Improving Formula Student Engine Performance Through Cam Profile Optimisation

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Improving Formula Student Engine Performance Through Cam Profile Optimisation

Henry C A W Neilson

Abstract

This project has been aimed at changing the camshaft of a 2007 Honda CBR 600 engine to increase its performance for use in the UCLR's Formula Student racing vehicle. Regulations imposed by Formula Student state that the engine intake must be restricted, depriving the engine of air and reducing its performance. In a bid to counteract this restriction, the engine has to be re-optimised for use in the Formula Student vehicle.

The camshaft is in integral part of the modern combustion engine. It governs the movement of the intake and exhaust valves in an attempt to optimise the gas flow in and through the engine. To match its level of importance, there is a near infinite scope for alterations one can apply in optimising the cam shapes. Modern cam profiles are traditionally broken down into sequential splines of varying polynomial order. For profile generation, not only is the lift curve of importance but also several derivatives of the curve too. Research into how cam profiles are generated showed that the optimal way of generating shapes has been the centre of debate for near a century. Combining a long established industry method, with a theoretical method has yielded a novel way of creating a profile.

A MATLAB script defines the whole profile using 14 input variables. Some of which are fixed due to the dimensions of existing components in the engine. Other variables can be altered freely for profile optimisation, with each one having a different effect on performance.

In addition to the fixed parameters and optimisation parameters, the theoretical profile has to be "plottable" on the cam. There are restrictions on the ratio of each variable to one another that if not followed, result in creating an "impossible" profile, that cannot be mapped onto a cam. This check was performed by a CATIA model.

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List of Tables and Figures

Nomenclature

Item	Description
OHC	“Over head camshaft”
DOHC	“Dual over head camshaft”
TDC	“Top dead centre”
BDC	“Bottom dead centre”
BTDC	“Before top dead centre”
ATDC	“After top dead centre”
BBDC	“Before bottom dead centre”
ABDC	“After bottom dead centre”
RPM	“Revolutions per minute”
BHP	“Brake Horse Power”
Alpha	Angle of cam profile, defined as 90 Degrees at the cam tip.

Chapter 1

Introduction

1.1 How the Camshaft Works

Figure 1.1: *An Example Camshaft. Constructed in CATIA, Rendered in KeyShot 5. Design based of a Suzuki Hayabusa intake camshaft. Dassault Systèmes 2010. Luxion, Inc. 2015.*

Figure 1.1 shows a render of a typical camshaft. At its heart, it is a shaft consisting of various cams positioned along its length. A typical engine contains two different shafts offset from one another and driven by interlocking gears. One shaft is for the intake valves and the other for the exhaust valves. The main shaft is constructed from one cast part that is machined to micron precision. Its lack of moving parts makes it look deceptively less complex and influential on engine performance than it really is.

The camshafts reside at the top of most combustion engines. They drive the valves in one of two ways; rocker arm contact or overhead camshaft (OHC). Rocker arm contact is a method where by the camshafts do not directly drive the valves. Instead, the lobes push against rocker arms that pivot about a rocker shaft and push the valves. An overhead contact mechanism, as its name suggests, puts the camshafts directly above the valves and drives them directly without any levers. Though OHC mechanisms are a newer development, both method have their advantages. Samarins 2013. The CBR 600 engine employs an dual overhead camshaft (DOHC) setup. Figure 1.2 shown below shows an example cross-section through the valves of a typical DOHC setup:

