



PRIFYSGOL CYMRU

Y Drindod Dewi Sant

UNIVERSITY OF WALES

Trinity Saint David

University of Wales Trinity Saint David

School of Engineering

Valve Train Friction Model and Feasibility Study of the Practical Deployment of Frictional Reduction
by Way of Rocker Arm Valve Actuation

Ben Stevenson

P143645

BEng Motorcycle Engineering

21st March 2018

Major Project M6X00417

Table of Contents

Introduction	1
Literature review.....	2
Manning, J. Internal Combustion Engine Design.	2
Cam Profile.....	2
Typical rocker ratios.....	2
Manning's' example of Suitable rocker design.	3
John, A and Agarwal, V. SAE International. Evaluation and Comparative Study of Valve Train Layouts with Different Rocker Ratio.	4
W Boulton. Mechanical science second edition.	4
Simon Biggs. The Future of 3D Printing in Education.	4
Tien-Chi Huang; Chun-Yu Lin. From 3D modelling to 3D printing: Development of a differentiated spatial ability teaching model.	5
Self-discovered 3D printing techniques, CURA how to.....	6
Nozzle diameter	6
Nozzle temperature	6
Bed temperature.....	6
Bed level.....	6
Print speed	6
Infill percentage	6
Outer shell speed	6
Layer height.....	7
Ambient temperate	7
Piece orientation within the slicer software.....	7
Part One - Methodology	8
Measuring cam profile	8
Achieving valve lift from cam profile.	9
360 x 360 Matrix Method.	10
Velocity mm/°	10
Acceleration mm/°/°	11
Acceleration m/s ²	11
Contact forces	11
Component mass	11
Valve Float.....	12
Frictional force	13
Frictional torque and FMEP	13

Results and Analysis of the valve train.	14
Valve lift	15
Velocity	16
Acceleration	17
Contact Force	17
Frictional Torque and FMEP.....	18
Part two – Modification of the valve train to actuate four valves with one camshaft.	18
Priority design considerations.	20
Iterative Design Approach Using the Rapid Prototyping Model.....	21
Classic iterative design example – rocker pivot mount	21
Design iterations	23
First rocker “duck” and “horseshoe” mounting bracket set up.....	23
Second rocker in the block and horse shoe mounting arrangement.	24
Third rocker with pivot sleeve and larger cut away.....	24
Fourth rocker with roller bearing and clearance adjuster	25
Analysis	26
Valve phasing difficulties	26
Final design with optimal best-case scenario characteristics.....	28
Valve duration difficulties.....	29
Lift profile discussion.	29
Existing geometry difficulties.....	32
Using moments to calculate the contact force between the cam and the rocker.....	33
Contact force experienced on the cam with new rocker arm design.....	34
Further work.	35
Conclusion.....	36
Evaluation.	37
References	38
Appendices.....	40

Table of Figures

Figure 1. Typical Valve Timing Illustration, (Manning, 2012).	2
Figure 2. Potential Rocker Designs from Manning's ICE Design.	3
Figure 3. CIDO Educational Framework.	5
Figure 4. Orientation affects tolerances as the filament struggle to print in mid-air.....	7
Figure 5. An Example of a Failed Print, in This Case the Part Became Detached from The Build Plate Because The Bed Temperature Was Too Cool.....	7
Figure 6. Spigots to Hold the Camshaft Central.....	8
Figure 7. Measuring of the RMZ-250 Exhaust Cam Profile. The Right Dial is Measuring the Lobe While the Left is Measuring Runout.....	9
Figure 8. Cam Making Contact with the Bucket Follower.....	9
Figure 9. Illustration of Cam Plots.....	10
Figure 10. Reciprocating Components and Spring Rate Testing.....	12
Figure 11. Measured Inlet Cam Profile.	14
Figure 12. Measured exhaust cam profile.	14
Figure 13. Valve lift (s) calculated from the 360 x 360 matrix method.	15
Figure 14. Valve velocity (s'). Constant velocity can be seen at the beginning but not at the end of the profile.....	16
Figure 15. Valve acceleration (s'')	17
Figure 16. Contact Force Generated from Excel.....	17
Figure 17. Contact Force at 6500 rpm Cam Rotation.	18
Figure 18. Honda Unicam ® (Motorcross Action Magazine, 2012).....	19
Figure 19. Honda CRF450R valve train, SOHC and rocker arm assembly (Vital MX, 2017)	19
Figure 20. Engine rotation, anticlockwise.....	20
Figure 21. Iterative design process employed when rapid prototyping.....	21
Figure 22. Iterative design. Top, first print. Bottom, alterations made.....	22
Figure 23. Design constraints - available volume and set length for the rocker.	23
Figure 24. Potential rocker intended to actuate both valves. Twin rocker driven from separate lobes.	26
Figure 25. Fulcrum positions was trial error to see which produced the closest lift profile.	27
Figure 26. Consequences of altering the fulcrum.....	27
Figure 27. Balancing acceptable rocker ratio with desired valve lift. Rocker ratio of 1.17:1, marginally less than Manning suggests to acceptable.	28
Figure 28. The lift profile of the factory set up and proposed design.	29
Figure 29. Comparison of exhaust results purely to prove the accuracy of the model.	30
Figure 30. Comparison of velocities.....	31
Figure 31. Comparison of accelerations.	31
Figure 32. Roller follower "falling" down the cam flank once it crests the nose.	32
Figure 33. The final Design Must Incorporate the Spark Plug Casting.....	33
Figure 34. Rocker Forces Illustration.	33
Figure 35. Contact force of standard bucket follower and rocker arm assembly.	35
Figure 36. Contact force Excel work.	40
Figure 37. Final page on the Excel workbook generating an end figure for FMEP.....	40
Figure 38. Components used to generate Fv.....	41
Figure 39. And in turn, to generate Fc.	41
Figure 40. Stilt type jig to allow access for the DTI measuring the valve displacement.....	42

Introduction

Both established and fledgling manufactures are constantly trying to improve their products and secure a strong hold in today's buoyant and competitive motorcycle market. While celebrity endorsements no doubt helped bolster BMW's ever-growing empire, their unprecedented growth in the adventure motorcycle market hasn't been without competition. Arguably, KTM have been the success story of late when it comes to purely off road focused machines, their dominance in the Paris Dakar and World Enduro Championship has no doubt helped them sell a total of 86,769 off road units in 2016 (KTM Group, 2017).

This ongoing rivalry sees manufactures redesign and reinvent their products, constantly adapting to market demands, whether that be consumer, environmental or political influence. A notable approach has been KTM's multi operational valve train aboard their 690 DUKE, this engine utilises a direct controlled and DLC coated (diamond like carbon) rocker arm to operate the exhaust, and direct controlled DLC coated finger followers to operate the inlet (Faistaure, 2015). This configuration is key to achieving their compact valve train layout and is also thought to offer performance improvements by reducing frictional losses associated with dynamic valve train forces.

Somewhat inspired by KTM, the author has set out to create an Excel based simulation package in an effort to understand the dynamic forces occurring within the valve train of his own personal Suzuki RMZ-250 single cylinder. Practical disassembly and mathematical study will form the first half of this work.

Continuing on from the academic results, a feasibility study will be conducted in an attempt to reconfigure the valve train lay out, by replacing the intake or exhaust (timing dependant) direct acting cams with a rocker arm assembly to reduce the frictional losses previously calculated. Particular attention will be directed towards the correct timing (if achievable) and valve duration. Every effort will be made to keep the original head dimensions with a view to produce an aftermarket kit that can be installed by a confident home mechanic. However, if timing and duration requirements cannot be achieved then redesigning of the cylinder head layout may be necessary i.e. relocation of the spark plug.

Summary of aims:

- To create a frictional simulation model within Microsoft Excel.
- To conduct a feasibility study with a view to reconfigure the inlet valves to be actuated by a rocker arm design while maintaining the direct acting bucket followers operating the exhaust or inlet.
- To understand 3D printing, its benefits, limitations and application suitability.

The following material has been reviewed in an attempt to gain an understanding of fundamental principles associated with this report.

Chapter 10.2 Camshaft and Valve Train – Design Features and Construction.

Engine timing illustration of gasoline engine valve overlap shows how there is approximately 107 degrees (measured from the diagram with a protractor) between maximum opening point of each valve. As such, a commonly referred value of 110 degrees has been adopted as a guide for this paper.

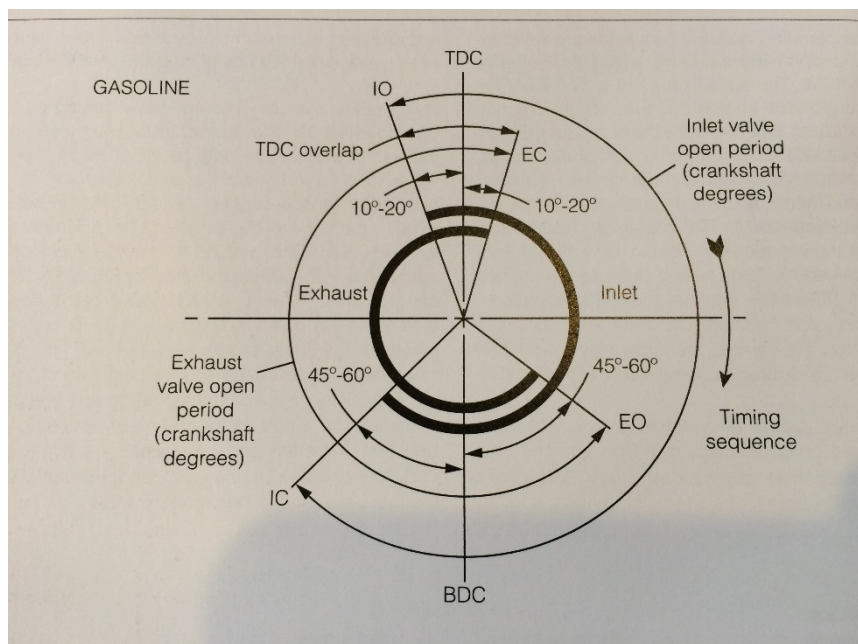


Figure 1. Typical Valve Timing Illustration, (Manning, 2012).

Cam profile is fundamental to achieving the desired breathing characteristics. As an example, cams that operate a finger follower design will require shaping to avoid dynamic instability because of the way in which they contact the follower (Manning, 2012). The cam profile to correctly actuate the roller follower will likely need broader cam flanks to achieve suitable breathing performance.

In a direct acting OHC the cam lift is equal to valve lift (negating the valve clearance), however in a rocker/finger follower design the cam lift is multiplied by the rocker/follower ratio to produce valve lift. As such a rocker/follower valve train will utilise a smaller cam (base circle and lift), often making it favourable when packaging space is at a premium. Rocker arm ratio, commonly governed by valve train stiffness and dynamic stresses, is typically between 1.2:1 and 2.0:1 (Manning, 2012). Since this exercise is looking to maintain existing components (i.e. exhaust cam shaft) the rocker ratio will need to be as small as possible to produce a similar lift characteristic.

Manning's' example of Suitable rocker design.

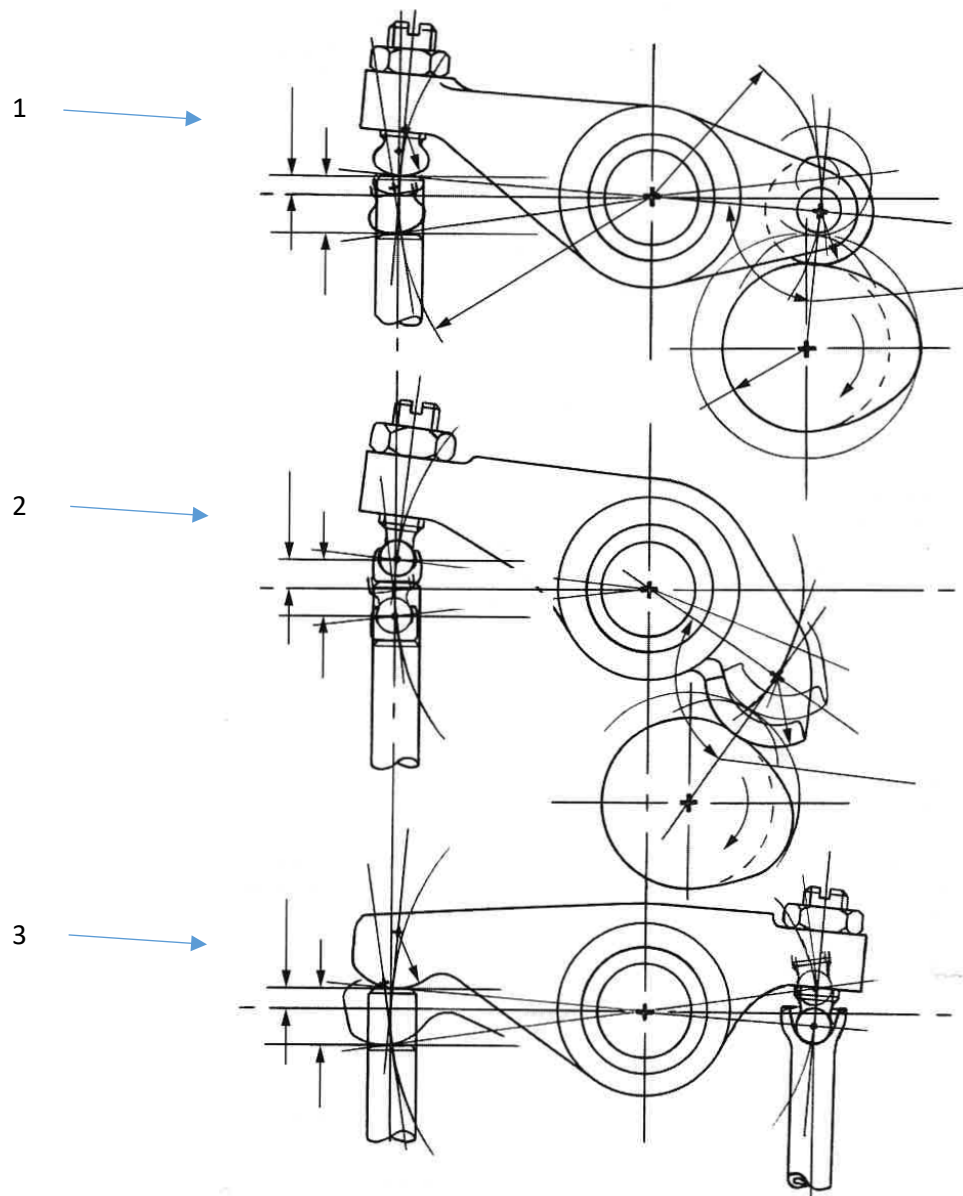


Figure 2. Potential Rocker Designs from Manning's ICE Design.

1) OHC roller follower design.

Suitable for the suggested application given the existing geometry. A roller follower employs a needle bearing to reduce frictional losses seen with sliding components.

Commonly used in modern vehicles and with proven reliability in light duty gasoline and diesel engines (Manning, 2012)

2) OHC pad adjuster design.

Although packing conditions would allow, it is unlikely this design would integrate with existing geometry. In addition, there would be no benefit when considering frictional losses given that the rocker is still operated by sliding contact with the cam lobe.

3) Camshaft in the block with pushrod mechanism with ball and cup adjuster.

