configuration (Solidworks, 2020).....	15
Figure 10 - V8 Configuration Crossplane	16
Figure 11 - Connecting rod showing centre of gravity location	17
Figure 12 - Crank balancing machine (Industries, n.d.)	Error! Bookmark not defined.
Figure 13 - 1.5l eco-boost flywheel	19
Figure 14 - Schematic of 4-Cylinder crossplane	20

Figure 15 - Schematic for taking parallel moments.....	20
Figure 16 - Schematic for taking perpendicular moments	21
Figure 17 - Resultant moment	22
Figure 18 - Visual representation of mass offset for external balancing.....	22
Figure 19 - Schematic showing forces.	23
Figure 20 - Overbalancing.....	24
Figure 21 – 2nd order forces.....	25
Figure 22 - 2nd order balance shafts and crank.....	26
Figure 23 - Cut crank demonstrating oil passages	27
Figure 24 - Schematics of oil holes in crankshaft	28
Figure 25 – Flow-chart of boundary conditions	31
Figure 26 - Ford 1.5l eco-boost crankshaft.....	31
Figure 27 - Forged crank on lathe (Koflach, 2019)	32
Figure 28 - Measurement of crank.....	32
Figure 29 - Box extrude window	33
Figure 30 - Fillet rolling, located at connecting rod journal.....	35
Figure 31 - Standard test conditions for investigation	36
Figure 32 - No web scenario	37
Figure 33 - Marginally overbalanced scenario	37
Figure 34 - Bob weight test - traditional method.....	38
Figure 35 - Bob weight - new balancing	39
Figure 36 - Stress plot of crank balanced without webs.....	40
Figure 37 - Stress plot of marginally overbalanced crank.....	40
Figure 38 - Bob weight balancing stress on bracketry.....	40
Figure 39 - Matlab script of data offsetting – single plane	42
Figure 40 - Matlab script of data offsetting – crossplane.....	42
Figure 41 - Single cylinder and resultant single plane torques	43
Figure 42 – single cylinder and resultant crossplane torque	44
Figure 43 - Inertia torque 6000RPM.....	44
Figure 44 - Gas torque - single and crossplane	45
Figure 45 - Composite torque - single plane.....	46
Figure 46 - Composite torque – crossplane	46
Figure 47 - Centrifugal forces.....	48
Figure 48 - 3D printed from CAD drawings	49
Figure 49 - Aluminium CNC crossplane crankshaft from CAD	49
Figure 50 - Turning the dowels from aluminium.....	50
Figure 51 - Individual parts	55
Figure 52 - Excel method of engine simulation.....	56
Figure 53 - Matlab initial method.....	56
Figure 54 - Piston.....	57
Figure 55 - Connecting Rod Cap	57
Figure 56 - Connecting Rod.....	57
Figure 57 - Crossplane crank (unbalanced) used for model production	57
Figure 58 - V8 crank configuration and reciprocating assembly.....	57
Figure 59 - Crossplane crank, balanced to centre of rotation	58
Figure 60 - Flywheel	58
Figure 61 - Full configuration parts	58
Figure 62 - Ford 1.5l Eco-Boost engine.....	59

Figure 63 - Various stages of engine striping	59
Figure 64 - Valves and Valve springs with buckets	60
Figure 65 - Connecting rod and piston assembly	60

3. Table of Tables

Table 1 - Reciprocating and rotating mass percentages.....	18
Table 2 - Material allocation	34
Table 3 – Motion simulation conditions to experiment	36
Table 4 - Crank rotation times	47

4. Table of Equations

Equation 1 - Centrafugal Force	10
Equation 2 - Mass of crank and length	12
Equation 3 - Balance shaft mass and length	14
Equation 4 - Bobweight mass.....	17
Equation 5 - moment one about R2	17
Equation 6 - moment two about R2	17
Equation 7 - Parallel moment	21
Equation 8 - 40% compensation force.....	21
Equation 9 - Perpendicular moment	21
Equation 10 - Required angle for global balancing	22
Equation 11 - Required angle for global balancing example	22
Equation 12 - Resultant moment	23
Equation 13 - Required moment for flywheel	23
Equation 14 - Overbalancing.....	24
Equation 15 - Resultant force when overbalancing	24
Equation 16 - Vertical resultant force	25
Equation 17 - Second order balance shaft force.....	26
Equation 18 - balance shaft mass and length	26
Equation 19 - Bob weight mass.....	38
Equation 20 - Inertia force	41
Equation 21 - Inertia torque	42
Equation 22 - Rod angle.....	42
Equation 23 - Gas Torque	42

5. Introduction

Noise, vibration and harshness is a key area to perfecting in the internal combustion engine (ICE), therefore it is of most importance that every component in a vehicle works together to provide a smooth drive, and the crankshaft is no exception.

The following project explores the theory and practice behind crankshaft balancing and the crossplane crankshaft with a consideration to its usage. Alongside this the project also explores the claims made by Yamaha about their YZF-R1M and why it utilises a crossplane crankshaft.

The initial stage of the project examines the research made by the operator exploring the key areas of the subject that have been chosen during the project. The research has been accumulated throughout the project as it develops and has made changes to the end objective.

The first stage of research considers the crankshaft and what its main purpose is, with respect to its design criteria. Following this, the theory of crankshaft balancing has been studied to gain a thorough understanding of what is needed to be considered when balancing this component of the engine.

The balancing aspect is based around the theory of a single cylinder crankshaft. This area of the research looks at both 1st and 2nd order crank balancing and the fundamental differences between them.

Following this, the hypothesis of balance shafts has been studied and why it is and is not always necessary to employ this component into the engine in different situations such as racing application and mass production.

The next section of research looks at the theory behind connecting rod masses when balancing a crankshaft using a bob weight, and whether the current method is accurate enough in comparison to researched techniques that use computer aided design to assist the procedure.

The crossplane crankshaft has also been researched to gain an understanding of the applications that different areas of the motor industry use it for, from the Yamaha YZF-R1M to the widely held amount of legendary muscle cars that utilise the configuration.

Following the trend of crank balancing, consideration has been given to two different methods used within industry. The first being external balancing and why it may be attractive to some engine builders and engine manufacturers to use this method and, secondly, a look into overbalancing and the purpose of the process.

The final area of the literature review looks at the lubricating techniques that are used by crankshaft designers and manufacturers. With thought to the changes made to the component having negative affects to the strength of the material.

The following section of the project looks at three different applications of software and manufacturing in respect to the overall project aims and objectives.

The first experiment is fundamentally a computer aided design (CAD) motion simulation that aims to prove the theory of crankshaft balancing and bob weight mass theory using the software provided.

This includes looking into the production of the parts made in CAD and the boundary conditions taken into place when simulating the real-life scenarios.

The second experiment undertaken follows proving a claim made by Yamaha in 2009 concerning the crossplane crankshaft being utilised in their superbike engine. The calculation study aims to show the fundamental reasons for Yamahas allegations concerning the development.

Penultimately, to gain an enhanced understanding of the final step of any design process, attempts of creating a prototype of the balanced crossplane crankshaft have been made with the intent to take these prototypes further and into real-life testing state.

Finally, the results of all three experiments and prototypes will be produced and analysed to achieve a sense of the real-life feedback that can be gained using computer aided software and production methods.

6. Aims objectives and time scale

- To gain a thorough understanding of the chosen software with the intention to take this understanding into future projects.
- To achieve a working understanding of crank balancing and the effects each method has on racing and production-built engines.
- To enhance knowledge of crossplane crankshafts in V8 engines, and further develop understanding of crossplane cranks used in motorcycles and the effects on the overall feel when riding.
- To obtain a greater understanding of the internal combustion engine, and the effects that changes made to the crankshaft have on other components within the motor.
- Investigate the claims made by Yamaha concerning the crossplane crank including its advantages and disadvantages.

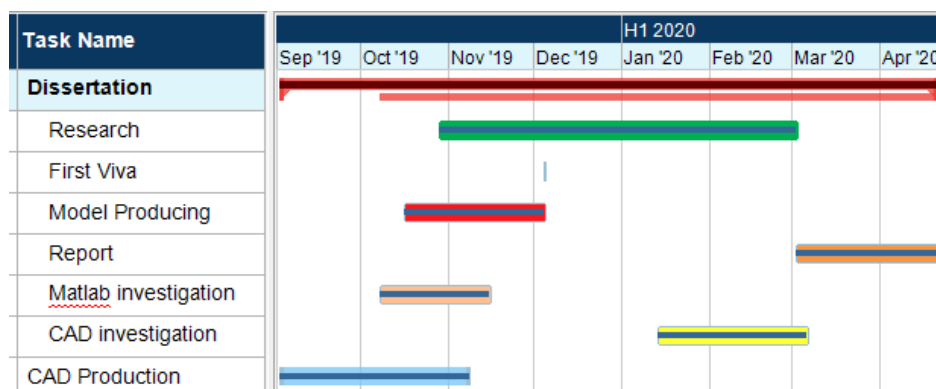


Figure 1 - Gantt chart of overall project

7. Literature Review

Software

Within this investigation significant amounts of calculations and simulations are going to be essential to understanding the subject selected. Moreover, to achieve this and with the resources inside the facilities, the appropriate software has been chosen to conduct the research and provide evidence for the findings.

Solidworks

Solidworks is a computer-aided design software that offers the ability to design and create virtual representations of any given piece of engineering; in this case the software has been applied to design components of the internal combustion engine (Solidworks, 2020). As a result of limited facilities within the place of learning, other software is unavailable.

Solidworks offers a motion simulation package; the comprehensive package within Solidworks allows the user to evaluate, simulate and improve design ideas, by using real-world scenarios to test design ideas prior to manufacturing.

Motion simulation gives the user the ability to design parts of engines and other engineering mechanisms and test these components to acquire results and data, rather than manufacturing for testing without an idea of the loads that the part is capable of sustaining (Trevelyan, 1986).

Matlab

‘MATLAB merges a desktop environment tuned for iterative analysis and design processes with a programming dialect that expresses matrix and array mathematics directly’ (MathWorks, 2020). The second allocation of software for the major project is Matlab. Matlab is a desktop calculator that allows the user to calculate large amounts of sums with great quantities of input. Using Matlab speeds up the calculating process and highlights any problems with the user’s code, therefore, producing an efficient and effective script. This software was chosen in comparison to excel as it provided ease of use in comparison, this is elaborated in appendix C.

Crankshafts

The main functionality of a crankshaft is converting reciprocating motion into rotating motion. Villalva (2010) explains that the crankshaft experiences load produced by the gas forces transmitted through the connecting rod, from the combustion chamber, these are called radial forces. The torsional loads also come from the combustion forces; the dynamic rotating motion force developed by the delay in torque at each crank causes the physical phenomenon call crankshaft vibration.

Whether running a two or four stroke cycle, the engine will always include a cyclic mode of operation. This results in varying gas forces during the combustion cycle, from this, acceleration and de-acceleration of the reciprocating assembly produces inertia forces (Ipci and Karabulut, 2018).

The inertia forces affect the engine internally and externally; the varying forces are transmitted through the crank train system resulting in the components following the crankshaft needing to withstand the loads without excessive distortion, noise, or eventually failure (Koszela *et al.*, 2018).

Externally, the engine experiences forces and moments that produce vibration frequencies, noise, and harshness (NVH). These areas are fundamentally what drives development in the engine application. Therefore, optimisation in the reciprocating and rotating assemblies is crucial (Kotter, Zirn and Wegener, 2016).

Design elements and criteria

The design stage of crankshaft production considers the fundamental areas of the ICE.

- Engine cylinder bore and piston stroke
- Engine RPM boundaries and over-revving
- Engine configuration (V, inline, boxer)
- Number of main bearings
- Typical gas pressures within the cylinder
- Connecting rod length, piston pin boss size
- Reciprocating and rotating masses/inertia forces acting on the crankshaft
- Crankshaft material and commonality across other engines
- Lubrication of bearings and pressure through oil feeds (Manning, 2012)

The torsional and radial forces are taken into consideration when designing a crankshaft, however, when designing a race engine such as the Petronas FP1, designers and engineers are more obligated to reducing the rotating mass of the crankshaft as the regulations allowed for higher engine speed in 3 cylinder 900cc motorcycles in world superbikes then, therefore, reducing the energy needed to rotate it. Zuhdi (2007) explains that drilling of the crank pins, smaller counterweights and tungsten inserts where incorporated to reduce the inertial forces.

Different engine configurations and sizes produce different amount of inertia forces. A 1 litre, single plane, 4-cylinder car engine will produce more inertia than a 1 litre single plane bike engine as the car engine has more to carry therefore the stroke is longer for more torque. For example, the Yamaha R1 has a bore and stroke of 77mm and 53.6mm respectively, however the Volkswagen Polo 1l has a bore

and stroke of 74.5mm and 76.4mm respectively (Carfolio, 2017), this results in greater amounts of mass for more connecting rod.

Crankshaft balancing

It would be beneficial to first ensure understanding of the fundamentals of crankshaft balancing within an internal combustion engine. It is also worth, at this point, noting that a crankshaft can never be perfectly balanced (Summit, 2016).

Static balancing is a term used to describe the balancing of in plane forces, and dynamic balancing, or external balancing are terms used to describe balancing the moments along the axis. The reciprocating assembly produces a forward and reverse vector presented in the figure below:

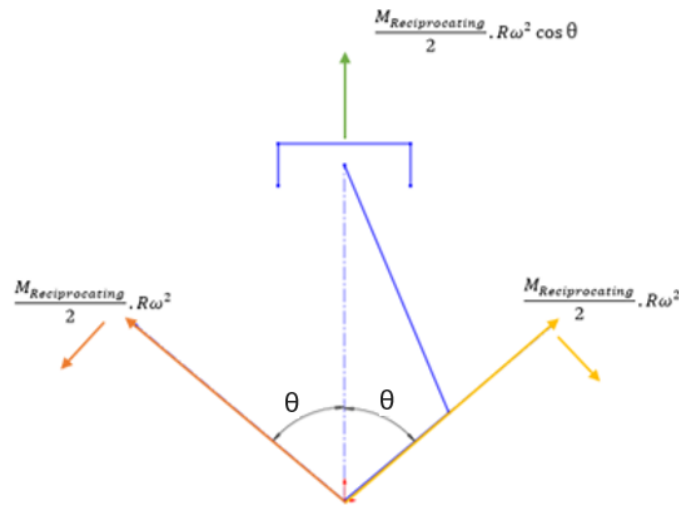


Figure 2- Schematic of forward and reverse vectors (Static balancing)

Figure 2 shows the forward vector (in yellow) and the reverse vector (in orange); the blue section of the schematic shows the piston and rod assembly and the force that is being produced from it. It can also be seen the forward and reverse vectors are at the same angle as one another located at crank angle ' θ '.