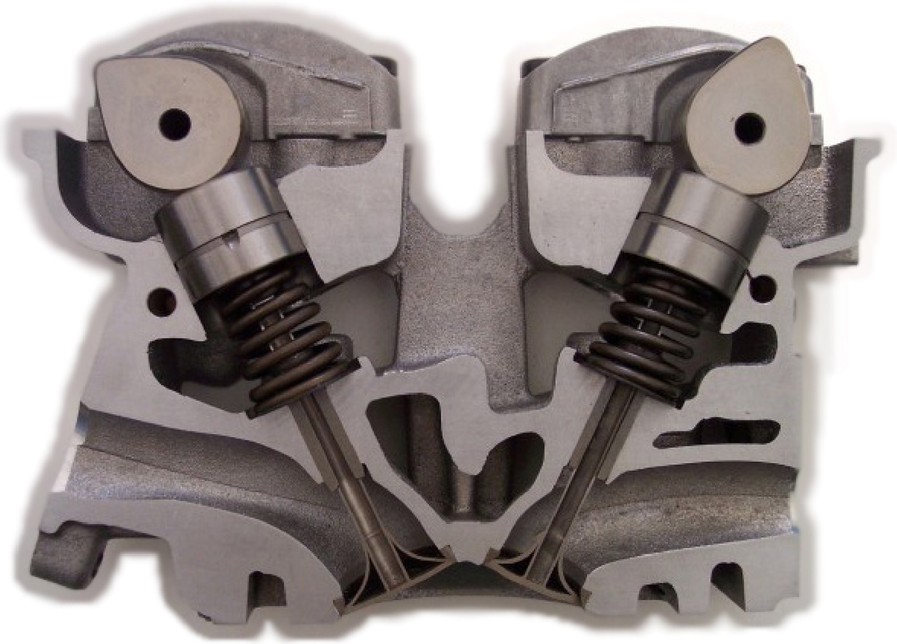

Figure 1.2: *An Example DOHC Setup. Wheels of Italy 2007. The left hand side shows an intake valve and the right hand side shows an exhaust valve. The combustion chamber has not been included but would be below the valves.*

In the CBR 600, the cross section seen in figure 1.2 is repeated 8 times, twice for each of the 4 cylinders. The springs are important in ensuring the valves accurately follow the movement of the lobe surface. Too high a tension and the valve follower will press unnecessarily hard against the cam causing more ware. Too low, and during high revolution operation, the follower and cam may separate in a problematic situation know as overshooting. Over time, overshooting wears the cam down when the follower comes crashing back to make contact. The seemingly small distance the separation creates, and low mass of the parts may make it hard to believe this would be a problem. However, taking into consideration that if overshooting does happen is occurs thousands of times a minute, the microscopic wear does add up.

1.2 The Importance of the Camshaft

The camshafts govern when the engine valves open and close. The opening and closing sequence for the valves is very complex and is set to be a balance between many factors. Timing is crucial as the valves control the gas and vapour flow throughout the combustion chamber. As internal combustion engines are driven by pressures, any unnecessary pressure losses are a large area for inefficiencies.

The cam profiles created by the intake and exhaust camshafts can be represented on a 2D plot. The 2D plot is far more useful to visually dictate the properties of the cam profiles than simply looking at the lobe shape. Figure 1.3 shown below presents a typical valve timing plot that shows both the intake and exhaust valve profiles:

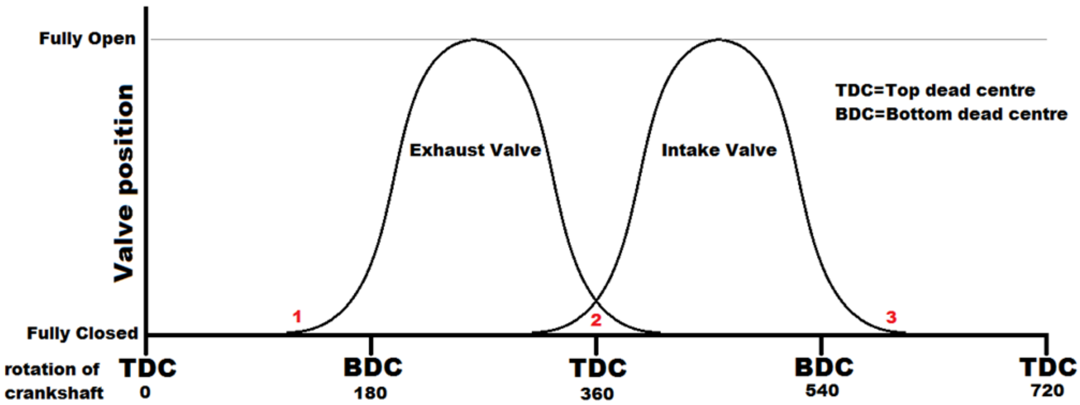

Figure 1.3: *An Example Cam Profile Plot. Labeled are the exhaust and intake valves lift profiles. Points 1,2 and 3 mark points of interest.*

Figure 1.3 shows the opening and close paths taken by both the intake and exhaust valves, in relation to the rotation of the crank shaft. A four-stroke engine takes two complete revolutions of the crankshaft to complete one engine cycle. This is why the crankshaft rotation goes up to 720° in figure 1.3. In a ideal situation, both the intake and exhaust valves would open and close instantaneously. This would facilitate greater gas and vapour flow in and out of the engine for a shorter time, aiding in engine performance. This, in the real world, is not possible due to mechanical restrictions. Increasing the opening and closing accelerations to square off the displacement curve creates the problem of overshooting, mentioned earlier.