This is likely to be very impractical given the existing geometry and relocation of the camshaft. Pushrod actuated valves suffer from poor stiffness characteristics as well as having a larger effective mass. As such, their use is reducing in modern engines and is unlikely to be suitable for the purpose of this exercise (Manning, 2012)

[John, A and Agarwal, V. SAE International. Evaluation and Comparative Study of Valve Train Layouts with Different Rocker Ratio.](#)

This paper describes a study that aims to increase valve lift by means of increasing the existing rocker arm ratio while remaining dynamically stable. The study found that while the increased valve lift was easily achievable, dynamic stability was compromised. The increased rocker length and mass created a larger negative acceleration and hence initiated valve float earlier in the rev range. To compensate the author has suggested a stiffer spring could be used to return the valve to its seat. However, this solution would have also increased the contact stress between the cam and follower.

In addition, the torque required to rotate the camshaft would increase and as such frictional losses would be larger (Agarwal & John P, 2014). This particular point is worth considering when approaching the rocker design phase.

[W Boulton. Mechanical science second edition.](#)

The term *plane bearing* is used to describe two faces whose bearing surfaces are plane and slide against one another. This is the case when analysing the cam lobe and bucket follower mechanism. When the load is high, or the relative speed between the two surfaces is low, it becomes difficult to maintain a thick layer of lubricant between the two surfaces. The resulting layer adhering to the surfaces, perhaps only a few molecules thick is known as boundary layer lubrication. In this case the coefficient of friction is typically between 0.05 and 0.10.

Friction only occurs when one body slides over another, if one body rolls over another then ideally there is no relative sliding at the point of contact and therefore no friction. With this being said, in reality pure rolling motion does not occur and surfaces will always deform to some extent, it is this deformation that gives rise to rolling friction.

The replacement of plane bearings with roller bearings can offer a considerable reduction in frictional resistance. The advantages of roller bearings is that they are much shorter for a given load-carrying capacity and they offer a lower effective (when in motion) coefficient of friction, usually between 0.001 and 0.004 (Bolton, 1998).

[Simon Biggs. The Future of 3D Printing in Education.](#)

Learning aids are constantly evolving, first the slide rule helped budding engineers with mathematical study then only to be superseded by the electronic scientific calculator. Years later the iPad transitioned from family entertainment to classroom helper. Nowadays 3D printing is finding its niche in the learning world and helping visual learners fulfil their potential.

The primary reason 3D printing is able to positively influence the learning process in design is the ability to learn through trial and error. Interrogating a physical object can make it easier for pupils to

spot mistakes or offer alternative design solutions, thus allowing students to gain valuable problem-solving skills in a creative, hands-on way. Without the ability to print prototypes, it would be considerably more difficult for students to identify weaknesses in their designs and improve upon them. Students are able to relate their learning in the classroom with practical applications in industry, a link that may otherwise be difficult to grasp.

“Given the rapid reduction in cost of materials and printers, and schools’ focus on active learning and addressing the skills gap, it would be logical for 3D printers to become a widely used educational tool in years to come” (Biggs, 2017).

Tien-Chi Huang; Chun-Yu Lin. From 3D modelling to 3D printing: Development of a differentiated spatial ability teaching model.

This paper discusses the merit of physically examining an object, and how rotating or changing perspective can help students increase their spatial ability. It was highlighted how differences among individual learners will affect learning outcomes, and how progression within 3D modelling relies heavily on the student’s innate spatial ability. Past studies have indicated that 3D modelling students need to view things from different angles to construct cognitive abstract spaces, thus highlighting the importance of spatial ability.

The main body tells of an experiment run over four consecutive weeks with tests conducted pre, mid and post phase and the results that followed. The participants were asked to design a product suitable for 3D printing by implementing the CDIO (Conceive, Design, Implement, Operate) framework. The report goes on show how solid models can improve learners’ spatial ability and also the ability to mentally rotate objects. The use of solid models results in 3D model learning outcomes better than those achieved using traditional three-view diagrams and suggests that 3D modelling can be of more value to vocational education (Tien-Chi & Chun-Yu, 2016).

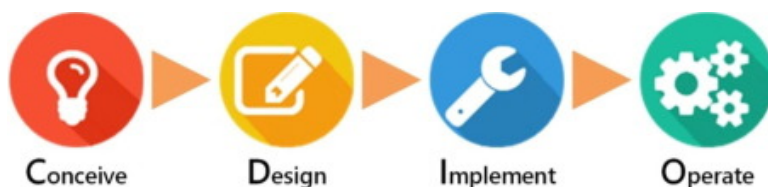


Figure 3. CDIO Educational Framework.

Self-discovered 3D printing techniques, CURA how to.

3D printing can be a frustrating affair and in reality, smooth, blemish free prints are seldom achieved. Using experience from multiple failed prints and How To articles the author suggests the following areas should be carefully considered to achieve best results: (Ultimaker, 2017)

Nozzle diameter

A common size aperture of 0.4mm is supplied by the majority of manufactures. The author has no personal experience with size variation but it is common knowledge within the 3D printing community that the smaller the nozzle diameter, the finer the print quality.

Nozzle temperature

Commonly used temperatures for PLA (polylactic acid) vary between 180°C and 220°C. This variation in temperature is needed to accommodate the various types of PLA offered by different manufactures. The author has found a temperature of 210°C produces the best results with the brand “Surreal PLA”, while 205°C is favoured for “Creality”.

Bed temperature

Bed temperature dictates how well the printed specimen adheres to the build plate. Again, experience has shown how varying filament can exhibit different adhesion qualities. Considering “Surreal PLA”, a bed temperature of 55°C is deemed best to combat the ‘waxy’ filament properties.

Bed level

Spending time to accurately level the build plate will greatly reduce the likelihood of print failure. A distance of 0.1mm is desirable across all four corners of the build plate.

Print speed

Print speed, measured in mm/s will dictate how quickly the print head travels along the X and Y axis. Naturally, there is a compromise between speed and finish quality, a suitable speed is considered to be 50 mm/s, however well-adjusted machines can reach speeds of 150mm/s.

Infill percentage

Infill (a honeycomb mesh) is responsible for the structural integrity of the final part. An infill of 100 percent will yield a solid part, however the author has found an infill of just 5 percent is suitable for demonstrative purposes and greatly reduces print time.

Outer shell speed

Printing at a slower speed will improve the appearance of the outer skin. The author has found a speed of 30mm/s produces acceptable results. It should be noted that having a large difference between inner and outer print speed can affect print quality in a negative way.

Layer height

Arguably the most important variable when considering finished print quality. Layer height of 0.2mm will produce a relatively smooth surface finish, above this and individual layers become more noticeable.

Ambient temperate

Ambient temperature will have effect how quickly the molten filament cools. Experience has shown how a cool room tends to produce a rougher surface finish. As such, professional machines are enclosed thus allowing for more accurate temperature control.

Piece orientation within the slicer software.

Print orientation can be awkward to fine tune and often results in compromised structural integrity. For a simple shape it's often clear which direction will achieve the strongest part, however as the complexity of a part increases the chances of maintain structural integrity throughout reduces. Each case should be considered individually and the part orientated suitably.

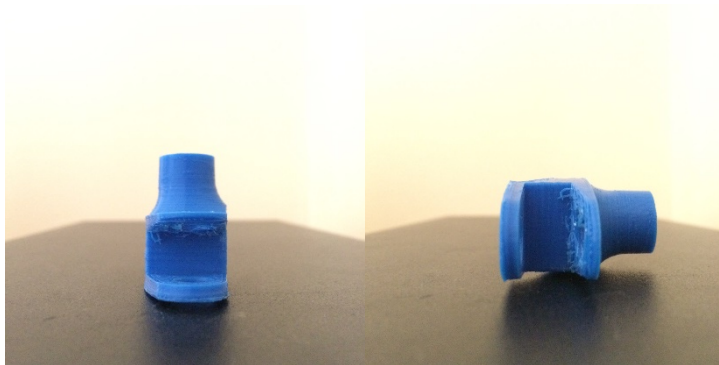


Figure 4. Orientation affects tolerances as the filament struggles to print in mid-air.



Figure 5. An Example of a Failed Print, in This Case the Part Became Detached from The Build Plate Because the Bed Temperature Was Too Cool.

The author feels compelled to consider a moral position given the recent media attention towards plastic pollution and its associated strains on our eco system. A test by California State University suggests that PLA (polylactic acid) did not show any measurable degradation while exposed to marine conditions during the 365-day study (Greene, J; California State University, 2012) In addition, a piece by Scientific American suggests that while PLA is considered the “friendly alternative” and certainly degradable under specific conditions (sunlight, moisture, controlled temperature, digestive microbes), in a real world scenario (landfill) PLA would take between 100 to 1000 years to fully degrade into its constituent parts (Royte, 2006)

Part One - Methodology

This section will detail the measuring and obtaining of the cam profile and the mathematical approaches used to generate frictional mean effect pressure from cam profile measurements.

Measuring cam profile

Spigots were machined to hold the camshaft steady in the optical dividing head. Aluminium was chosen as a stock material for its abundance and machineability.

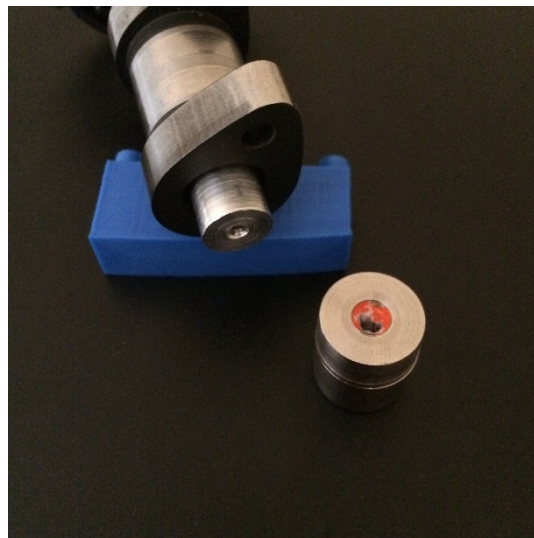


Figure 6. Spigots to Hold the Camshaft Central.

Without accurate measuring of the cam lobes the ensuing mathematical study cannot take place. The apparatus must be prepared so that all dials are easy to read without interfering with the equipment and that the camshaft rotates “in sync” with the dividing head. In this case a small rubber disc was used to grip the camshaft and eliminate any rotational slip. Over time, the author discovered the camshaft was not running true in the measuring apparatus, likely caused because of the centre relief inaccurately being machined into the spigots. A solution was to set a second gauge to measure the run out of the camshaft journal and subtract this value. The machine was rotated one degree at a time and the readings collected in Excel.

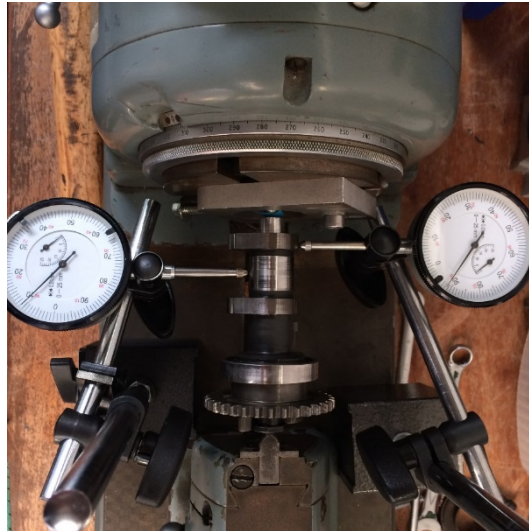


Figure 7. Measuring of the RMZ-250 Exhaust Cam Profile. The Right Dial is Measuring the Lobe While the Left is Measuring Runout.

Achieving valve lift from cam profile.

As the cam lobe rotates the point at which the cam contacts the bucket follower will change (point A). As such, the valve displacement profile will differ from the measured cam profile. The diagram below illustrates this process and shows how the lift L (Y direction) will vary with cam angle.

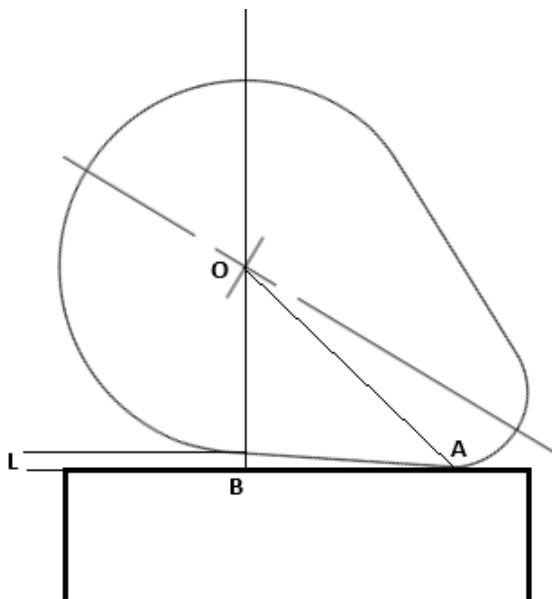
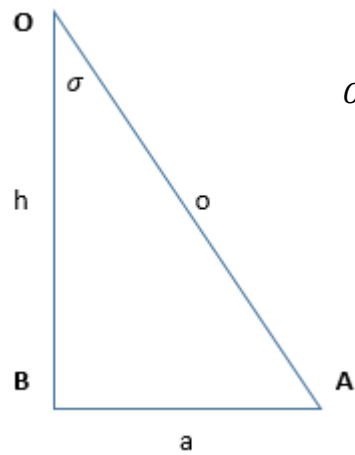


Figure 8. Cam Making Contact with the Bucket Follower.

Trigonometry can be employed to calculate OB for every angle of rotation, since OB will change as OA changes.

$$\begin{aligned}
 h &= OB & \text{Where: } S &= \text{displacement} \\
 \text{Let: } a &= OA & R &= \text{base circle radius} \\
 o &= BA & l &= \text{length}
 \end{aligned}$$



$$OB_{max} = S \cdot \cos\{180 - (\varphi + \theta)\}$$

$$OB_{max} = R + l \cdot \cos\{180 - (\varphi + \theta)\}$$

360 x 360 Matrix Method.

By allowing the cam rotation (notation theta) to run in the vertical cells and Phi in the horizontal cells, excel can execute the above formula to generate a result of lift for every instance of rotation. At every cam angle φ , θ is changed from zero to 360 to determine the maximum value of OB.

$$OB = OA \cos \alpha \text{ Where: } \alpha = 180 - (\theta + \varphi)$$

$$OB = OA \cos\{180 - (\theta + \varphi)\}$$

$$OB = S \cos\{180 - (\theta + \varphi)\}$$

The largest figure in each column is then considered, for a greater level of accuracy the author has subtracted recommended valve clearances to achieve a final lift value.

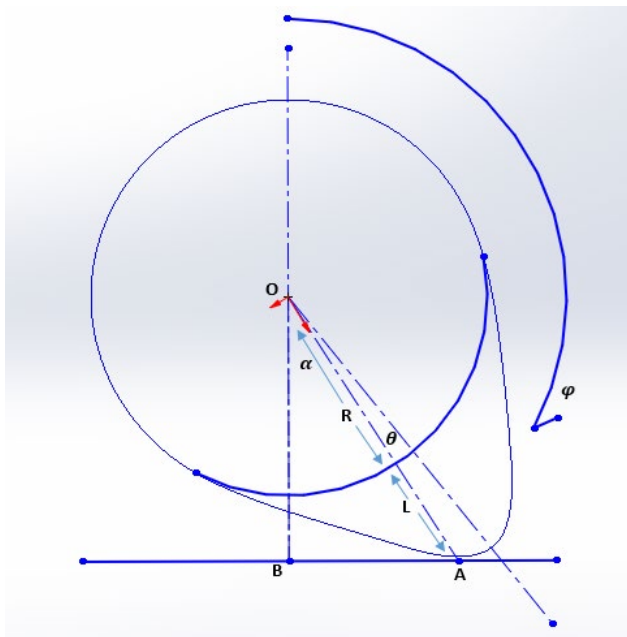


Figure 9. Illustration of Cam Plots.