The forward vectors are balanced by the counter webs on the crank, whereas the reverse vectors are usually balanced using a balance shaft, depending on the configuration. This will be explored further in the literature review.

A centrifugal force is produced from 2/3 of the connecting rod mass ($M_{Rotating}$) given as:

$$F_{Centrafugal} = M_{Rotating} \cdot R \cdot \omega^2;$$

Equation 1 - Centrafugal Force

Where R is the distance between the centre of rotation and the centre of the rod journals, also known as 'throw', and ω is the engine speed in radians/sec.

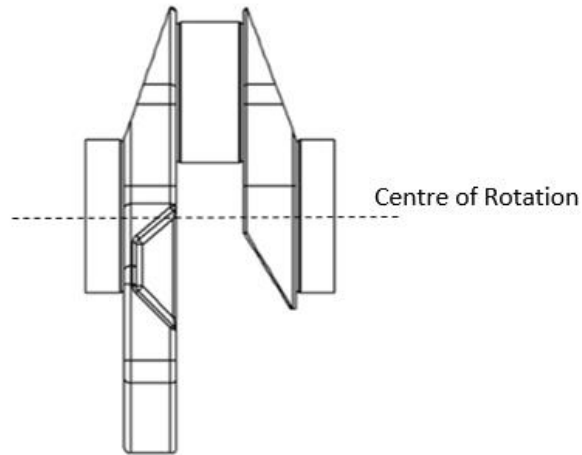


Figure 3 – Single cylinder crankshaft

To balance the forward vector of a single cylinder crankshaft, it is necessary to move the centre of gravity in line with the centre of rotation. The crankshaft can only produce a centrifugal force if the centre of gravity is offset from the centre of rotation (Infante, Freitas and Fonte, 2019). Halderman (2011) wrote that if a crankshaft has counterweights on both sides of the connecting rods down to the envelope, it is considered 'fully counter weighted'.

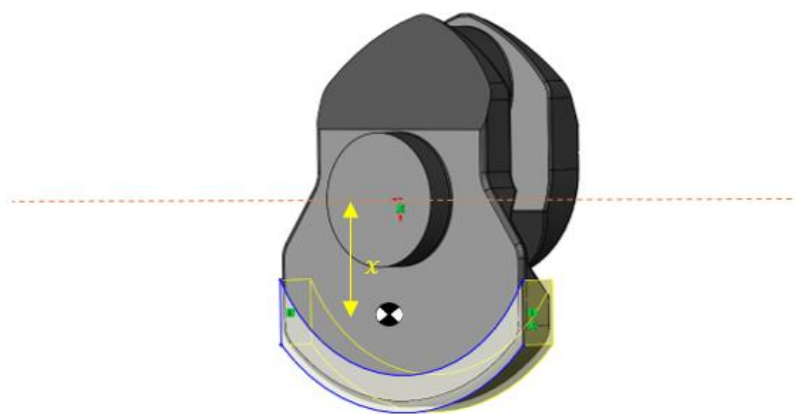


Figure 4 - Weight added to single cylinder crankshaft

To achieve 100% balance between the crank web and the connecting rod assembly of the forward vectors, it is essential to move the centre of gravity in the opposite direction to the mass being added.

$$M_{Crank} \cdot x = M_{Rotating} \cdot R;$$

Equation 2 - Mass of crank and length

Using the above equation, it is possible to determine the amount of mass (M_{Crank}) needed to move the centre of gravity at distance (x). This is demonstrated in the above figure and can be seen that the mass is being added beneath the web to lower the centre of gravity. The crank web size is however restricted by the envelope. Ward (2010) writes of different materials being used such as tungsten and other heavy metals to counterweight without using too much engine space.

The envelope is the restricting layer of engine casing that contains the crankshaft. In the case of exchanging a single plane crank for a crossplane crank, larger webs are necessary for balancing purposes which can be compromised by the envelope.

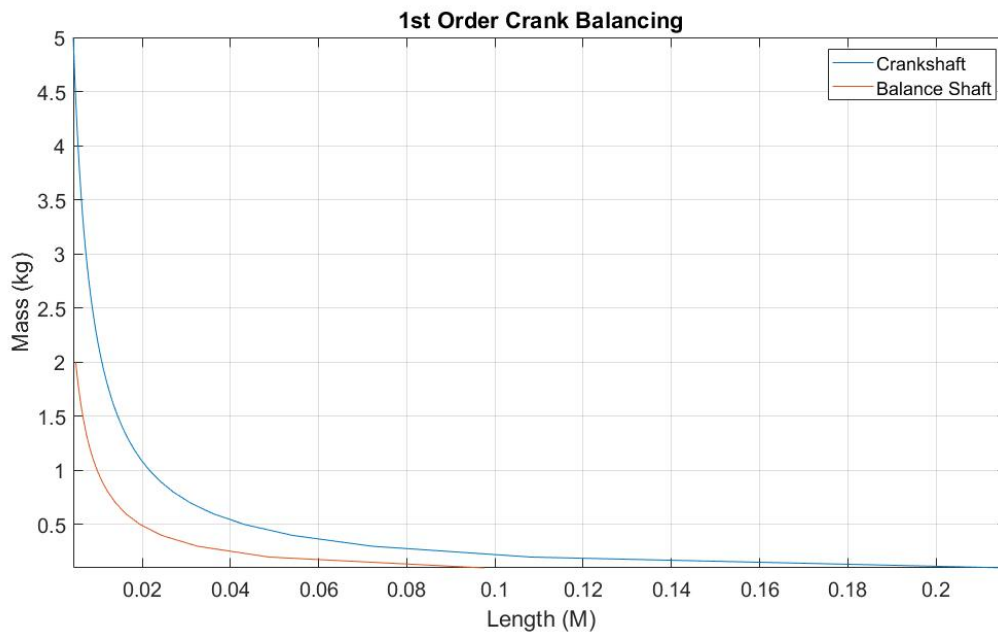


Figure 5- 1st order crankshaft balancing

Figure 5 shows array of masses and lengths to demonstrate the general trend of balancing. The y-axis is the mass of crank web/balance shaft and the x axis shows the length that the centre of gravity needs to be from the centre of rotation.

Balance shafts

Balance shafts are a key participant in reducing vibrations in an engine. The balance shaft compensates for the reverse rotating vector in an engine; it is *not* always necessary to have a 1st and 2nd order balance shaft.

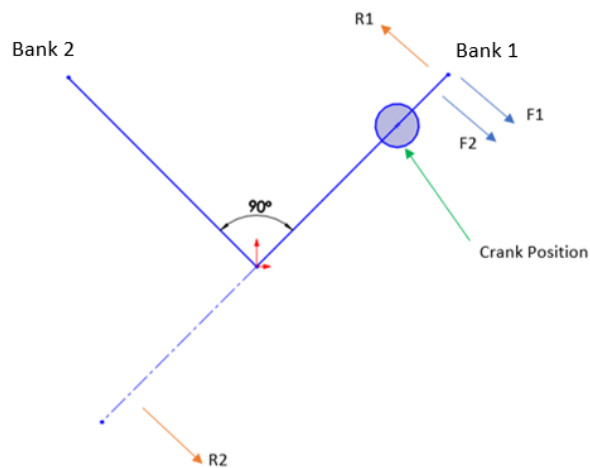


Figure 6 – 90-degree V-Twin

Figure 6 shows the 1st order balancing of a 90-degree V-Twin. In 1st order balancing, both the forward vectors will be located along the angle of the crank, in this case, along bank 1. The reverse vectors will be found at the same angle anti-clockwise from its original bank.

As can be seen from the figure the reverse vector for the first force is along bank one along with the first forward vector, however, the reverse vector for bank 2 is 90-degrees anti-clockwise of bank two. This is because the forward vector for bank 2 is located at bank 1 (90-degrees on), therefore, the reverse vector must travel 90-degrees in the opposite direction. With having the two vectors opposite one another the forces are cancelled out and thus, a 1st order balance shaft is not necessary.

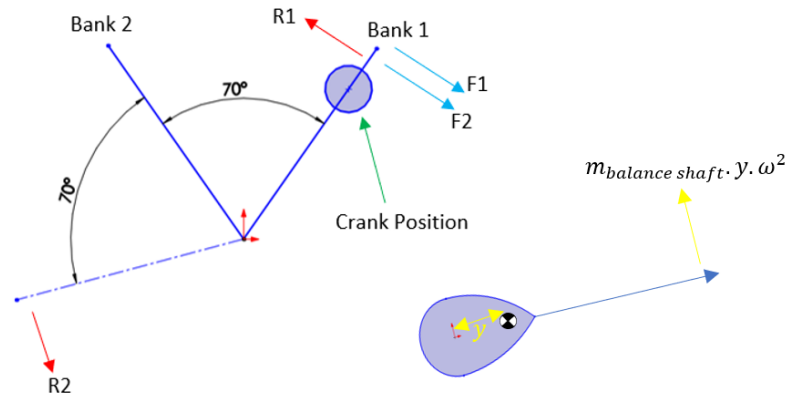


Figure 7 - 70-degree V-Twin

Figure 7 shows a 70-degree V-Twin on a single crank pin much like the previous diagram. The difference, besides the bank angle, in this configuration is the requirement for a balance shaft. Unlike the 90-degree set up, the reverse vector of bank 2 does not cancel out the reverse vector for bank 1, therefore a balance shaft is essential to cancel out this vector.

$$m_{bal} \cdot y = \frac{m_{rec}}{2} \cdot R ;$$

Equation 3 - Balance shaft mass and length

Equation 4 then shows the required mass and distance from the centre of rotation to the centre of gravity of the balance shaft that is required to fully balance the reverse vector. As the reverse vector rotates anti-clockwise, the balance shaft must also rotate in the same direction. To do so, it may be possible to run gears off the crank, of the same size, to rotate the balance shaft.

Halderman (2011) writes that balance shafts are more commonly used in larger engines such as 2-litre blocks, and that the balance shafts are often located underneath the crank, driven by a gear or chain. This is, however, in contrast to Clarke (2008) as he writes of a 4-cylinder 1200cc BMW motorcycle that utilises twin balance shafts and also suggests that any large displacement motorcycle motors (1000cc – 1300cc) will benefit from utilising balance shafts.

Cross plane crankshaft

The crossplane crankshaft has been around for many years, originally led by American muscle cars, the crossplane crankshaft dissipates torque response more efficiently when accelerating out of corners smoother delivery. To achieve smoother vibration and to distribute the torque factor more evenly, the crossplane configuration has been utilised in V8 engines in muscle cars for much longer than seen in motorcycles such as the Yamaha YZF-R1M (Ensanian, 2016).

The crossplane crank shaft possesses a crank pin perpendicular to one another at every crank angle. When looking at the crossplane, the ends of the crank pins make a cross (+) shape, therefore assigning the name 'crossplane' to the configuration. The firing order of a crossplane crank, in a 4-cylinder motorcycle engine is 1-3-2-4, which leaves a gap of 270-degrees between 2 and 4 firing. Chhangani (2009) writes of the crossplane crank giving the tyre the time between firing to re-grip to the track.

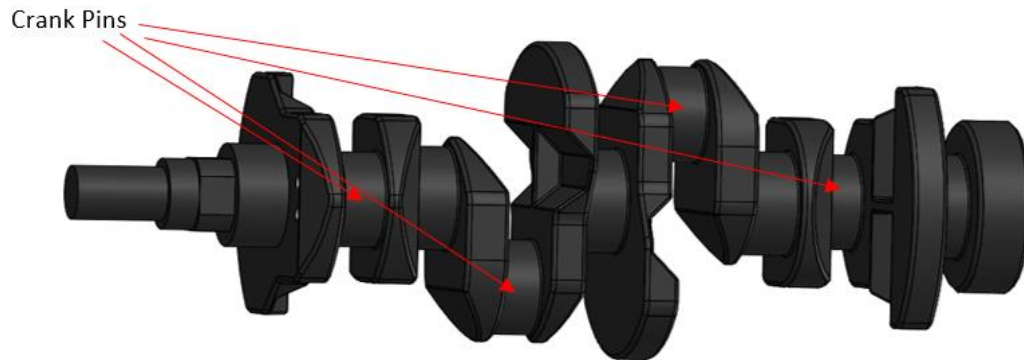


Figure 8 - Crossplane crankshaft (Solidworks, 2020)

The flat-plane configuration assembles with the 1st and 4th pin being at the same position, and the 2nd and 3rd pins located 180-degrees from here. When looking from one end, it is seen that there is only a single plane (I) through the configuration. The firing order for a standard flat plane is 1-3-4-2 which leaves evenly spaced and uniform 180-degree gaps between firings.

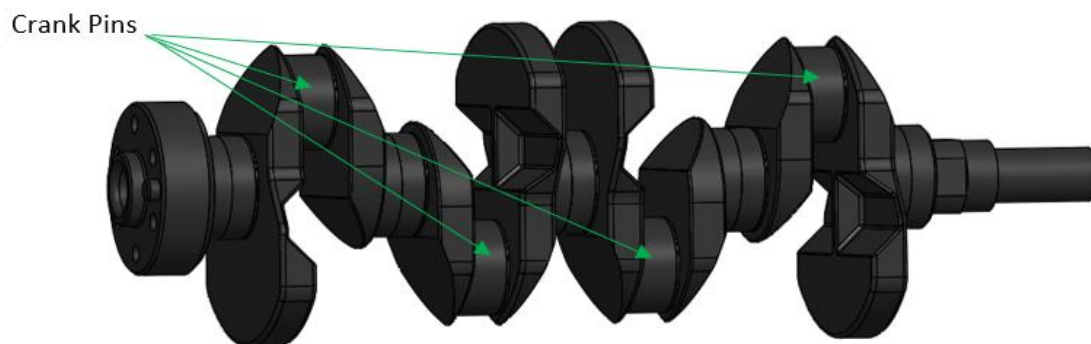


Figure 9 – Flat-plane configuration (Solidworks, 2020)

When discussing muscle cars, the crossplane crankshaft has led the world of V8 engines for decades. Due to the balancing on the crossplane crank, the engine experiences far less vibration, however it does mean a larger amount of rotational mass due to the balancing, which is unattractive to engineers building especially high-revving V8 motors, due to the inertia forces (Fuller, 2015). Another major disadvantage to the crossplane is the exhaust scavenging. Scavenging arises as the rush of gas leaving

the cylinder helps to pull more air/fuel mixture back in, due to the crossplane's non-linear firing order this is somewhat compromised.