Figure 1.3 also shows how the timing of opening and closing the valves are not directly linked to the position of the piston. The opening and closing of valves occurs before and after TDC and BDC respectively. This is mainly because of two factors:

Firstly The "analogue" nature of the the valve movement:

Due to the finite time it takes to open and close the valves, starting the exhaust valve opening right at the TDC would mean that for the first few degrees of rotation after TDC, the valve would be open a minute amount. This would cause the exhaust gases to form a high-pressure cushion, which would impede the movement of the piston. The compromise is to begin the opening sequence a little before the TDC to ensures that the valve is mostly open at the start of the exhaust stroke. This has the detrimental affect of allowing some of the pressurised gases to escape during the "power stroke". This downside is more than made up for in allowing the exhaust to escape more efficiently after TDC.

Secondly The inertial properties of the gas and vapour flowing through the valves:

Though the mass of the gasses may be low, they do have an affect. The masses contribute to the gasses having an inertia. Considering the intake valve: As the piston reaches BDC it no longer creates a negative pressure to draw in more combustible vapour. The inertia of the fast inward flowing gas keeps the flow going into the cylinder despite the now negative pressure that is being created due to the piston rising. The intake valve is designed to close at the same time the inertia and negative pressure are equal and thus creating no inward flow. This also has the benefit of packing more combustibles into the chamber as the gasses are compressible.

1.3 The Purpose of Altering the Profiles

The optimal opening and closing times for the valves is something that changes constantly during and engines use. Changing the RPM and load torque alter when the best time to open and close the valves is. Due to this, camshafts are a compromise between these changing factors. They are designed to work at the modal engine RPM and modal loading conditions. Depending on what the initial criteria during engine design was, camshafts are designed to balance fuel efficiency, power output and engine longevity.

The CBR 600 engine was originally designed to be used in the Honda CBR 600 motorcycle. This is a road vehicle, that to meet customer expectations has to achieve a certain mileage (fuel efficiency), last a certain amount of time (engine longevity) and go a certain speed / accelerate to that speed in a certain time (engine power). The very same engine is now to be used in a formula student racing vehicle. The criteria for this is different.

On top of "conventional" requirement alterations, a air intake restrictor is attached to the engine to limit it's power output. This is a restriction imposed by the Formula Student governing body. The restrictor deprives the engine of air, lowering the performance. To counteract this, the camshaft profiles have to be modified to allow greater flow.

Chapter 2

Theory

2.1 Breaking Down the Cam Lobe

Figure 2.1 shown below, depicts a typical cam shape used for the intake valve:

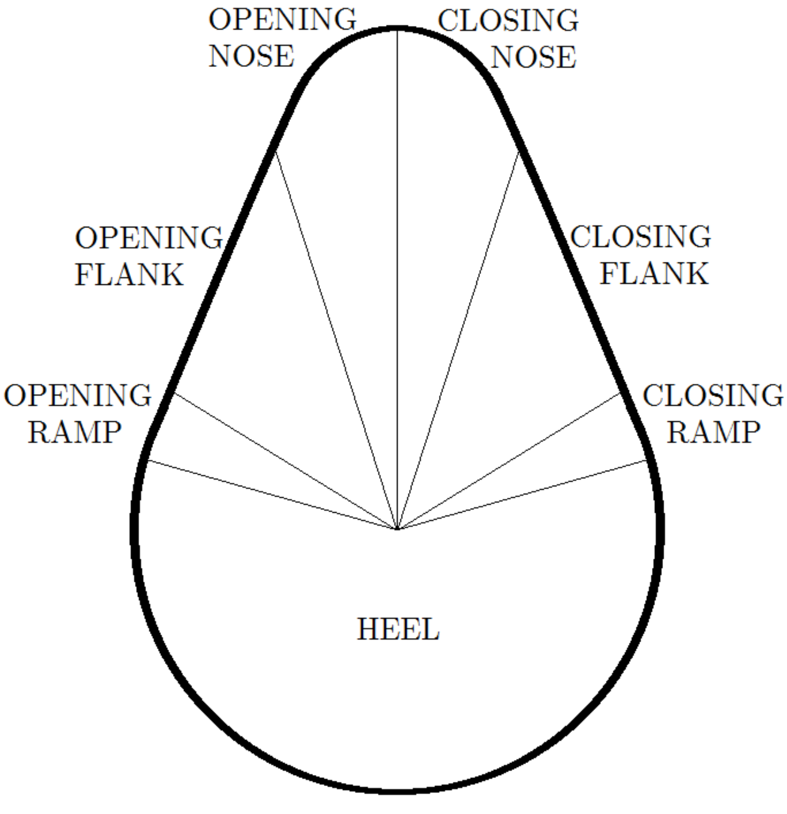

Figure 2.1: *Depiction of an Intake Valve Cam. Labeled are the names of typical lobe sections.*

