

Numerical Modelling of Mechanical Methods of Increasing the Efficiency of an Internal Combustion Engine

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'This thesis is the result of the author's 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.'

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Abstract

Increasing the thermal efficiency of an engine is a key strategy in the development of passenger vehicles for the future. This report details work done to understand non-traditional mechanical methods of increasing internal combustion engine efficiency and a method of deriving valve lift profile from camshaft profile for follower actuated valves. Numerical and CAD models of a linkage type engine arrangement as produced by Nissan and an eccentric on the connecting rod journal as being developed by Gomecsys were produced to analyse these as methods of adjusting the compression ratio of an engine. Models were also produced of sliding lobe camshafts as a method of cylinder deactivation. These systems are assessed as methods to be implemented on the Suzuki SV650 90-degree V-twin engine. As an extension of the conversion to follower valve actuation as part of the cylinder deactivation system's design, a numerical model to return valve lift from cam profile is developed. Some challenges to implementing the linkage type engine as a modification of existing architecture, especially for a V configuration engine, are identified. The Gomecsys system was modelled to compare piston motions and the numerical methods derived for this were proved accurate with the corresponding CAD model. The numerical model for calculating valve lift from camshaft profile was successfully built and tested. Both systems of compression ratio variation show promise and this work forms a useful basis for further work, providing tools to quickly model and assess any designs produced.

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Contents

Abstract	2
Acknowledgements.....	2
List of Charts.....	4
List of Equations	4
List of Figures	5
List of Tables.....	5
1.0 Introduction	6
2.0 Variable compression ratios.....	7
2.1 Introduction; Downsizing and Turbocharging.....	7
2.2 Compression Ratios.....	7
2.3 Overview of Systems Considered.....	8
2.4 Gomecsys	10
2.4.1 Overview	10
2.4.2 Modelling	11
2.4.3 Numerical	12
3.4.4 Verification	13
2.5 Nissan Linkage Type Engine	13
2.5.1 Overview	13
2.5.2 Nomenclature	13
2.5.3 Review of Patents.....	14
2.5.4 Modelling	16
2.5.5 Validation	19
2.5.6 3D Modelling	21
2.5.7 Fitting to SV dimensions.....	23
2.5.8 Motion Analysis.....	24
2.6 Comparison of Piston Motions.....	25
2.7 Conclusions	28
3 Cylinder Deactivation	28
3.1 Introduction	28
3.2 Overview/systems.....	28
3.3 VW/BMW style system	30
3.3.1 Overview	30
3.3.2 Initial ideas and packaging Issues	30
4.3.3 Forces and accelerations.....	32
3.4 Conclusions	33

4 Camshaft Profile and Valve Lift	33
4.1 Introduction	33
4.2 Cam Shaft Measurement	34
4.3 Bucket and Shim.....	34
4.4 Rocker	36
4.5 Methodology.....	36
4.6 Validation	42
4.7 Conclusions	46
5 Project Conclusions	46
References.....	48

List of Charts

Chart 1; Gomecsys numerical model validation	13
Chart 2; Linkage engine numerical model verification	19
Chart 3; Piston motion of a single linkage group in V configuration engine	22
Chart 4; Piston motion of a V configuration linkage engine	23
Chart 5; Piston motion comparison	25
Chart 6; Piston peak velocity comparison.....	26
Chart 7; Big end motion comparison	27
Chart 8; Cam and Valve lift comparison, bucket and shim	35
Chart 9; Rocker and Cam profiles	37
Chart 10; Cam and rocker slopes	38
Chart 11; Cam and Rocker slopes, method 2.....	38
Chart 12; Comparison of results from rocker and cam surface slopes.....	40
Chart 13; Valve lift model verification	42
Chart 14; Change in Lift around max lift	43
Chart 15; Comparison of method accuracies.....	45

List of Equations

Equation 1; Change in clearance volume for a change in CR.....	8
Equation 2: Change in TDC for a change in CR.....	8
Equation 3; Linkage engine Piston Height.....	16
Equation 4, Equation 5, Equation 6; Horizontal displacements of linkage engine	17
Equation 7, Equation 8; Rearranged horizontal displacements.....	17
Equation 9; rearrangement of equation 5	17
Equation 10; Equation for γ	17
Equation 11, Equation 12; Final equations used for γ and α	18
Equation 13; Time available for cam slide actuation	32
Equation 14; Sliding lobe acceleration.....	32
Equation 15; Newton's second law of motion	32
Equation 16; Lift calculation.....	34

Equation 17, Equation 18; Cam local coordinate calculations.....	36
Equation 19, Equation 20; Cam global coordinate calculations	36
Equation 21; Cam surface slope.....	36
Equation 22; Rocker slope from cam surface coordinates	39
Equation 23; Valve lift from rocker slope	40

List of Figures

Figure 1; Diagram of Gomecsys geometry.....	13
Figure 2; Nissan VCT engine geometry diagram	16
Figure 3; Representative solidworks model of linkage engine	19
Figure 4; Representative CAD model, second version	21
Figure 5; BMW Shiftcam system (BMW, 2018).....	30
Figure 6; CAD model of splined camshaft	31
Figure 7; Lift actuated valve sliding mechanism	31
Figure 8; Cylinder deactivation system with separate actuation cam	32
Figure 9; Bucket and shim arrangement cross-section.....	34
Figure 10; Bucket and shim arrangement trigonometry diagram	34
Figure 11; Local and global coordinates for cam lobe	36
Figure 12; Close up of chart 10 intersection	38
Figure 13; Excel table for rocker slopes	39
Figure 14; Excel sheet for rocker lift	40
Figure 15; Excel macro	41
Figure 16; Solidworks model for returning valve lift.....	42

List of Tables

Table 1; Changes in TDC for Changes in CR.....	8
Table 2; Variation of measured and calculated linkage engine angles.....	19
Table 3; Measured and calculated linkage engine angles	20
Table 4; peak piston velocity comparison.....	26

1.0 Introduction

The internal combustion engine continues to see constant development, with today's main focus of research and development being on lowering emissions to meet tightening regulations worldwide (Delphi Technologies, 2018). This has led to solutions looking to increase the range of conditions at which the engine will operate in an efficient manner. Recently this has meant that turbocharging has become common fitment for almost all manufacturers (Ulrich, 2015), bringing with them increased specific power outputs allowing for the use of smaller engines without compromising on driveability (Borg Warner, 2019). Turbocharging does however bring with it some compromises/challenges, these include issues with low efficiencies when no boost is available, knock issues at higher RPM and loads, and a compromise between high compression ratios for low load efficiency or low compression ratios to make the most of the turbo (Hartman, 2007).

The genesis of the project undertaken for this paper was an interest in engine geometries. From this, a goal of redesigning the Suzuki SV650 90-degree V-twin engine incorporating major changes to its operation in the pursuit of increased efficiency was set. Initially, the effect of crankshaft offset as a means of reducing friction was considered and a simple model to determine the difference in piston motions in relation to crank offset was created. Then, the linkage type engine being released by Infiniti was considered as a system to incorporate a variable compression ratio. This led on to research of other such systems, mostly centred around variable compression or over-expansion, with other mechanical methods of improving overall efficiency also being considered.

Models for both the Gomecsys and Infiniti variable compression ratio systems were developed and designs for cylinder deactivation systems that could be incorporated into the existing cylinder head castings were developed. It became apparent that these cylinder deactivation designs would not be feasible, so an alternative of redesigning the valve train to incorporate rocker arms, thus increasing the number of viable cylinder deactivation systems, became the focus of the project. This led to a need for a numerical model which could calculate the resultant change in valve lift profile due to the change in valve actuation methods.

Whilst this project's focus has changed throughout its development, it has stuck to the principle of investigating methods of increasing engine thermal efficiency through the design and development of non-traditional mechanical systems. It has provided a greater knowledge and understanding of the systems considered, mechanical design and especially the geometry of

linkages and levers etc. The models developed can be improved upon and expanded as tools to be used in the design of crankshaft and valvetrain assemblies.

2.0 Variable compression ratios

2.1 Introduction; Downsizing and Turbocharging

In reaction to strict emissions regulations being measured by the NEDC test, automotive manufacturers in Europe have focused on downsizing including turbocharging as the predominant method of meeting the demands of regulators (Ulrich, 2015). This solution has performed well but a change to the WLTP with a randomised portion of the emissions testing procedure means that these small engines are no longer the best solution available (Davis, 2018). Whilst forced induction is good for better specific power outputs, the increased cylinder pressure and temperatures associated with it mean that often the engine design will be a compromise of high compression for good slow speed performance (off boost) and low compression to reduce knock and extract the most from higher engine speeds and boost levels (Hartman, 2007). To combat this, a number of variable compression ratio systems have been designed and developed, though until recently none had made it to the market in any great numbers. However, with the announcement of the 2019 Infinity QX50 being fitted with Nissan's new VC-T or Variable Compression Turbo engine, a variable compression ratio system is now available to consumers. With a major manufacturer having the confidence in the systems durability and reliability, it seems likely that other manufacturers will follow suit in the near future.

2.2 Compression Ratios

As variable compression ratios are to be implemented, a range of viable ratios must be established from research; Heywood lists typical compression ratios of 16:1 for compression ignition and 9:1 for spark ignition engines (Ideal Models of Engine Cycles, 1989), although an older source, these figures are close to those found from Nissan who uses a variation from 8:1 to 14:1 (Infiniti, 2019), Gomecsys at 9:1 to 14:1 (Gomecsys, 2019) and Mazda who's Skyactiv-X engine has a static compression ratio of 16:1 (Carney, 2018). This range can be used as targets to be considered in future designs. As the project centred around the Suzuki SV650 engine, a simple model to determine the necessary changes in volumes and TDC positions to achieve a given compression ratio was made in excel. To get a range that matched the Nissan system for example, it is assumed that the maximum static compression ratio would be set to 14:1, then the system would need to change the stroke by 3.4mm giving an additional 17.3cc of clearance volume to reach a compression ratio of 8:1. Once an understanding of the systems to be used has been

established, the viability of this figure can be determined. The changes in TDC height and additional clearance volumes required for each step in compression ratio are shown in table 1.

The additional clearance required was calculated using the following formula;

$$V_c^a = \text{Swept} \frac{V_s}{CR} - V_c^s$$

Equation 1; Change in clearance volume for a change in CR

Where V is volume and the subscript denotes whether it's swept or clearance and the superscript denotes either additional or static clearance volume. From this, the change in TDC is calculated by dividing the additional clearance volume by the area of the bore.

$$\Delta TDC = \frac{V_c^a}{A_b}$$

Equation 2: Change in TDC for a change in CR

Compression Ratio	Change in TDC, mm	Additional Clearance, cc
8:1	3.4	17.3
9:1	2.5	12.8
10:1	1.8	9.3
11:1	1.2	6.3
12:1	0.8	3.9
13:1	0.4	1.8
14:1	0	0

Table 1; Changes in TDC for Changes in CR

2.3 Overview of Systems Considered

Variable compression ratio systems have been used to test the knock characteristics of different fuels for decades and are an attractive proposition for automotive applications (The Fuel Expert, 2019), especially forced induction engines where ignition mapping tends to be knock limited across more of the rev range than in a naturally aspirated engine (Walker, 2001). Being able to reduce the compression under higher boost and load conditions would allow the timing to be closer to max break torque (MBT) timing. Several systems have been proposed since the inception of the four stroke Otto cycle, with the Atkinson cycle being representative of an early system of over-expansion with simple modifications being possible to allow for changes in compression (United Kingdom Patent No. 367,496, 1887). Since then, the most common method of over expansion/variable compression has been a modified Atkinson cycle, commonly known

as the Miller cycle, where the inlet valve is left open during the start of the compression stroke to allow some the fresh charge to be expelled, reducing the actual compression stage of the stroke hence the amount of material being compressed (United States Patent No. 2817322, 1956).

This system is used to reduce compression on the Mazda Skyactiv-X engine, a SPCCI (SPark Controlled Compression Ignition) engine, prized for its high efficiency and low NO_x and particulate emissions (Zurschmeide, 2018), where an extremely high compression ratio is required to approach auto-ignition temperatures in the cylinder. As this is not always the desired running condition, the miller cycle is used to drop the compression ratio (incidentally increasing the expansion ratio) so that the engine can transition between SPCCI and conventional spark ignition. They have used electronic variable valve timing actuators to switch between Miller and Otto cycles, with a 16:1 compression ratio in their prototype vehicles. The main advantage of this system being that it uses established and readily available parts with the only item less common on modern production engines being cylinder pressure monitoring devices (Mazda, 2017).

SAAB debuted a system at the 2000 Geneva show using a two-part block which allowed the cylinders to hinge away from the crankshaft as a method of reducing the compression ratio. This system was first patented by Gregory Larsen in 1992 (Larsen, 1992) but never made it to production, partially due to development issues due to requiring more flexible exhaust and intake couplings, and partially due to SAAB's financial issues. It gave compression ratios of 8-14:1 through four degrees of rotation of the upper block (SAAB Planet, 2016).

Another solution explored by various manufacturers is a piston in the cylinder head, whereby the piston can be set flush with the combustion chamber or retracted to increase the clearance volume of the cylinder. Such a system has been commercially available for years in two stroke compression ignition model engines of less than around 8cc whereby compression is adjusted by changing the position of a plate from the top of the cylinder towards the top of the piston's travel. This system is ideally suited to two stroke engines without valves in their cylinder heads, where the piston varying the clearance volume can account for a larger percentage of the cylinder area than is possible when fitted around valves. Notably, this formed part of the Lotus Omnivore prototype engine designed to run on several different fuels, necessitating some method of changing compression ratios (8-50:1). This was also a two-stroke design, using a variable volume cylinder head in conjunction with variable exhaust port timing to vary the timing

of pure compression in the cylinder, a system patented in 1978 by Yamaha (United States Patent No. 4121552, 1978).

Two systems of variable piston geometries were patented by Ford in 1993 (United States Patent No. 5178103, 1993) and 1997 (United States Patent No. 5755192, 1998), reminiscent of those used in Teledyne Continental Motors military engines in the 1970's (Grundy, Kiley, & Brevick, 1976). These did not make it into production by Ford. Possibly as they would have added significant mass to the rotating assembly and as they used hydraulic signals sent through the crankshaft and connecting rods, concerns over reliability issues from using varied pressures or supply routes through vulnerable plain bearings. Similarly, AVL has researched telescopic connecting rods but have not put them into production (Fraidl, et al., 2016).