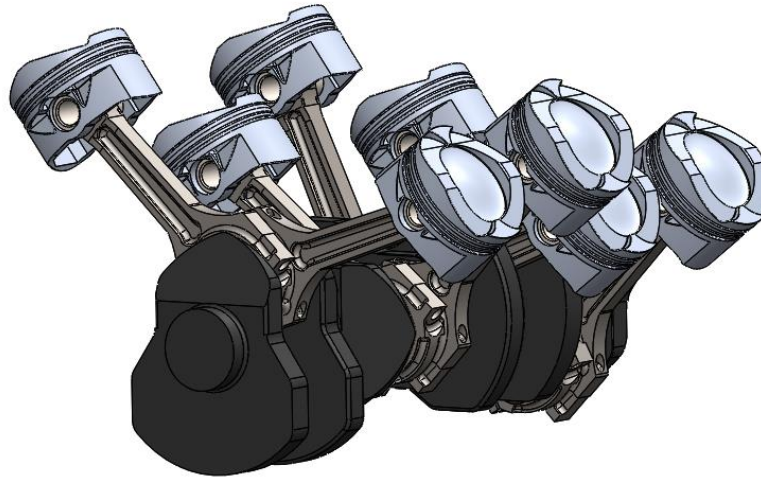


Figure 10 - V8 Configuration Crossplane

Recently, however, the flat-plane crankshaft has been used in the new Ford GT350 Mustang and although it is uncertain as to why this change has come about, there are several reasons that may suggest why Ford has decided to deviate from the standard configuration. Alongside the even firing order, excluding the need to have the primary header cross-over from bank to bank, the flat-plane crank is a lot lighter than the crossplane.

The reason for the lighter crank is due to the balancing, which does come at a price of increased vibration, this is due to the two-up-two-down configuration as opposed to the one at 90 degrees format, which results in larger moments produced, due to the inertia forces.

In comparison to the crossplane crank however, the flat plane can be built with much lower rotating mass, making it very desirable to race engine builders; this will mean less energy used revving up the engine and less stress caused at high RPM. Race engines will then be able to rev higher but with, however, a large amount more vibration which is a consequence that is put up with by the drivers.

[Connecting rod theory](#)

Balancing cranks, much the same as balancing anything, involves adding mass to the opposite side of the system in question, in this case the system is the reciprocating and rotating masses of the piston and connecting rod assembly.

To find balance in a crank, counterweights known as 'bobweights' are assembled with the crank on the balancing machine to act as the recommended mass (piston and connecting rod assembly).

$$M_{Bobweight} = M_{Rotating} + \frac{M_{Reciprocating}}{2};$$

Equation 4 - Bobweight mass

Where the masses are as previously explained, Singh (2006) writes of the reciprocating mass being 1/3 (33.3%) of the connecting rod mass including the piston, Gudgeon pin and piston rings. This then, leaving the rotating mass being represented as 2/3 (66.6%) of the connecting rod mass. This contrasts with Chen (2009) as he writes of using a CAD package to first create the connecting rod to then determine the centre of gravity of the part, followed by determining the mass percentages of the rotating and reciprocating areas by taking moments of it.

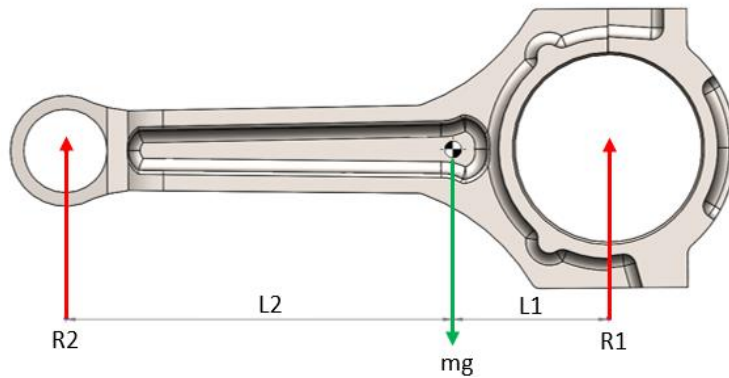


Figure 11 - Connecting rod showing centre of gravity location

Figure 11 shows the Ford 1.5l eco-boost connecting rod and cap produced using Solidworks. To accurately determine the mass percentages of the reciprocating and rotating areas, it is first essential to take moments about R2 of the connecting rod.

$$R_1 = \frac{L_2}{L_1 + L_2} \cdot mg;$$

Equation 5 - moment one about R2

Equation 6 gives the mass moment of 'R₁'.

$$R_2 = (m_{rod} \cdot g) - R_1;$$

Equation 6 - moment two about R2

By then using this equation the mass moment of ' R_2 ' can be found where ' m_{rod} ' is the overall weight of the connecting rod, with this, the percentages of the overall weight can be found.

Table 1 - Reciprocating and rotating mass percentages

<u>Area</u>	<u>%</u>
Reciprocating	28.7
Rotating	71.3

Table 1 shows the mass percentages of the reciprocating and rotating masses which are clearly different to the more commonly used 33.3% and 66.6%. Which raises the issue of which method is more effective? This is a question which will be investigated further into this report.

Balancing machine

Balancing machines are sometimes made by engine builders however, the Mutli-Bal 5000 is purpose made for crankshafts.



Figure 12 - Crank balancing machine (Industries, n.d.)

The 'Multi-Bal 5000' is a prime example of a balancing machine. Balancing machines work like tyre and wheel balancing machines, however, adding weight isn't always the answer; it is normal to see any crank drilled into to find balance (Kareem, 2017).

Once the crankshaft has been mounted to the balancing machine it can be run up to around 500-600RPM to find out if it is out of balance and by how much. Tolerances are always taken into place, and the monitor located to the right of the machinery reports how unbalanced the part is.

Methods of balancing

There are many methods of balancing for different applications, from racing, to production vehicles, to the mass produced low powered commuting vehicles. This section of research investigates two methods of balancing and the applications they are used for.

External Balancing

Balancing the engine does not stop at the crank. Some engine producers often build the crankshaft to less than 100% balance to save weight. This method is achieved by reducing weight from the crank webs while still accounting for the stresses applied to the main bearings (Massingill, 2016). But how is the vibration then accounted for? In a case where the crank was balanced to 60%, it is possible to use the pulley and flywheel in compensation for the unbalanced 40%. This is achieved by creating a centrifugal force through offsetting the centre of gravity of the flywheel and the pulley.

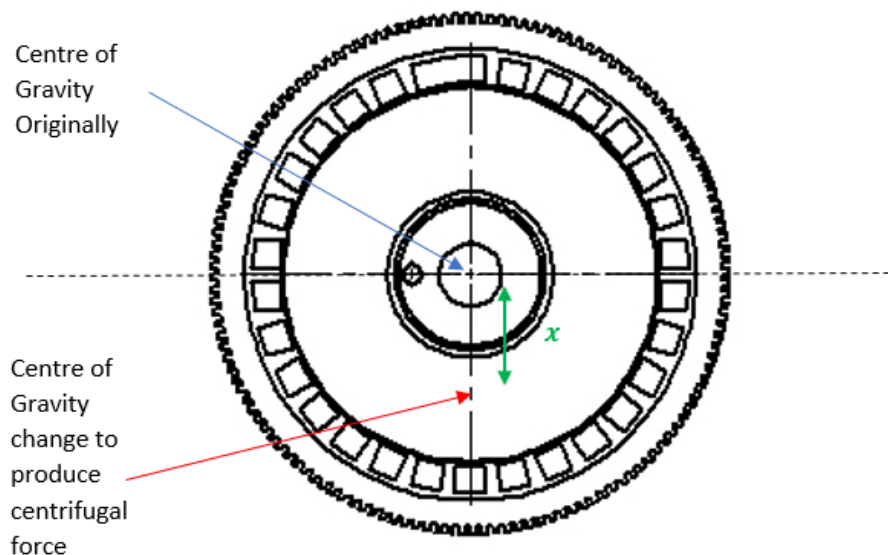


Figure 13 - 1.5l eco-boost flywheel

As can be seen from figure 13 above, the centre of gravity is originally manufactured around the centre of rotation of the flywheel, thus producing no centrifugal force. In the case of the balancing being carried out externally, the centre of gravity will be changed by a distance ' x '.

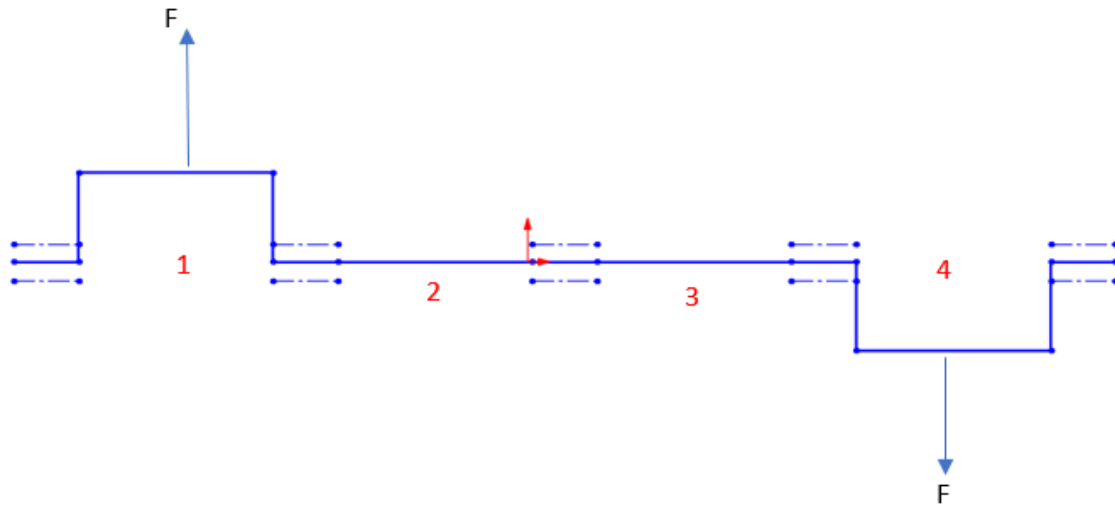


Figure 14 - Schematic of 4-Cylinder crossplane

Figure 14 shows a side-view schematic of the crossplane 4-cylinder crankshaft, where the dotted line represents the main bearings, and 'F' signifies the forces due to inertia. This representation would be similar in both the Yamaha R1 and most V8 muscle cars. As discussed previously, from the front plane, the schematic would resemble a cross. As can be seen from the figure, there are vertical moments being produced at the 1st and 4th pins, however the 2nd and 3rd are producing no vertical moments in this position as they are perfectly horizontal.

If 60% local balance was already achieved, then 40% is left over.

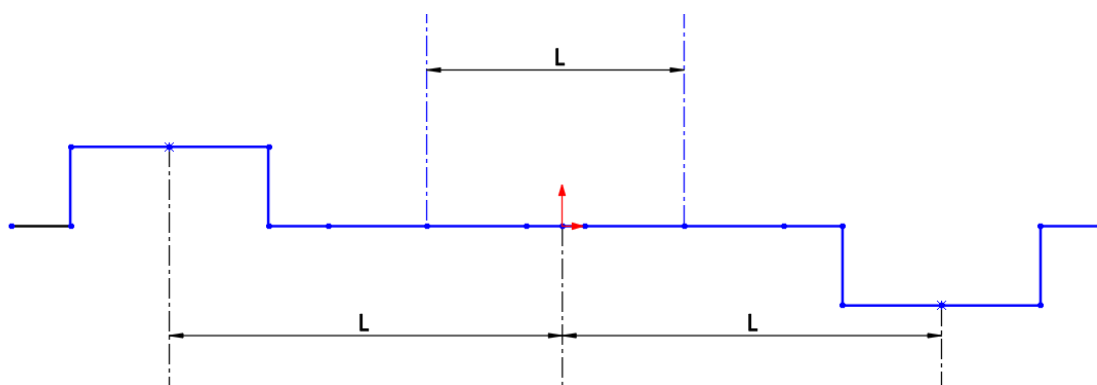


Figure 15 - Schematic for taking parallel moments

Using Figure 15, it is essential to take parallel moments about the centre.

$$M_{Parallel} = F \times 1.5 + F \times 1.5 = 3FL$$

Equation 7 - Parallel moment

Where:

$$F = 0.4 \left(m_{rot} + \frac{m_{rec}}{2} \right) . R;$$

Equation 8 - 40% compensation force

With this the parallel moment has been found. Furthermore, the perpendicular moment must be found.

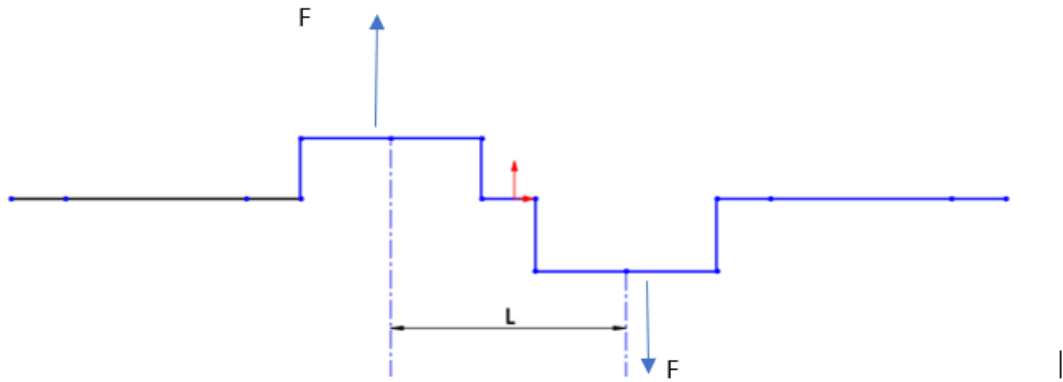


Figure 16 - Schematic for taking perpendicular moments

By then taking moments about the centre:

$$M_{perpendicular} = \frac{FL}{2} + \frac{FL}{2} = FL ;$$

Equation 9 - Perpendicular moment

Equation 10 then gives the moment perpendicularly. The two moments then rotate with the crank.

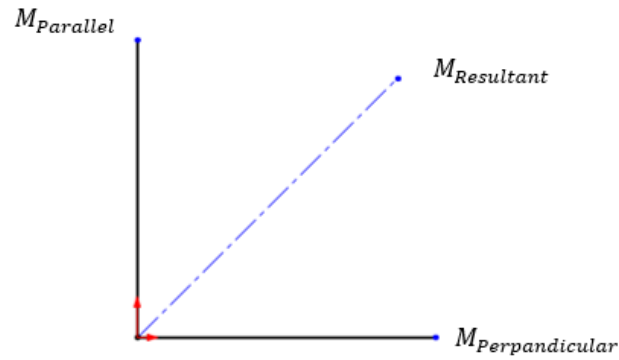


Figure 17 - Resultant moment

Figure 17 shows the perpendicular and parallel forces with a resultant force.

$$\theta = \frac{M_{\text{Perpendicular}}}{M_{\text{Parallel}}};$$

Equation 10 - Required angle for global balancing

By then inserting the correct information for θ .

$$\theta = \frac{FL}{3FL} = \frac{1}{3} = 18.435^\circ;$$

Equation 11 - Required angle for global balancing example

Using equation 12, the ideal angle is found that the weight should be offset by and located from the centre.