Figure 2.1 shows the lobe broken down into typical sections. It can be seen that the left and right side are for opening and closing the valves, respectively. Each section of the lob performs a different action in opening and closing the valve:

The Heel of the cam is the constant radius portion of the cam that is active when the valve is closed. The heel is dimensioned as to have a “tappet” clearance of roughly 0.2mm. This clearance when the valve is fully closed is for three main reasons:

- To accommodate for tolerances in manufacturing and assembly. Slight inaccuracies in the parts and mechanisms could cause the valve to not be fully closed when it was meant to be fully closed.
- To reduce unnecessary wear on the cam and follower through friction.
- To accommodate for the movement of the vehicle. A sudden jolt to the engine could cause the cam to move slightly, this runs of the risk of opening the valve when it should be closed.

The Opening Ramp ensures a smooth pick up of the follower. As the heel of the cam is not in contact with the follower, the opening ramp ensures that at the point of contact, the collision between the follower and cam causes as little damage as possible.

The Opening Flank opens the valve in a continuous acceleration motion.

The Opening Nose decelerates the opening of the valve until it is fully open.

The Closing Nose begins the closing of the valve in an increasing acceleration path.

The Closing Flank decelerates the valve closing to almost stationary.

The Closing Ramp much like the opening ramp, ensures the transition from contact to non-contact is as smooth as possible. Allowing the valve to “drop” too quickly will cause it to bounce off its seat. Causing problems such as pressure losses due to the valve being open for too long.

Although the description for the flanks and noses may be lacking, they are the most important part of the cam as they majorly govern the movement of the valve. To ensure the valve follows the path of the cam accurately the valve spring is used to ensure constant contact between the follower face and the cam (apart from when the heel is active), figure 1.2 back on page 10 shows these springs. The compression force applied by these springs is an important variable that governs the shape of the cam. As mentioned earlier, the valve spring tension can be too high, this creates a larger friction force between the valve follower and the cam, causing premature wearing. Though this is bad, it is the lesser of two problems, the worse of which being a too low a tension . . .

As stated previously, the cam lobe is usually represented on a 2D plot as it is easier to understand what is going on. In figure 2.2 shown below, the lobe has been divided up into the aforementioned sections:

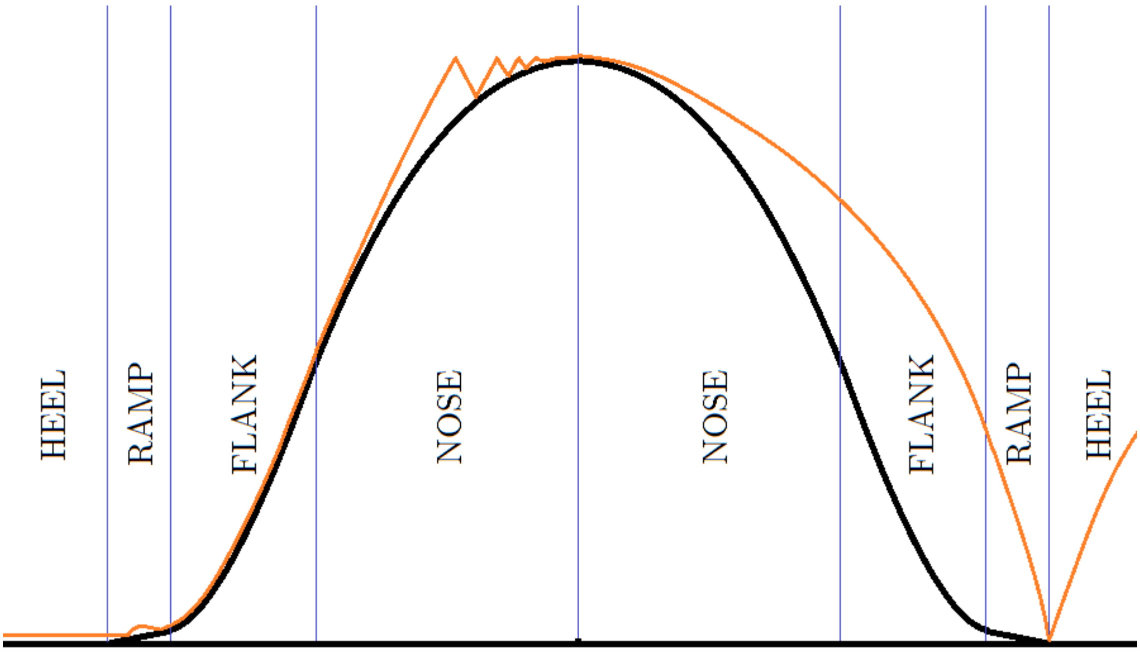

Figure 2.2: *An Example Segmented 2D Cam Plot. Shown in orange is a theoretical path taken by a valve if the spring tension is too low. It can be seen that there are three main areas of separation.*

Figure 2.2 depicts what might occur if the spring tension were too low. The events have been exaggerated and are not to scale, they just act as a visual aid. Tracing the path from left to right as the cam rotates the following events take place:

The follower is not in contact with the heel. As the two make contact the follower has a tendency to bounce back. As the spring tension is too low, the follower looses contact. After a short time the follower regains contact, there is the possibility of another bounce (not depicted).

The follower and cam remain in contact throughout the flank portion as the cam

surface is accelerating into the follower.
At the start of the nose portion, the surface of the follower is decelerating away from the follower. This is nearly matched by the follower but as the spring tension is low, the follower cannot keep in contact with the surface. The valve keeps opening separate from the cam until is reaches a physical limit. At this point it bounces off back towards the cam, it reverberates between the two until there is no gap due to the encroaching cam face.
After the peak of the cam, the surface begins moving back away from the follower, the spring cannot keep up with the movement so the follower separates from the surface, and remains so for quite some time. As the cam surface begins to dwell, the follower catches up and comes crashing back to the surface. As seen earlier in the cycle, it bounces off. The heel of the cam is now active so the valve will bounce in its seat until the cycle starts again.

It is important to note that the three mentioned points of separation would not occur proportionally in magnitude to each other as they have been depicted. Another point of information is that the follower is unlikely to bounce quite as much as has been depicted. This is due to newtons coefficient of restitution (a property that partially governs collision properties) between the follower and cam, not being particularly large.

2.2 Further Cam Classification

Like any high level commercial product, camshafts have consumer relevant specifications. To understand the common specifications used, one is first referred to figure 2.3, Shown below:

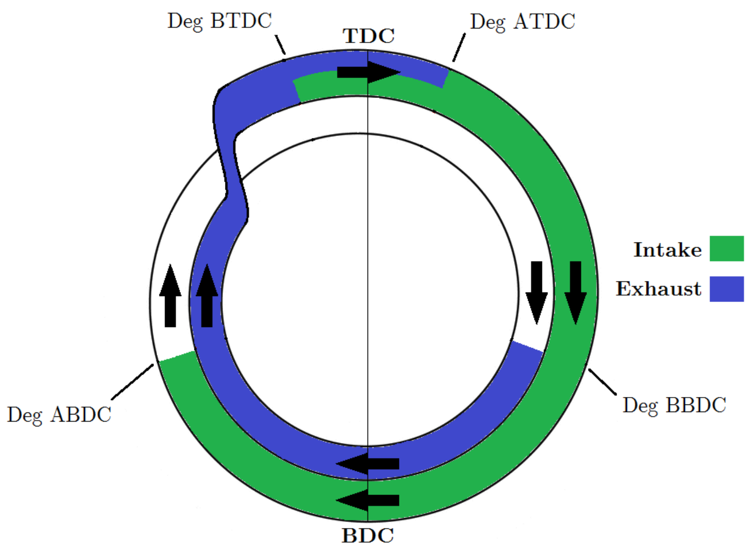

Figure 2.3: *Timing of Valves Relative to Crank. The double continuous loop represents the motion of the crankshaft that rotates twice in a 4 stroke sequence.*

Figure 2.3 shows the 4 stroke engine sequence. The valve opening and closing times have be arbitrarily chosen to demonstrate the cycle pattern. The four key points are the opening and closing times of the intake and exhaust valves. Ordering the 4 events as BTDC, ATDC, BBDC and ABDC gives the camshaft timing matrix. The timing matrix is a key commercial specification for camshafts as it shows when each valve opens and closes in relation to the crankshaft.