It is worth noting that Peugeot unveiled a system with MCE-5 called the VCRI, offering 7-18:1 CRs, which is reminiscent of the Nissan linkage-type design (Green Car Congress, 2009). Other manufacturers looking to adopt such a system may be able to use a design with elements of both in a bid to compete without having to license either design. Whilst MCE-5 also claim very good fuel efficiency and specific power figures from their engine (220hp from 1.5l @ 6.7l/100km NEDC) (Green Car Congress, 2009) they also claim reduced engine friction and improved oil consumption by using a combined piston/con-rod assembly that sees greatly reduced horizontal loading of the piston (MCE-5 Developments, 2019). Yet it has not seen production since its unveiling at 2009 Geneva motor show, though development is ongoing.

The two main systems having been considered during this project then are the Nissan linkage-type engine system, which varies the position of the piston's stroke within the cylinder and the Waulis derived Gomecsys system which varies both this and the ratio of inlet and exhaust to compression and expansion strokes. These systems are discussed in further detail further on in this paper.

2.4 Gomecsys

2.4.1 Overview

The Gomecsys system varies the stroke of the piston with each half rotation of the crankshaft, hence compression and expansion ratios. It accomplishes this through a planetary gear fixed to the eccentric shaft on the big end, which continuously rotates at half crank speed. This is geared via an intermediate shaft, rotating at twice crank speed, which is geared to halve the eccentric's rotation in relation to the crank (International Patent No. WO 2013/110700 A1, 2013). Adjusting the input angle at the sun gear adjusts the stroke lengths so that the eccentric will be at its 'top' position at TDC either for the end of the exhaust or compression strokes, or some point in

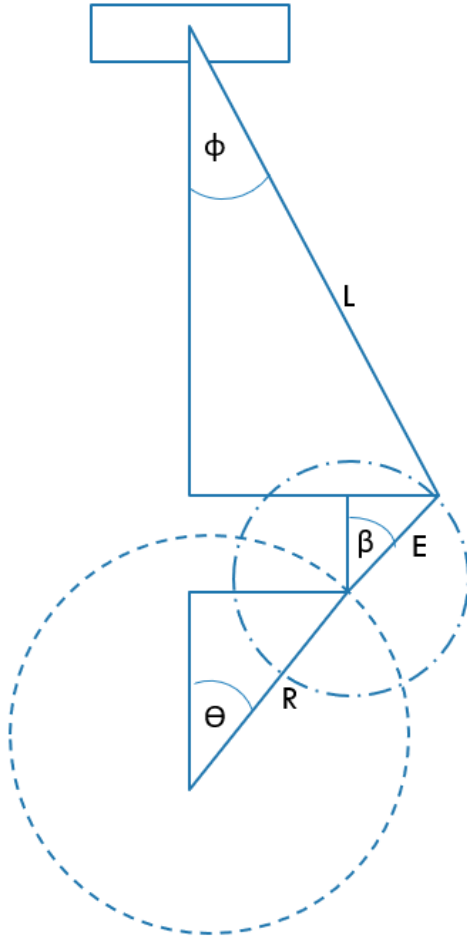
between these two extremes. This system is currently being developed by Gomecsys and has previously undergone development work at Prodrive (Corfield & Harrison, 2006). Currently, they have one test vehicle fitted with this system (Gomecsys B.V., 2019) and are hoping to market it for aftermarket fitment requiring a change to their crankshaft assembly and installation of control system but with very minor modifications required to the base engine, and as an OEM fitment.

2.4.2 Modelling

The models developed to assess this system include a numerical model in excel to study the motion of the piston and effect of input shaft changes and a simplified 3D model to verify the numerical model and more easily explain the system's operation. The basic motion of the parts was determined by studying promotional material including animations of the system released by Gomecsys. To develop the model, the dimensions were set to match that of the ford fox engine, with the eccentric set to an arbitrary dimension for expediency. Once the model had been verified as accurate, these dimensions were changed to fit the SV650 and the proposed variation of compression ratio.

2.4.3 Numerical

To determine the equations of motion for the Gomecsys system, it can be thought of as a set of two circles and a triangle as in figure 1. The crankshaft is the lowest of the two circles, its radius



is the crank throw, R , and its angle of rotation is θ . The position of the end of the line R determines the origin of the line E , the offset length of the eccentric, at an angle, β , which changes with θ as they are geared together in a ratio of 2:1. β can be calculated from the angle that the input shaft is turned through as follows;

$$\beta = \frac{\theta - \text{Input Angle}}{2}$$

If these two angles and all three lengths are known, then ϕ can be calculated via the following formula;

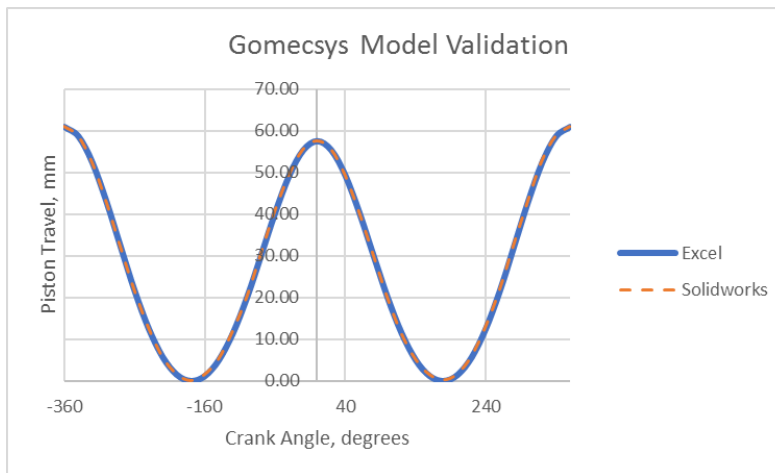
$$\phi = \sin^{-1} \left(\frac{R}{L} \times \sin \theta \right) + \left(\frac{E}{L} \times \sin \beta \right)$$

So that the height of the piston from the centre of the crankshaft can be calculated as;

$$H_p = R \times \cos \theta + E \times \cos \beta + l \times \cos \phi$$

These were input into excel to model the motion of the piston, allowing the effect of changes to the input angle on expansion ratios and compression ratios to be studied.

Figure 1; Diagram of Gomecsys geometry



3.4.4 Verification

The results from the numerical model were verified by building a simplified model in Solidworks, this model was also 3D printed as a means of explaining the systems. The values from both

showed little variation (see chart 1), so the numerical model is an accurate model of the system.

Chart 1; Gomecsys numerical model validation

As this part of the project was discontinued at this point, further refinements such as including crank offsets were not implemented and further research not carried out.

2.5 Nissan Linkage Type Engine

2.5.1 Overview

The Nissan VC-Turbo engine is a linkage-type engine which varies the compression ratio by keeping the same (or near identical) stroke length throughout its range of adjustment, whilst moving the maximum height the piston reaches in the bore, varying the clearance volume. This is achieved by having a fixed crankshaft throw dictating the motion of a beam, with one end attached to the connecting rod and the other connected to a lower control arm. The position of the pivot at the lower end of this control arm is controlled by an eccentric arm on a stepper motor (Infiniti, 2019).

It is now commercially available in the Infiniti QX50, an SUV aimed towards the higher end of the market with Nissan announcing plans to release the system across various platforms in the next few years (Infiniti, 2018).

2.5.2 Nomenclature

The components for this engine type are referred to by the names found in the patents filed by Nissan. The part analogous to the connecting rod of a traditional engine is called the upper link, the linkage connecting the crankshaft to the upper link and control link is called the lower link, the lowest link is called the control link and it is pivotally mounted on an eccentric shaft called the input shaft (United States Patent No. US 7228838 B2, 2007).

2.5.3 Review of Patents

To gain an understanding of the system, patents assigned to Nissan were searched and analysed.

In 2007, a patent based on the system with the input shaft being driven at half crank speed to allow for overexpansion type operation with variable compression ratio was filed (United States Patent No. US 7228838 B2, 2007). It is not known why this system was not implemented; it may be introduced in later variable compression engines.

One patent found during research detailed a design of the lower linkage that distributes the rigidity throughout the part when under load to stop excessive wear at the crankshaft bearing, extending the life of the part's bearings (Europe Patent No. EP 3 040 535 A1, 2014). This may be patented as a means of stopping competitors developing similar systems from using this method of overcoming excessive bearing wear. This suggests that this point, where the forces from the piston and the reaction forces from the rest of the system meet at the crank, may have been causing longevity issues during development. Therefore, this part should be investigated in detail if an engine with a similar system is to be produced. Further details of the lower linkage are found in another patent from 2007 in which some information about the construction of such a linkage and the directions in which forces are acting upon it from each of the three pins passing through it are described (United States Patent No. US 7290507 B2, 2007). This is likely to be useful in further development of a linkage type engine when the magnitude of these forces should be calculated to estimate the loads seen at each bearing so that their ability to withstand them can be assessed. It also provides further diagrams of parts which may provide information on the inherent weaknesses of the system.

Nissan also filed a patent around the management of the fuelling and ignition timing of an engine wherein the piston acceleration is lower around TDC than BDC. This patent describes the engine as having a range of fuelling modes (including stratified and homogenous and lean or stoichiometric modes) in response to engine load and speed as well as a prediction of the compression ratio from input shaft position (United States Patent No. US 7159543 B2, 2007). This patent may be useful research for developing control strategies for variable compression ratio engines, however it is quite unspecific in its description of operation and application and may be intended purely as a means of protecting general methods for controlling a linkage type engine. This patent is most useful in the context of this project as a source of information on linkage system types explored in the development of their production engine. The difficulties of managing an engine with a variable compression ratios are also discussed in a patent for a system of linking the input motor to the control shaft such that ratio of input to change is highest

when moving from a high compression to intermediate compression (United States Patent No. US 8087390 B2, 2012). The patent claims that a linkage type engine suffers with knock more frequently during this transition if it is slow, requiring a fast transition from high compression to intermediate, with a slower transition from intermediate to low being inferred, possibly to allow finer control during low demand running.

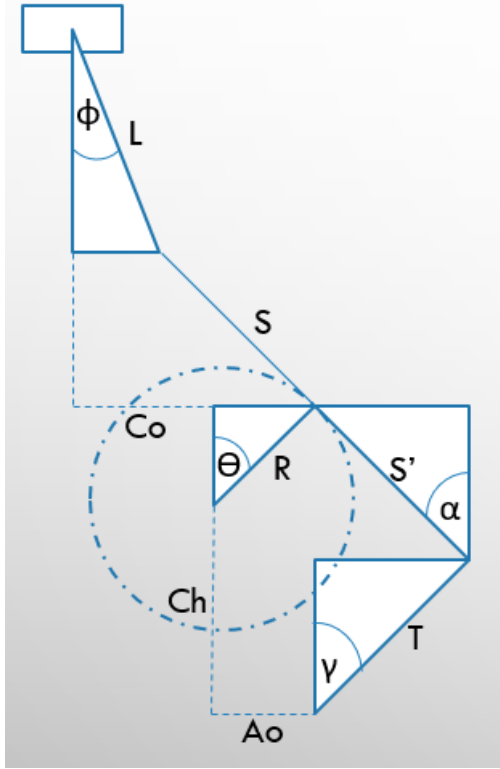
Another patent detailing an engine with increased thermal efficiency, being a linkage type but not solely focused on this method of adjusting compression ratio, details a piston with improved resistance to piston ring sticking and oil caulking (United States Patent No. US 7594467 B2, 2009). This design may have been made in reaction to an increase in piston temperature caused partially by the reduced piston acceleration during combustion, allowing more time for the combustion heat to transfer into the piston's crown.

Further issues encountered during the development are covered in another patent wherein linkage geometries that avoid the misalignment of the control shaft's bearing which was found to occur when the forces acted down upon the control rod (United States Patent No. US 7980207 B2, 2011). Another patent (United States Patent No. US 8100098 B2, 2012) also includes geometric constraints which reduce the side thrust of the piston around bottom dead centre in reaction to an issue with liner damage or oil film being removed from the piston by the liner arising from the need for a cut out in the lower edge of the liner to allow clearance for the crankshaft (United States Patent No. US 2009/0031994 A1, 2009). The number of inventors listed on these patents suggest that significant resources were devoted to solving these issues. Therefore, the ratios and constraints outlined may be implemented as checks in further work on the numerical model developed for this project as a means of informing design changes and ensuring that a dimensions determined by the use of the model would be a useful starting point for the development of such an engine. Due to the shortening of the crank throw over a conventional engine, higher torsional stresses are seen at the crank pins and a system of creating oil passages that are not on planes 0° , 90° , 180° or 270° from the plane of the crank as a means of alleviating torsional stress concentrations around the bearing side openings are detailed in a 2012 patent (United States Patent No. US 2012/0111143 A1, 2012).

As an unconventional engine type, an issue with transverse vibrations, not commonly considered an issue in traditional engines, is described in a 2017 patent along with a means of reducing the magnitude of such vibrations through the use of counterweights as a part of linkage designs (United States Patent No. US 9617912 B2, 2017).

2.5.4 Modelling

To develop the numerical model for the VCT engine, dimensions were taken by scaling an image released by Nissan of a cutaway model displayed during the launch of the engine. This was done in Solidworks by importing an image and scaling it by using the bore measurement from an older 2l inline 4-cylinder petrol engine from Nissan. This allowed measurements between centres of the main linkages and offsets which were included in the model. These measurements were



limited in accuracy by the resolution and perspective of the image. As such they were considered as a guide and a representative model built in Solidworks was used to check their validity. Another issue with this method was that it was not possible to measure the crank throw of the engine as it was obscured so an estimate was made by examining frames from a promotional animation of the system (Nissan, 2018), these were then put into the Solidworks model and adjusted until the model worked. These figures were then used as the base for the numerical model until the model was tested against Solidworks data, first by scaling these dimensions then by inputting the dimensions used in the Solidworks model created to represent the

system in the Suzuki engine.

The aim of the numerical model was to be able to input the dimensions of the linkages and have an output of the motion of each part for each degree of crank rotation. This was then verified against the Solidworks model.

Figure 2 is a diagram of the system with the notation used in this model.

Piston height is given by the addition of all vertical measurements from the crank centre upwards;

Figure 2; Nissan VCT engine geometry diagram

$$Piston\ Height = L.\cos\phi + s.\cos\alpha + R.\cos\theta$$

Equation 3; Linkage engine Piston Height

However, to calculate this, all other angles must be determined as a function of crank angle, θ . And as all intersections of the parts are pin jointed, they can all rotate so only the height and offset of the control shaft (the lowest joint of control rod, T), and the offset of the bore, using the crankshaft as the origin are known.