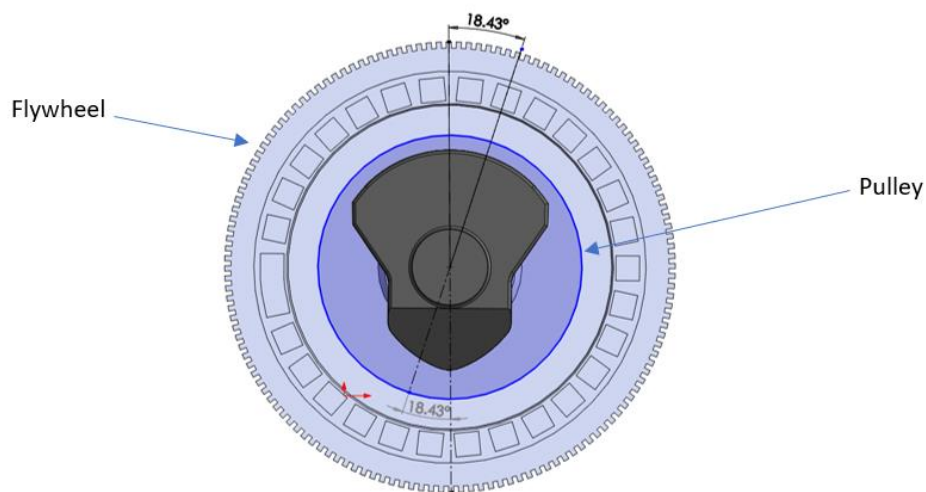


Figure 18 - Visual representation of mass offset for external balancing

Figure 18 shows a visual representation of where the extra mass is located. This would be the same for the pulley on the opposing side, however, it would be situated 180-degrees opposite to the flywheel's extra mass.

$$M_{Resultant} = FL\sqrt{3^2 + 1^2} = 3.16FL;$$

Equation 12 - Resultant moment

Equation 13 gives the resultant moment in kg.mm.mm. As the mass moment of the flywheel is equal the mass moment of the pulley.

$$M \cdot M_{Flywheel} = \frac{M_{Resultant}}{L_{Overall}};$$

Equation 13 - Required moment for flywheel

Where the ' $L_{Overall}$ ' is the overall length of the crank and half width of the flywheel and pulley. Using this equation, the mass moment of the flywheel and pulley can be found in kg.mm.

Halderman (2011) writes of the 350 cubic-inch Chevrolet V8 **internally** balances the rotating assemblies (all rotating parts are balanced individually), whereas the 400 cubic-inch Chevrolet externally balances the engine, which consequently increases weight.

Overbalancing

Overbalancing is a method used by crank balancers. The aim behind overbalancing is to remove the vertical 1st order vibration without a balance shaft. The horizontal vibrations are less apparent and therefore do not matter as much.

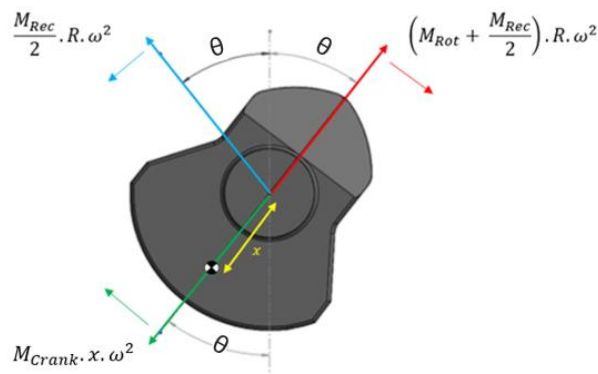


Figure 19 - Schematic showing forces.

Figure 19 shows the forces produced from the reciprocating and rotating mass's, along with the centrifugal force produced by the crank web ' M_{Crank} '. To overbalance is described in the following equation:

$$M_{Crank} \cdot x = \left(M_{rot} + \frac{M_{rec}}{2} \right) \cdot R;$$

Equation 14 - Overbalancing

By doing this, the resultant forces produced by the green and red forces becomes the following:

$$F_{Resultant} = \frac{M_{rec}}{2} \cdot R ;$$

Equation 15 - Resultant force when overbalancing

As can be seen from equation 16, the rotating speed is cancelled out, proving that the balancing cannot change with engine speed.

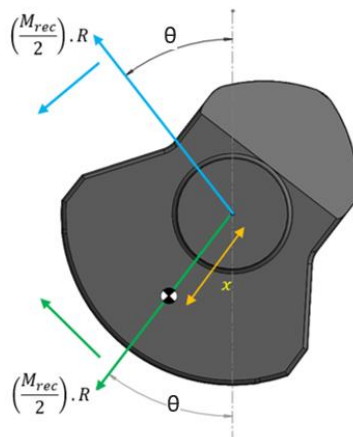


Figure 20 - Overbalancing

Finally, the two forces rotating in different directions cancel each other out vertically. Thus, eliminating the need for a 1st order balance shaft. Although this does save weight by not needing a 1st order balance shaft, it does add rotational mass onto the crank. Therefore, overbalancing may be desirable to everyday cars that want to save on space and build time but not so much for high revving race engines as vibration is less of an issue (Carley, 2014).

2nd order balancing

2nd order balancing consists of resolving the 2nd order vibrations due to engine speed. One of the key differences between 2nd and 1st order crank balancing is that 2nd order vectors move twice as far than 1st order.

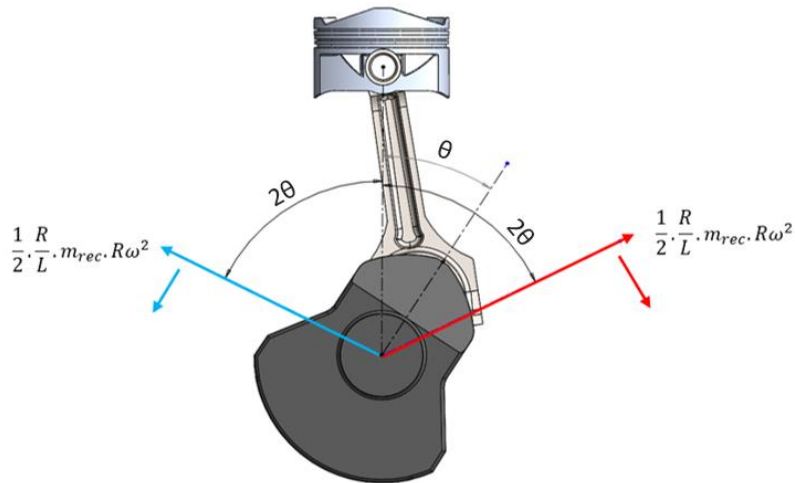


Figure 21 – 2nd order forces

Figure 21 shows the forward and reverse 2nd order vectors produced by double the engine speed, which correlates to the angle located at twice the original crank position.

$$R_{Vertical} = 2 \cdot \frac{R}{L} \cdot m_{rec} \cdot R \omega^2 \cdot \theta ;$$

Equation 16 - Vertical resultant force

' $R_{Vertical}$ ' gives the resultant force of the 2 vectors from the 2nd order inertia forces. As explained by Halderman (2011), secondary vibration is a weak, high frequency vibration caused by a slight difference in inertia at top dead centre in comparison to bottom dead centre.

2nd order balance shaft

To balance the 2nd order forces in a single cylinder, a balance shaft is needed for both rotating vectors.

$$F_{Balance\ Shaft} = m_{Balance\ Shaft} \cdot y \cdot (2\omega^2);$$

Equation 17 - Second order balance shaft force

Equation 18 gives the centrifugal force produced by the 2nd order balance shaft, where y is the distance from the centre of rotation to the centre of gravity.

$$m_{bal} \cdot y = \frac{1}{8} \cdot \frac{R}{L} \cdot m_{rec} \cdot R;$$

Equation 18 - balance shaft mass and length

Equation 19 then shows that the force at a certain length produced by the balance shaft must be equal to the 2nd order force at length 'R' produced by the reciprocating mass. The rearrangement of the mass of balance shaft equation can be found in appendices A. in the configuration of an inline 4-cylinder, the magnitude of the 2nd order forces are multiplied, thus making it hard to ignore, especially in larger engines with heavier parts (UKCAR, 2018).

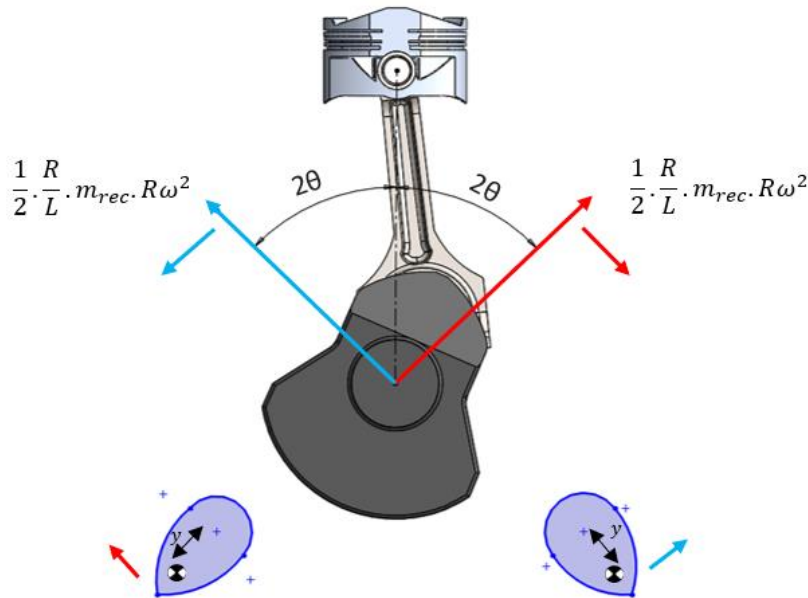


Figure 22 - 2nd order balance shafts and crank

Figure 22 shows the 2nd order balance shafts, which show rotation in the same direction to the vector being balanced, this would mean a configuration of gears would be needed to drive the balance shafts

at twice engine speed. Ruckle (2014) explained that without twin balance shafts, engines such as the Porsche 944 S2, 3-litre inline four would find it impossible to reduce vibration.

Lubrication

The crankshaft plays a significant part in the lubricating side of the internal combustion engine. Using 'oil passages', the spinning crankshaft is able to transfer oil to the reciprocating assembly.

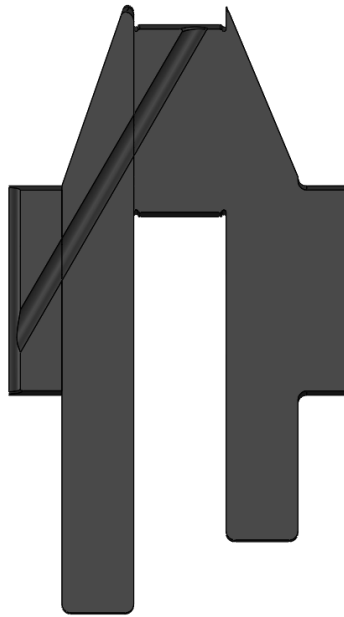


Figure 23 - Cut crank demonstrating oil passages

Figure 23 shows the oil passages on a single crankshaft. Drillings are provided to feed oil from the main bearings to the crankpin bearings. The bearings use a hydrodynamic oil film to lubricate the two solid surfaces efficiently. Hydrodynamic lubrication is used to reduce friction or wear of rubbing solids with the aid of a liquid, therefore, it is imperative to maintain a constant supply of oil to the bearings, connecting rod, and piston to prevent any unnecessary wear (Manning, 2012).

Cross angled drilling allows oil to lubricate both the grooved and un-grooved sides of a main bearing shell. Intersections of drilling helps to prevent bearing damage as a careful manufacturing procedure allows the crankshaft to avoid swarf traps, burrs, and sludge traps that can lead to passage block.

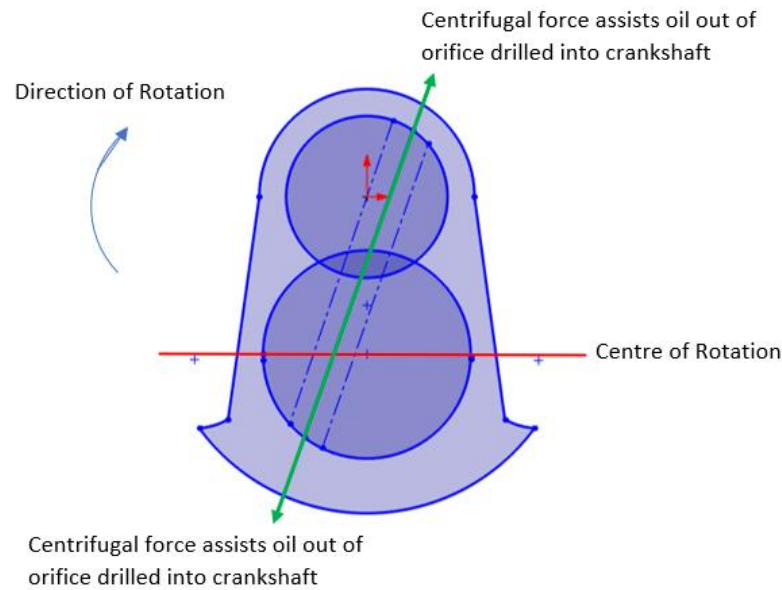


Figure 24 - Schematics of oil holes in crankshaft

Figure 24 shows the oil drillings in the crankshaft. The lubricating holes must be a respectable length from the edges of the crank web walls when passing through the structure, this is dependent on the size of the crankshaft. If the drillings are too close to the edges of the structure, the strength of the part could be compromised.

It can also be seen from figure 24 that the drillings are not located at the 12 and 6 o'clock position, this is to prevent the connecting rod from implementing peak force directly onto the location of the lubrication passage. Halderman (2011) writes of the holes being found where the stress and loads are at their weakest.

The edges of the holes are normally sharp from being drilled, this produced stress concentrations and can be prevented by carefully chamfering the whole. Alternatively, a ball bearing (slightly larger than the holes diameter) can be knocked into the hole, without infiltrating, to create a denser collar to the hole and thus eliminating the stress concentrations (Wahid Ferdous, 2020).

Conclusion to literature review

In comparison to the original aims and objectives, the literature review raised more detailed questions to be investigated within the project, this adds more intricacy into the project which then contributes and adds depth to the initial aims.

The crankshaft balancing explored the fundamentals of the theory, although, however good the theory may prove, the practical element will be investigated to show another point of view on the subject, that way a better and more thorough understanding of the topic can be obtained.

From this, the question of how much mass should be considered when accounting for the reciprocating and rotating areas of a connecting rod has been raised and thus, how it effects the

overall balancing. Furthermore, a conclusion can be drawn as to which method is more effective when balancing a crankshaft with a bob weight.