Figure 2.4 shown below, is an example specification table for a camshaft. In this case, the camshaft is a PH10, used in the CBR 600 engine:

Part No.	Application	Duration @ 1mm		Cam Lift		Valve Lift		Timing	Full Lift		Lift @ TDC		Valve Clearance	
		Inl	Exh	Inl	Exh	Inl	Exh		Inl ATDC	Exh BTDC	Inl	Exh		
PH10	RACE	250°	242°	0.363" 9.22mm	0.330" 8.38mm	0.355" 9.02mm	0.320" 8.13mm	23-47 45-17	102°	104°	0.127" 3.23mm	0.097" 2.46mm	0.008" / 0.20mm 0.010" / 0.25mm	info

Figure 2.4: *Specification table for PH10 Camshaft. Piper Cams 2015.*

Figure 2.4 at first glance shows quite a lot of information. Each point of interest will be discussed below:

- The timing matrix can clearly be seen in the middle of the table. Shown as BTDC-ATDC and on the second line BBDC and ABDC. As the opening and closing sections on a camshaft are so shallow, it is very difficult to define when the portion of actual lift occurs. For this reason, the timing matrix defines not when the follower first makes contact with the cam, but when the valve has opened a certain distance. This distance is chosen by the manufacturer yet common distances are 1 mm or 0.050 inches of lift. In Figure 2.4, Piper Cams use 1 mm of lift.

- The Duration is a somewhat redundant fact, as it can be worked out from the timing matrix:
  "Inlet Valve Duration"$= BTDC + 180 + ABDC$
  "Exhaust Valve Duration"$= ATDC + 180 + BBDC$
  The duration is the total time each valve is open (again taken between certain values of lift, 1 mm in this case). It is only included to help a less informed consumer find the correct cam at a glace.

- Cam lift is the height difference of the surface of the cam between the heel and the tip of the nose, shown in inches and mm for both inlet and exhaust lobes.

- The valve lift is the lift experienced by the valve. As can be seen, the values differ from the cam lift. This is because of the "slack" distance between the follower and the heel that has been previously discussed.

- Full lift gives explanation as to how many crankshaft degrees of rotation before or after TDC the nose tip (full lift point) occurs at.

- Lift at TDC, as expected, is the height of cam lift for both the exhaust and intake valves at TDC.

The information provided in figure 2.4 though detailed, still leaves the shape of the profile under-defined. The key bits of information have been specified that would be enough for a consumer to decide what profile they need. As with most consumer products, the consumer-friendly specifications only provide an outline to the product. Tabulated below in table 2.5 are a list of factors that change with duration and angular separation between the full lift points of the intake and exhaust lobe (lobe centre):

Longer Duration	**Tighter Lobe Centres**
Increases peak power	Decreases peak power
Widens the power band	Narrows the power band
Raises the power band	Lowers the power band
Causes peak power to occur later	Causes peak power to occur earlier
Causes rougher engine idling	Causes rougher engine idling
Decreases maximum torque	Increases maximum torque
Decreases bottom end torque	Increases bottom end torque
Increases emissions	Increases emissions

Table 2.5: *How Performance Changes due to Cam Changes.*

These factors listed in table 2.5 are majorly, but not exclusively, influenced by changing the duration and lobe separation. As stated earlier the shape of the profiles also play a hugely important role in the performance of the engine as well as the key timings. To fill in the gaps in information provided by the specification table in figure 2.4, a range of calculations are needed.

2.3 How Profiles are Constructed

Cam profiles can be infinitely adjustable. Computational power just isn't at the stage where millions of different profiles can be generated and simulated to try and find the best one. To cut down the number of variables, common limitations are often undertaken. A typical practise taken up by cam designers is to employ symmetric cams, this nearly halves the work needed in designing an efficient camshaft. The difference required in the closing sequence and opening sequence of a cam is so minute that it is sometimes seen as rather esoteric to create asymmetric cams.