Resultant horizontal displacements were first considered and returned the following equations;

$$L \cdot \sin \phi + s \cdot \sin \alpha = R \cdot \sin \theta$$

$$T \cdot \sin \gamma + A_o = s' \cdot \sin \alpha + R \cdot \sin \theta$$

$$s' \cdot \cos \alpha + T \cdot \cos \gamma = C_h + R \cdot \cos \theta$$

Equation 4, Equation 5, Equation 6; Horizontal displacements of linkage engine

To separate the known values from those that needed to be calculated, equations 4 and 6 were rearranged as follows;

$$s' \cdot \sin \alpha - T \cdot \sin \gamma = A_o - R \cdot \sin \theta$$

$$s' \cdot \cos \alpha + T \cdot \cos \gamma = C_h + R \cdot \cos \theta$$

Equation 7, Equation 8; Rearranged horizontal displacements

As $\cos \alpha = \sqrt{1 - \sin^2 \alpha}$, equation 4 could be further rearranged to give $\sin \alpha$ so that it could be used to remove α from equation 6, leaving γ as the only unknown value in that equation. So, equation 5 was rearranged as;

$$\sin \alpha = \frac{T}{s'} \cdot \sin \gamma + \frac{A_o}{s'} - \frac{R}{s'} \cdot \sin \theta$$

Equation 9; rearrangement of equation 5

And combined into equation 10;

$$s' \sqrt{\left(1 - \left(\frac{T}{s'} \cdot \sin \gamma + \frac{A_o}{s'} - \frac{R}{s'} \cdot \sin \theta\right)^2\right)} + T \cdot \cos \gamma = C_h + R \cdot \cos \theta$$

Equation 10; Equation for γ

As a rearrangement for γ by itself was not found, this equation was rearranged for 0 so that a "Goal seek" function could be used in excel to iterate γ until a value that satisfied the equation was found. This was implemented using a macro, which took several minutes (4.5 when timed)

to iterate through all 360 degrees of Θ . The same approach was then coded in Matlab and was found to be significantly faster (25.2 Seconds). It would, however, return a motion plot based on impossible inputs using irrational numbers. Also, as it was not an integrated solution, the overall decrease in time required to return a motion profile when the inputs were corrected was not significant enough to warrant the added complexity of having the model split between two platforms.

For final method implemented in excel, equations 5 and 6 were again rearranged for γ and α respectively.

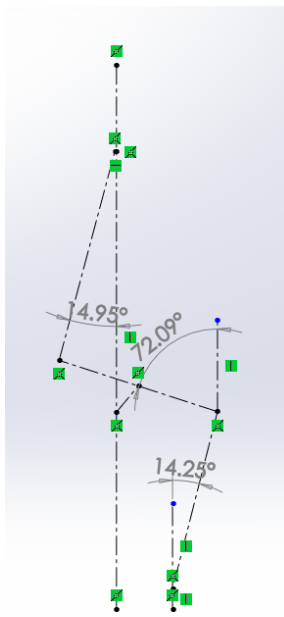
$$\gamma = \text{asin}\left(\frac{s' \cdot \sin \alpha + R \cdot \sin \theta - A_o}{T}\right)$$

$$\alpha = \text{acos}\left(\frac{C_h + R \cdot \cos \theta - T \cdot \cos \gamma}{s'}\right)$$

Equation 11, Equation 12; Final equations used for γ and α

These were then put into separate columns, referencing each other, and with iterative calculations enabled the results for both angles at each degree of Θ were calculated. This process was much faster than either the Goal seek or Matlab approaches, finishing calculations within 2 seconds. This method is not ideal as it will sometimes result in errors, so the Matlab script is then used. The calculated angles using this method match those calculated via Goal seek exactly and the maximum difference between the values calculated by excel and Matlab was 0.02° . The same step change of 1×10^{-4} degrees was used in both platforms, the exact method of determining the result through iterative calculations in excel is unknown, whereas in Matlab, a while loop was used to return a value when the equation returned a value of $0 \pm 1 \times 10^{-3}$. Inputting tighter tolerances and smaller steps offered no significant change in results with additional time required to calculate. See appendix for full Matlab script.

2.5.5 Validation



For validation, a representative line model was produced in Solidworks, figure . It was constructed with an angle mate on the crank shaft to allow adjustment through 360° of Θ , 20° increments were used. Chart 2 shows the calculated angles with the measured angles overlaid. This verifies that the calculated angles are an accurate representation of the motions of the system.

Figure 3; Representative solidworks model of linkage engine

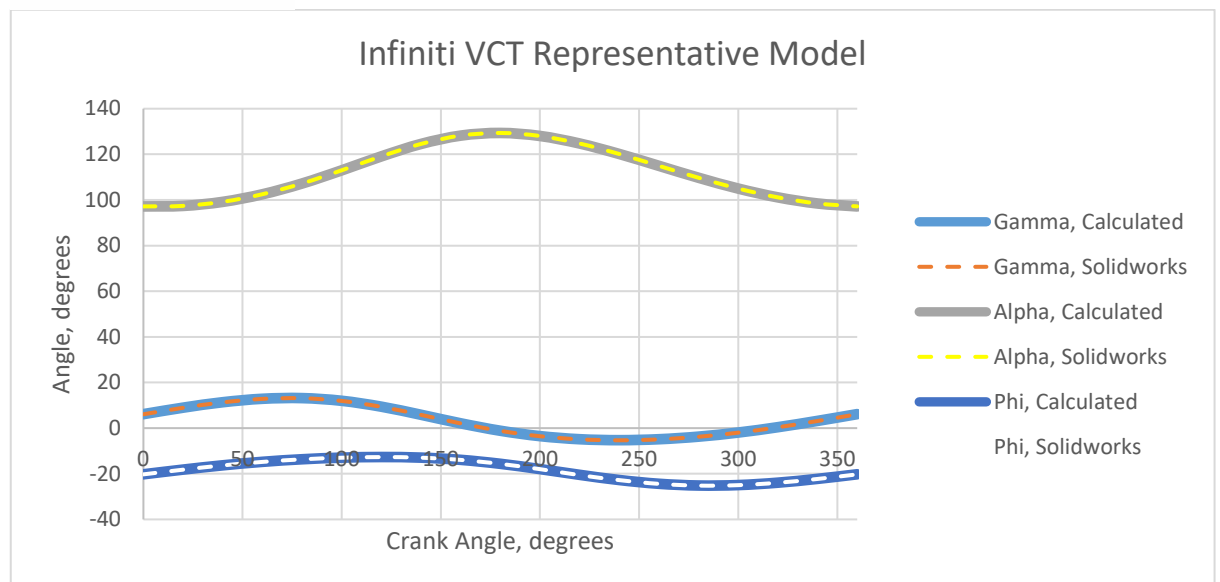


Chart 2; Linkage engine numerical model verification

	Gamma	Alpha	Phi
Minimum	0.032	0.001	0.028
Maximum	0.063	0.027	0.123
Average	0.052	0.012	0.039

Table 2; Variation of measured and calculated linkage engine angles

The above table (2) contain the maximum, minimum and average, difference between calculated and measured angles for the system, the angles were measured to 2 decimal points in Solidworks so the differences were formatted to three to show the correct accuracy.

	Calculated (Excel)			Solidworks		
Theta	Gamma	Alpha	Phi	Gamma	Alpha	Phi
0	6.13	97.16	-20.23	6.07	97.18	-20.26
20	8.97	97.38	-18.17	8.91	97.40	-18.21
40	11.35	99.16	-16.30	11.29	99.19	-16.34
60	12.86	102.49	-14.77	12.81	102.51	-14.80
80	13.16	107.20	-13.64	13.10	107.23	-13.68
100	11.98	112.98	-12.94	11.92	113.00	-12.97
120	9.40	119.10	-12.66	9.34	119.12	-12.69
140	5.85	124.55	-12.93	5.81	124.56	-12.96
160	2.11	128.20	-13.90	2.06	128.20	-13.93
180	-1.15	129.37	-15.63	-1.19	129.36	-15.66
200	-3.51	128.05	-17.95	-3.55	128.04	-17.98
220	-4.86	124.74	-20.47	-4.91	124.74	-20.59
240	-5.28	120.15	-22.74	-5.33	120.14	-22.78
260	-4.86	114.95	-24.39	-4.92	114.94	-24.43
280	-3.73	109.73	-25.15	-3.78	109.72	-25.19
300	-1.93	104.99	-24.96	-1.98	104.99	-24.99
320	0.41	101.13	-23.91	0.38	101.13	-23.94
340	3.18	98.46	-22.23	3.13	98.46	-22.27
360	6.13	97.16	-20.23	6.07	97.18	-20.26

Table 3; Measured and calculated linkage engine angles

2.5.6 3D Modelling

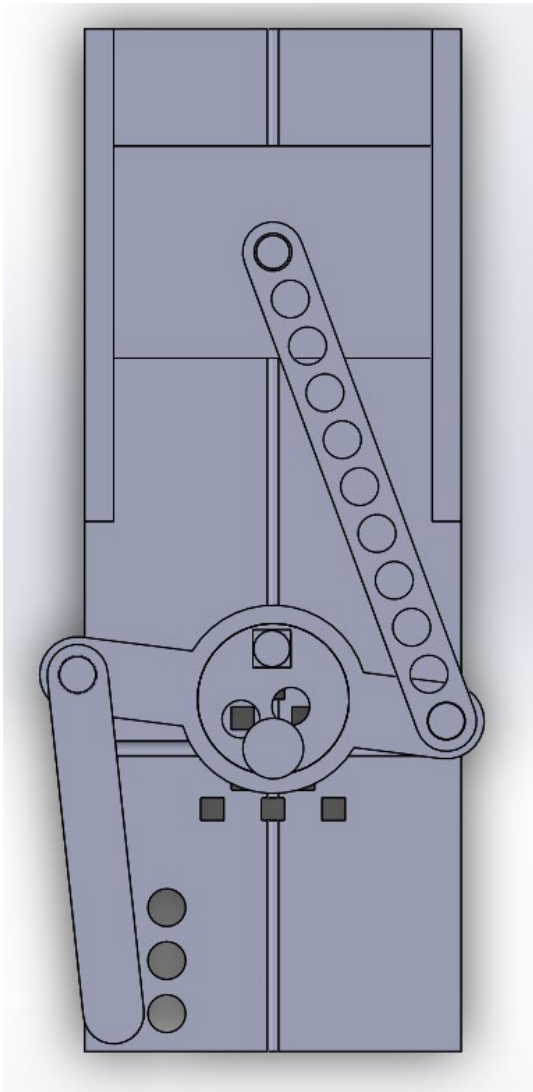


Figure 4; Representative CAD model, second version

A simplified model was produced in Solidworks and 3D printed to demonstrate the operating principles of the system, at $\frac{1}{2}$ scale of the measurements taken from the press release image. The process of modelling the system revealed inaccuracies in the measurements, so measurements that worked in the CAD model were then used to inform the numerical model. Once the first model was printed and assembled, issues with its operation, tolerances used, and construction were identified and a second representative model was produced with amendments to fix the issues found.

The tolerances initially used were 0mm for an interference fit and 0.5mm for rotating parts, it was assumed that these tolerances would be required to account for the inherent inaccuracies of the 3D printing machine being used (Ultimaker 3+). The parts produced using these tolerances required significant finishing and shaping for the interference fits and the rotating assemblies were found to be loose. The

revised tolerances were then taken from the Schaeffer Technical Pocket Guide (Schaeffer Technologies GmbH, 2014) and produced parts that required little or no adjustment to fit together.

The first model used threaded steel pins which were cut from bolts as a means of avoiding printing round parts parallel to the print bed so that the raft material would not need extra work on reshaping these parts to be able to rotate freely. On the second model, a redesign of the crank shaft, lower link and control rod meant that the additional pins were no longer required and all round surfaces could be printed vertically. Printing the round surfaces vertically reduce the amount of work required in removing raft material but did require additional finishing on some parts due to parts settling on the build plate, leading to a greater diameter at the bottom than the top.

The second model was an improvement on the first, though further work to simplify the design, adjust printing and finishing techniques and refine tolerances would produce a more useful demonstrator.

Once the numerical model had been verified and the Suzuki SV650 engine had been decided upon as the base engine, models of the crankshaft, connecting rods, pistons and crankcase were produced and made into an assembly so that the standard piston motions could be simulated and measured. The parts modelled to replace these kept standard bearing dimensions as the change in forces for each part was not known. The maximum space available was used to assess the available build space for further refinement.

To make the system work in a V layout, a single control rod and lower linkage group cannot be used as it produces different displacements on either bank (see chart 3). As such, the simplest solution possible was to attach one control rod and lower linkage for either bank in such a way that the lower linkages crossed the crankshaft in opposite directions. This was used as a first step to assessing the viability of the system in a V layout. The crankshaft throw was halved and the linkage parts were designed to be as short as possible whilst retaining bearing sizes and cross-sectional areas based on the standard SV650 parts.