Velocity mm/°

The velocity at which the valve travels through its path can be calculated by differentiating the previously calculated lift values, $v = L_2 - L_1$ (BOSCH, R, 2000)

Where: v = velocity

L_2 = lift at second interval

L_1 = lift at first interval

Acceleration mm/°/°

The rate of change of velocity, or acceleration is similarly calculated, $a = v_2 - v_1$ (BOSCH, R, 2000)

Where: a = acceleration

v_2 = velocity at second interval

v_1 = velocity at first interval

Acceleration m/s²

To convert the rate of acceleration from degrees to time the following equation can be used. Since there are 360 degrees in one revolution and 60 seconds in one minute.

$$\text{Acceleration m/s}^2 = \text{acceleration} \times \frac{6\text{rpm}^2}{1000}$$

Contact forces

As the cam rotates and makes contact with the bucket follower, its' motion is resisted by the valve spring, as such a force must be exerted to overcome the resistance and allow the cam to continue its revolution. This formula can be considered as:

$$F_c - k(l_0 + l) = m \cdot a$$

Where: k = spring rate

l_0 = installed compression

l = lift

m = reciprocating mass

a = acceleration

Rearranged to make F_c the subject:

$$F_c = m \cdot a + k(l_0 + l)$$

To first calculate contact force, it is necessary to know the reciprocating mass, acceleration of said mass and the spring rate. The individual components were weighed while both intake and exhaust springs were measured on the spring rate testing jig.

Component mass

Component	Weight (gm)
Valve	16.6
Bucket	16.7
Spring	26.9
Retainer	4.5

Collet x2	0.5
Shim	0.7
Total	65.9

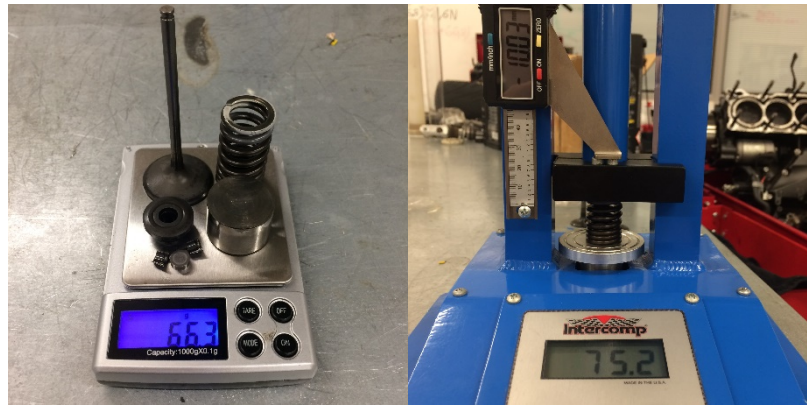


Figure 10. Reciprocating Components and Spring Rate Testing.

Reciprocating mass should be considered as:

$$\text{Valve} + \text{bucket} + \text{retainer} + \text{collets} + \text{shim} + \frac{1}{2}\text{spring} = 52.45\text{gm}$$

Acceleration in this case must be considered as the maximum negative acceleration during the 360° of cam rotation:

$$\text{Maximum negative acceleration} = -0.005 \text{ mm}/^\circ/^\circ$$

Table 1. Inlet Spring Measurements.

Inlet Spring rate K	Measured				
Displacement (mm)	Force (lbs)	Force (kg)	kg per mm	Average	Newtons
10.00	74.00	33.57	3.36	3.14	30.85
10.00	75.00	34.02	3.40		
1.00	5.90	2.68			

$$\begin{aligned}
 \text{Spring force} &= k(l_0 + l) \\
 &= 30.85(5 + 8.64) \\
 &= 420.8\text{N}
 \end{aligned}$$

Valve Float

Most commonly arising at higher rpm, valve float occurs when the inertia of the reciprocating parts overcomes the force exerted by the valve spring and the bucket follower fails to maintain contact with the cam lobe. This undesirable condition can lead to combustion gasses passing the valve in the wrong direction and increased wear on the valve seat/lip as it crashes back to rest.

Theoretically increasing RPM to understand at which point $F_c=0$.

Remembering there are 6.RPM° in a second

$$= \frac{0.005}{1000} \times (6. rpm)^2 mm/s^2$$

$$= \frac{0.005}{1000} \times 36rpm^2 mm/s^2$$

$$Rpm^2 = \frac{k(l_0 + l) \times 1000}{m \times a \times 36}$$

$$Rpm^2 = \frac{420.8 \times 1000}{0.05245 \times 0.005 \times 36} = 44571549$$

$$Rpm = \sqrt{44571549}$$

$$= 6676 Rpm(cam) \times 2$$

$$Crank Rpm = 13352$$

Frictional force

Contact force multiplied by Mu will yield a frictional force. Coefficient of friction for sliding, lubricated surfaces is considered to be 0.05-0.10 (Bolton, 1998)

Coefficient of friction for axial needle roller bearing, 0.0035-0.004 (Beardmore, 2012)

Frictional torque and FMEP

The moment of a force applied from a distance represents the torque experienced per degree.

$$\begin{aligned} \text{Frictional torque Nm} &= OB \times F_f \\ &= \frac{R + l}{1000} \times \text{frictional force} \end{aligned}$$

Work done (J) per degree of cam rotation = $OB \times F_f \times \text{angle in radians per degree}$

$$= OB \times F_f \times \frac{2\pi}{360}$$

Total amount of work done over 360 degrees is the sum of every rotation.

FMEP per lobe can be considered as $\frac{\text{Total work done}}{\text{engine displacement } (\times 10^{-6})}$ and is measured in bar.

Results and Analysis of the valve train.

To smooth data as best as possible the author has averaged each column of results in Excel. While this has reduced the overall accuracy of the figures it is arguably more suitable than trying to interpret 'messy' or fluctuating data.

Plotting inlet cam lobe profile

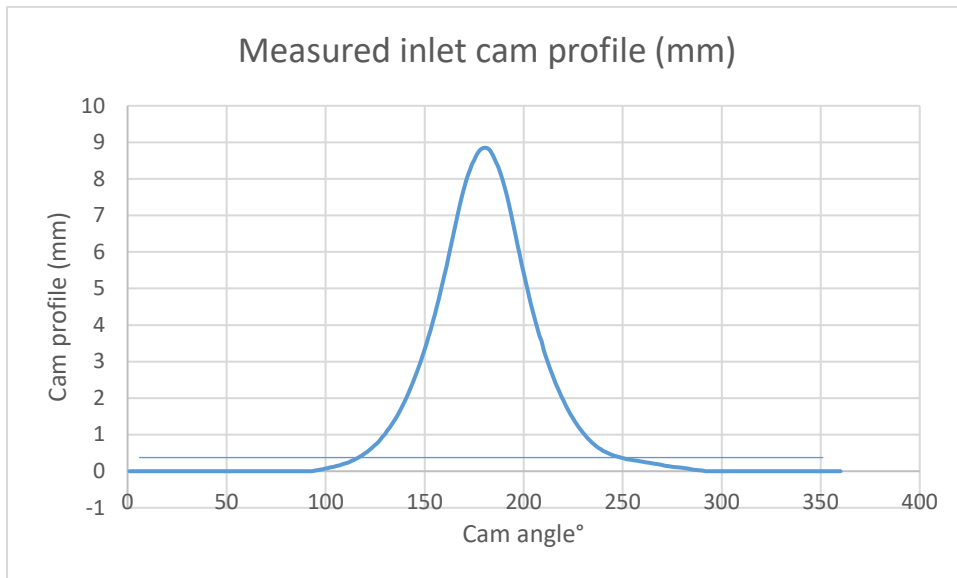


Figure 11. Measured Inlet Cam Profile.

Plotting exhaust cam lobe profile

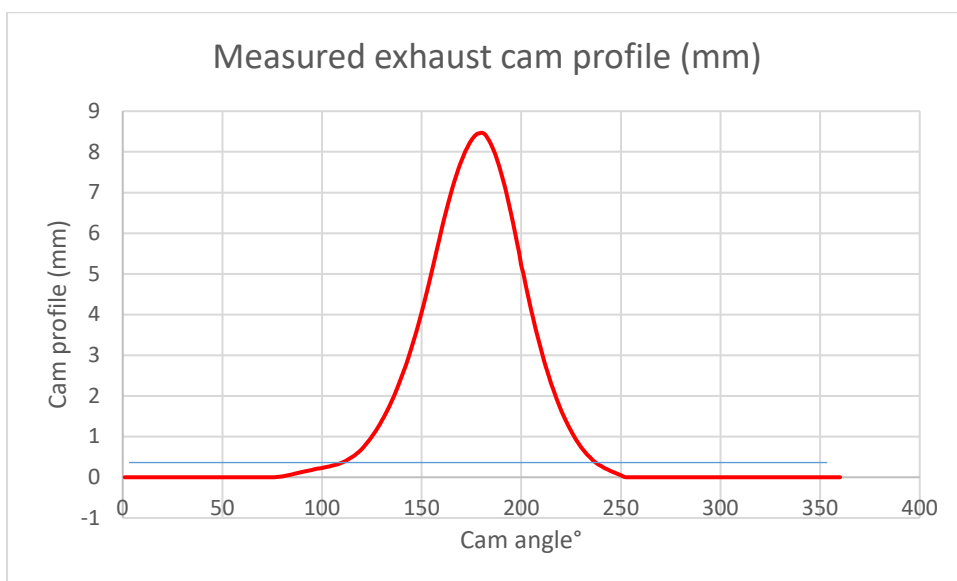


Figure 12. Measured exhaust cam profile.

Graphing the measured results allows for an easy visual interpretation of the lobe profile. It can be seen how the inlet cam will generate approximately 0.3mm additional max lift. Both profiles have a gradual opening flank to minimise excessive acceleration, this is perhaps more so noticeable in the exhaust profile.

The inlet valve spring has been measured at 30.8N/mm, this is approximately 13% stiffer than the exhaust. As such, the inlet profile appears to have a more gradual closing ramp, therefore allowing the stiffer spring to seat the valve in a gentler fashion.

Although difficult to make out from the above graphs, Excel results have shown the exhaust cam to have an additional ten degrees (approx.) of duration.

Valve lift

Maximum inlet lift: 8.64mm, with a calculated duration of: 160°

Maximum Exhaust lift: 8.47mm, with a calculated duration of: 171°

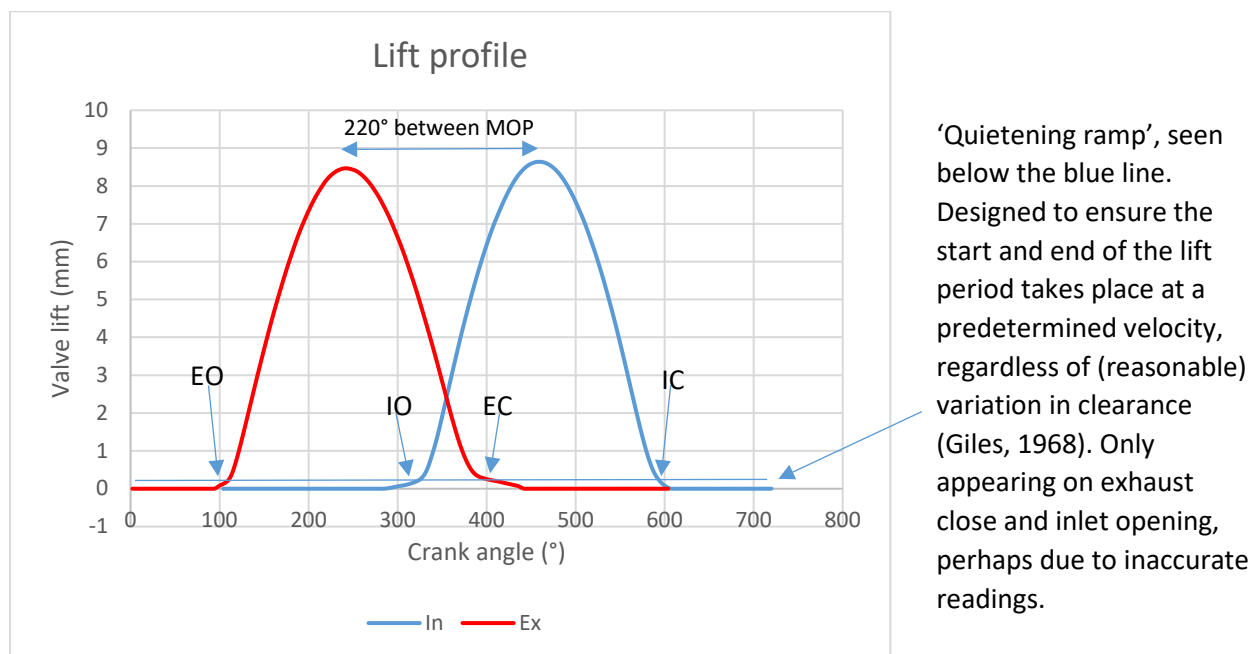


Figure 13. Valve lift (s) calculated from the 360 x 360 matrix method.

The resulting figures from the 360 matrix generate the above valve lift profile. The profiles were aligned to highlight the duration between maximum opening point (MOP) exhaust and inlet. The above profiles will be used as a base line when designing the rocker assembly. It is interesting to note how the opening and closing ramps are only partially visible in the results, ideally they would be present on each side of the curve (although not necessarily equal). The author understands that valve durations are usually taken from the top of these constant velocity periods however, because they are not present at both ends of each curve, the author has chosen to measure lift simply from start to end of the profile for continuity moving forward.

The Suzuki RMZ-250 is a competition motorcycle and as such would have timing more suited towards the discipline of racing. Consequently, the calculated duration fits closely to the recommended values shown in the table below.

Table 2. Typical petrol and diesel engine valve timings (Giles, 1968).