As discussed earlier, an ideal cam would open and close the valves instantaneously, or close to. As this is not possible, the next best option is to open and close them as quickly as possible. To do this, the acceleration of the valve must be maximised without causing too high a force on the valve train components. Already it is apparent that not only the lift curve is of importance, but also several derivatives of the curve too. Figure 2.6 shows a plot of a theorised cam profile with several derivatives labeled:

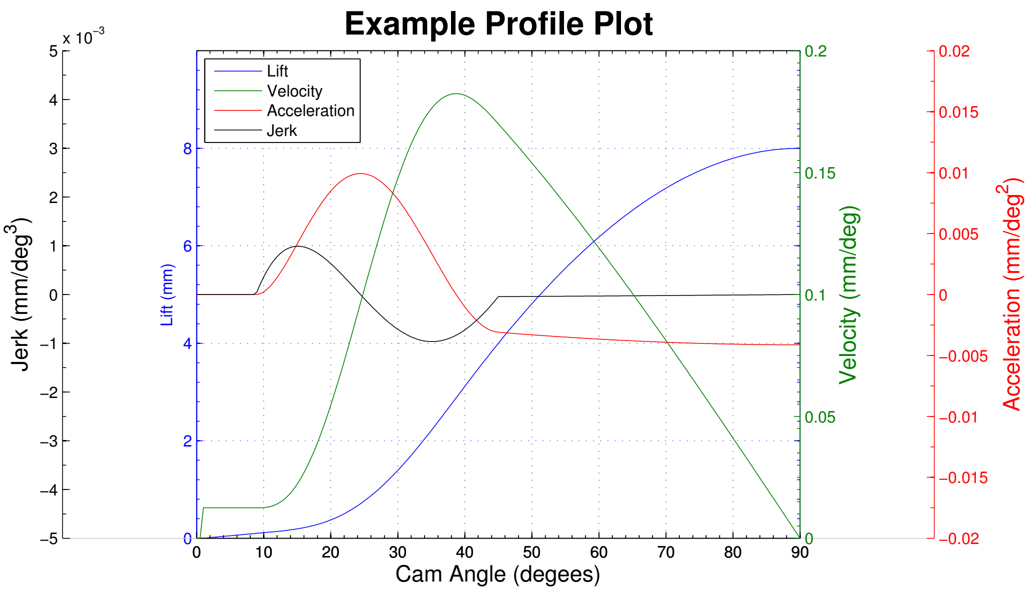

Figure 2.6: *A Theoretical Profile Plot. Shown are the magnitudes of the lift derivatives that have a surprisingly large importance in cam profile design. MathWorks 2013.*

Figure 2.6 depicts a plot of only one side of the cam as it is a symmetrical lobe. Each derivative has bounding factors that govern the shape of the profile.

2.3.1 The Importance of the Derivatives

Velocity:

The velocity is limited by the dimensions of the follower tappet. Figure 2.7 helps explains why this is the case:

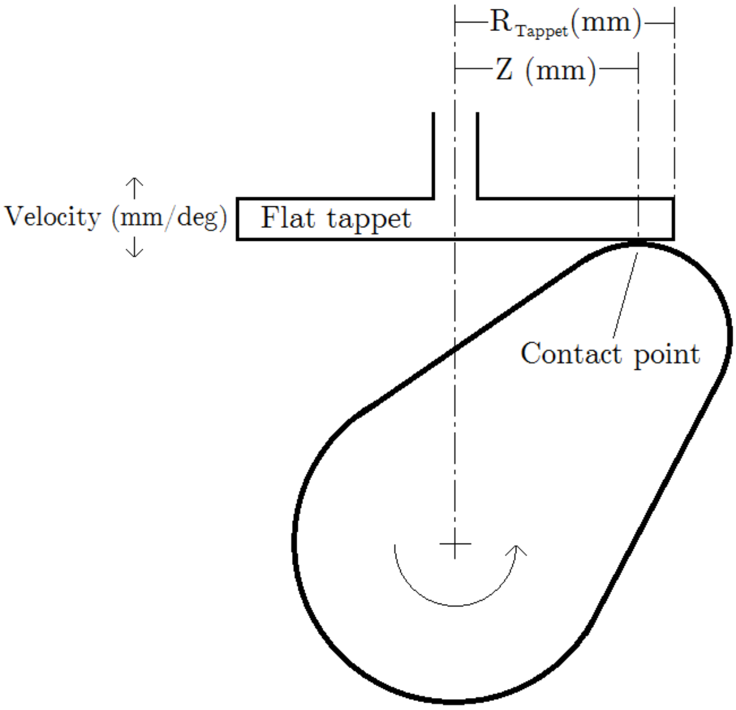

Figure 2.7: *Diagram to Explain Velocity Limit. This is a side view of a lobe and follower segment. Labeled are the dimensions of importance.*