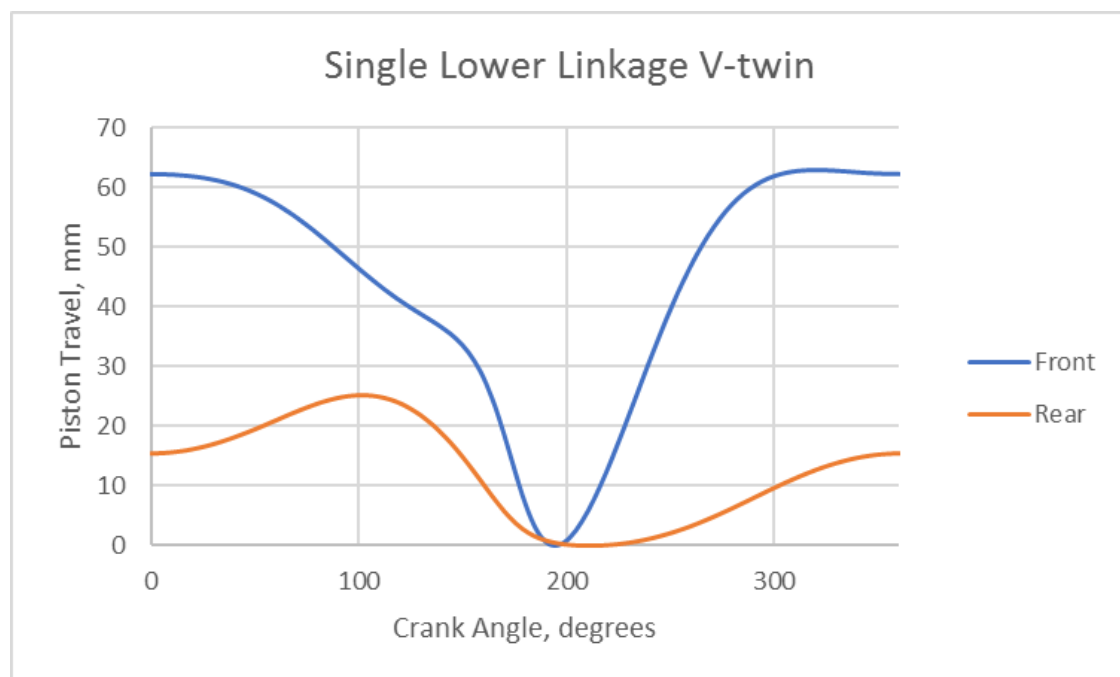


Chart 3; Piston motion of a single linkage group in V configuration engine

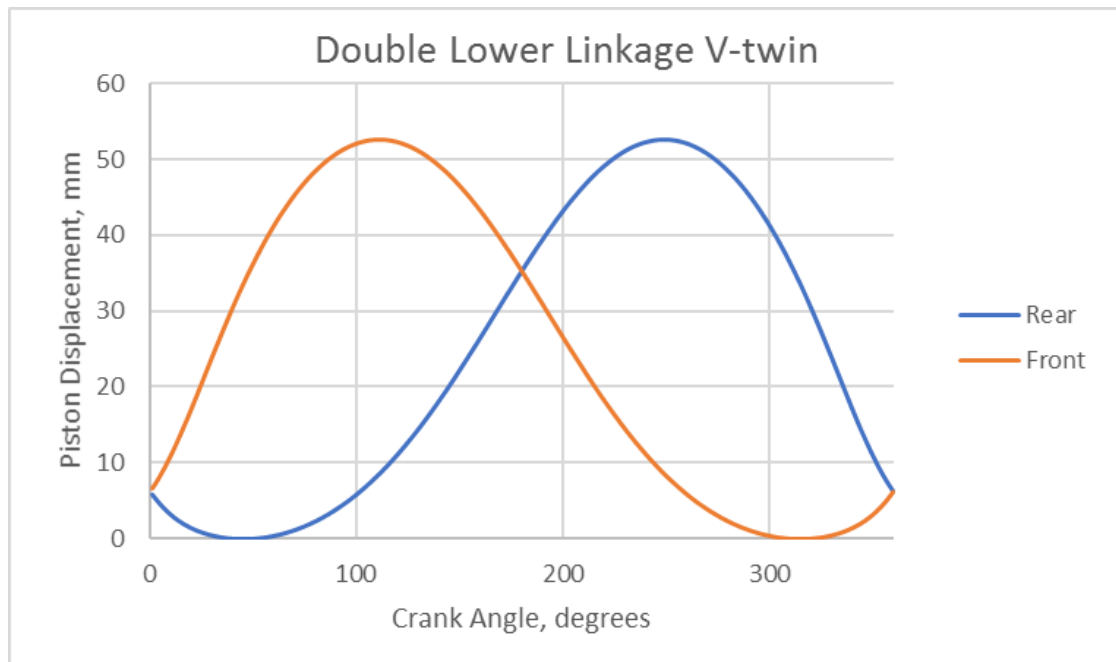


Chart 4; Piston motion of a V configuration linkage engine

This chart (4) shows the resultant piston displacement from the first iteration of the engine using a system with a separate set of linkages for both cylinders. It gave the same stroke on both, 52.7mm, before being optimised to match the standard stroke of 62.6mm and a suitable range of input at the eccentric to allow for the desired range of compression ratios. The correct stroke can be achieved by simply increasing the crank throw, however it can also be achieved by changing the ratios of lower linkage length either side of the crank pivot, with the added advantage of changing the ratio at which control shaft inputs affect the compression ratio so that it can be optimised either for finer control, lower required input force or a more compact system.

2.5.7 Fitting to SV dimensions

To optimise the dimensions for the SV650 engine, the limitations of the dimensions were first considered.

The rod length is limited by contact with the lower link at BDC and should be as short as possible. The length of the lower linkage must fit within the cases at all angles of Θ and allow enough material between the pin bores for strength. The control link's length is constrained by the need to allow the full motion of the system. The crank throw has no practical lower limit and its upper limit is based on the dimensional constraints of fitting the system in the engine.

These rules were followed to fit the parts into the Solidworks model, then the crank throw was adjusted until the standard stroke was matched. The standard con rod length of 119.5mm was

used for the upper link, the final crank throw was 17.14mm, the lower linkage was set to 60mm length with the crank pin bore in the centre, the control link length was 40mm. The vertical height from the control shaft to the centre of the crank was set to 35mm.

2.5.8 Motion Analysis

In addition to allowing for a variation of compression ratios and removing the need for a balance shaft, this system changes the piston's motion throughout each cycle. This change in motion is due to the path being traced by the lower end of the upper link (connecting rod) being near an ellipse as opposed to the circle of the standard engine (see chart 7).

This reduces the variation in connecting rod angle throughout the cycle, especially as the piston is moving down the bore. In the Infiniti engine, this reduces the rod angle and so the friction generated by gas forces during the expansion stroke. This change in motion also changes the pistons velocity, increasing the maximum velocities throughout the stroke, meaning higher inertial forces.

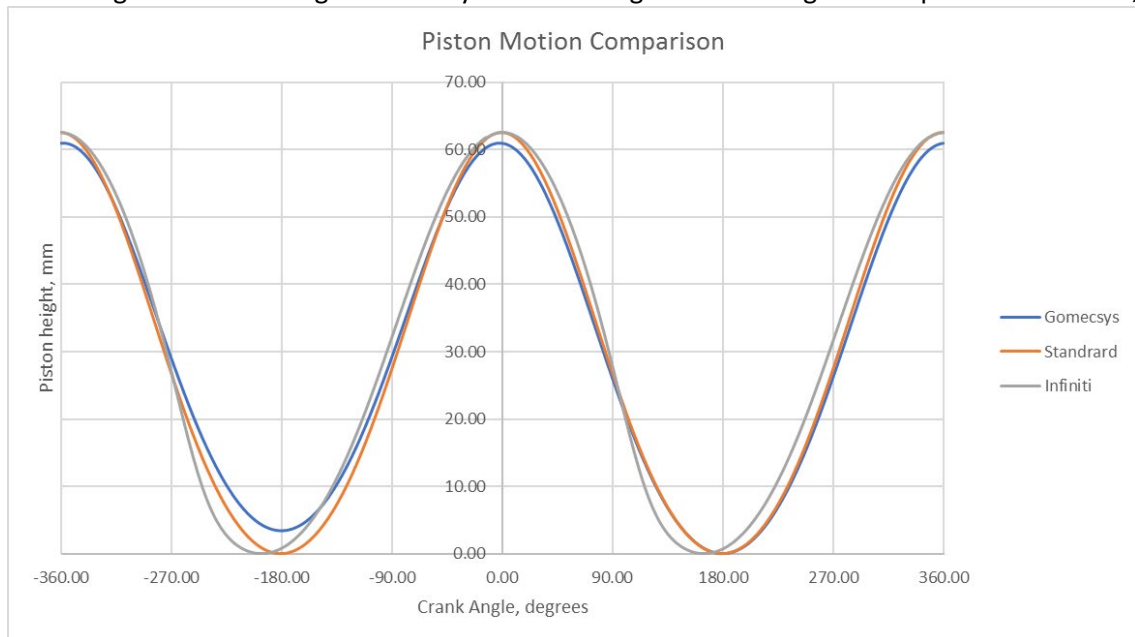
The piston motion is not a sinusoidal wave as per a conventional engine and the asymmetry is mirrored between the two banks due to the linkages being on opposite sides of the crank, so that the rotation is 'against' the linkages for the front cylinder and 'with' the system for the rear. This may be an issue for balancing unless the components can be made to satisfy the dimensions and ratios detailed in patent US 9,617,912 B2, though these ratios are based on their testing with an inline configuration and therefore may not be applicable to a V configuration. It is possible that adjusting the angles at which the counterweights shown in the patents are positioned upon the components may be enough to adapt them to the new layout. It may also cause issues for power deliver and engine management as the piston velocities will be asymmetrical, meaning that ignition and fuelling will need to be adjusted for either bank individually. The valve timing would vary between cylinders to account for the difference in piston velocities and difference in time between exhaust and induction strokes.

The difference in crank rotation in relation to both sets of linkages also means there is an asymmetry of con rod angles, where the system provides a low rod angle across the power stroke for the front cylinder, this is replaced with a high rod angle for the rear cylinder.

These differences between each bank may mean that a different approach to incorporate a linkage type system into the engine would be more suitable. Re-arranging the linkages may be the only option to make the system work in a V configuration.

2.6 Comparison of Piston Motions

As the two systems for varying compression ratio effect the relationship of the crank to the piston, they will both have effects on piston travel throughout the cycle, not just the height of TDC and BDC. These changes to piston motion will mean a change in piston velocities and accelerations, leading to changes in the inertial forces experienced by the reciprocating masses in the engine. These changes will likely lead to changes in the design of the piston assemblies,



as evidenced by Nissan's patents related to the system featuring pistons and engine management systems.

Chart 5; Piston motion comparison

This chart shows the piston motions for the standard, Gomecsys and Infiniti models. TDC for each system is aligned to 0° so that the phasing and position of BDC can be seen. This shows the key difference between the Gomecsys and standard models is not the phasing but the position of BDC in the bore. In this case the Gomecsys system is set for most over expansion, so the intake and compression strokes are shorter than the expansion and exhaust, allowing for greater extraction of the thermal energy produced during combustion. The system does not allow equal lengths across all four strokes, whereas the Infiniti system does not allow changes of length per stroke (although a patent was filed by Nissan for a means of gearing the input shaft to the cranks to achieve this). The Infiniti model shows that the difference between it and the standard system is the phasing of BDC, this was mirrored in chart 5 above which showed that the piston motion did not follow the sinusoidal wave of a standard engine. These changes in phase result in changes in piston velocity.

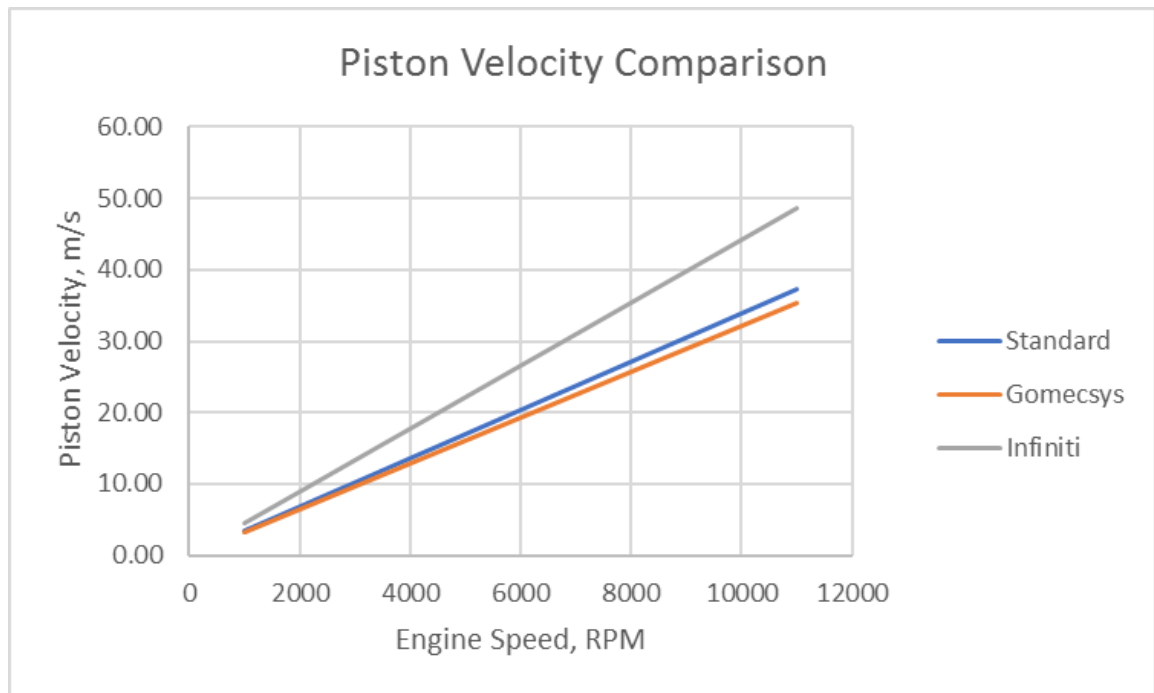


Chart 6; Piston peak velocity comparison

This chart shows the peak piston velocities (absolute) for each system up to the 11000RPM redline. The Gomecsys system reduces the piston velocity by reducing the distance being covered in a stroke, whilst a significant increase is seen in the Infiniti system due to its effect on the motion of the big end of the connecting rod/upper link. The difference in velocities is listed in the following table (4) and the following chart (7) shows the differences in the paths traced by the big ends.

	Gomecsys	Infiniti	
Velocity	35.25	48.73	m/s
Difference	-2.04	11.45	m/s
	-5.46	30.7	%

Table 4; peak piston velocity comparison

This table shows that the Infiniti system results in a 31% increase in piston velocity which will require a redesign of the piston and gudgeon pin to match the safety factors used before.