Investigating into the crossplane crankshaft raised several questions which will affect the actions of the project. Firstly, how is the crank to be balanced? Secondly, whether the crossplane crankshaft can offer a smoother torque delivery? Finally, does the gap in firing change the way a rider feels about the grip levels on a 4-cylinder crossplane motorcycle? Two of these questions can be answered somewhat mathematically, however, the final question is very much subject to the opinion of the rider.

The hypothesis of the hole drilling into the crankshaft demonstrated a constructive theory to lubricate the main bearing and reciprocating assembly within engine, however, further investigation and testing will expand the knowledge how the drilled holes affect the strength of the component.

8. Computer Aided Design (CAD) simulation

Introduction to simulation experiment

The following experiment is an investigation into providing evidence of the research undertaken concerning crank balancing and the mass of bob weights when balancing crankshafts.

The initial test investigates how to show a comparative difference between a marginally overbalanced single cylinder crankshaft and a crankshaft made with no counter webs. The aim of this research is to show a distinct trend difference when balancing the crankshafts.

Finally, this will be an inquiry into proving the difference between traditional method bob weights and the recently investigated CAD assisted bob weight approach.

Methodology

To first utilise the Solidworks motion simulation programme; it is essential that the user is competent and has a basic understanding of the operations and considerations taken into place when using the software. These considerations for optimal operating consist of the boundary conditions taken into effect for real life simulation and efficient data collection.

These boundary conditions consist of:

- Appropriate application of load bearing faces is vital within motion simulation, without correct allocation of loads the system is oblivious to forces being implemented, and therefore ignores them.
- Correct and accurate mates are an essential contribution to the motion simulation as it prevents any faults or errors in the system while contributing real-life features.
- Contacts between parts in an assembly produce collisions in a motion simulation if applicable, which present errors and problems to be solved, rather than the parts being intangible and idealistic.
- Changing the frame rates to higher frames per second increases accuracy in data but consequently also increases run time, a certain amount of compromise is therefore necessary.
- Operation speed, such as engine speed, can also influence the running time of a scenario, thus the need for compromise is essential.
- Considering gravity in a scenario adds realistic measures when collecting data.
- Material allocation is a very crucial step in conducting a motion simulation especially when the scenario is based on an engine in real life conditions; the materials of the components have different masses depending on the density and this can affect the results significantly.
- Accounting for frictional losses increases the accuracy within the experiment, thus providing more precise results.

To sum up, the boundary conditions add realistic measures and increase reliability and accuracy in data collecting, however, this does come at a cost of time. Therefore, an iterative approach to the investigation informs the operator of problems with simulation in hand.

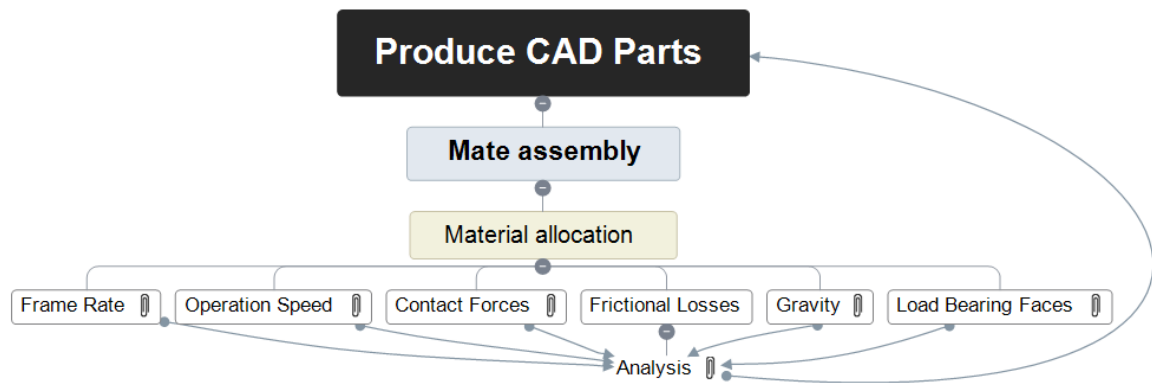


Figure 25 – Flow-chart of boundary conditions

Production of CAD parts

Through following the methodology, it is necessary to produce CAD models using Solidworks as a validation study to prove the theory of balancing the crank. To begin at a reasonable starting point, a 1.5l eco-boost engine has been acquired to familiarise the operator with the internal combustion engine and measure certain components to implement in the motion simulation.

The parts are constructed in CAD using a Vernier calliper, micrometre, and a radius gauge to accurately measure the lengths, diameters, and radii for the most precise outcome. It is essential to make sure the measurements being taken are correct, as this can alter the force being transmitted through the parts and affect the results.

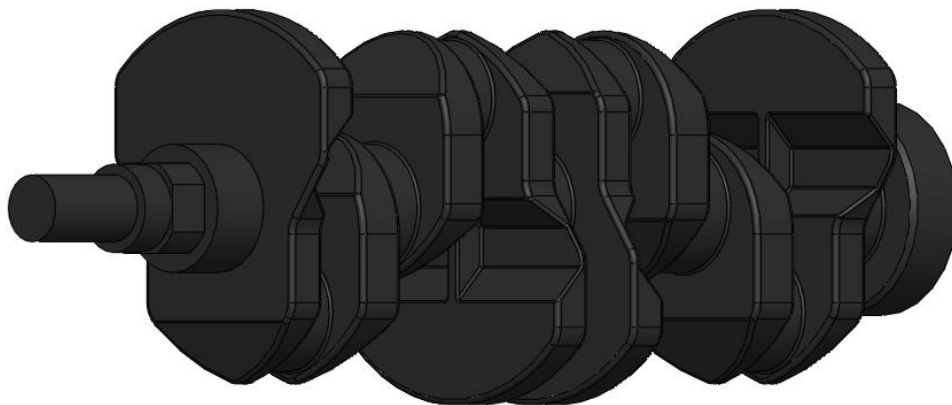


Figure 26 - Ford 1.5l eco-boost crankshaft

Figure 26 shows the Ford 1.5l eco-boost crankshaft created using Solidworks. The standard crank is cast out of steel. Casting process involves the utilisation of molten materials, usually metal. This molten material is then poured into a mould cavity that takes the form of the finished part. Advantages are that casting is relatively inexpensive and is used by mass produced engine manufacturers (Thomas, 2019).



Figure 27 - Forged crank on lathe (Koflach, 2019)

The main journals and connecting rod pins are then machined to the correct dimensions and finally the crank is balanced to specification.



Figure 28 - Measurement of crank

To build the crankshaft up on Solidworks, each section of the crank has been measured and produced individually, from the webs to the journals as shown in figure 28. The reason for following this

procedure of production is to ensure the detail of the original crank is carried through the project when making adaptations. If the original model is as close to how Ford designed it, the operator will be more aware of the minor details when re-designing it for the crossplane configuration.

Assembly

To ensure the most precise results, the computer designed crankshaft must show significant similarities to the original model, therefore, combining the crank components together is an essential step in the production process. To combine the crank components together, it is first essential to untick the '*merge results*' box in a boss extrude, this allows the user to create a solid body from the parts being connected.

Once this has been carried out, the individual parts can be combined into a solid body and saved as a complete part. With the product being saved as a solid body, the motion simulation will have less to process, and the results will be more accurate due to fewer seams and breaks in the model.

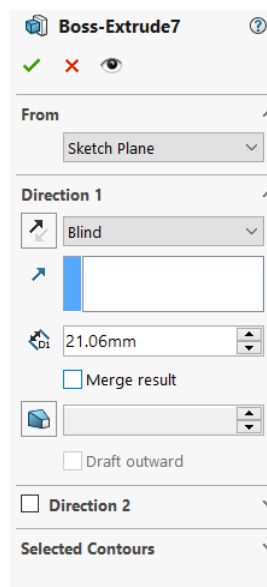


Figure 29 - Box extrude window

Material allocation

The material allocation is essential to creating a realistic motion simulation, as the materials possess different densities producing a range of masses, this has a significant effect on the inertial forces.

Table 2 - Material allocation

<u>Component</u>	<u>Material</u>
Crankshaft	Cast steel
Flywheel	Steel
Piston	Aluminium
Gudgeon pin	Steel alloy
Connecting rod	4340 steel
Connecting-rod cap	4340 steel
Cylinder	Cast iron
Sump	Cast iron

Table 2 shows the materials allocated to the parts being utilised in the motion simulation. As can be seen from the table, the materials used in the Ford engine are generally quite common and easy to source. This is because engines like the 1.5l eco-boost, made by OEM's, are mass-produced and if every engine was using lightweight materials the cost would be too high to justify or profit from, in addition, a significant portion of the consumer base would be priced-out.

Racing teams demand high performance and lightweight engines that can withstand high temperatures. In the case of drag race engines, the connecting rods must withstand extreme temperatures but for a small amount of time, which is why con-rod failure is quite common through stretch, this means teams must change them more frequently (BME, 2020).

For racing crankshafts, T. Rolvag (2020) explains that use of AISI 8620 steel can be used with carburising treatment. Various other treatments are also used such as fillet rolling, diamond like coating, and material annealing to harden the crankshaft, and increase fatigue life.

Halderman (2011) explains that SAE 4340 steel is used in high performance crankshafts. The product is made from hot steel billet through a series of forging dies, with die refining the structure of the crankshaft more than the last.

These materials have been allocated to this simulation to replicate commonly used materials for mass produced engines, such as the 1.5l eco boost engine. By using the same materials allocated to the same engine, the masses will be in proportion to each other for precise results.

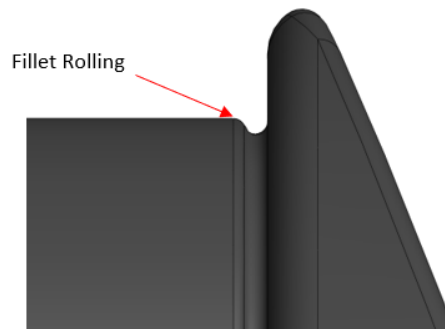


Figure 30 - Fillet rolling, located at connecting rod journal

Inadvertently, the Ford 1.5l eco-boost crank possesses fillet rolling around the connecting rod journals where it connects to the crank web, this is to reduce stress concentrations. Bending moments produced about the rod journals cause stresses in these areas (HPM, 2012).

Motion simulation

Initially, an experiment will take place to test the difference between a crank which has no webs, in comparison to an overbalanced crankshaft. The aim of this test is to prove the reason for crank balancing and the consequences of no balancing on the crankshaft within an engine.

Following this, an investigation in to two different methods of bob weight balancing has been carried out to gain further understanding of the significance and importance of the correct bob weight masses when balancing a crankshaft.

Test conditions

For the experiment to be fair and provide valid results, all tests undertaken must follow a standard to ensure the findings are reasonably accurate.

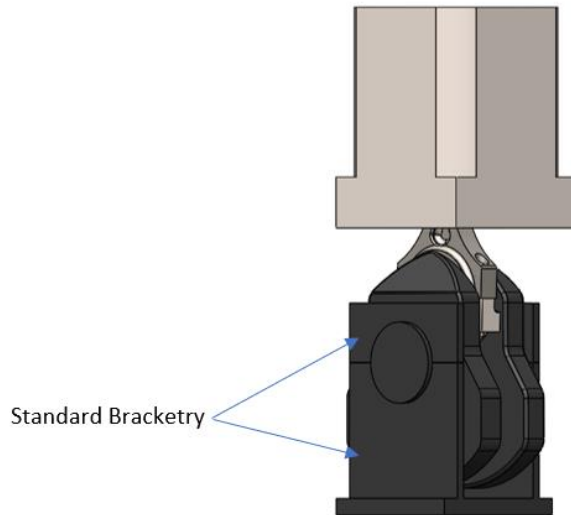


Figure 31 - Standard test conditions for investigation

Figure 31 shows the standard format of assembly to be followed when undertaking the tests. The only exception is that the cylinder is unnecessary when carrying out the bob weight testing.

Table 3 – Motion simulation conditions to experiment

<u>Condition</u>	<u>Parameter</u>
g	9.81 m/s ²
Bracketry	Standard
Operation speed	600RPM
Analysis	Load bearing faces
Contacts	Steel (greasy)
Frames per second	1000

Table 3 shows the conditions for the experiments being undertaken using Solidworks, and the parameters to be considered when commencing the tests.

Contrast test

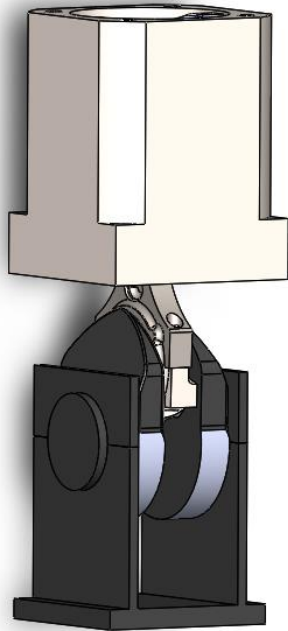


Figure 32 - No web scenario

Figure 33 shows the second test's assembly. The simulation is being carried out with a single cylinder assembly, this is to reduce redundancies seeing as the theory is the same for a single cylinder in comparison to multiple cylinders. This assembly uses a crank which has been marginally overbalanced to the envelope layer, therefore the crank is not perfectly balanced, however it is not considered completely overbalanced.

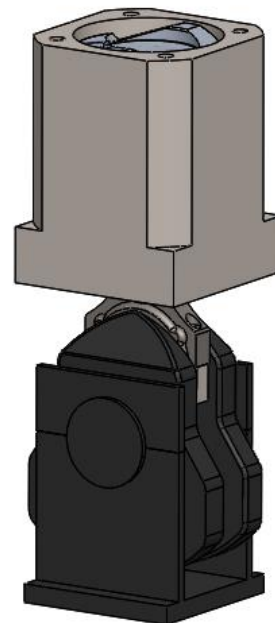


Figure 33 - Marginally overbalanced scenario

Bob weight testing

Traditional method

The initial area of testing investigates the theory of bob weight masses. By conducting this investigation, a conclusion can be drawn to prove which method is more effective.

$$m_{Bob} = m_{rot} + \frac{m_{rec}}{2} ;$$

Equation 19 - Bob weight mass

Bob weight masses are commonly produced using the above equation. However, just as previously discussed, there are questions as to what percentage of mass should be reciprocating and rotating, therefore, a test using Solidworks motion simulation has been undertaken.