Type of engine	Exhaust		Inlet	
	Opens b.t.d.c°	Closes a.t.d.c°	Opens b.t.d.c°	Closes a.t.d.c°
Diesel: Low speed	40	8	8	35
High Speed	50	12	12	42
Petrol: Family cars	45	10	10	45
Higher performance	55	15	15	50
Sports	60	22	28	68
Racing	80	60	60	80

Velocity

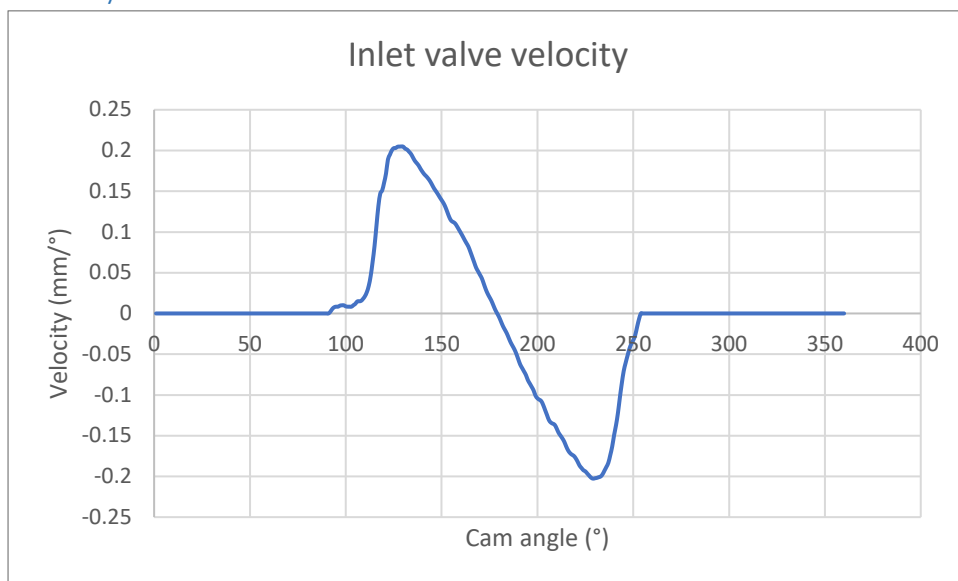
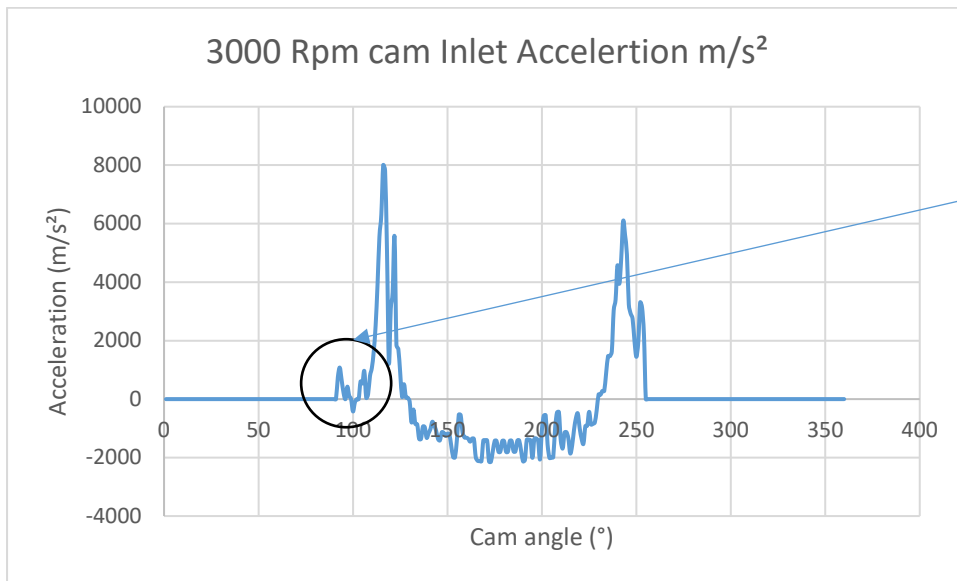


Figure 14. Valve velocity (s'). Constant velocity can be seen at the beginning but not at the end of the profile.

The 'quietening ramp' designed to open and close the valve at a constant velocity is visible in the graph shown above, although not present at closing.

Acceleration



An initial acceleration is seen but then drops away as the cam transitions through its constant velocity ramp.

Figure 15. Valve acceleration (s'')

Acceleration spikes when the lobe contacts the bucket face and begins the actuation. The valve is rapidly accelerated away from its seat, acceleration drops back down to zero before a deceleration (negative acceleration) occurs. The valve has now reached its maximum lift and is momentarily stationary before the cam continues on its rotation and the process is reversed.

The BOSCH handbook suggests typical valve acceleration rates for passenger car OHC valve assemblies should be larger than or equal to 7900m/s^2 at 6000 rpm for overhead bucket assemblies (BOSCH, R, 2000)

Contact Force

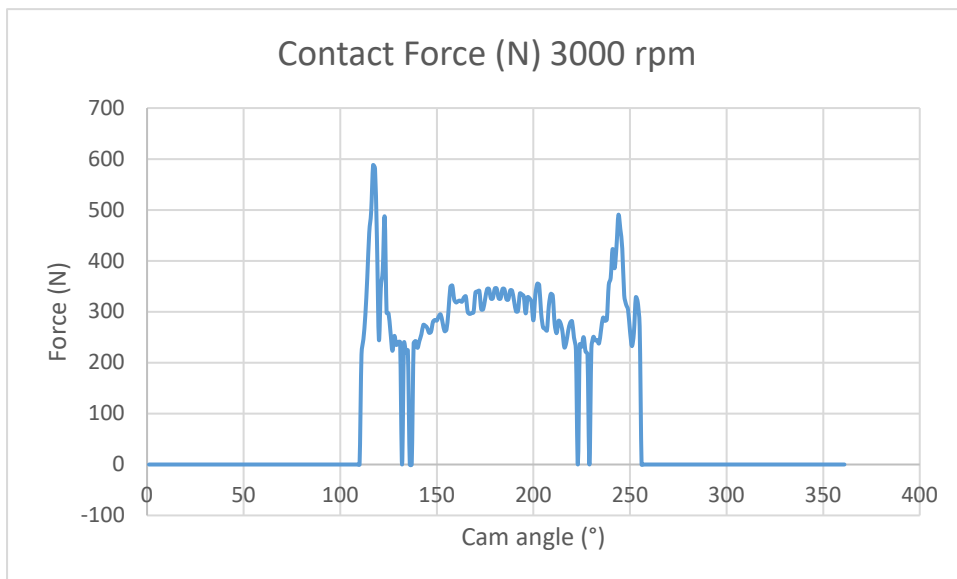


Figure 16. Contact Force Generated from Excel.

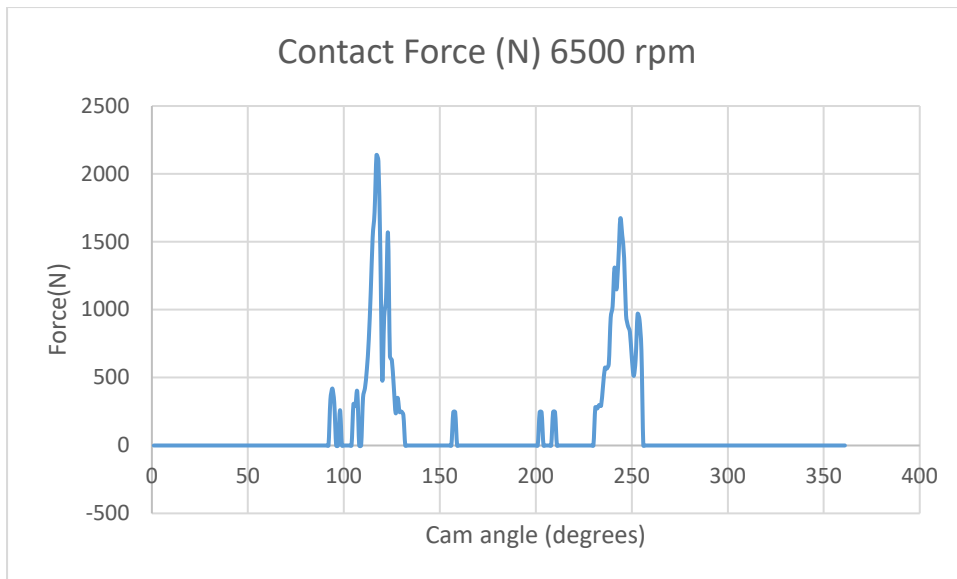


Figure 17. Contact Force at 6500 rpm Cam Rotation.

Contact force between the cam and bucket follower will always peak at the start of opening and the end of closing. This is because this is where the maximum acceleration is experienced as a result of the cam profile. Figure 16 shows how the contact force has reduced to zero at 6500 rpm, previous hand calculations suggest a similar figure of 6676 rpm at the cam.

Frictional Torque and FMEP

Lost mechanical work is regarded as “friction” work. It has the generic name “friction” but it contains all the engine losses: pumping losses, actual friction (mechanical rubbing) losses and auxiliary devices (accessory) losses (X-Engineer, 2018). The resulting FMEP figures are only considering the valve train friction which makes up approximately seventeen percent (of the mechanical rubbing) of the overall frictional losses within the engine (X-Engineer, 2018).

The total amount of work over one revolution at 3000 rpm has been calculated to be 1.337 Joules. The resulting FMEP losses at 3000 rpm to two cam lobes is 0.107 Bar.

A report by MAHLE Powertrain outlines their findings from a frictional test of a V/W Audi 2.0 TFSI engine. The report shows how the cylinder head with active valve train produces 13.7 kPa FMEP (Makay, 2009). The results from the earlier mathematical study suggest an FMEP of 10.7 kPa, which at first appears to be a similar figure. However, the V/W Audi powerplant uses DOHC and has a total of 16 cam lobes, whereas the Suzuki has only been calculated with two cam lobes. If this is to be believed then the mechanical friction calculated for the Suzuki is far greater than that of the V/W Audi.

Part two – Modification of the valve train to actuate four valves with one camshaft.

Part two of this report will focus on a feasibility study looking to reduce mechanical friction within the valve train by replacing sliding frictional components (bucket follower mechanism) with roller bearing rocker arm assembly. This will be done by way of designing a rocker arm assembly and evaluating its suitability.

Before KTM gained their current popularity, Honda were busy pushing frontiers in the off-road market themselves. 2002 heralded a change as competitors transitioned from two stroke single cylinders to four stroke machines (American Honda, 2013). At the time (and very much still employed), DOHC (dual over-head camshaft) was the common approach when an engine featured four valves per cylinder, however, since weight and handling are so desirable in the off-road sector, Honda set about redesigning conventional valve train layout. 2002 saw the birth of Hondas Unicam® design. A single over-head camshaft directly actuates the two intake valves, while the two exhaust valves operate through low-friction roller rocker arms. A roller bearing on the rocker arms reduces friction and therefore wear, which allows the cam lobes to be narrower, smaller and lighter than conventional rocker arms (American Honda, 2013). The image below shows how the cam profile operating the rocker arm has been broadened to maintain an acceptable valve profile and maximise exhaust gas flow.



Figure 18. Honda Unicam ® (Motorcross Action Magazine, 2012)

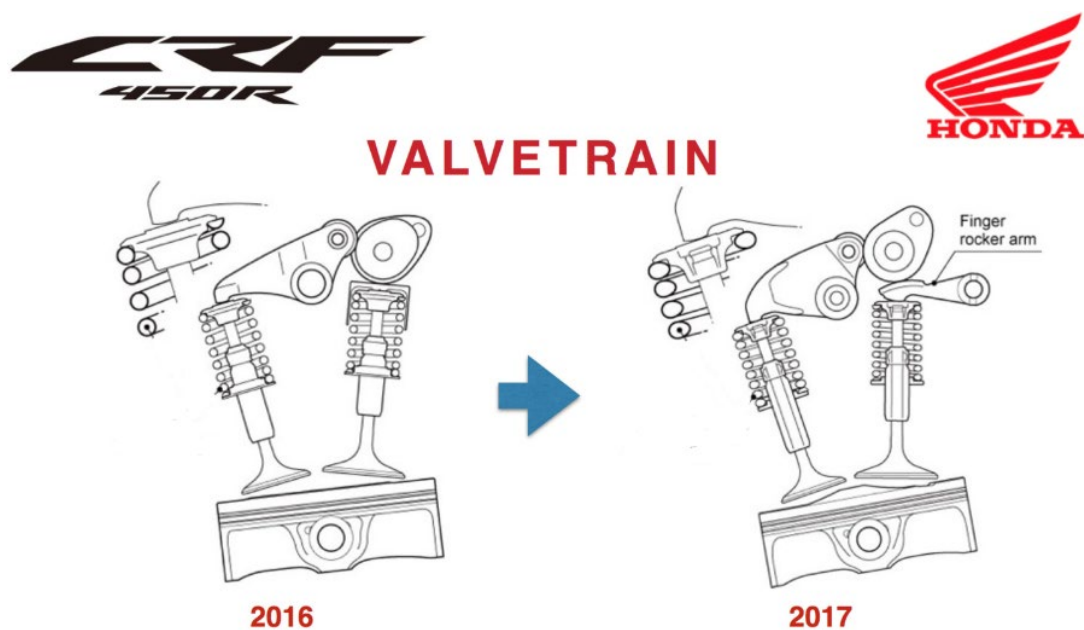


Figure 19. Honda CRF450R valve train, SOHC and rocker arm assembly (Vital MX, 2017)

Priority design considerations.

- Valve timing
- Valve duration
- Existing geometry
- Packaging constraints

Could an existing cam lobe actuate a rocker, be it either inlet or exhaust? This first design problem is dependent on the direction of engine rotation. Timing must be considered if the design is to stand any chance of operating correctly.

Firstly, it must be understood which way the engine rotates since this will influence which set of valves is most likely to work with a rocker arm design. By physically removing the engine from the bike and examining the valve train, preliminary studies showed the engine to run in an anticlockwise direction thus making the intake cam a more suitable candidate for replacement with a rocker arm. Efficient combustion dictates there should be approximately 110° of cam rotation between MOP of each valve (Manning, 2012). If the exhaust cam were to be replaced, then the best possible outcome would be for the exhaust valves to be actuated (approx. 130°) AFTER the intake.

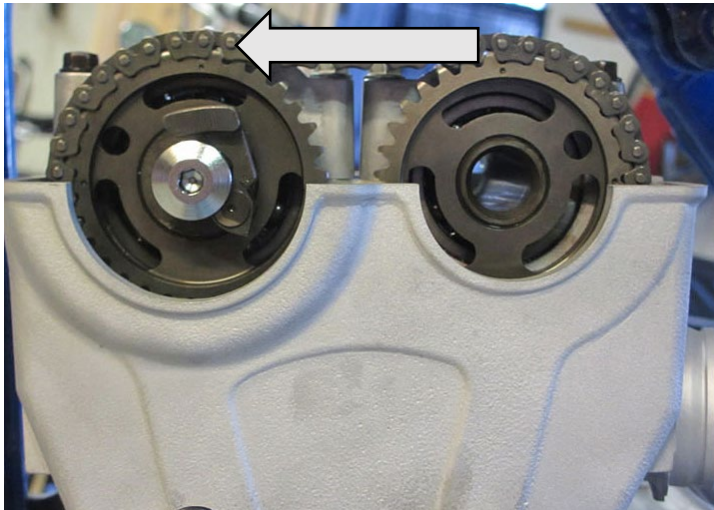


Figure 20. Engine rotation, anticlockwise.

Iterative Design Approach Using the Rapid Prototyping Model.

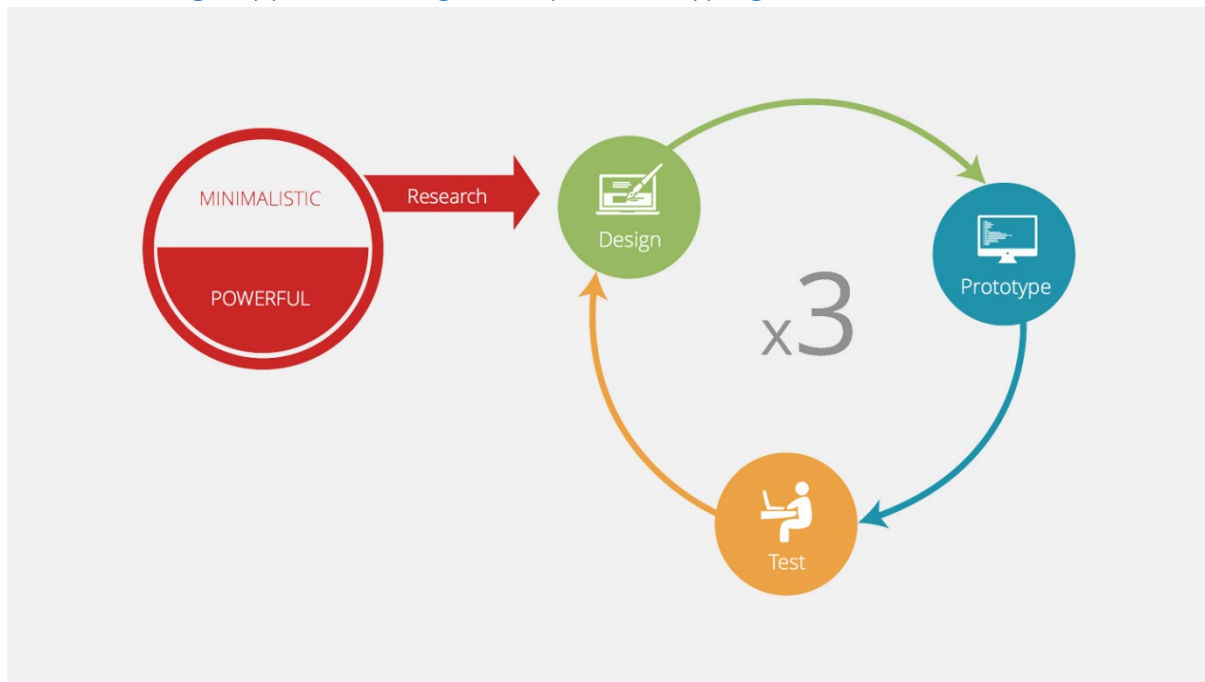


Figure 21. Iterative design process employed when rapid prototyping.