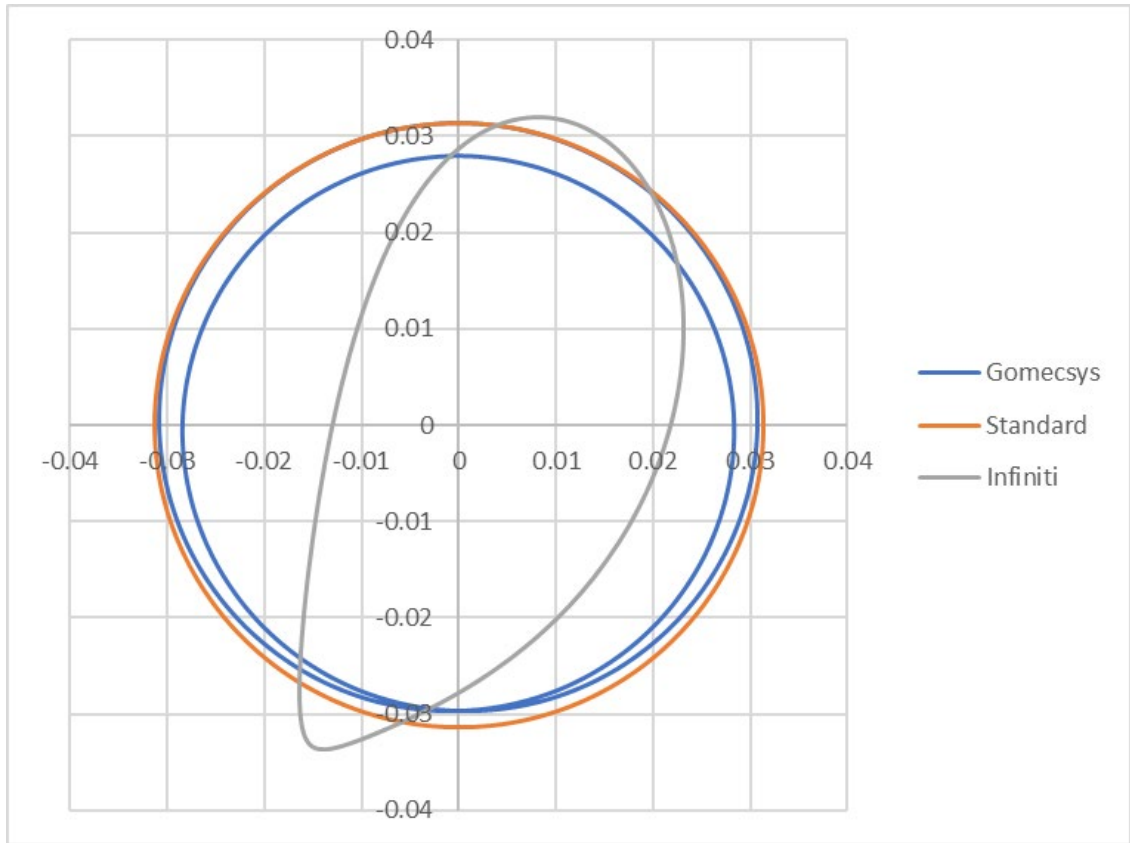


Chart 7; Big end motion comparison

These traces show the driving forces in the differences in piston velocity of each system. It is important to note that as the Infiniti trace is asymmetric, the left hand side is the downward stroke's path. This also shows some of the other benefits of the Infiniti system. The narrow path traced by the big end reduces con rod angle which reduces the amount of the thermal energy from combustion which is wasted on overcoming friction between the piston and cylinder. This is not true, however in the design used to fit the system to the SV engine, where the crankshaft is in line with the cylinder's axis. The crankshaft would need to be offset to centre the big ends motion around the cylinder's axis to take advantage of the reduced rod angle. The offset of this motion also means that in this arrangement, the big end has more vertical motion than the other two systems to achieve the same stroke, increasing parasitic losses. The Gomecsys also has the effect of reducing rod angle, but to a lesser extent. This trace also shows how the piston acceleration will be lower at TDC, this will mean that combustion occurs in a smaller volume over a longer period of time which may improve the effective expansion ratio of the combustion gasses at the expense of time under high pressure and temperature which may lead to issues with knock.

2.7 Conclusions

The Gomecsys system was studied to a basic level as an expedient means of understanding non-traditional engine assemblies and to inform the development of the numerical model of the Infiniti system. When compared to other systems studied though promising in its simplicity and means of operation, the lack of a production version and concerns over the longevity of the gear teeth in the system meant that it was considered not to be worthy of further investigation.

The Infiniti system is in production and may be a sign of the direction of engine development to come if it proves reliable. It has several benefits above being able to vary compression ratios and a version geared to allow for over expansion may be produced in the future, however its complexity and limitations due to increased piston velocities and possible cooling issues may limit its range of applications.

3 Cylinder Deactivation

3.1 Introduction

For this section of the project, cylinder deactivation systems were researched with the goal of producing a final design based on the SV650 cylinder head and valve train assembly. One of the core targets of this part was to assess whether any of these systems could be feasibly incorporated without significant changes to the cylinder head itself. Several designs were developed and assessed until it was determined that this was not possible.

3.2 Overview/systems

Cylinder deactivation strategies currently in production keep all valves in the cylinder closed during deactivation (Halvorson, 2018). This negates the pumping losses that go with an open cylinder, which are especially high at the operating conditions when the cylinder will be inactive (i.e. low load, part throttle cruising). This comes at the expense of some negative work being done expanding a closed volume of gas in the cylinder however the negative work done here is largely reclaimed as the piston returns, when the trapped gas acts as a spring. This means that the work done in moving the piston in the deactivated cylinder is lower than if the valves were left in operation, increasing the overall thermal efficiency of the engine. Simply cutting fuel does not have this effect and will in most cases decrease the overall efficiency (Kortwittenbourg & Walter, 2013).

Several systems are currently available in production vehicles. Production systems decouple the cam lobe from the rest of the valve train by some means or separate cam lift from valve lift. Though electronic valve actuation systems in development such as Koenigsegg's Freevalve

system can be programmed to run cylinder deactivation in addition to variable valve lift, these systems were not considered as an approach based around current mass production technologies was being sought (Hu, Chang, Xu, & Liu, 2018).

Other systems not considered in this project include Chevrolet's Active Fuel Management system as this relies on the space afforded by using pushrods as part of the valvetrain, though it may be feasible to implement it by increasing the distance between camshaft and valve, this was deemed an unacceptable compromise on the design goal of minimal change to the existing engine. It uses a hydraulic cam follower between the camshaft and pushrod which switches between acting as a solid unit to a soft spring which absorbs the motion of the cam's surface without imparting it upon the pushrod (Stabinsky, et al., 2007).

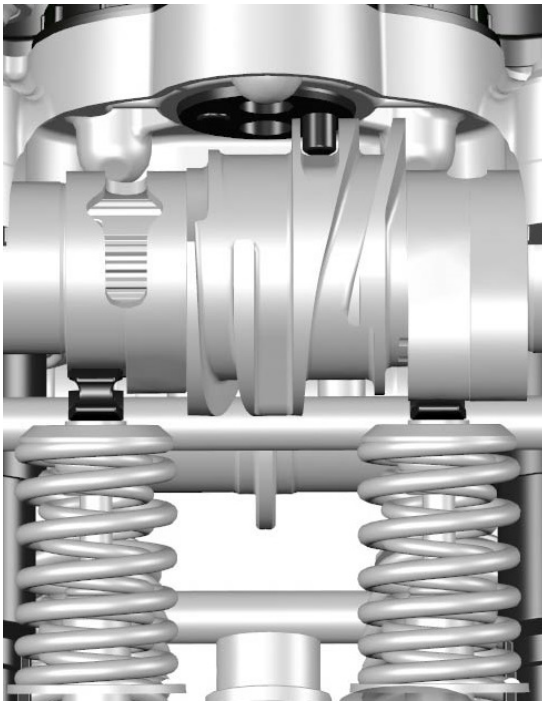
Another system not considered is the Delphi Skip Fire system currently used by Ford which uses a hinged follower in which a pin latches both halves together to allow valve actuation, and removes the pin allowing the central roller to follow the path dictated by the cam lobe without transferring this to the valve (Halvorson, 2018). This system appears to be relatively low impact in terms of changes required to the basic design of the valvetrain assembly (assuming rockers are already being used and sufficient space) as it may be possible to incorporate simply by changing to this design of rocker and adding a method of supplying signal.

Both systems are not mechanically timed to valve events, meaning that the valves may be open when the system is activated if the programmed timing is not accurate, reducing the consistency of cylinder contents at the time of deactivation.

The most promising system was deemed to be that which is currently used by Volkswagen for cylinder deactivation and BMW Motorcycles for variable valve lift.

3.3 VW/BMW style system

3.3.1 Overview



This system uses splined camshafts with lobes which can slide along the shaft to move them out of contact with the rockers. As these sliding lobes are being rotated by the splines, with contact against the rockers opposing their rotation, they can only be moved along the shaft when not in contact with the rocker. This requires that the motion along the shaft is timed to only occur when the valves are closed, with a failure of this timing possibly leading to engine damage. The actuator pins drop in to one of the two slots in the centre of the figure 4, this causes the whole assembly to slide along the cam shaft, switching lobes (RHS of figure 4).

Figure 5; BMW Shiftcam system (BMW, 2018)

3.3.2 Initial ideas and packaging Issues

This system was selected as the basis for this part of the project as it appeared the simplest to implement in a bucket and shim arrangement within the existing cylinder head design. It also was the only system researched that was inherently timed to the valve timing events, meaning greater predictability of behaviour in use, reducing the impact to driver experience.

Initially, representative CAD models of the standard cylinder head and camshafts were produced as a basis to begin modified designs. The measurements gathered in this process soon revealed that it would not be possible to closely follow the designs of VW; once a splined camshaft with the necessary bearing faces had been produced, measurement of the available space in which the lobe would be able to slide ruled out the possibility of having the track for the actuation pin on the lobe part itself (see figure 6). As the buckets are 25mm in diameter and the cam lobe 9mm in width, the track would need to move the lobe 17mm. This would leave the lobe assembly being at least 27mm wide if the lobe width was kept to match the contact force and assuming a wall thickness of 1mm would be enough on the track. The available space for the lobe assembly on the splined shaft was 34.4mm, leaving 7.4mm of space for actuation.

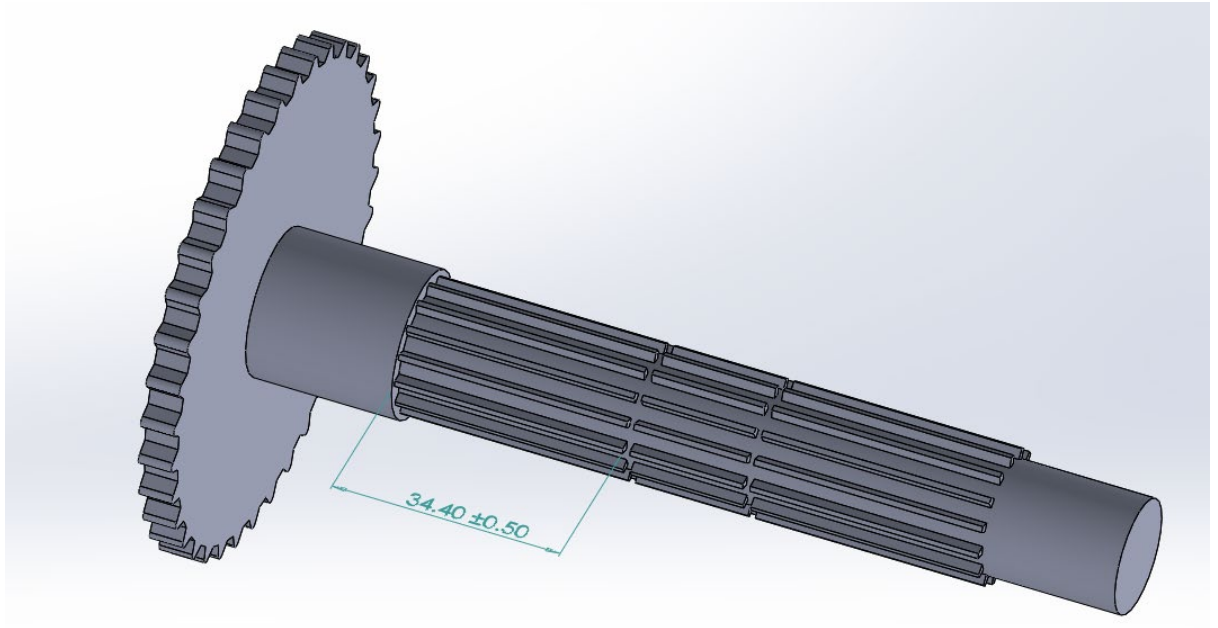


Figure 6; CAD model of splined camshaft

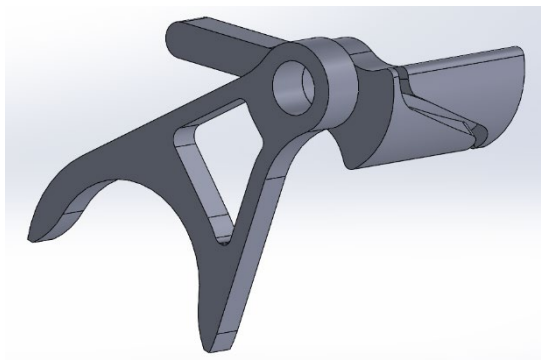
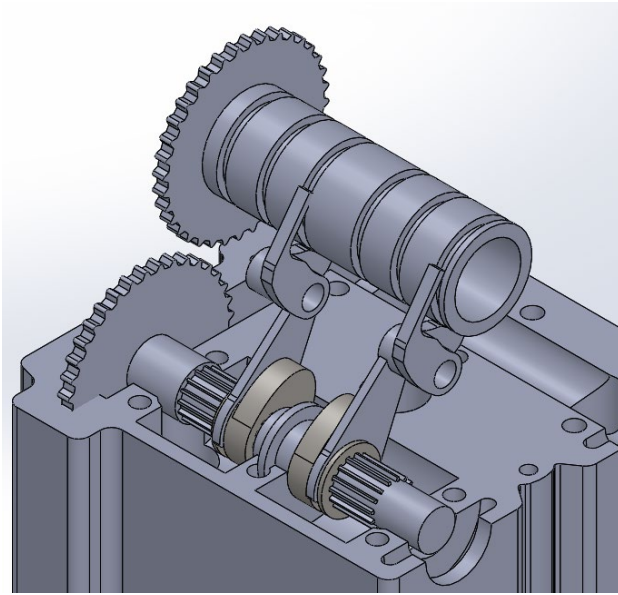


Figure 7; Lift actuated valve sliding mechanism

A slip ring with an actuation fork as typically found in a motorcycle gearbox was chosen as a method of reducing the width of the lobe part and this was found to be an acceptable solution to fit the lobe into the space at both the extremes of the necessary travel. The method of sliding the lobe part and timing it to valve timing events then needed to be redesigned.

The first solution considered consisted of a follower (top of figure 7) pivoting on the slider fork that would be engaged by the actuator pin. This would limit the arc of the sliding cam's slotted face (to the right of figure 7) to the ratio of follower length multiplying the cam lift.



The second solution moved the slots for the actuation pins on to a separate barrel cam (figure 8), similar to the selector drum found in a motorcycle gearbox. This also simplified the sliding fork design to a single piece that would mount onto sliding rails that would be raised or lowered to select which pin was interacting with the barrel cam and thus the direction of cam lobe slide.

Figure 8; Cylinder deactivation system with separate actuation cam

4.3.3 Forces and accelerations

Had the decision been made to continue with the development of this system, the next step in development would have been to calculate the forces that would be seen at the actuation pins. Initial calculations were made after the first iteration was designed, assuming that the maximum RPM the system would be active at would be 6000rpm to coincide with 80mph in 6th gear on the SV650, the time available to slide the lobe whilst not in contact would be 0.01 seconds.

$$T = \frac{1}{2} \times \frac{60}{3000} = 0.01s$$

Equation 13; Time available for cam slide actuation

The acceleration required to cover the 17.5mm distance to move the lobe out of contact with the bucket was calculated from;

$$a = \frac{s - ut}{\frac{1}{2} \times t^2}$$

Equation 14; Sliding lobe acceleration

Giving 350m/s². With the sliding lobe being assigned steel as a material in Solidworks, its mass was 47.7 grams. This was used to calculate the required force for the lobe from;

$$F = m \times a$$

Equation 15; Newton's second law of motion

And a force of 17.17N was calculated. This seems like a reasonable figure for the system to deal with, however more calculations to include the full assembly would be required to judge the viability of it if the decision had not been made to switch to rocker actuation. Especially regarding the strength of the actuation pins.