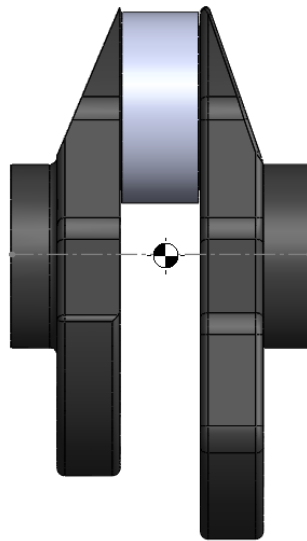


Figure 34 - Bob weight test - traditional method

Figure 34 shows the bob weight and single cylinder crankshaft that has been balanced, including the centre of gravity located at the centre of rotation. The test has been undertaken as if a crossplane crankshaft has been balanced, therefore, the centre of gravity should be located at the centre of rotation. The crankshaft in this test has been balanced traditionally, using the rotating mass of the connecting rod to be 2/3 of the overall mass, and the reciprocating mass to be 1/3 of the overall mass, plus the mass of the piston and gudgeon pin.

CAD Method

The following test has been undertaken by first utilising the previously calculated mass percentages of the connecting rods' reciprocating and rotating masses using CAD.

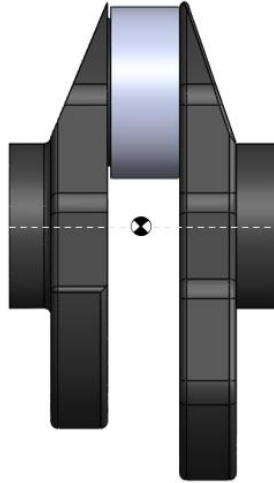


Figure 35 - Bob weight - new balancing

As can be seen from figure 35, there is little difference, visually, between the new method of bob weight mass balancing, and the traditional method of bob weight mass balancing; however, the results will be the determining factor in deciding which is more effective.

Results

As Solidworks does not provide a package or channel to provide the amount of vibration created by a moving part, the results have been found by measuring the stresses through the bracketry over a single cycle of rotation at 600RPM engine speed.

Comparative test

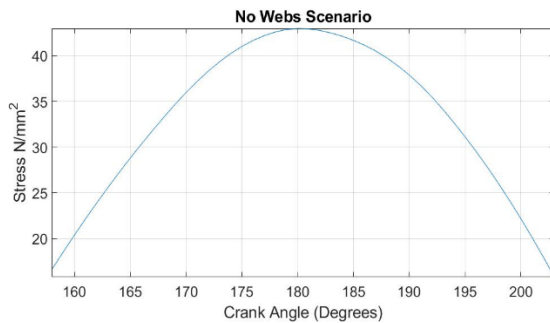


Figure 36 - Stress plot of crank balanced without webs

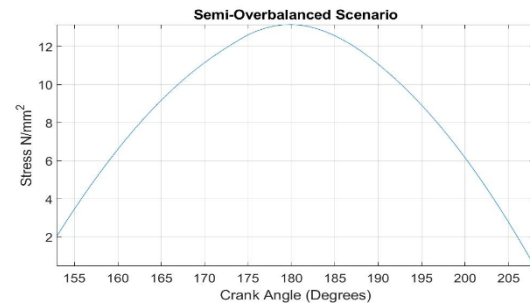


Figure 37 - Stress plot of marginally overbalanced crank

Figure 36 shows the test results utilising a crankshaft with no counterweights, figure 37 shows the results to the test utilising a marginally overbalances crankshaft. The large amount of contrast shows a clear difference between the two extremes of balancing, figure 36 presents much higher stresses than that of figure 37. If the 'No webs scenario' was undertaken, large amounts vibration would have to be overcome using the engine mounts, Whereas the 'semi-overbalanced scenario' results in much more bearable vibrations to dampen.

Bob weight balancing comparison

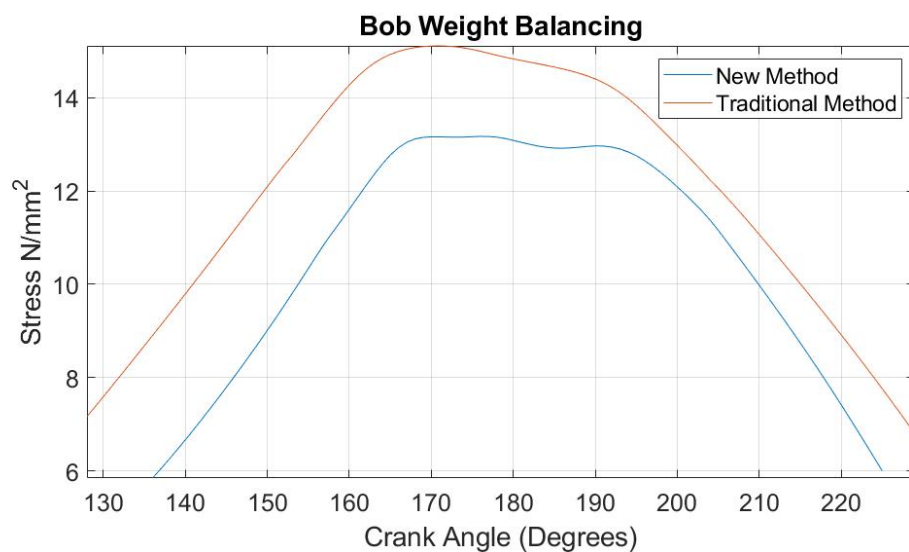


Figure 38 - Bob weight balancing stress on bracketry

Figure 38 shows both tests of balancing methods undertaken using Solidworks. It can quickly be seen from the graph that the 'new method' of balancing is implementing less stress on the bracketry, thus creating less vibration though multiple cycles. There are, however, two schools of thought on the matter. Although it may seem evident, in this case, that the results show positive feedback from the new balancing method, many crank balancers of V8 engines still prefer to use the more traditional method (Carley, 2014).

Conclusion to motion simulation

The investigation showed positive results that agreed with the research and theory, this however does raise a further question of whether the configuration of the engine can affect the choice of which balancing method should be implemented?

Both tests show stresses being implemented on the bracketry which shows that there is still a force being produced by the rotating crankshaft. This may be due to the reverse vector implementing force on the bracketry. Interestingly, the test utilising the slightly overbalanced crankshaft resulted in less stress applied to the bracketry than the bob weight testing, this may be a product of the over balancing cancelling out a portion of the reverse rotating vector.

9. Matlab investigation

Introduction to the Matlab investigation

The following section of the report aims to explore the theory of the crossplane crankshaft in comparison to a single plane crankshaft by examining the theory claimed by Yamaha in their YZF-R1, 1000cc motorcycle that utilised this configuration.

Yamaha (2009) stated the intervals between firing in the crossplane crankshaft gave the tyre enough time to re-grip to the tarmac being rode on. Yamaha also claimed that the new technology improved delivery of torque at high RPM. With this said, the theory of how the crossplane crankshaft achieves this has been considered and investigated to further understand the benefits of using this configuration.

Methodology

To first find the trend of torque produced by the crossplane crankshaft, the forces produced by the engine speeds are essential to move forward with the investigation.

$$F_{Inertia} = R\omega^2 \cos \theta + \frac{R}{L}R\omega^2 \theta \cdot (m_{recip});$$

Equation 20 - Inertia force

Equation 21 gives the inertia force at a certain engine speed ' ω ' and crank angle ' θ ', considering the connecting rod length ' L ', crank throw ' R ' and the reciprocating mass ' m_{recip} '.

$$T_{Inertia} = F_{Inertia} \cdot \tan \tan \phi \cdot R \cdot \cos \cos \theta - F_{Inertia} \cdot R \cdot \sin \sin \theta ;$$

Equation 21 - Inertia torque

Following this, equation 22 can be used to find the inertia torques considering the connecting rod angle ' ϕ '.

$$\phi = \frac{R}{L} \cdot \cos \cos \theta ;$$

Equation 22 - Rod angle

Equation 23 then gives the rod angle. Following this, finding the gas torque is necessary.

$$T_{Gas} = F_{Gas} \cdot \tan \tan \phi \cdot R \cdot \cos \cos \theta - F_{Gas} \cdot R \cdot \sin \sin \theta ;$$

Equation 23 - Gas Torque

A combination of the inertia torques, and the gas torques is called composite torque. Furthermore, the Matlab function '*circshift*' has been applied to offset the data being used to simulate the varied crank angles. Prior to this, other methods were used, and elaborated in appendix C.

```
%% Cyl offset Torque inertia inline standard

Cyl_3_IT = circshift(Torque_inertia,[0 180]);
Cyl_4_IT = circshift(Cyl_3_IT,[0 180]);
Cyl_2_IT = circshift(Cyl_4_IT,[0 180]);
Crank_IT = (Torque_inertia)+(Cyl_3_IT)+(Cyl_2_IT)+(Cyl_4_IT);
```

Figure 39 - Matlab script of data offsetting – single plane

Figure 39 gives the data offsetting of a standard inline four cylinder. As can be seen from the script, the operator has offset the original data by 180-degrees and then proceeded to do the same with the following cylinders. The matrices are then combined to produce a full cycle of inertia torques.

```
%% Cyl offset Torque inertia xplane

Cyl_3_ITx = circshift(Torque_inertia,[0 270]);
Cyl_2_ITx = circshift(Cyl_3_ITx,[0 180]);
Cyl_4_ITx = circshift(Cyl_2_ITx,[0 90]);
Crank_ITx = (Torque_inertia)+(Cyl_3_ITx)+(Cyl_2_ITx)+(Cyl_4_ITx);
```

Figure 40 - Matlab script of data offsetting – crossplane

Figure 40 gives the data offsetting of the crossplane configuration. What is different in this script is the amount that the data has been offset by. Cylinder 3 is offset by 270-degrees from the standard inertia torque, cylinder 2 is offset by 180-degrees of cylinder 3, and cylinder 4 is offset by 90-degrees from cylinder 2. This then creates a simulation of the crossplane crank inertia torques.

Results

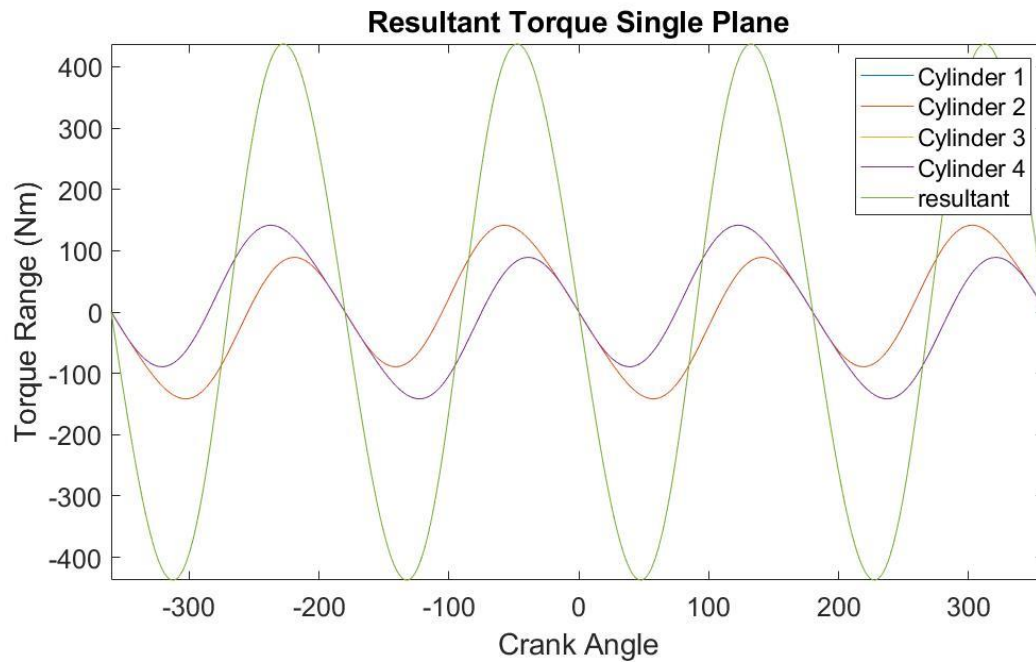


Figure 41 - Single cylinder and resultant single plane torques

Figure 41 shows a single plane's instantaneous inertia torque produced on each cylinder, with a resultant torque shown also at 6000RPM. As the configuration is two-up two-down, cylinders 1 and 4, and 2 and 3 share the same trend. The result, however, is much higher than the individual produced torques as the cylinders are effectively working together to generate a greater torque.

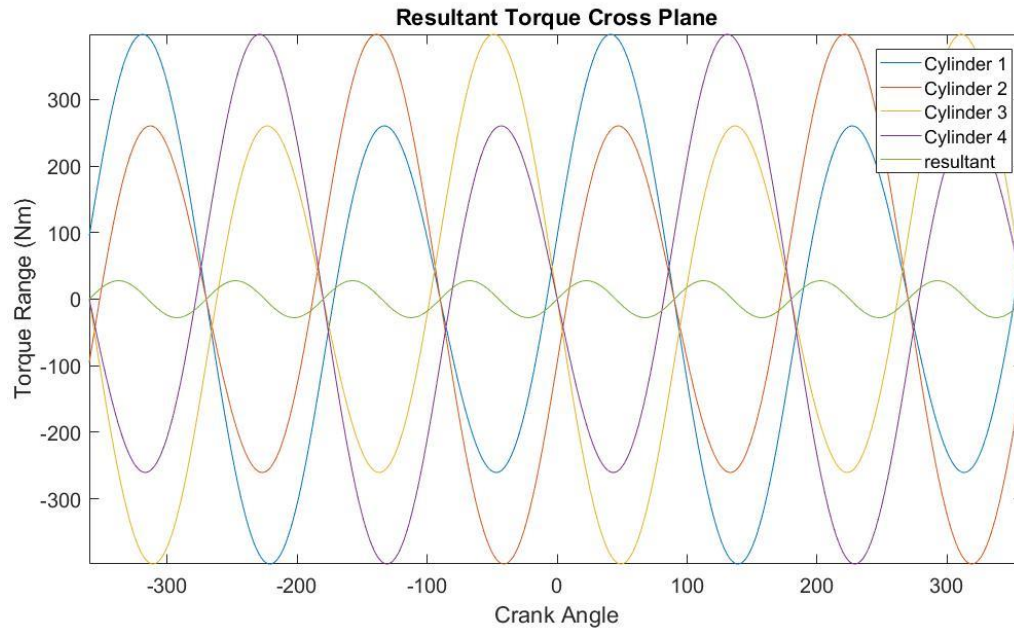


Figure 42 – single cylinder and resultant crossplane torque

Figure 42 shows the individual cylinders' inertia torques produced over a cycle, and the resultant torque from the crossplane crank. The first difference that can be seen from the graph is the result is lower than the individually produced torques, this is due to the perpendicular configuration of crank pins, cancelling one another out.

Figure 42 also shows that a crossplane engine does not put out anymore/less torque than a single plane engine, but displaces the torque in a different manner, due to the configuration.