Based on the above-mentioned model the author moved into the design phase. Typically the design process had three iterations, with design being followed by prototyping and then product testing/fitment. With each iteration, the design was refined based on the findings and feedback from the previous iteration.

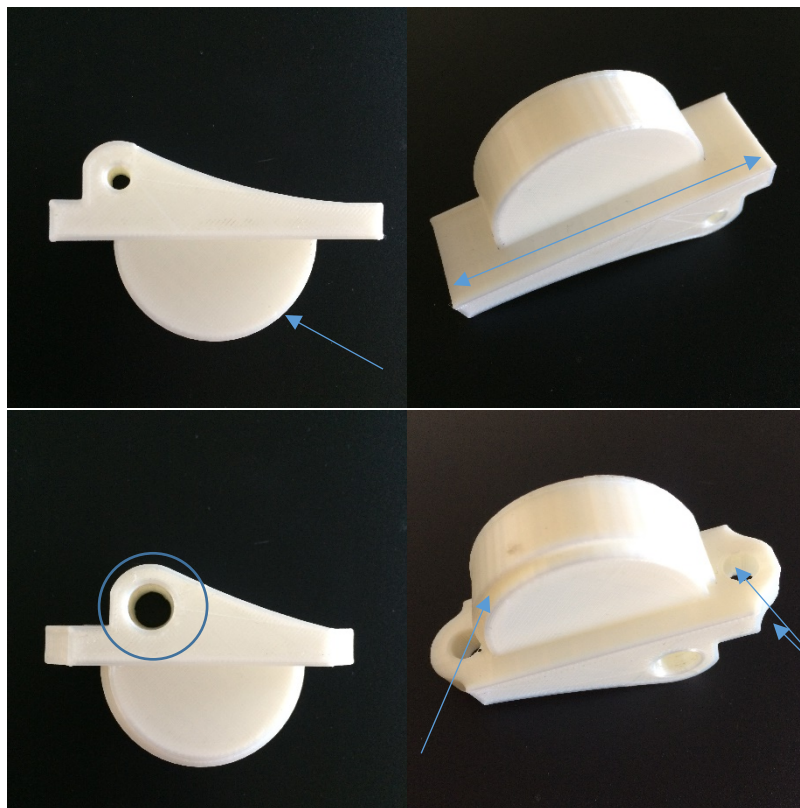
Classic iterative design example – rocker pivot mount

- Initial rocker mount pivot must be free standing/self-supportive
- Must fit within existing fixed geometry

The image below shows the iterative process of a pivot mount designed to fix the rocker in place so as to test the suitability of the rocker arm. In keeping with the design model, the part was conceptualised, prototyped and tested. Positive outcomes from printing this part were: the semicircle designed to locate into the cam recess was the correct size and fitted well. The overall length was correct, confirming measurement details.

Parameters that could be improved were: the bore for the rocker shaft was too small to be of any real use and would need to be enlarged. The position of this hole was also unsuitable and would require relocating by approximately ten millimetres. The semicircle could be altered slightly so that the whole piece sat closer to the metal casting of the head, a small recess along the outside edge would be suitable. Finally, holes for more permanent fixing could be included and the unsightly hard corners could be softened to fit with existing contours.

3D printing this small part took only thirty to forty minutes and used very little material, yet the value of having a physical piece to examine and assemble with other components was incredibly beneficial especially when the CAD drawing looked suitable.



Semicircle fits well to the aluminium casting and the overall length is correct.

Holes for fixings created, hard edges smoothed to fit existing casting profile. Recessed cut for better fitment and repositioned & enlarged pivot bore.

Figure 22. Iterative design. Top, first print. Bottom, alterations made.

Understanding set constraints that exist within the valve train was critical if the existing geometry was to remain unchanged. The governing factors when considering design possibilities were:

- Available design volume
- Set parameters, i.e. the distance the rocker arm needed to span.

The rocker arm design must take advantage of the available volume once the exhaust camshaft was removed. The rocker must be designed in a way so that it wouldn't interfere or foul on existing components. This particular obstacle was well suited to rapid prototyping with a 3D printer. Being able to physically offer up a printed part to the assembly allowed for errors to be instantly noticed, alternative solutions developed and immediately redesigned. Since the material being used was relatively soft this meant any small changes where material needed to be removed could be easily sanded or cut away with a hobby knife.

Design necessity dictated the rocker arm must contact the cam lobe during its entire rotation and its length, or span must be sufficient enough to reach the valve stem. As such it was possible to understand where the rocker pivot would be situated to deliver an acceptable rocker ratio.

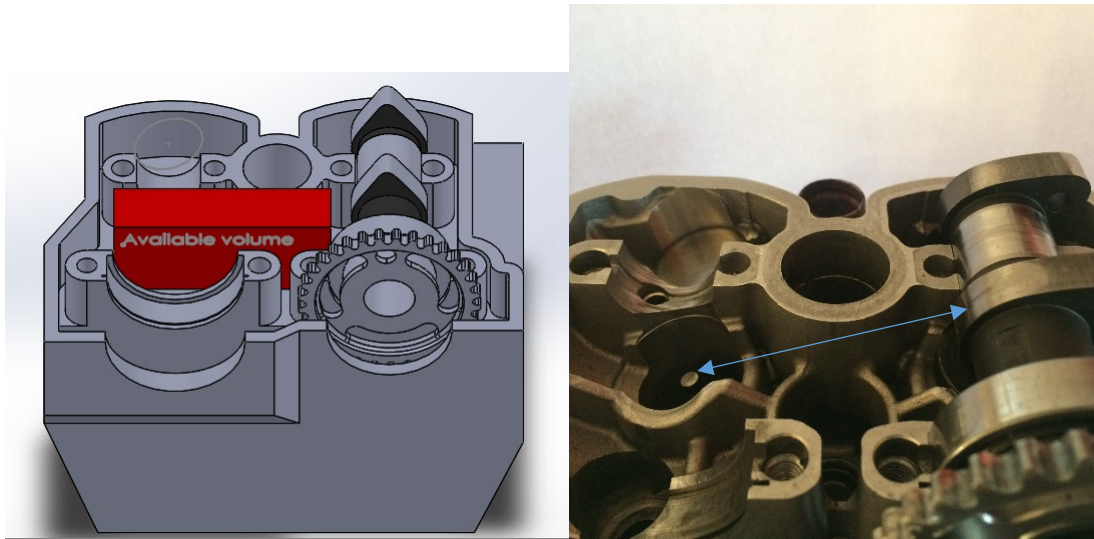
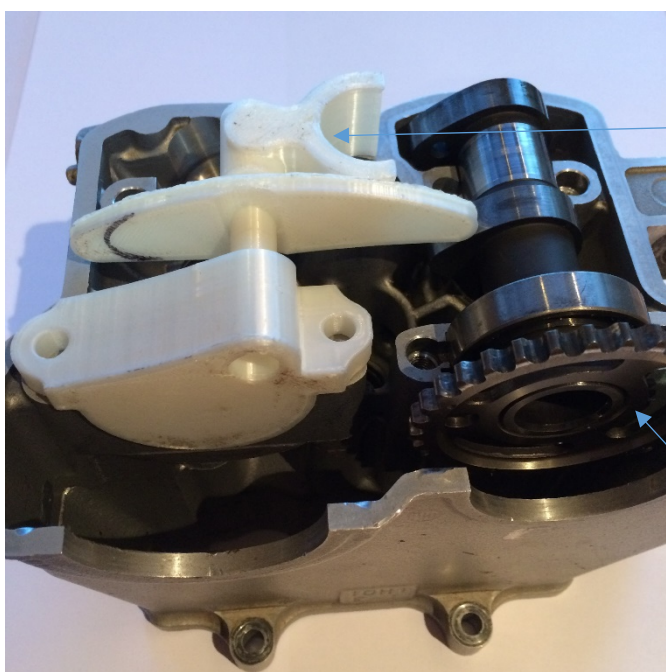


Figure 23. Design constraints - available volume and set length for the rocker.

Design iterations

First rocker “duck” and “horseshoe” mounting bracket set up.

The image below is the first complete assembly. It is comprised of two rocker mounts, a cylindrical rocker pivot and rocker arm. They have been designed to attach to the existing geometry by way of press fit locating pins. The rocker arm features a ratio of 1.7:1. This particular assembly was constructed in advance of any preliminary studies and as such is being operated by the inlet camshaft. This design combination is of little worth to the project aims since correct valve timing will never be achieved. Although arguably it does lend itself to a valuable understanding tool and is a prime example of learning through trial and error.

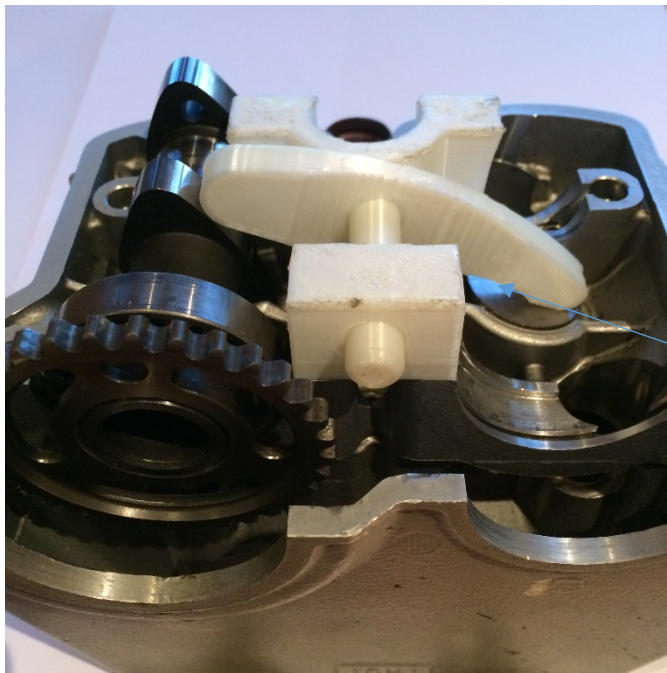


Firm mounting for the rocker shaft and still allows access for the spark plug.

Incorrect camshaft being used to drive the rocker.

Second rocker in the block and horse shoe mounting arrangement.

This assembly features a lesser rocker ratio of 1.56:1 and has been redesigned to shift the pivot mechanism closer towards the valve in an effort to make better contact with the cam lobe. The semi-circular rocker mount has been abandoned for a simpler rectangular block with two dowel pins extending into the casting for fixing. Similarly the opposite side has been reconfigured to hold the rocker pivot straight and still leave space for the spark plug. Perhaps hard to see in this image, a small cut out has been made on the right-hand side of the rocker arm to avoid fouling on the bucket guides when the valve is depressed.

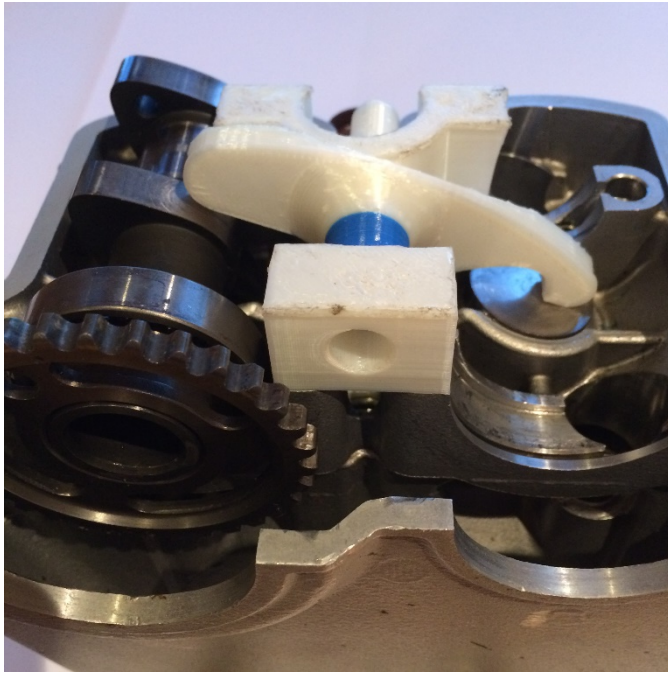


Rectangular block offers a simpler design and allows for easy modification within Solidworks, this is particularly important when repeatedly altering the design trying to chase timing targets.

The small cut away incorporated to allow for improved testing.

Third rocker with pivot sleeve and larger cut away

This assembly features a similar rocker with revisions having been made to the cut away on the valve end to eliminate contact with the existing bucket retaining walls. This rocker has also received a conical sleeve type addition to reduce unwanted movement during actuation. The mounting bracketry has been retained.

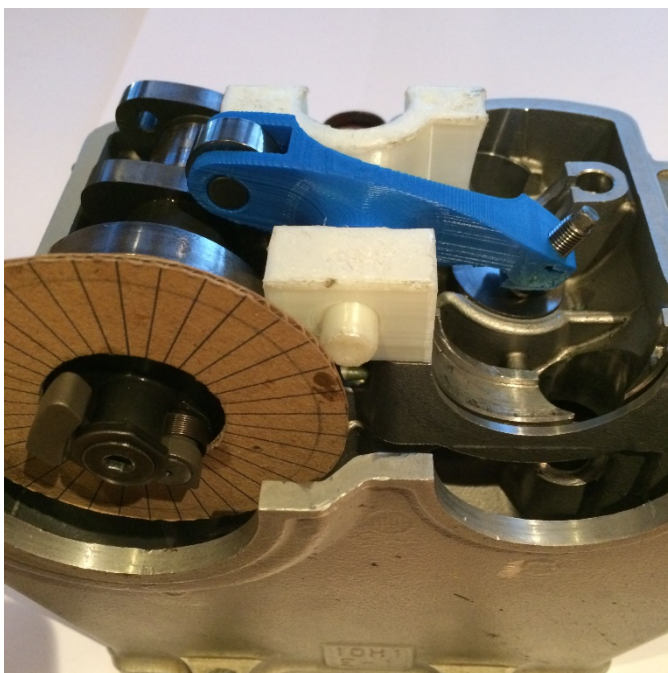


The large fillet provides greater stability when testing the rocker. The blue material can be attributed to a filament change during the print process.

The cut away has also been enlarged once again.

Fourth rocker with roller bearing and clearance adjuster

This rocker arm received revisions to the side profile to reduce contact with the spark plug casting and mounting bracket. Alterations have been made to accept a roller bearing at one end and a tappet clearance adjuster at the other. A disk marked with ten degree increments was attached to the camshaft sprocket to physically measure the rotation between valve openings.