3.4 Conclusions

As the initial goal was to design a system which would necessitate the least modification to the base design of the engine, systems which required the least changes to the original equipment were first considered. This meant retaining bucket and shim valve actuation, which later proved to be impractical due to the travel of the cam lobe necessary to clear the bucket. This would have required significant changes to the cylinder head. As such, it was decided that a change to rocker valve actuation would be a suitable solution, leading on to the final part of this project; development of a numerical model of valve lift from cam profile with the use of rockers.

An important factor not considered in this part of the project is the affect it might have on engine sound. This is of lower importance on most small cars where cylinder deactivation is currently found, but on a motorcycle may not be accepted by end users as noise and performance are generally much stronger influences than economy and emissions. Trends in the market towards such demands may however make this an inevitable development.

4 Camshaft Profile and Valve Lift

4.1 Introduction

Valve lift profiles are an important part of engine development, heavily influencing engine characteristics, efficiency and longevity (Blair, 1999). As the proposed cylinder deactivation system using sliding cam lobes are reliant on a change from bucket and shim type actuation to rockers, an effect on valve lift profiles and hence camshaft lobe design is inherent and must be understood to be able to arrive at a suitable final design.

Due to the geometry of the surface of the cam lobe, the actual valve lift at a given point is not simply that of the lift measured on the cam's surface above the base circle diameter. Therefore, it must be modelled to understand the changes brought about by changing from bucket and shim to rockers. The models presented here were built in excel, though an alternative method such as Matlab may have proved more suitable to aid accuracy and automation of processes, especially in further work to return a cam shaft profile from a valve lift profile.

4.2 Cam Shaft Measurement

The cam's profiles were measured directly by positioning them in the optical dividing head to allow more accurate rotation measurement than a degree wheel. A dial test indicator was used to measure the lift at each degree of cam rotation. This resulted in a set of cam lift measurements against cam angle, Θ . These range from 0 to 8.7mm for inlet and 0 to 7.3mm for exhaust.

4.3 Bucket and Shim

A bucket and shim arrangement is used as standard on the SV650 engine (see figure 9). This places the cam lobe (1) directly above the valve (4), with a shim (3) underneath a steel bucket (2) as viewed from normal to the valve stem. The bucket allows enough surface area above the stem to account for the changing position of the contact point laterally about the centre of the stem. This change in position dictates the difference between the cam and valve lift profiles.

The contact point in this system is the lowest point on the surface measured vertically from the axis of rotation of the cam, regardless of its horizontal position in relation to this axis.

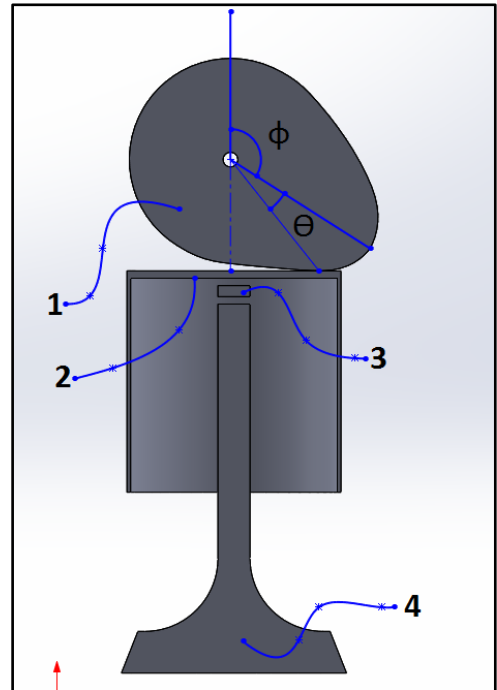
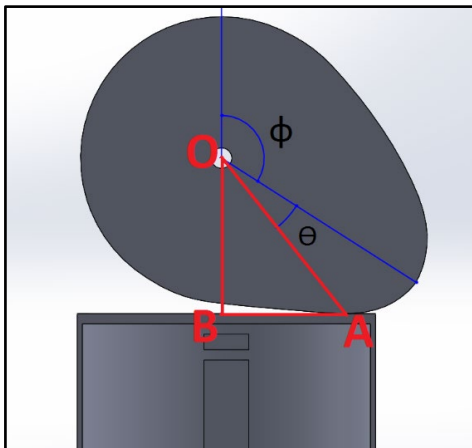


Figure 9; Bucket and shim arrangement cross-section



This necessitates the calculation of the vertical position of each measured degree of Θ at any given angle of cam rotation, ϕ . This was achieved by creating a matrix of 0 to 360 for both Θ and ϕ , multiplying the distance from the centre of the cam to the surface (length OA in figure 10) by the cosine of the angle from vertical in relation to the valve stem

(length OA being the sum of the measured cam lift and the radius of the base circle).

$$OB = OA \times \cos (180 - (\phi + \theta))$$

Equation 16; Lift calculation

Figure 10; Bucket and shim arrangement trigonometry diagram

From this, the greatest value in each column of ϕ can be returned as the longest possible length of the side OB of the triangle. Once this overcomes the valve clearance value (ignored in this case for simplicity, but easily implemented), the lowest value in the series can be removed from each in turn to determine the lift value for each degree of ϕ .

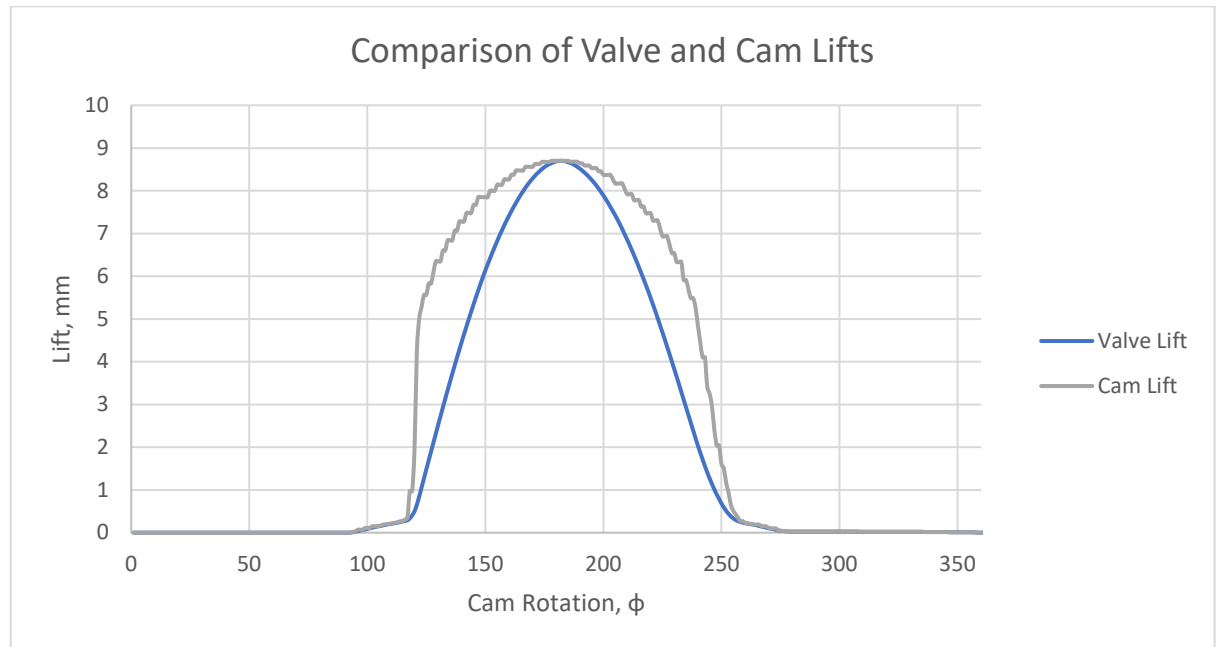


Chart 8; Cam and Valve lift comparison, bucket and shim

Chart 8 shows the comparison of valve and cam lift in the standard bucket and shim arrangement. It shows the clear difference in the rate of change of lift due to the horizontal displacement of the contact point. This chart shows a narrower margin between cam and valve lift after peak lift, this is due to the asymmetry of the cam profile used to create a lower velocity on valve closing to promote valve and seat longevity (Lewis, 2000).

It also shows that as this displacement reaches 0 at the nose of the cam (i.e. maximum lift), both lifts match. This would not be true if valve clearance had been accounted for, the difference at $\phi=180^\circ$ would then be the clearance value.

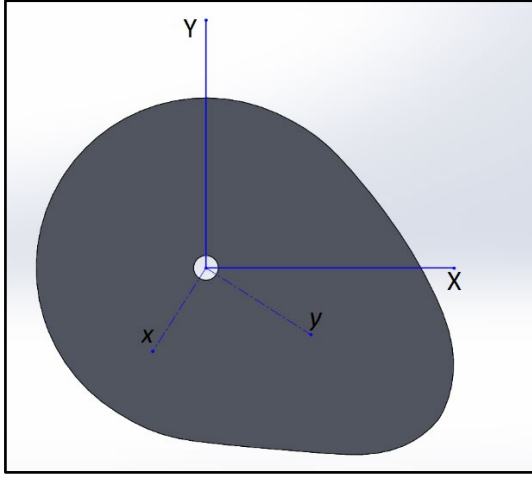


Figure 11; Local and global coordinates for cam lobe

4.4 Rocker

As the slope of the rocker surface changes as a function of lift, the contact point is no longer simply the lowest point on the cam's surface but is where the cam and rocker surfaces have same slope and are at the same position in a global coordinate system. For both the local and global coordinates systems, the cam's centre is used as the origin (see figure 11). Therefore, from the local x and y coordinates, all possible global X

coordinates for the cam's surface must be calculated for each degree of ϕ . The global Y coordinates must also be calculated to allow the calculation of the slope at each of these points. This proved to be a major hurdle in terms of automating the process of mapping valve lift from cam profile as matrices of 360 degrees were used, limiting the accuracy by fixing the degrees of ϕ to whole degrees.

Local Coordinates;

$$x = OA \times \sin \theta$$

$$y = OA \times \cos \theta$$

Equation 17, Equation 18; Cam local coordinate calculations

Global Coordinates;

$$X = (x \times \cos \phi) + (y \times \sin \phi)$$

$$Y = (y \times \cos \phi) - (x \times \sin \phi)$$

Equation 19, Equation 20; Cam global coordinate calculations

Slope;

$$Slope = \frac{\Delta Y}{\Delta X}$$

Equation 21; Cam surface slope

4.5 Methodology

Two functional methods of determining the valve lift and contact point were found. The first required manual alignment of two lines representing the rocker slope (one at a given lift value, the other at a division of the rocker's length) on a chart of cam slope against X coordinates. The

second automatically returned both lift and contact point coordinates by linking rocker slope to ϕ .

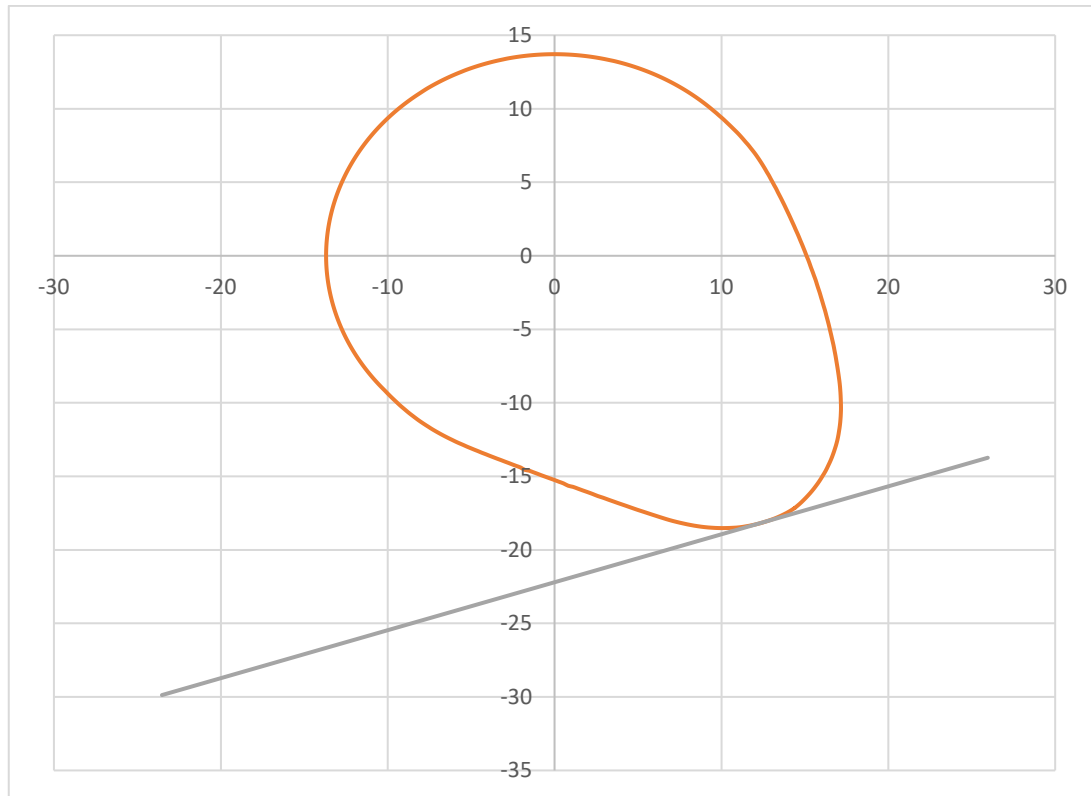


Chart 9; Rocker and Cam profiles

For method 1, to determine the lift value both cam and rocker XY coordinates were charted (chart 9) so that ϕ and lift could be altered. This allowed a visual alignment of their lines. This then determined the position of the rocker's slope against X line, at which point the division of rocker length could be adjusted until the curve of rocker slope against lift at a point along its length coincided with the intersection of the slope against X lines for both cam and rocker (chart 10). This method was not only time consuming but also introduced the possibility of user error and uncertainty. This uncertainty was around intersection alignment due to the limitations of excel's chart controls, as can be seen in figure 12 where the length of intersection is relatively long, instead of a single point of interaction as in an ideal situation. This meant that the alignment had to be made in terms of the best match over the largest area available with the divisions of lift selected. This method did however produce reasonably accurate results when benchmarked against the model in Solidworks, measured at steps of 10 degrees of ϕ from 90 to 270, with a maximum of 0.09mm and average of 0.03mm difference.