Inertia torque

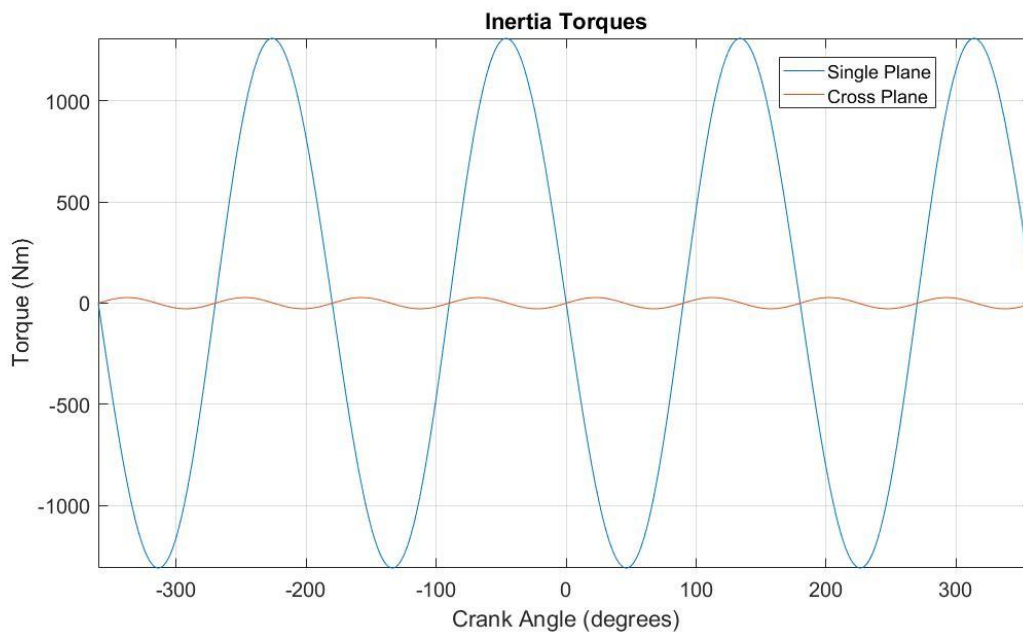


Figure 43 - Inertia torque 6000RPM

Figure 43 shows the inertia torques produced by the single plane and crossplane crankshaft at 6000RPM. The first noticeable difference in the graph is the magnitude of torque being produced by both crankshaft configurations, whereas the crossplane reaches barely above 20Nm, the single plane exceeds 1000Nm per revolution.

The crossplane configuration produces twice as many oscillations of torque per crank revolution due to the crank pins being located perpendicular to one another, which delivers a low-magnitude yet high-frequency vibration.

The single plane produces higher amounts of magnitude, half as much as the crossplane, this means excessive amounts of vibration as the engine speed increases.

Gas torque

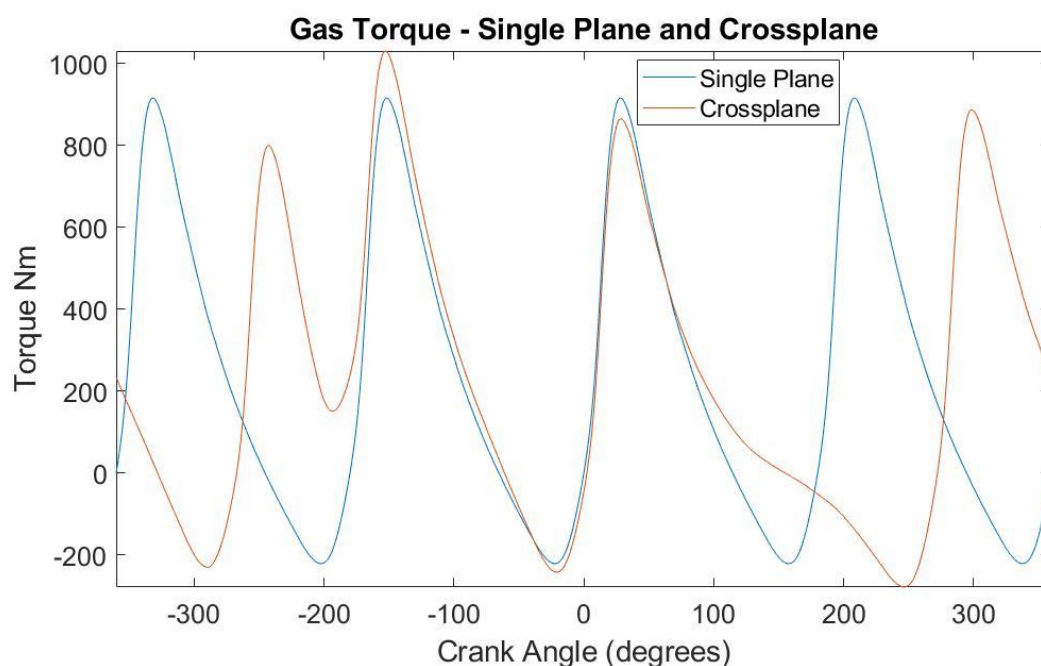


Figure 44 - Gas torque - single and crossplane

The gas torques are proportional to the gas forces. Figure 44 shows the gas torques in firing order of single plane and crossplane; the figure shows an obvious difference between the two configurations and how this can affect the gas torques. Whereas the single plane returns to the same amount of torque each rhythm, the crossplane crank has intervals of torque close enough to one another that the previous pulse of torque assists the following, producing an individual spike greater than the others.

Composite torque

Composite torque is a term used to describe the combination of the inertia torque and gas torque. Or from a rider's/driver's point of view, how the torque is being output when applying the throttle.

Single plane

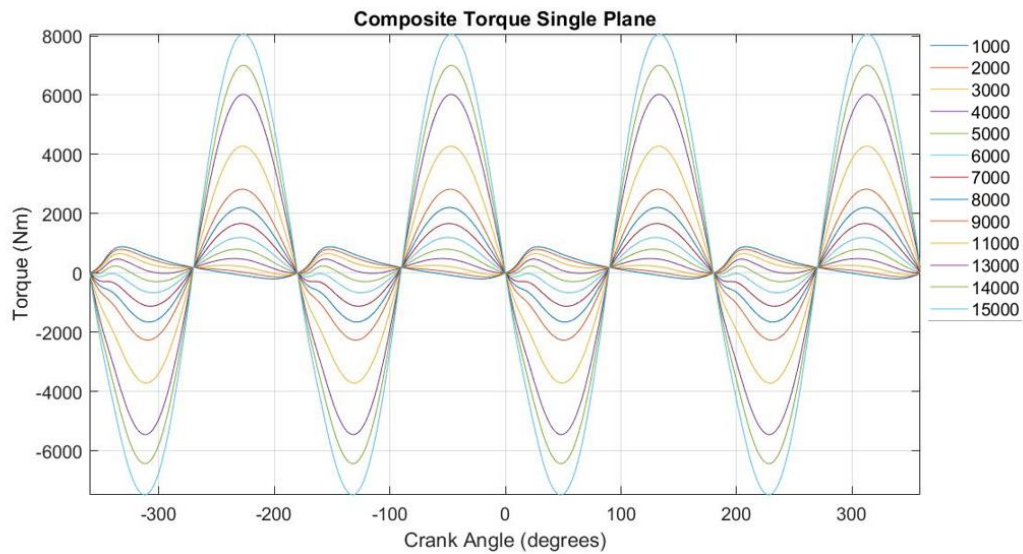


Figure 45 - Composite torque - single plane

Figure 45 shows the composite torque of the single plane crankshaft. The single plane crank produces linear-pattern torque output from each cylinder every 180-degrees of crank rotation. The magnitude of the torque output on the single plane increases significantly with every increase in RPM.

Crossplane

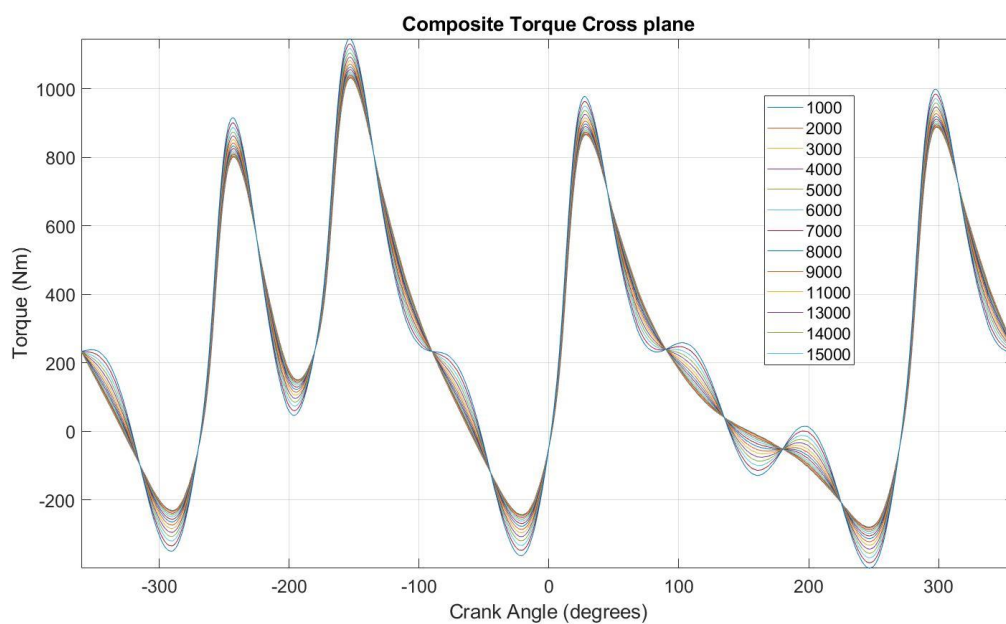


Figure 46 - Composite torque - crossplane

Figure 46 shows the composite torque of the crossplane crankshaft. As can be seen from the schematic, there are noticeable differences between the crossplane and single plane crankshafts. The first being the non-linear pattern of torque that is a product of the firing order, additionally, the 270-degree gap, that Yamaha claim is for tyre grip to be regained, can be distinctly seen in this diagram.

Another distinct difference is that the inertia torques do not have as much of an influence on the composite torque as the single plane, meaning the feedback that the rider/driver is receiving from the vehicle is a greater amount from the gas torques in the crossplane, rather than the inertia torques masking the input from the throttle in the single plane.

Table 4 - Crank rotation times

<u>Angle between firing (degrees)</u>	<u>Time taken to rotate (s)</u>
180	0.002
270	0.003

Following the claim of the time between firing being enough to allow the tyre to re-grip, table 4 shows two different scenarios of angles between firing and the time it takes to complete this rotation at **13500RPM**.

180-degrees is a common angle between firing used in single planes and results in being completed in 2 thousandths of a second, the crossplane crank has one gap of 270-degrees within its full cycle and Yamaha claim that the extra 1 thousandth of a second is enough to re-grip the tyre to the tarmac.

Table 4 demonstrates a basic understanding of what procedure is taking place towards Yamaha's claim; however, the tyres are the main contribution to maintaining grip on a road. This said, to prove this theory made by Yamaha, a test will have to consider certain factors.

- Tyre compound
- Tyre pressure cold and warm
- Tyre temperature
- Ambient temperature
- Road temperature
- Mass of vehicle (Equal amounts of fuel every test)
- Tyre life
- Tyre condition (New set of tyres every test)
- Tyre age
- Heat cycles
- Manufacture of tyre (Prevorsek, 1980).

Prevorsek (1980) explains that these variables play a very important part in tyre deflection testing; however, in an open circuit or road test, these variables cannot be schooled properly as there is not a way of depending on them.

Centrifugal forces

The following diagram shows the centrifugal forces produced by the rotating masses of a Kawasaki ZX10R and a Yamaha YZF-R1M in relation to engine speed and crank radius, the purpose of this test is to gain an understanding of Yamaha's thought process by comparing similar engines with different crank configurations and masses.

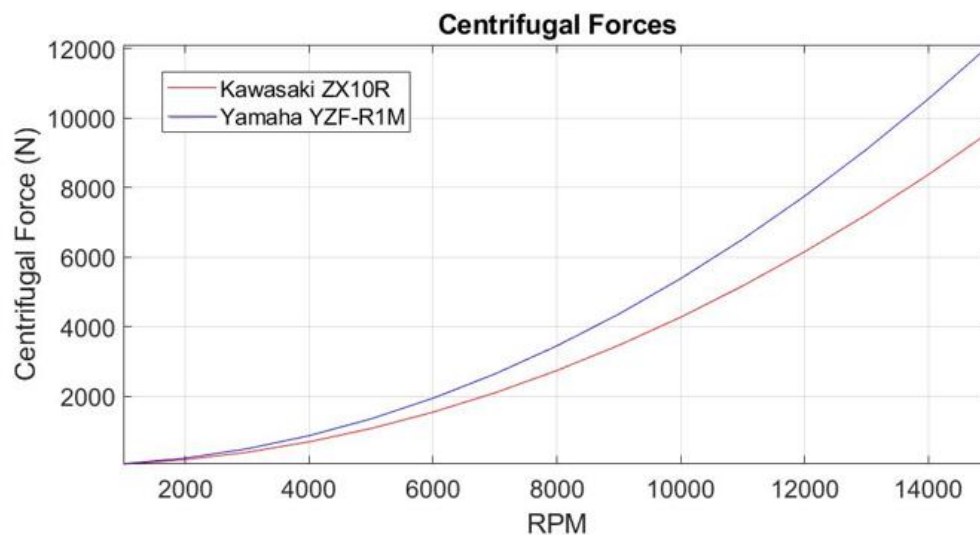


Figure 47 - Centrifugal forces

The graph presents a higher rotating force produced by the Yamaha in comparison to the Kawasaki, this is due to the balancing method of the crossplane producing a higher rotating mass as previously explained.

It is evident from the results gathered that Yamaha sacrificed having more rotating mass than most, with the advantages of smoother torque delivery. However, the supposed ability to re-grip the tyre in 3 thousandths of a second does seem improbable.

Conclusion to Matlab experiment

Initially, the results depicted positive results from the claims Yamaha had made concerning the torque delivery. The trend produced by the crossplane crank shown in figure 43 shows significantly lower values than that of the single plane.

Furthermore, the tests proved the assertions made by Yamaha during their promotion of the YZF-R1 on paper, however without any real testing the evidence remains a theory. Further real-life testing in ideal conditions would prove whether the theory of extra grip is present in a noticeable way. Moreover, tyre testing would enlighten the operator as to whether the tyres are able to re-grip in the timeframe between firings.

10. Model production

During the project, computer aided designs were used to produce and develop a physical model of the crossplane crankshaft. Starting with a detailed CAD model, the part was then generated using a 3D printer, followed by a CNC machined aluminium part.