In reality SolidWorks proved to be more useful in this area. It allowed for quicker results because there was no print time or plotting hand recorded results in excel. In addition SolidWorks also proved to be more accurate than an awkward dial gauge and inaccurate readings. (Plastic springs were designed, printed and inserted to return the valves but weren't quite strong enough to always perform correctly)



The author has been focused on getting one valve to actuate correctly before addressing the challenge of actuating both intake valves simultaneously. A forked rocker arm could be a potential solution but the torsional forces experienced when depressing the second valve would likely cause the part to fail under any kind of load. A more reasonable solution would be to have two identical rocker arms each operating off separate cam lobes. In this case it was not possible because of limited space within the valve train and location of spark plug.

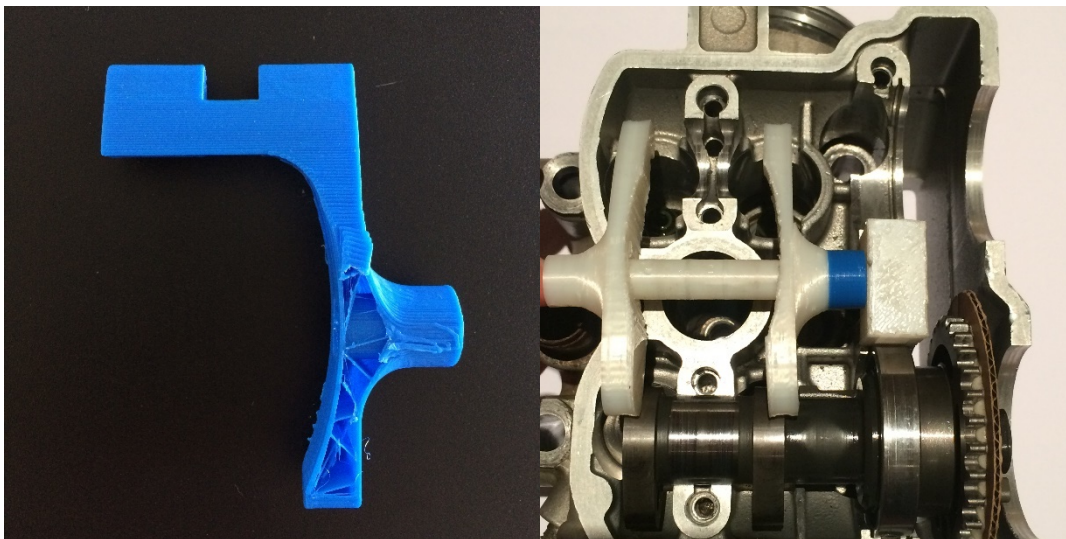


Figure 24. Potential rocker intended to actuate both valves. Twin rocker driven from separate lobes.

Analysis

Valve phasing difficulties

Understanding the valve duration of each design iteration has proven to be a complicated affair. On the one hand physical parts are pleasantly tactile and have allowed the author to engage with the

project in ways a CAD approach couldn't. On the other hand, making multiple parts, each with minor geometry changes and then trying to achieve accurate readings with hand equipment simply wasn't practical.

Phasing considerations were tackled by moving the fulcrum left or right in two millimetre increments from a chosen centre, thus changing the point at which the cam and rocker made contact. Using SolidWorks Motion Study each scenario was analysed and the results exported to Excel.

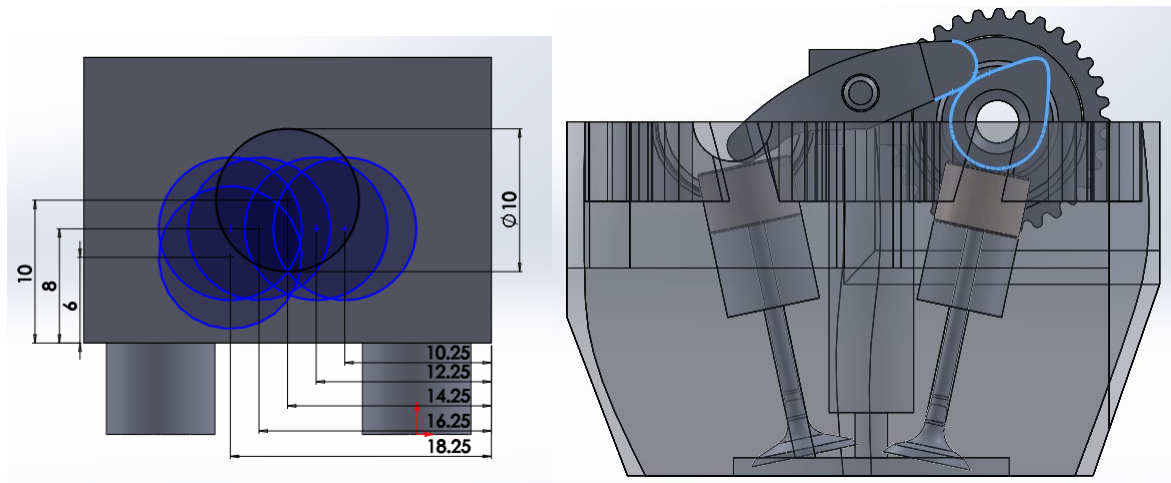


Figure 25. Fulcrum positions was trial error to see which produced the closest lift profile.

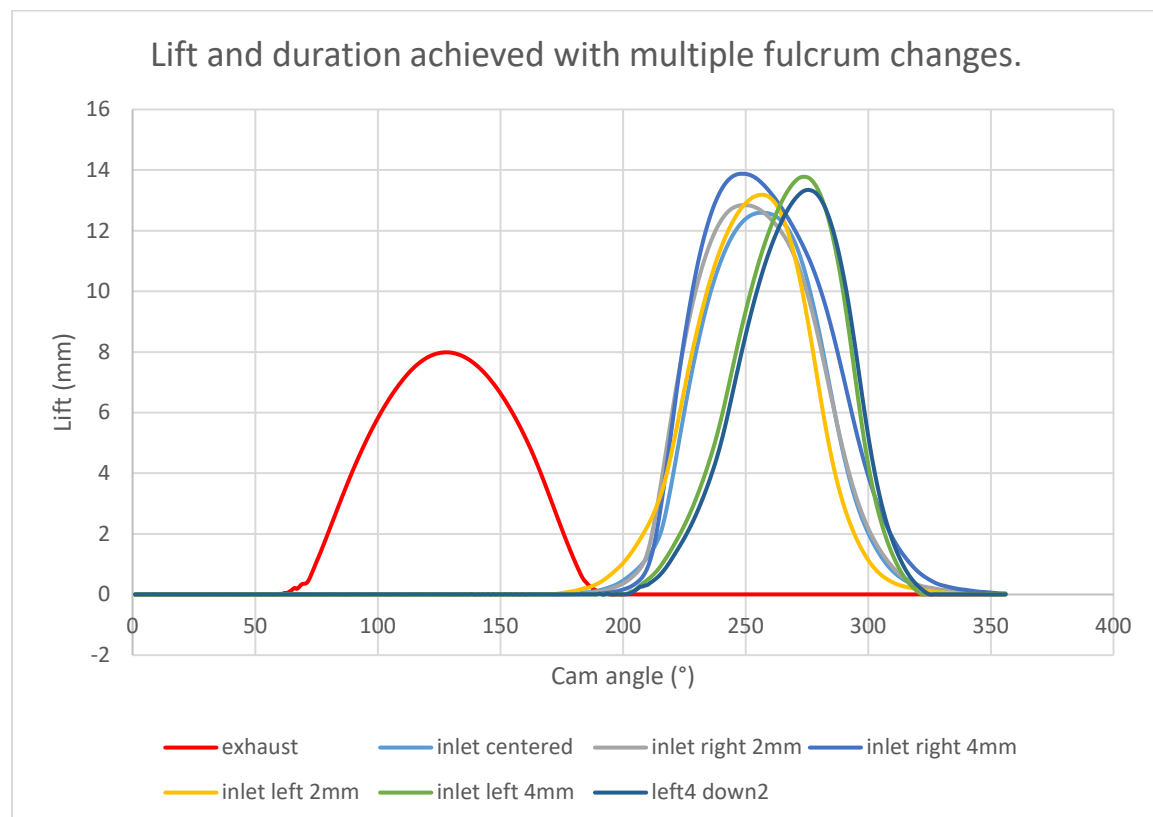


Figure 26. Consequences of altering the fulcrum.

Results show how altering the fulcrum influences the duration between full lift of exhaust and inlet. With MOP exhaust occurring repeatedly at 130 degrees and the closest MOP inlet occurring at 247 degrees thus achieving the best-case scenario of approximately 117 degrees between valves. Valve overlap has been all but removed with the rocker arm assembly, regardless of where the fulcrum was positioned. Although the inlet MOP is reasonably close to being acceptable, the opening and closing of the valve is too abrupt. The opening flank of the cam is almost being bypassed completely as the nose quickly becomes responsible for most of the lift profile. Valve overlap is desirable because with a suitable length exhaust manifold it is possible to scavenge the clearance volume from the cylinder (Giles, 1968). The gas velocity of the exhaust is high enough to initiate the drawing in of the inlet charge and convert as much as possible of the incoming air's kinetic energy into pressure before the piston begins to move down on its suction stroke. Without this overlap the proposed design would suffer because of the delayed introduction of the intake charge into the cylinder.

Final design with optimal best-case scenario characteristics.

With fulcrum position, rocker ratio and existing geometry all carefully considered the author has arrived at the final iteration delivering the best valve profile characteristics with the existing cam shaft.

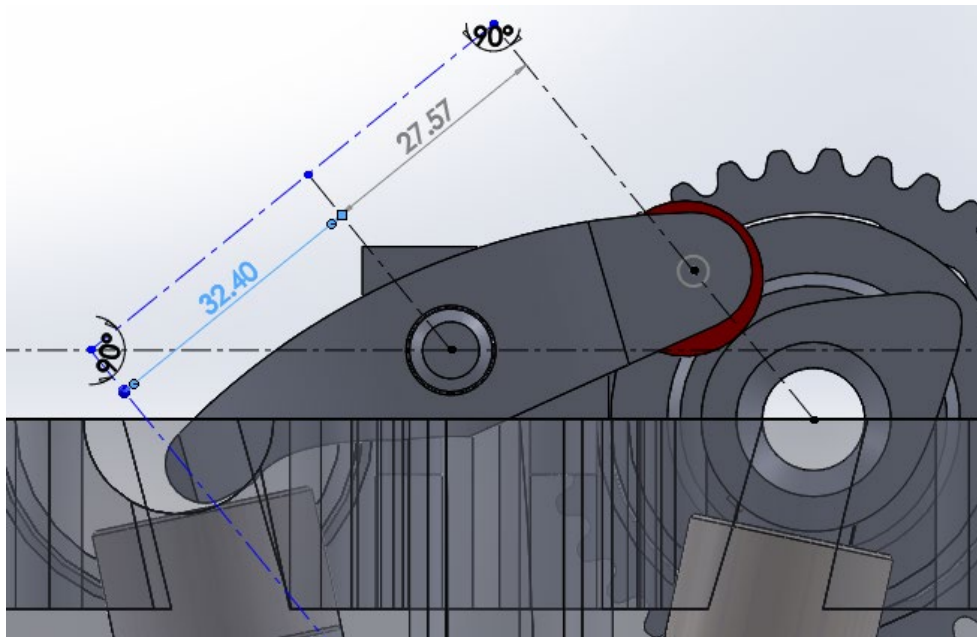


Figure 27. Balancing acceptable rocker ratio with desired valve lift. Rocker ratio of 1.17:1, marginally less than Manning suggests to acceptable.

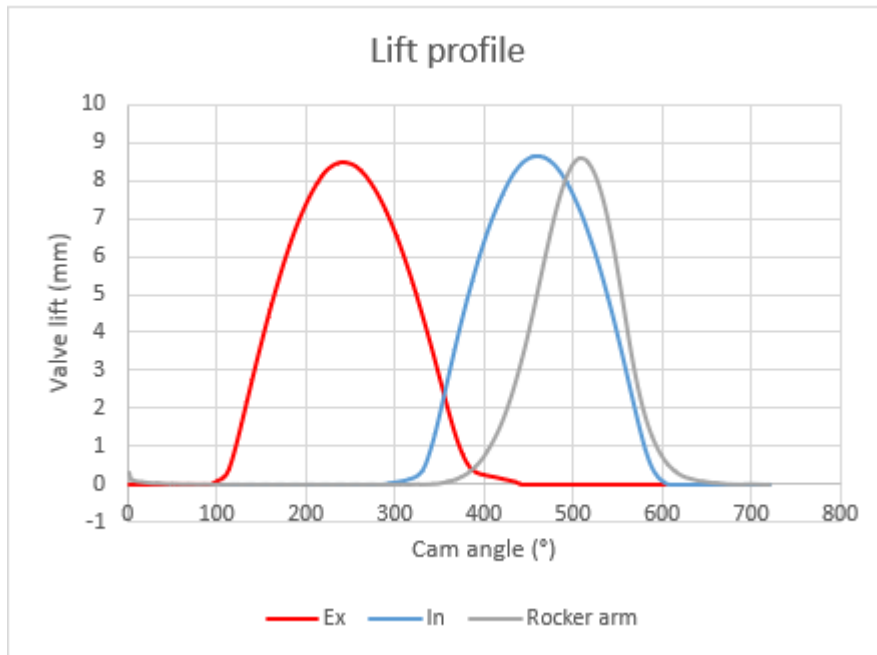


Figure 28. The lift profile of the factory set up and proposed design.

Valve duration difficulties.

The inlet duration of the proposed rocker design is very similar to that of the original bucket follower mechanism. Work carried out suggests a discrepancy of only six degrees additional duration, a margin that could be classed as negligible (Manning, 2012). The total duration is so similar because rocker arm follower is contacting the cam lobe correctly as it rotates through the start and end of its profile. With the addition of the roller follower (modelled above in red), the rocker arm now experiences the entire sweep of the cam lobe, as opposed to the cam “skipping” across the arm.

Lift profile discussion.

Despite the duration of the inlet valve profile being acceptable, the rate of change of the profile is sluggish and the gradual opening and losing ramps suggest a slower acceleration along the opening flank. The area under the curve is significantly less than the bucket follower design and as a result will hamper breathing performance. The valve is reaching full lift but doing so in a much slower manner. It is moving away from its seat but not accelerating away as rapidly and therefore its “productive range” is limiting the ingress of combustion charge.

As a tool for comparison and validity the author has graphed the hand calculations for the exhaust against the accelerations suggested by SolidWorks for the exhaust. At 3000 revolutions per minute there is a discrepancy of approximately ten percent. The author has deemed this acceptable and as such believes the results obtained for inlet to be of a similar accuracy.