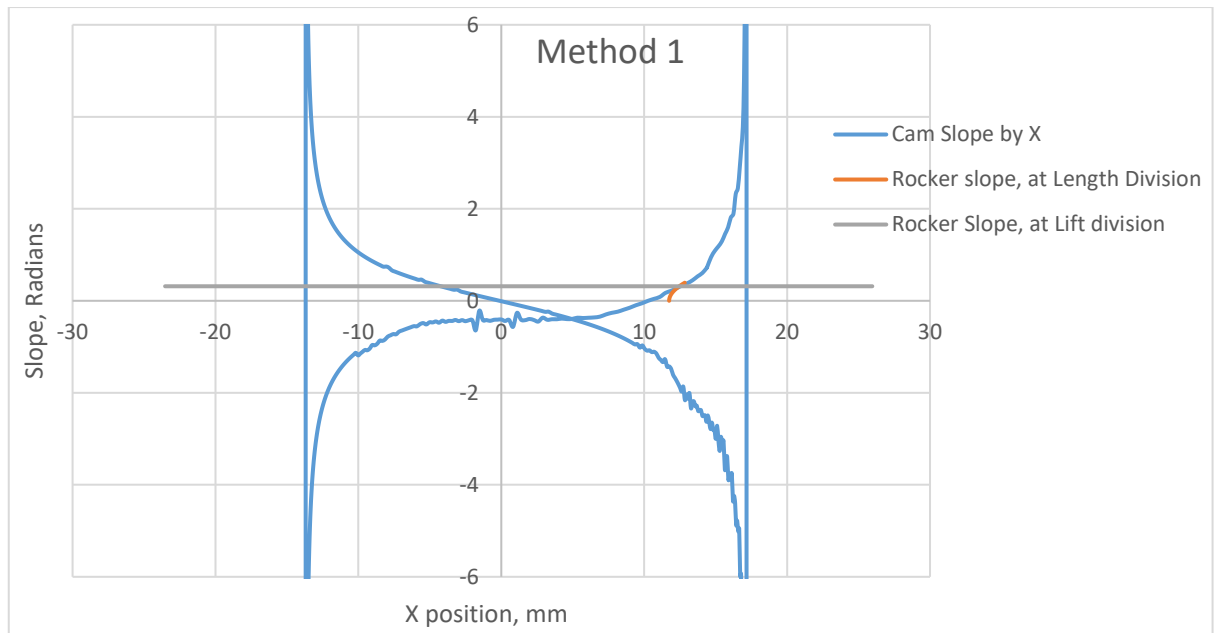


Chart 10; Cam and rocker slopes

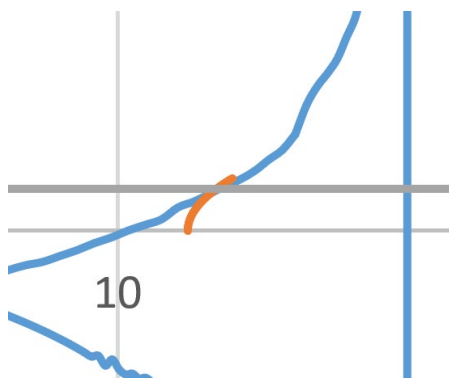


Figure 12; Close up of chart 10 intersection

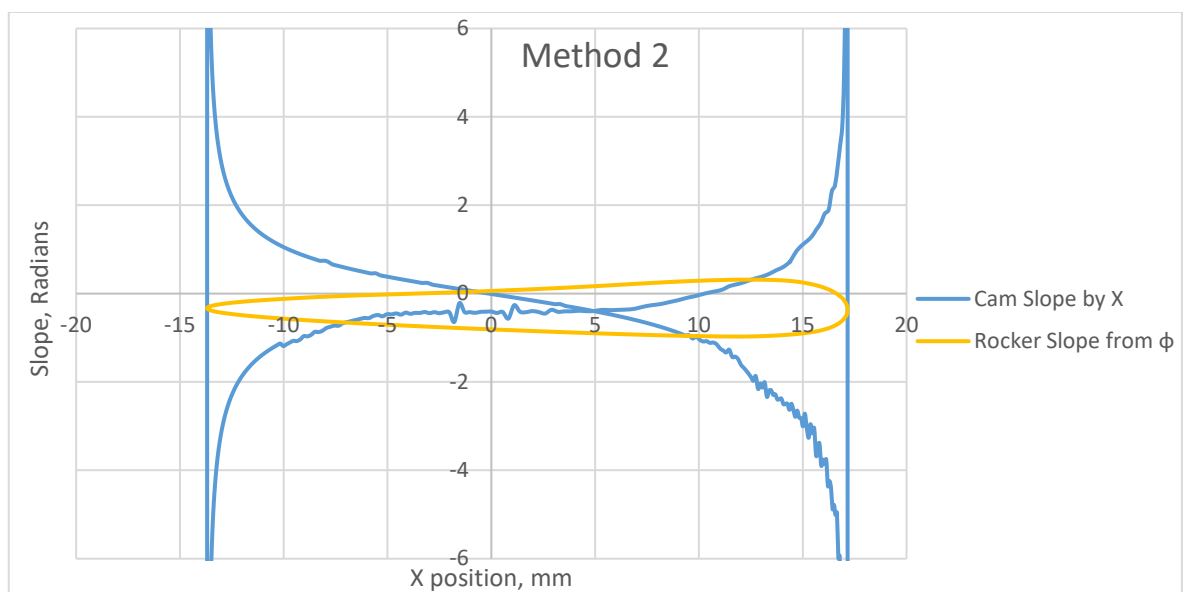


Chart 11; Cam and Rocker slopes, method 2

The second method implemented linked the change in ϕ with rocker slope (α), hence lift, by using the assertion that the global coordinates must be the same for both cam and rocker and the point of contact so that given each XY coordinate of cam surface was known for any degree of ϕ , this allows the slope of the rocker to be calculated from the magnitude of its X and Y displacements.

The slope of the rocker as a function of cam angle is calculated by making a triangle in global coordinates using the pivot point of the rocker as the origin and the cam's global coordinates to determine the opposite and adjacent sides to the angle. This gives;

$$\text{Rocker Slope} = \text{atan}\left(\frac{Y_{\text{pivot}} - Y_{\text{cam}}}{X_{\text{pivot}} - X_{\text{cam}}}\right)$$

Equation 22; Rocker slope from cam surface coordinates

=ATAN((\$B\$3-'Cam Y 360 matrix'!D5)/(\$B\$2-'Cam X 360 matrix'!D5))												
	A	B	C	D	E	F	G	H	I	J	K	L
1	Rocker len	52.2 mm										
2	Pivot X	26.1 mm										
3	Pivot Y	-13.69										
4												
5												
6												
7		Phi →										
8	Theta ↓	0	1	2	3	4	5	6	7	8	9	10
9	0	-0.94454	-0.95164	-0.95873	-0.9658	-0.97284	-0.97987	-0.98689	-0.99388	-1.00085	-1.0078	-1.01472
10	1	-0.95131	-0.95839	-0.96545	-0.97249	-0.97951	-0.98652	-0.9935	-1.00046	-1.00741	-1.01433	-1.02123
11	2	-0.9577	-0.96475	-0.97178	-0.97879	-0.98578	-0.99275	-0.9997	-1.00663	-1.01353	-1.02042	-1.02728
12	3	-0.96384	-0.97085	-0.97784	-0.98482	-0.99177	-0.9987	-1.00561	-1.0125	-1.01936	-1.0262	-1.03302
13	4	-0.96964	-0.97661	-0.98355	-0.99048	-0.99739	-1.00428	-1.01114	-1.01798	-1.0248	-1.03159	-1.03836
14	5	-0.97514	-0.98207	-0.98897	-0.99585	-1.0027	-1.00954	-1.01635	-1.02314	-1.0299	-1.03664	-1.04335
15	6	-0.98042	-0.98729	-0.99414	-1.00097	-1.00777	-1.01455	-1.02131	-1.02804	-1.03474	-1.04142	-1.04807
16	7	-0.98538	-0.9922	-0.99899	-1.00575	-1.0125	-1.01922	-1.02591	-1.03257	-1.03921	-1.04582	-1.0524
17	8	-0.99001	-0.99676	-1.00348	-1.01019	-1.01686	-1.02351	-1.03014	-1.03673	-1.0433	-1.04984	-1.05634
18	9	-0.99436	-1.00104	-1.0077	-1.01433	-1.02094	-1.02751	-1.03406	-1.04058	-1.04707	-1.05353	-1.05996
19	10	-0.99809	-1.00469	-1.01127	-1.01782	-1.02435	-1.03084	-1.03731	-1.04374	-1.05014	-1.05651	-1.06284

Figure 13; Excel table for rocker slopes

A HLOOKUP function is then used to return the slope of the cam and rocker at a user input degree of ϕ . An if statement is used to determine the difference between the cam and rocker slopes for each degree of Θ (at the specified ϕ) only when the slope of the cam's surface is positive (as dictated by the rocker slope only being positive in this instance). Another column is then populated with a logical test to determine at which angle of Θ the best match in slope occurs. This step could be cut out in later developments, especially to minimise the number of steps used in this process to make it feasible to implement a single sheet determining the match for each degree of ϕ but is currently necessary for error correction as discussed further on. This column is then used to return the calculated rocker slope at the crossover point via a nested index and match function.

Once the rocker slope, α , has been returned, it can be used to calculate the lift by using trigonometry with the length of the rocker as hypotenuse and lift as the opposite side to the slope angle;

$$\text{Valve Lift} = \text{Rocker Length} \times \sin \alpha$$

Equation 23; Valve lift from rocker slope

=Rocker!\$B\$4*SIN(K2)						
G	H	I	J	K	L	M
ϕ	140	Degrees	Lift	16.19	mm	
			alpha	0.315259416	Radians	
Theta	Cam Slope	Y	X	Rocker Slope from ϕ	Slope Difference	Match finder
0	0.722219733	-17.15173508	14.39201458	0.28748235	0.434737382	0
1	0.605447209	-17.38086943	14.07475055	0.297799335	0.307647875	0
2	0.53197157	-17.58445997	13.73848582	0.305203729	0.226767841	0
3	0.447972284	-17.7696401	13.39038427	0.310599339	0.137372945	0
4	0.383372795	-17.93186168	13.02826012	0.313785281	0.069587514	0
5	0.335010428	-18.07458986	12.65596407	0.315259416	0.019751012	1
6	0.275169825	-18.20151991	12.2770802	0.31547925	0.040309425	0
7	0.2173959	-18.3081785	11.88947013	0.314216388	0.096820488	0
8	0.17303871	-18.39416321	11.49394884	0.311579059	0.138540349	0
9	0.084361607	-18.46338366	11.09392013	0.307975436	0.223613829	0
10	0.041760433	-18.49830262	10.68	0.302267506	0.260507073	0
11	-0.000688431	-18.5156992	10.26341966	0.295780063	n/a	0
12	-0.053453723	-18.51541102	9.844818672	0.288567804	n/a	0
13	-0.096020278	-18.49284041	9.422572822	0.280397269	n/a	0

Figure 14; Excel sheet for rocker lift

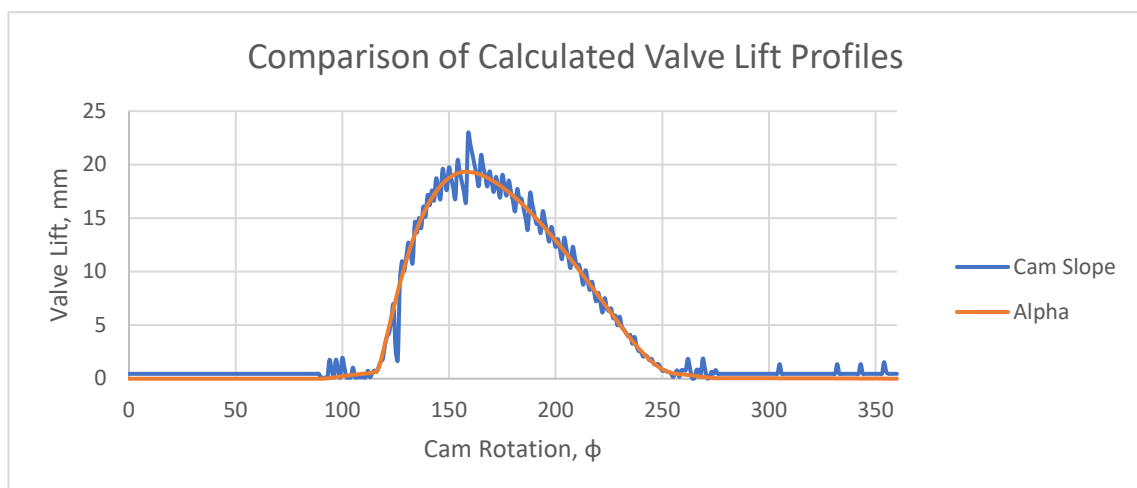


Chart 12; Comparison of results from rocker and cam surface slopes

The calculated rocker slope is used here rather than the cam slope as it provides much less noisy data, the noise arising from the use of cam slope appears to be carried forward from cam profile measurement. Chart 12 shows a comparison of the graphs produced from both slopes.

Method 2 has proved to be more accurate and substantially less time consuming, with a maximum error of 0.06mm and an average of 0.02mm. It has also been automated for all 360 degrees of ϕ , though it does occasionally return erroneous data. This is due to a circumstance where there are two intersections when both slopes are positive, and the erroneous intersection has a closer slope match than the correct one. So far, this has only affected a maximum of two degrees of ϕ , so has been easily rectified but this may become more of an issue if it occurs more frequently when using other data and may prove to be an issue if the valve lift profile is to be used as the driver for the model. The automation for all degrees of ϕ is currently achieved via a simple macro which iterates the input degree from 0 to 360 and copies the results into the output column (see figure 15), this was done as an expedient measure as re-writing the excel to do this on another matrix page would likely have been too time consuming at this stage in the project. This is not an ideal solution as it is comparatively slow (around 40s to run), acting as a limiting factor in the speed at which new cam profiles can be evaluated in the current model and as a barrier in the process of modifying or re-writing the model to allow for returning cam profile from valve lift profile.

```

Dim i As Long
Dim n As Long
For i = 0 To 360
    n = i + 3
    Sheets("Slopes @Length").Select
    Range("H1").Select
    ActiveCell.FormulaR1C1 = i
    Range("K1").Select
    Selection.Copy
    Sheets("Lift profile").Select
    Range("B" & n).Select
    Selection.PasteSpecial Paste:=xlPasteValues, Operation:=xlNone, SkipBlanks _
        :=False, Transpose:=False
    On Error Resume Next
Next
End Sub

```

Figure 15; Excel macro

4.6 Validation

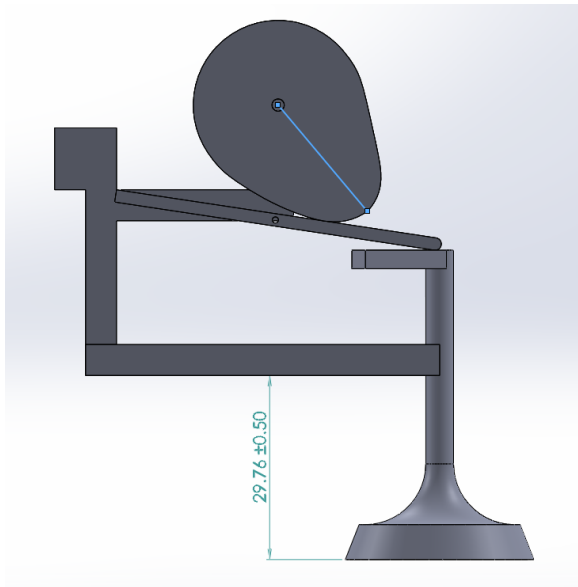


Figure 16; Solidworks model for returning valve lift

To validate the lift values being returned by the excel model, a representative model was prepared in Solidworks. This model used the measured cam lift data to create the lobe profile by using the 'curve through XYZ' tool and the local x and y coordinates used in the excel model. Initially, an angular separation mate between an upright element on the simple fixture piece used as a base for the assembly and a centreline drawn on the cam lobe (highlighted in blue) was used to set the cam rotation to a specified angle (set to 140° in figure 16), this was repeated at 10°

intervals between 90° and 270°. This data was initially compared to the output of the excel model for validation. This assembly was then used as a basis for a motion simulation which was used to output continuous lift values which were then compared. This simulation was carried out at a very low RPM as it was only intended to validate the basic motion of the valvetrain and not any issues expected at higher RPM sites.