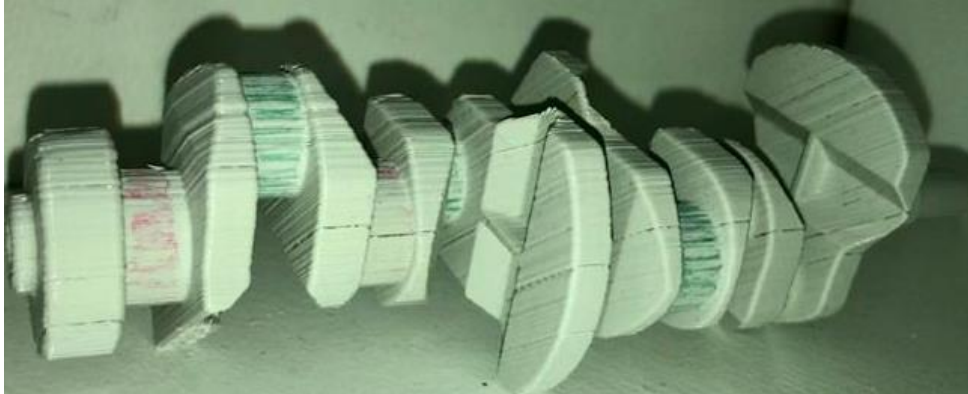


Figure 48 - 3D printed from CAD drawings

Figure 48 shows the 3D printed model with the rod journals highlighted in green and the main bearings highlighted in red. The model is 158.24mm and was used as a base 3D model to support more detailed and mechanically accurate production.



Figure 49 - Aluminium CNC crossplane crankshaft from CAD

Following the 3D printed model, and with material ready to be used and easily sourced, an aluminium model was created. The model was manufactured using a 'Mazak smooth VCN 430A' computer numerical control (CNC) machine. As there was a lack of the specific machinery required the model had to be produced in separate parts, to create the individual part, it was essential to split up the CAD model into webs and bearing then combined using dowels. This can be seen in figure 49 and is expanded on in the appendices.

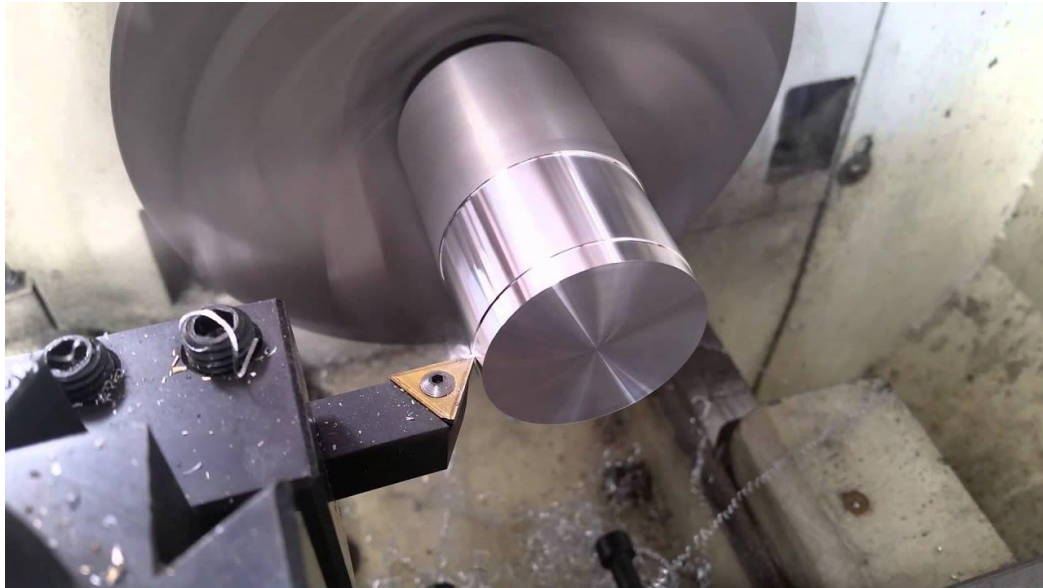


Figure 50 - Turning the dowels from aluminium

Several of the individual parts were turned down using a 'Quickturn 200MB' lathe provided in-house, the dowels are 28mm in length and 8mm in diameter. The parts were then built up together using a C clamp to compress each individual section together and araldite to prevent the components from twisting out of configuration.

Conclusion to model production

The physical model gave a clearer understanding of the manufacturing element of the engineering process. The 3D printed model gives a good idea to the configuration without wasting man hours producing it, however the aluminium model is much more presentable and shows the capability of the Solidworks software and what can be made with it.

Additional models would expand the understanding of what needs to be considered when creating a physical part. In an ideal world a life size model could be created to test in any given engine, however, due to joint limitations of machines and materials this is not possible at this stage.

11. Conclusion

The motion simulation, using Solidworks, provided evidence for two theories, the initial theory proven was the need for the counter webs when balancing a crankshaft, the evidence shown in figures 36 and 37 presents an obvious difference between the two scenarios. However, the test using a marginally overbalanced method showed stresses being produced on the bracketry, this is believed to be a consequence of the reverse rotating vector and raises question to if these forces can be eliminated further.

The second theory proven showed that the traditional method of bob weight balancing in the situation of a single cylinder was not as effective as the CAD assisted technique. The results shown in figure 38 present an evident difference between the two techniques. This can then be taken into further work concerning balancing with a bob weight to gather more efficient and accurate results.

The Matlab assisted area of the project showed sufficient evidence in proving that the torque delivery on the Yamaha R1M superbike was far smoother than that of a single plane superbike. This however came at a price of extra rotating mass, as shown in figure 47. A stress analysis study on the two methods would be beneficial as to showing how much additional stress the higher rotating mass puts on the system.

The results show a correlation to what Yamaha claim to being 'enough of a gap between firing' to de-stress the tyre and re-gain grip, however, with the results of table 4 and the insufficient evidence otherwise, the idea remains a theory. With further research and tyre testing this hypothesis can be investigated into greater detail, therefore a future project would be beneficial to concluding on a definite verdict.

As discussed within the report, the counter rotating vectors can be resolved using a balance shaft. The results produced from the motion simulation proved to be effective however a stress was, nevertheless, being applied to the bracketry from what can only be the reverse rotating vector. Therefore, a repeat attempt of the motion simulation, including a balance shaft, and calculating a resultant stress acting on the bracketry would further enhance the understanding of the results gathered.

Due to lack of facilities, a physical test was unable to take place. This would have been beneficial towards gaining a thorough understanding of what the results may have come out as in comparison to the simulated findings. Although a prototype of the crossplane crankshaft was produced, casting machinery is unavailable in house, therefore, the ability to manufacture a product would have not been feasible.

Limitations due to current situation

Unfortunately, due to lack of software towards the end of the project, several experiments could not take place and further physical models were unable to be produced. However, a large amount of the work had been completed before the covid-19 outbreak, therefore, not a lot of disruption occurred.

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13. Appendices

Appendix A

$$4. m_{balance\ shaft} \cdot y \cdot \omega^2 = \frac{1}{2} \cdot \frac{R}{L} m_{reciprocating} \cdot R \cdot \omega^2 ;$$

The ω^2 cancels out and re-arranged becomes.

$$m_{balance\ shaft} \cdot y = \frac{1}{8} \cdot \frac{R}{L} \cdot m_{reciprocating} \cdot R ;$$

Appendix B

Full Part	13/11/2019 13:22	SOLIDWORKS Part...	2,148 KB
Part1	13/11/2019 13:35	SOLIDWORKS Part...	143 KB
Part2	13/11/2019 13:37	SOLIDWORKS Part...	339 KB
Part3	13/11/2019 13:37	SOLIDWORKS Part...	145 KB
Part4	13/11/2019 13:37	SOLIDWORKS Part...	204 KB
Part5	13/11/2019 13:38	SOLIDWORKS Part...	116 KB
Part6	13/11/2019 13:38	SOLIDWORKS Part...	204 KB
Part7	13/11/2019 13:38	SOLIDWORKS Part...	135 KB
Part8	13/11/2019 13:39	SOLIDWORKS Part...	381 KB
Part9	13/11/2019 13:39	SOLIDWORKS Part...	111 KB
Part10	13/11/2019 13:40	SOLIDWORKS Part...	372 KB
Part11	13/11/2019 13:46	SOLIDWORKS Part...	133 KB
Part12	13/11/2019 13:46	SOLIDWORKS Part...	206 KB
Part13	13/11/2019 13:46	SOLIDWORKS Part...	114 KB
Part14	13/11/2019 13:47	SOLIDWORKS Part...	205 KB
Part15	13/11/2019 13:47	SOLIDWORKS Part...	118 KB
Part16	13/11/2019 13:47	SOLIDWORKS Part...	359 KB
Part17	13/11/2019 13:48	SOLIDWORKS Part...	89 KB

Figure 51 - Individual parts

Figure 51 shows the individually saved parts for input for the CNC machine. As the tooling was unable to be acquired, a full non-separated block was unavailable to create, therefore, separations had to be accounted for by constructing the components together with dowels.

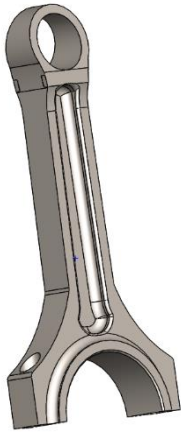


Figure 56 - Connecting Rod



Figure 55 - Connecting Rod Cap

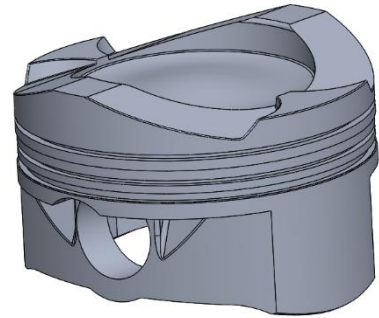


Figure 54 - Piston

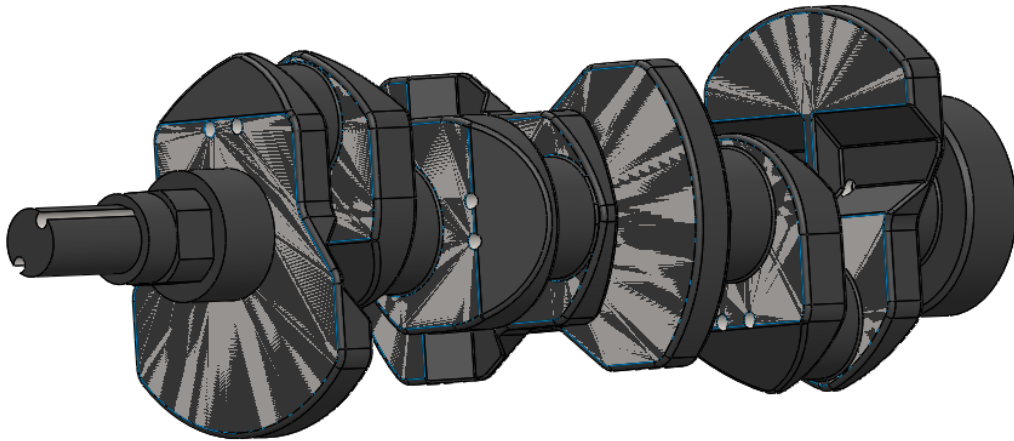


Figure 57 - Crossplane crank (unbalanced) used for model production

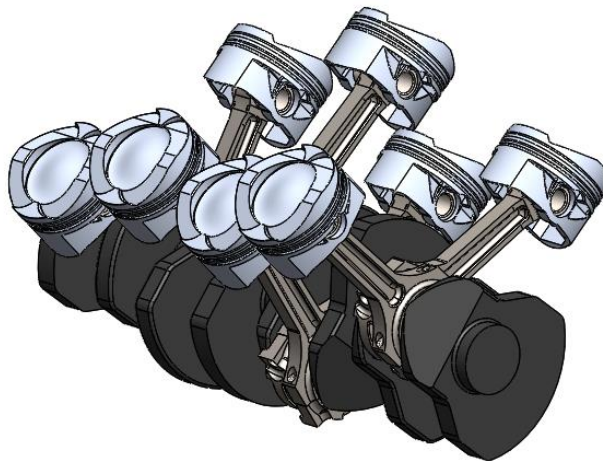


Figure 58 - V8 crank configuration and reciprocating assembly

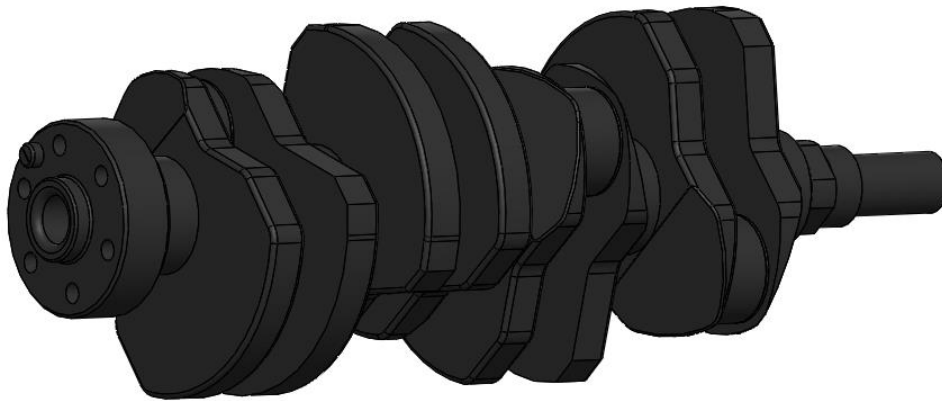


Figure 59 - Crossplane crank, balanced to centre of rotation

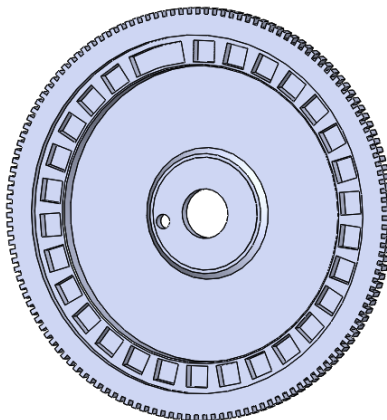


Figure 60 - Flywheel

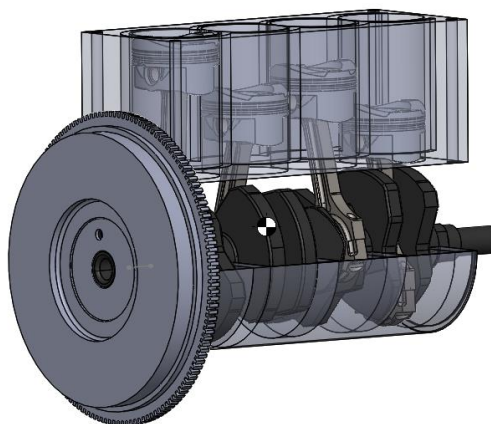


Figure 61 - Full configuration parts

Engine Strip and re-build



Figure 62 - Ford 1.5l Eco-Boost engine



Figure 63 - Various stages of engine stripping



Figure 64 - Valves and Valve springs with buckets



Figure 65 - Connecting rod and piston assembly