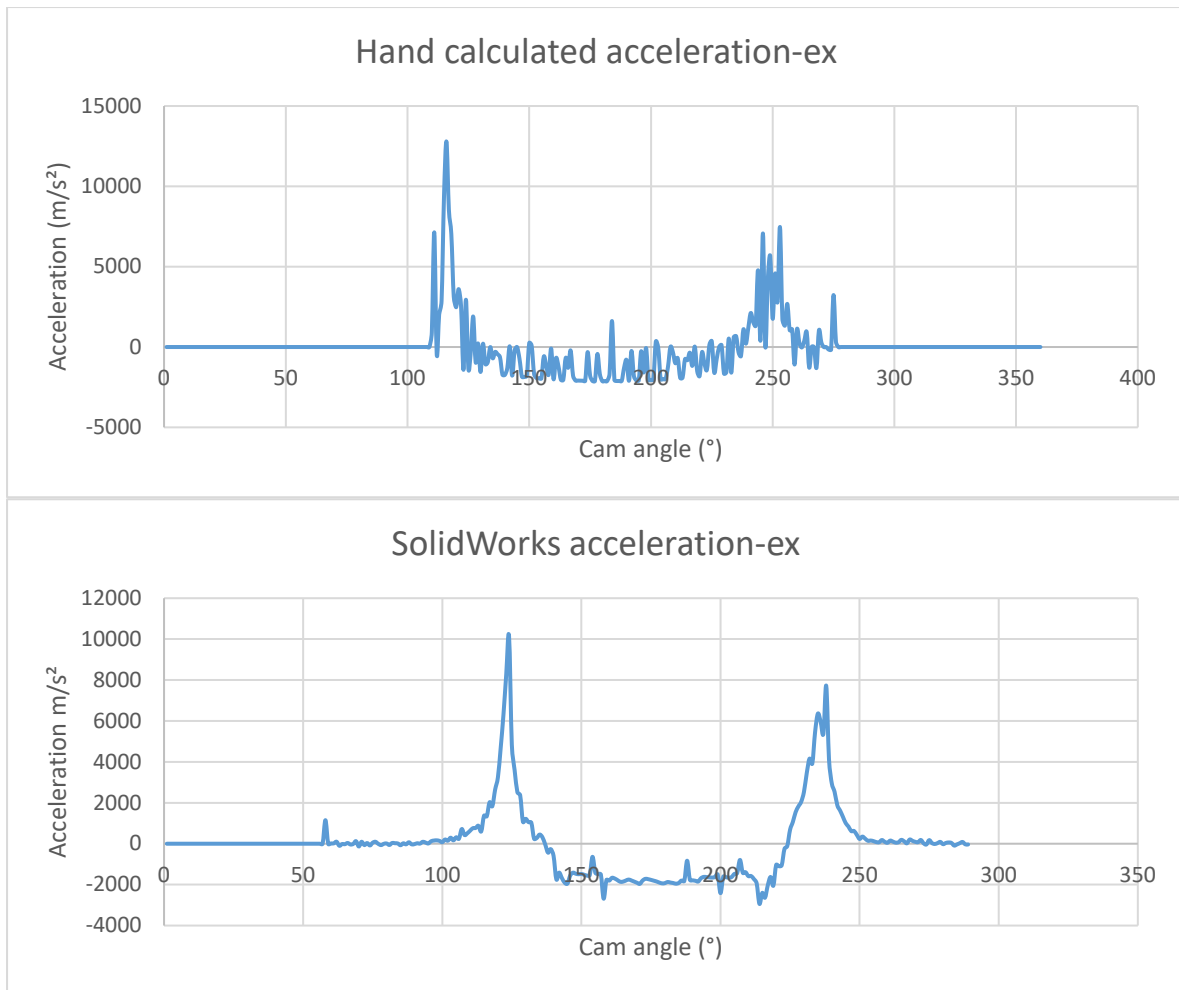


Figure 29. Comparison of exhaust results purely to prove the accuracy of the model.

The maximum acceleration the inlet valve experiences in standard form at 3000 rpm is approximately 8000 m/s² (Fig 12). The results from the motion simulation suggest the maximum acceleration for achieved by the rocker arm actuated valve is approximately 6000 m/s².

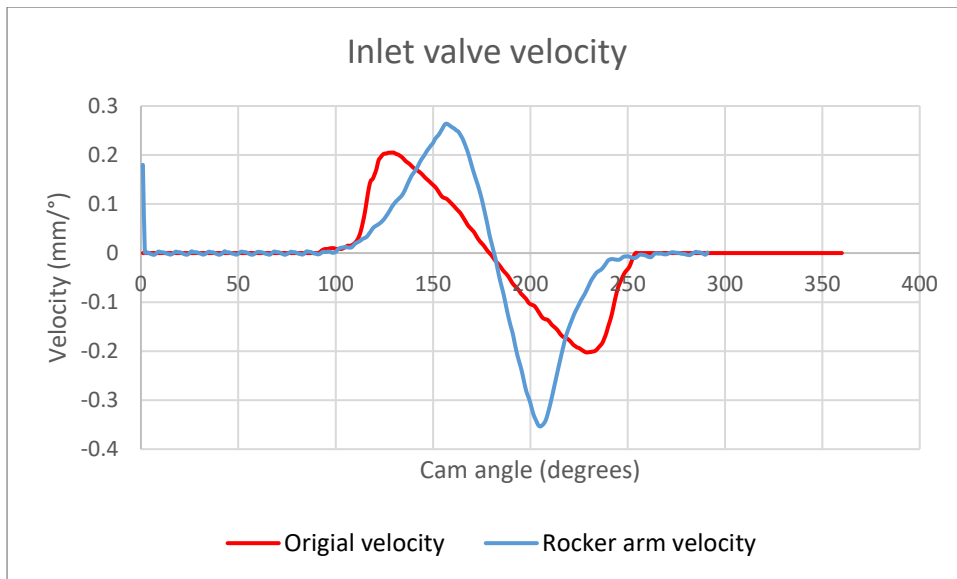


Figure 30. Comparison of velocities.

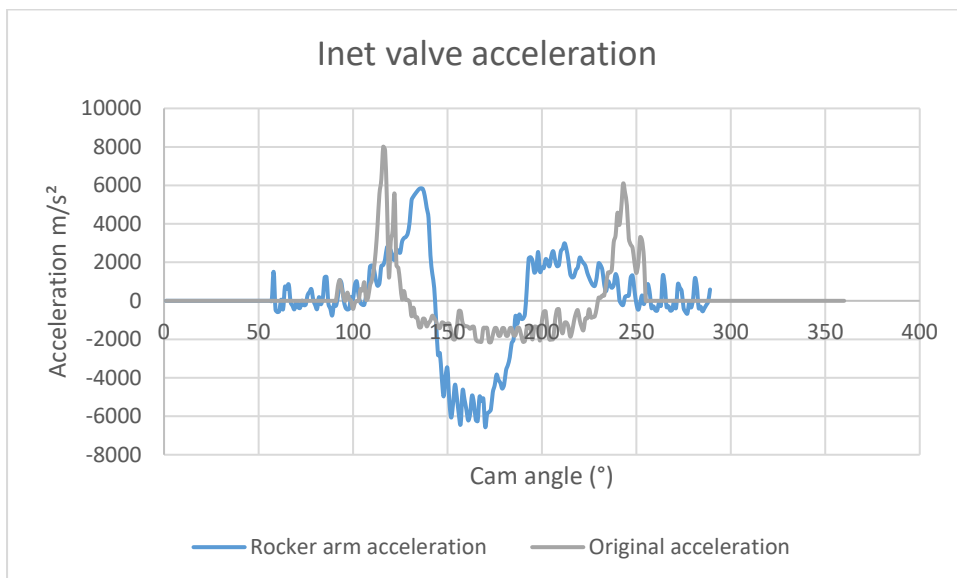


Figure 31. Comparison of accelerations.

It can be observed It can be seen how the maximum positive velocity of the rocker arm occurs approx. fifty degrees later in the rotation and the maximum negative velocity occurring 25 degrees earlier. As a result the rocker arm is suffering with a substantially larger negative acceleration. This is particularly undesirable because it will advance the likelihood of valve float earlier in the rpm range. The author believes this can be attributed to the way in which the roller follower and cam are off set. Once the cam passes through full lift, because of the orientation of the follower from cam centre, velocity and therefore acceleration increase rapidly as the follower crests over the cam nose and runs down the flank. If the roller follower were positioned at 12 o'clock to the cam then this would not occur and values would be equal both sides of the cam. However, it is important to remember this location was chosen to optimise exhaust/inlet timing.

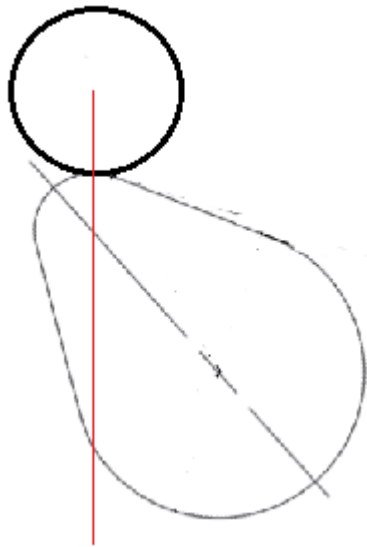


Figure 32. Roller follower "falling" down the cam flank once it crests the nose.

Existing geometry difficulties.

Attempting to package the proposed design into the existing cast cylinder head possesses several difficulties that would hinder the practical deployment (if the valve characteristics were appropriate). Most notably is the awkwardly positioned spark plug casting. Even combustion and propagation of the flame front during combustion is best achieved when the spark is centralised (Giles, 1968) and is likely why Suzuki have favoured this position. The current position of the casting limits the options available for mounting the rocker shaft and also forces the removal of material on the rocker arm (*fig 27*). The rocker arm has been designed to fit around the spark plug casting but will have compromised integrity in doing so.

Cast guides for the bucket follower have also influenced the design of the rocker arm. The final design still needs material removed to avoid contacting these castings. Again, material that would usually be present. The bearing surface for the inlet camshaft has also been awkward, now the inlet cam is removed this additional material is unnecessary and causes difficulties in allowing both inlet valves to work from one rocker.

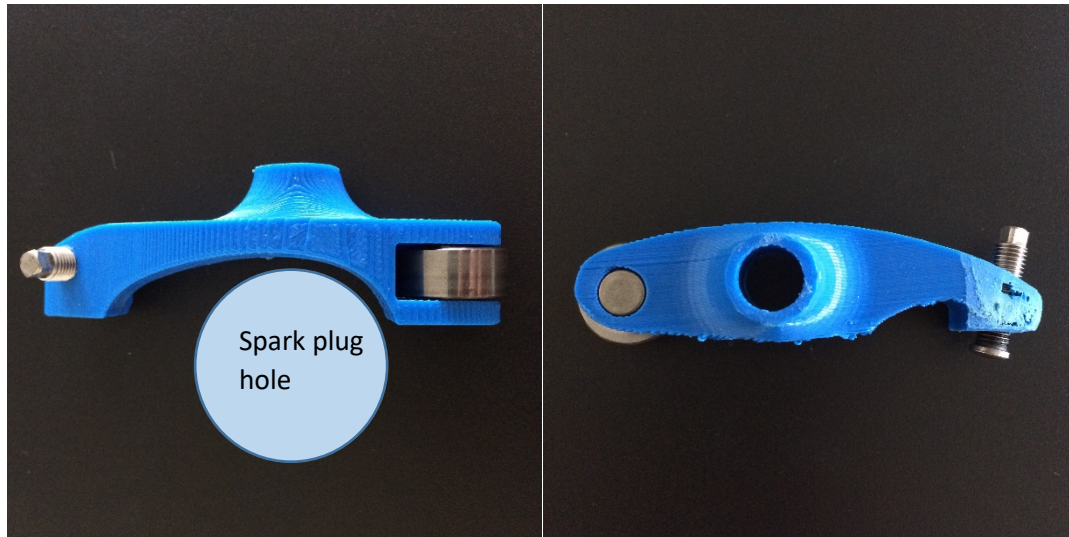


Figure 33. The final Design Must Incorporate the Spark Plug Casting.

Using moments to calculate the contact force between the cam and the rocker.

The diagram below demonstrates the forces acting on the rocker caused by the contact between the rocker and cam, and rocker and spring.

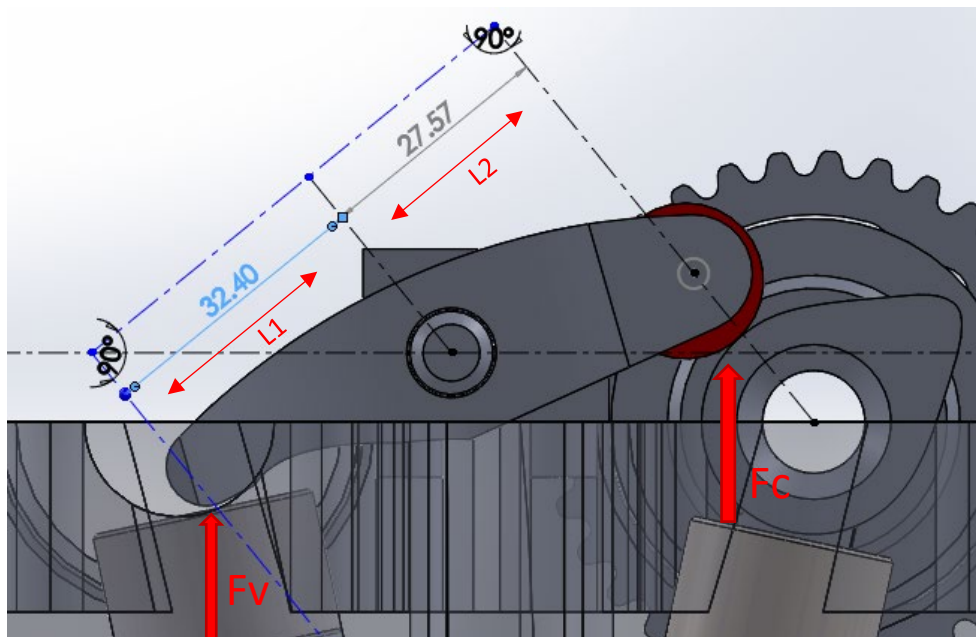


Figure 34. Rocker Forces Illustration.

F_v = force required to open the valve

F_c = contact force between the cam and rocker

The resistive force from the valve spring is applying an upward force onto the rocker, which in turn applies an opposite force in the vertical direction to the cam lobe. The following equation can be used to represent F_v :

$$F_v = m \frac{d^2y}{dt^2} + k_1(y + y_0)$$

Where: $m = \text{recipricating components}$

$$\frac{d^2y}{dt^2} = \text{acceleration}$$

$k = \text{spring rate}$

$y = \text{lift}$

$y_0 = \text{amount of compression on the spring}$

The above represent s the rocker in a horizontal position, however as the valve opens the angle of the rocker will change. The angle the rocker can represented by:

$$\begin{aligned}\varphi &= \frac{y}{l_1} \\ \tan &= \frac{\text{opp}}{\text{adj}} \\ \therefore \tan \varphi &= \frac{y}{l_1} \text{ (radians)}\end{aligned}$$

Equation of motion for rocker

$$\text{Torque} = F_c \cdot L_2 - F_v \cdot L_1 = I \cdot \frac{d^2\varphi}{dt^2}$$

The above equation states that the torque experienced by the rocker arm is equal to the moment of the contact force minus the moment of the valve force, which is also equal to the moment of inertia ($I=mr^2$) multiplied by acceleration of the rocker arm.

The linear acceleration of the valve has a relationship with the angular acceleration of the rocker and can be used as a substitute. If the rocker arm acceleration is unknown, the following relationship can be used:

$$\frac{d^2\varphi}{dt^2} = \frac{1}{l_1} \cdot \frac{d^2y}{dt^2}$$

Since SolidWorks provides a lift profile, valve acceleration can be derived.

The acceleration of the valve is known and can therefore be used to help calculate the contact force using:

$$\begin{aligned}F_c \cdot L_2 - F_v \cdot L_1 &= \frac{I}{L_1} \cdot \frac{d^2y}{dt^2} \\ \text{Rearranged to make } F_c \cdot L_2 &= F_v \cdot L_1 + \frac{I}{L_1} \cdot \frac{d^2y}{dt^2} \\ \text{Making } F_c \text{ the subject } F_c &= \frac{F_v \cdot L_1}{L_2} + \frac{I}{L_1 \cdot L_2} \cdot \frac{d^2y}{dt^2}\end{aligned}$$

Contact force experienced on the cam with new rocker arm design.

Resulting Excel work has lead to the following figures:

Contact force at the cam at 3000 rpm: Peak contact force of approx. 9000N.