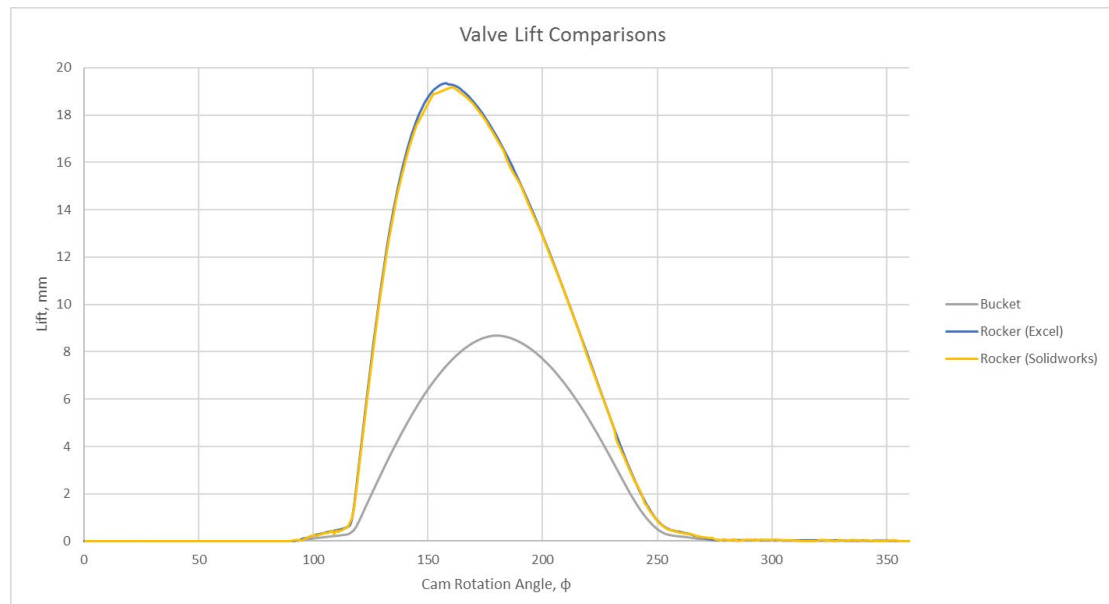


Chart 13; Valve lift model verification

This chart (13) compares the valve lift profiles from the numerical models for both bucket and shim and rocker systems, and the Solidworks motion simulation of the rocker system. The

differences between the numerical model and Solidworks motion simulation suggest that the error lies in the motion simulation, this was expected due to inexperience with the software package used. The differences in trace suggest that the Solidworks is incorrect, the numerical model produces a smooth curve as would be expected of a rigid system and the Solidworks data shows variation throughout the range of motion, especially around maximum lift and at changes in velocity. This suggests that the arrangement of springs and dampers used to keep the parts of the assembly in contact throughout the simulation were not acting as expected. Further time spent refining the simulation would likely reduce these issues and return a better match between the two methods. These observations are backed by the correlation of the static Solidworks test data, where the lack of simulation means that objects in the assembly are treated as solid bodies and without a physics model, meaning that the parts will not come out of contact during motion as they are fully constrained by the mates built into the model.

These observations suggest the necessity of development work centred around further effects on motion under real world conditions; i.e. valve spring rates, parts tolerances and clearances, flex in the system. These effects are beyond the scope of the project at this stage, as this model is only intended to study the basic relationship between cam lift and valve lift.

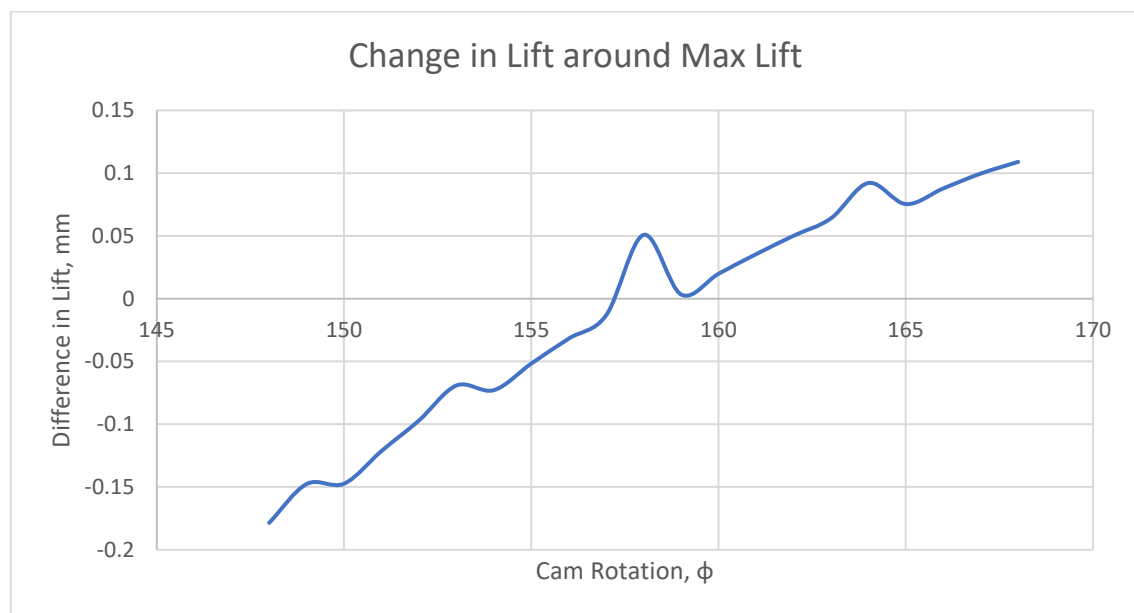


Chart 14; Change in Lift around max lift

There is also a small but noticeable step in the trace as the lift passes its maximum value on the numerical model's trace. This equates to a step of 0.05mm of lift between the degree of maximum lift and the one immediately after, 158° and 159° respectively. Chart 14 shows the change in lift per degree of cam rotation for the 20° around max lift and clearly shows the step in the rate of change at the point of max lift. The trend of changes in lift shows 4 such spikes on

this chart, each coinciding with a change in cam measurement angle Θ used to calculate the lift values, highlighting the key weakness of the current format's accuracy.

This suggests that the accuracy issue is not in the division of cam rotation but is inherent in the resolution of the measured data. The optical dividing head used to measure the cam profile is capable of accurate rotation down to 0.1° so the limiting factor in accurately measuring the cam to a higher resolution would be the contact point of the DTI used, as this already contains cosine error which has not been accounted for. This is exacerbated by the fact that the DTI was only perpendicular to the surface at the base circle and nose of the cam, this was not accounted for at the time of measurement. To properly account for this, a new tip using a finer point or a jig to keep the gauge close to perpendicular to the surface may have helped but as a higher degree of accuracy could have been gained by using the CMM (coordinate measuring machine) or 3D scanning facilities available, using one of these methods would have been a better solution. This may have hindered the project as it was found during development that increasing the divisions of ϕ from 1° to 0.25° steps increased a simple list in excel of both Θ and ϕ from 13.3kb to 22.9kb and converting both would result in a file size of 29.2kb, without any calculations referencing these cells. As the current iteration already takes several seconds to return a lift profile and is a simplified model, it seems likely that this increase in size and complexity would slow it further, meaning that a switch to a faster platform such as Matlab would yield greater benefits.

On chart 13 it is also clear to see the multiplying effect the rocker has on the lift. It also shows an amplification or a similar effect to that of the horizontal displacement of the contact point about the centre of the valve stem seen in the bucket and shim lift profile. This effect gives the higher velocity (slope) opening the valve and shifts the maximum opening point earlier in the rotation of the cam. It can be seen from this that the asymmetry found on the cam as standard would need to be increased to counteract this effect and may not be suitable if the rocker remains straight, as the relief curve gets so extreme that the rocker can no longer effectively follow the cam's contour. This may necessitate a change to a roller follower, curved rocker or some type of offset cam follower surface if a straight rocker is to be kept.

Both curved and roller follower rockers would come with benefits, and a curved rocker for increased stiffness with a roller to reduce friction is expected to be the optimal solution.

Phi	Lift			Difference	
	Solidworks	Method 1	Method 2.2	Method 1	Method 2
0	0	0	0	0	0
90	0	0	0	0	0
100	0.24	0.22	0.22	0.02	0.02
110	0.47	0.44	0.46	0.03	0.01
120	3.27	3.33	3.26	0.06	0.01
130	11.01	11	11.02	0.01	0.01
140	16.15	16.22	16.19	0.07	0.04
150	18.69	18.78	18.75	0.09	0.06
160	19.26	19.33	19.3	0.07	0.04
170	18.49	18.56	18.55	0.07	0.06
180	17.04	17.11	17.1	0.07	0.06
190	15.13	15.17	15.17	0.04	0.04
200	12.88	12.94	12.92	0.06	0.04
210	10.4	10.44	10.41	0.04	0.01
220	7.76	7.78	7.76	0.02	0.00
230	5.07	5.06	5.07	0.01	0
240	2.6	2.61	2.61	0.01	0.01
250	0.89	0.89	0.88	0.00	0.01
260	0.4	0.39	0.39	0.01	0.01
270	0.16	0.17	0.15	0.01	0.01
280	0.05	0.06	0.05	0.01	0.00
360	0	0	0	0	0
			Average	0.03	0.02
			Max	0.09	0.06

Chart 15; Comparison of method accuracies

This table shows the results from both Solidworks static testing, and method 1 and 2 used in excel. The columns containing the error between both excel methods and the Solidworks data are colour coded to highlight trends in the deviation from the benchmarking data. They both show the same increase in deviation around maximum lift, suggesting that these are linear differences in error rather than error specific to certain angles of cam rotation, this gives confidence that the results will be a useful tool for prediction of real-world system behaviour.

The current model assumes that the rocker is straight and that the surface in contact with the cam is on the axis of the rocker pivot. It is also set to 0 thickness, though this parameter has been included, it makes no difference to the valve lift output. Further work would see the addition of pivot axis offset from the contact surface and allowance for curved or roller followers. It is expected that both would be reasonably simple to implement and would make for a more useful model.

4.7 Conclusions

The main aim of this section of the project was to create a model which can take either type of profile, be it cam or valve lift, and produce the resultant counterpart dependant on which actuation method is to be used and geometry-based inputs (i.e. either measurements or desired ratios) supplied by the user. It is believed that such a model would be a powerful tool in the development and design of valvetrains and other cam actuated systems. Whilst this model currently can only return a valve lift profile from cam lift, it has provided the necessary understanding and framework to build a more complete model. This basic model is useful for understanding some of the major differences in cam actuation methods and fits into the wider scope of the project by allowing changes made to the existing data to be assessed for viability as part of proposed designs.

This model has also given an insight into techniques for modelling mechanical systems and valve lift profile design. Furthermore, as new mechanical innovations in engine design become apparent, the techniques used to build it may help to assess new systems being released and the background research conducted as part of the work will likely inform further work.

5 Project Conclusions

It is expected that the ICE will continue to be a major part of personal transport over the next decade and innovations like those discussed in this paper will continue to build on their current advantages over alternative power sources like electric and hydrogen. Therefore manufacturers will continue to invest in systems which may extend their viability and an understanding of unconventional mechanical systems will be increasingly important, especially as these unconventional systems bring forth new, unforeseen challenges.

The aim of the project was to investigate mechanical methods of improving the overall thermal efficiency of an engine, specifically to assess their suitability as modifications for the Suzuki SV650. The project identified two key methods; variable compression ratios and cylinder deactivation. These were modelled and designs to fit them into the SV650 engine were assessed.

The Gomecsys system was found to be promising, however a lack of production model meant that it was not fully explored as an option. The Infiniti system of variable compression was modelled and a design which produced the standard stroke on both cylinders was made. The design does not however produce symmetrical piston motions on both cylinders so further development would be required to assess its true viability in a V configuration.

Cylinder deactivation methods to be fitted to the SV650 were considered and none were found to be viable without significant change to the original designs, with a conversion to rocker valve actuation being the most suitable solution found for adapting the platform. As such, a numerical method returning a valve lift profile from the measured cam profile was developed as a tool to inform further work on valvetrain design.

In total, five numerical models of assessing engine geometries were produced for this project; piston motions for a conventional engine with crankshaft offset, a Gomecsys type variable compression ratio engine and a linkage type engine and; valve lift profiles from cam profiles for bucket and shim, and rocker actuation methods. All numerical models were verified to be accurate descriptions of the systems. These models form a strong basis for more powerful tools to model mechanical systems and have improved the authors understanding of numerical modelling, engine and valve lift profile designs.

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Appendix

```
clc, clear;
Gamma=-45.000001;
Target=50;
R=17.14;
T=40;
sp=30;
Ch=24.7487;
Ao=Ch;
Error=1e-3;
G=zeros(361,1);
for i=1:361
    while Target>Error || Target<=-Error
        Gamma=Gamma+(Error/10);
        Target=1-((T/sp*sind(Gamma)+Ao/sp-R/sp*sind(i-1))^2)-(((Ch+R*cosd(i-1)-T*cosd(Gamma))/sp)^2);
    end
    G(i,:)=Gamma;
    Target=50;
    Gamma=-45.000001;
end
```