Overall FMEP has been reduced to: 0.0046 bar.

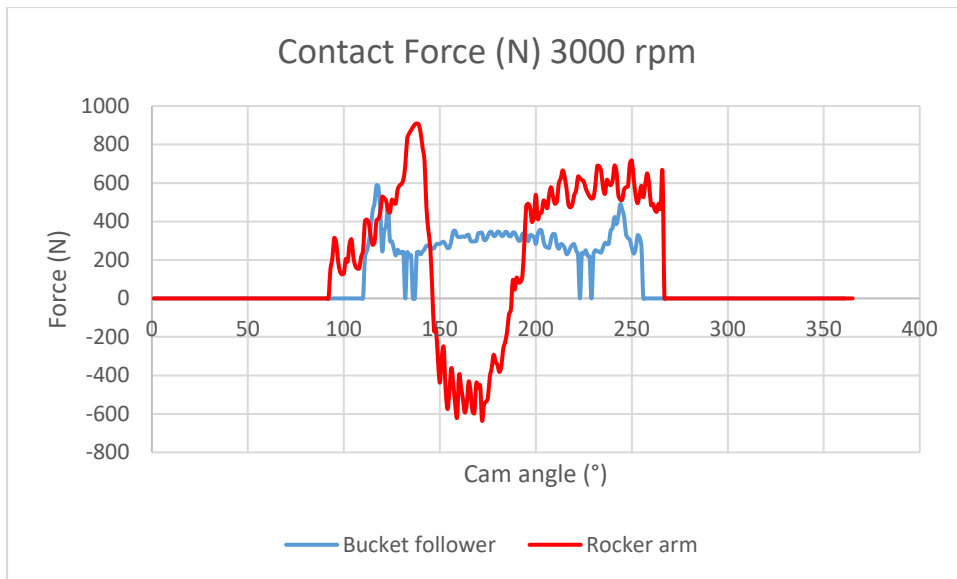


Figure 35. Contact force of standard bucket follower and rocker arm assembly.

Rocker arm; With a larger acceleration and increased mass from the rocker, the cam is now experiencing a larger contact force. This was expected and is understandable.

Since the frictional force (a governing factor in FMEP calculations) is a direct product of the contact force multiplied by the coefficient of friction, it stands to reason that a roller bearing would still provide less frictional resistance even with the added load generated by the additional reciprocating mass and acceleration.

Further work.

Areas of further work the author feels necessary to undertake are:

- FEA of the rocker arm
- FEA of the rocker pivot
- Design an alternative cam shaft with an intake lobe incorporated in the correct position

The most likely and perhaps most warranted circumstance for the application of FEA would be when considering the rocker arm itself. The rocker arm will experience significant accelerations likely causing the part to flex undesirably, as such any design would need to undergo additional testing if the part were to ever make it to market. A static evaluation would suit this purpose where the user can study stresses, displacements, strains and factor of safety for components with linear material properties.

The author suggests the most likely cause for failure (in this case as the rocker attempts to depress both valves simultaneously) is a bending of the part. It is thought that because of the increased length of the part (a necessity to reach from exhaust to inlet) and the forces exerted by the valve spring the part could likely twist and permanently fail. This would be particularly suitable to an FEA analysis.

In a similar approach the rocker pivot shaft and mounting bracketry could be evaluated in the same manner. Deflection of the shaft could be studied since the real life rocker arms pivot on needle roller bearings or tolerance fit copper sleeves (Chin-Sung & Ho-Kyung, 2010). Any movement would alter the valve lift profile and hence would be of interest.

It is important to remember that FEA isn't just a tool for analysing the strength, rigidity, durability or thermal resistance of a part. If a design can be optimised from the very beginning then in effect those concerns could be eliminated from the process. The part would be the best design it could be without the need to revisit and redesign multiple iterations. A well-suited tool for this particular approach is topology optimisation study (Stevenson, 2018)

Topology optimisation finds the best distribution of material given an optimisation goal and a set of constraints. Essentially taking a solid block of material and removing or migrating material to more or less critically effected areas in order to meet predefined objectives such as displacements or maximum stress levels. The results or offered design is often of complex geometry and has, in the past been difficult to manufacture but with the emergence additive manufacturing and in some cases even 3D printing is making these complex parts more practical.

When applying FEA to the rocker arm it is crucial to consider the following:

- **Variable** – Loading
- **Objective** – Minimal strain energy
- **Constraint** – Material volume

Variable – part loading or forces on the rocker arm.

From the earlier mathematical study the author has been able to determine how much force the cam lobe exerts on the bucket follower. In addition the returning spring rate is also known and as such can be applied to a basic model or even a block of material.

Objective – minimal strain energy across the rocker arm.

As previously mentioned the author believes the part is likely to deform under loading especially during periods of increased rpm. To make the part best suit its application the software can be asked to consider particular objectives, in this case to make the part as stiff as possible while remaining within the allowed envelope. Parameter can be entered asking for the software to achieve a percentage reduction in strain energy.

Constraint – material volume

Since existing geometry and necessary parts dictate the volume available in which to fit the rocker arm this would be classed as the constraint. Figure 22 shows how the exhaust cam shaft and surrounding material govern the amount of build space. The part must span the distance from cam lobe to inlet valve but cannot infringe or conflict with the spark plug surround or rocker cover (Stevenson, 2018)

Conclusion

Part one of this project, building a mathematical model to arrive at an end figure for FMEP has proved to be useful tool when comparing the proposed rocker arm assembly. Frictional torque was shown to be directly affected by the acceleration generated from the cam profile and coefficient of friction.

The iterative design process to understand which, if any, rocker arm would provide a suitable replacement highlighted the importance of being able to handle physical objects when tackling a project of this nature.

Using one existing cam shaft to drive both intake and exhaust was always going to be problematic since intake and exhaust cam have a marginally different profile to begin with. The proposed design

has been shown to be unsuitable in its current form. While the maximum lift can be replicated with an appropriate rocker ratio (1.17:1), the resulting duration and area under the curve are not acceptable. The accelerations caused by the positioning of the roller bearing, particularly the negative acceleration are too great and will lead to dynamic instability earlier in the load range. While single camshafts have been proven to work effectively, in this case the existing geometry hindered the progress. Ideally a new camshaft comprising of two individual lobes, both with a different profile would be a better solution.

FMEP was shown to reduce with the roller bearing despite the added acceleration and mass.

Evaluation.

This project has been rewarding as well as frustrating. I find designing parts and printing them out is extremely engaging, far more so than mathematical study. I'm certainly glad I bought my own 3D printer, at times relying on the university equipment just isn't practical especially when multiple parts are necessary. Spending time fine tuning the settings and seeing an improvement in print quality is equally satisfying. Regrettably my personal computer on which I'd stored the entirety of my CAD work failed on Sunday 22 April and a professional computer shop suggests the material is irrecoverable. Fortunately the word document was saved elsewhere but lack of access to CAD drawings will explain some of the recycle images within the latter half of the report.

The overall results were perhaps better than expected, for weeks I agonised over which cam would be suitable to drive a rocker and if correcting timing was ever a remote possibility.

I would like to extend a sincere thank you to the readers and my project supervisor.

References

- Agarwal, V. K. & John P, A., 2014. *Evaluation and Comparative Study of Valve Train Layouts with Different Rocker Ratio*, s.l.: SAE.
- Americian Honda, 2013. *How it works: Honda Unicam Engines*. [Online]
Available at: <http://powersports.honda.com/experience/articles/090111c0811a9fc2.aspx>
[Accessed 18 March 2018].
- Beardmore, R., 2012. *Roymechx*. [Online]
Available at: http://www.roymech.co.uk/Useful_Tables/Tribology/Bearing%20Friction.html
[Accessed 13 March 2018].
- Biggs, S., 2017. *The Engineer*. [Online]
Available at: <https://www.theengineer.co.uk/3d-printing-education/>
[Accessed 4 April 2018].
- Bolton, W., 1998. *Mechanical Science*. 2nd ed. Cambridge: Blackwell Science.
- BOSCH, R, 2000. *Automotive Handbook*. 5th ed. Plochingen: Robert Bosch GmbH.
- Chin-Sung, C. & Ho-Kyung, K., 2010. *Safety evaluation of the rocker arm of a diesel engine*, Seoul: Elsevier.
- Faistaure, S., 2015. *The LC4 wizard, how the KTM 690 DUKE flies along the road* [Interview] (16 December 2015).
- Giles, J., 1968. *Engine Design*. 2nd ed. London: Liffé Books Ltd.
- Greene, J; California State University, 2012. *PLA and PHA Biodegradation in the Marine Environment*, Chico, CA: California State University.
- KTM Group, 2017. *KTM AG Annual Report 2016*, Mattighofen: KTM AG.
- Makay, S., 2009. *Friction Teardown*, Worcester: MAHLE.
- Manning, J., 2012. *Internal Combustion Engine Design*. 1st ed. Shoreham-by-Sea: Ricardo UK Limited.
- Motorcross Action Magazine, 2012. *The complete, exhaustive detailed guide to the 2013 CRF 450*. [Online]
Available at: <https://motocrossactionmag.com/the-complete-exhaustive-detailed-guide-to-the-2013-crf450-more-than-anyone-else-will-ever-tell-you/>
[Accessed 18 March 2018].
- Royte, E., 2006. *Environmental Impact of Corn Based Plastics*. [Online]
Available at: <https://www.scientificamerican.com/article/environmental-impact-of-corn-based-plastics/>
[Accessed 1 April 2018].
- Stevenson, B., 2018. *Applying FEA to a final year project*, Swansea: N/A.
- Tien-Chi, H. & Chun-Yu, L., 2016. *From 3D modeling to 3D printing: Development of a differentiated spatial ability teaching model*, Taiwan: Elsevier.

Ultimaker, 2017. *Manuals: Interface Overview Ultimaker CURA*. [Online]
Available at: <https://ultimaker.com/en/resources/51947-interface-overview>
[Accessed 1 April 2018].

Vital MX, 2017. *Vital MX - First look*. [Online]
Available at: <https://m.vitalmx.com/photos/features/First-Look-2017-Honda-CRF450R-and-CRF450RX,38912/First-Look-2017-Honda-CRF450R-and-CRF450RX>
[Accessed 18 March 2018].

X-Engineer, 2018. *Mechanical efficiency and friction mean effective pressure (FMEP)*. [Online]
Available at: <https://x-engineer.org/automotive-engineering/internal-combustion-engines/performance/mechanical-efficiency-friction-mean-effective-pressure-fmep/>
[Accessed 18 March 2018].

Appendices

B3 ✕ ✓ f_x =0.05245*'Acceleration s.s'!B3							
	A	B	C	D	E	F	G
1			Example 1			Bucket follower	
2	Cam°	Inertia		Spring force		Contact force	
3	1	0		154.25		0	
4	2	0		154.25		0	
5	3	0		154.25		0	
6	4	0		154.25		0	
7	5	0		154.25		0	
8	6	0		154.25		0	

Figure 36. Contact force Excel work.

B3		✕		✓		f_x		=('Frictional force'!B3*(13+Lift!B2))/1000	
	A	B	C	D	E	F	G	H	
1	Example 1								
2	Cam°	Frictional torque N/m	Work done(J)	Total amount of work over 360°	1.337895367 J				
3	1	0	0	Δwork / engine displacement	5373.073762 Pa				
4	2	0	0						
5	3	0	0						
6	4	0	0	FMEP x 2	10746.14752 Pa				
7	5	0	0						
8	6	0	0						
9	7	0	0	Per lobe	0.053730738				
10	8	0	0	Inlet	0.107461475 Bar				
11	9	0	0						

Figure 37. Final page on the Excel workbook generating an end figure for FMEP.

D2				=30.85*(B2+5)				
	A	B	C	D	E	F	G	H
1	Cam angle	Y displacement	Accel	Spring Force	Fv			
2	1	0.43	0	167.60	0.00		$I=mr^2$	0.00005632
3	2	0.19	0	160.17	0.00		m	0.055
4	3	0.17	0	159.49	0.00		r	32.4
5	4	0.16	0	159.04	0.00			
6	5	0.14	0	158.61	0.00		L1	0.0324
7	6	0.13	0	158.20	0.00		L2	0.0276
8	7	0.12	0	157.83	0.00			
9	8	0.10	0	157.47	0.00			

Figure 38. Components used to generate Fv...

=((E2*H\$6)/H\$7)+(H\$2/(H\$6*H\$7))*C2							
D	E	F	G	H	I	J	K
Spring Force	Fv					Fc	Frictional force
167.60	0.00		$I=mr^2$	0.00005632		0	(
160.17	0.00		m	0.055		0	(
159.49	0.00		r	32.4		0	(
159.04	0.00					0	(
158.61	0.00		L1	0.0324		0	(
158.20	0.00		L2	0.0276		0	(
157.83	0.00					0	(
157.47	0.00					0	(

Figure 39. And in turn, to generate Fc.

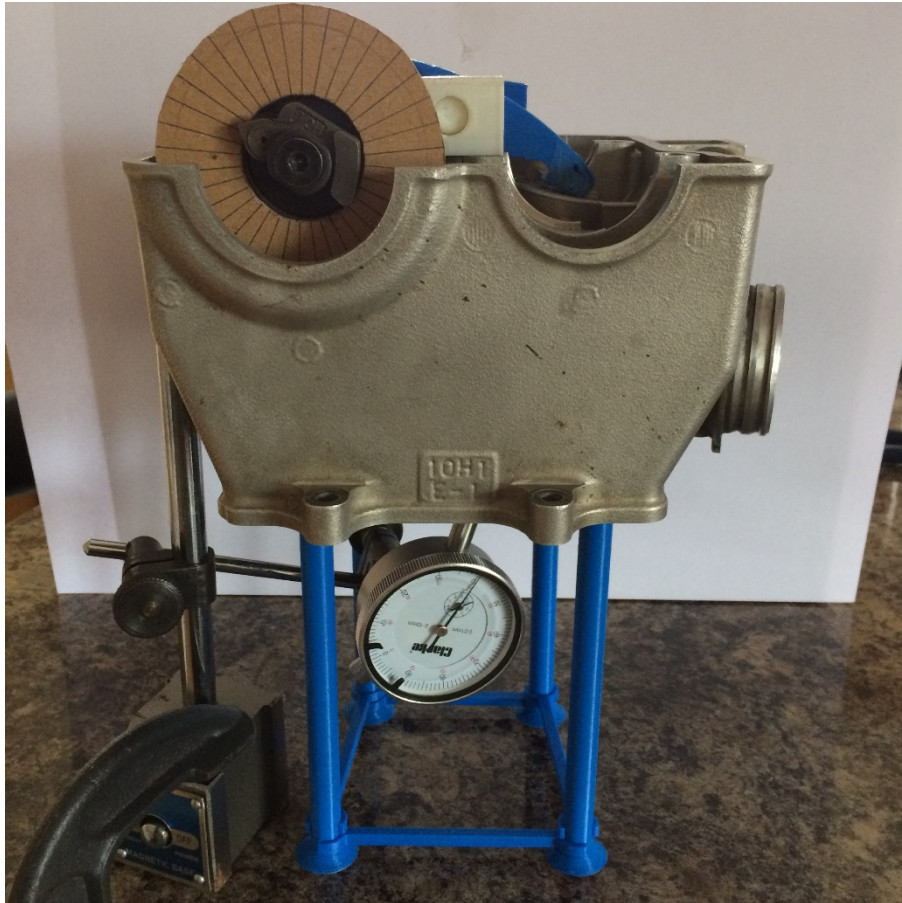


Figure 40. Stilt type jig to allow access for the DTI measuring the valve displacement.