The horizontal distance between the contact point and the tappet, Z, is directly proportional to the velocity of the tappet. The following equation shows the link:

$$Z = Velocity(mm/deg) \times (\pi/180)$$

Re-arranging gives a limit to the velocity as the tappet is a finite radius:

$$Velocity(mm/deg) < (\pi/180) \times R_{Tappet}$$

Finally, a safety distance is added to the tappet width to account for misalignment and edge chamfer:

$$Velocity(mm/deg) < (\pi/180) \times (R_{Tappet} - 5mm)$$

Acceleration:

The maximum acceleration, as discussed earlier, is primarily limited by the valve springs and the macro properties of the cam material. Too high a negative acceleration and the follower looses contact. Too high a positive acceleration and the cam may wear excessively. A field of research known as valve train dynamics, looks into how the deeper derivatives affect cam performance. Recalculating and going in depth into this field provides minimal gain compared to using established limits learnt from research.

Jerk:

Jerk is the first "complicated" derivative. Day to day life doesn't see jerk considered much. An easy explanation to jerk is the feeling one experiences when in a vehicle that is in the process of braking. A constant force is felt due to the deceleration of the vehicle, as the vehicle comes to a complete halt a high jerk is felt. This is the vehicle changing from constant deceleration to zero deceleration. Seasoned drivers may strive to reduce jerk by decreasing the rate of deceleration gradually by lifting their foot off the pedal as the vehicle slows. Jerk, like this, that can be felt, makes for an uncomfortable ride. Similarly, jerk in components causes issues such as accelerated wear and high jerks are prone to causing valve train vibrations. Once again, valve train dynamics can explain why this is the case.

It has been shown that velocity and acceleration have somewhat hard limits. Positive acceleration (that experienced in the flank portion) and jerk, do not have certain limits. Standardised limits found from the plethora of existing research into valve train dynamics and physical experimentation, can be employed. Setting these figures as hard limits reduces the number of variables used to create the best lobe profile.

2.3.2 “Modern” Methods of construction

We now know that simply making a lift curve to design the cam profile isn't enough. As the derivatives of the lift curve are important, is it important to be able to find their direct valves. This is only accurately possible though formulaic curve generation. In other words, using a polynomial to create the lift curve, and mathematically calculating its derivatives. This is commonly known as the “modern” way of creating cam profiles, yet has been present since at least 1985. Macarthy and Burns. 1985.

As different requirements for the different derivatives need to be met at different stages along the profile, the profile is commonly split into splines. These splines join at points known as “knots”. The positioning of the knots usually coincide with the aforementioned sections of the cam, yet sometimes more can be added. Knots are where jumps or infinite spikes of the derivatives occur. Depending on the degree of the polynomial, the two splines at each end can be matched. A quadratic spline matches the velocity but not the acceleration. A cubic spline will match the acceleration but not the jerk. This process can go on to a quintic spline where the first three derivatives of displacement match but not the 4th derivative. These derivatives only match one end of the spline. The other end doubles the variables that need to be matched so doubles the order of the polynomial required.

Splines offer a huge amount of flexibility when designing the profile. Unfortunately this freedom is not always a good thing as there are usually more variables than equations to affirm them. Experience from a designer is needed to select the positioning of knots effectively and even so, a lot of it is a guessing game. The problem with pure splining is under specification.

The method employed to reconstruct the CBR 600 camshaft is a modified method of the Triple Curve Contour method, first presented by Michael C. Turkish in 1946, Turkish 1946. This method typically sees the profile broken down into 3 splines, each of varying polynomial order, the highest of which being order 4. This means the highest order polynomial can match lift and velocity (gradient of the lift curve) at either end only. This creates jumps in the acceleration and infinite spikes in jerk at the knots, not a desirable trait. To better this, the method had to be modified.

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Chapter 3

Design

3.1 Triple Curve Contour Improvement

It has been explained that the Triple Curve Contour method, though useful, lacks in creating a smooth enough profile for the theory that has been discussed. The method was first theorised in 1934 and is somewhat out of date now. Current vehicle/engine manufactures keep their methods of cam profile generation to themselves. For this reason, the old method had to be used and modified to meet the more modern criteria.

The improvement involved increasing the order of the spline from 3 to 7. creating continuity at the knots right through to jerk. An order seven spline has 8 variables, this equates to 4 equations at each end of the spline, one for each derivative of importance.

This modified Triple Curve Contour method consists of a harmonic nose portion with the general equation:

$$Lift = a + \Big(b \times cos\Big((Alpha - 90) \times \pi \times c/180\Big)\Big)$$

The ramp portion is a linear function with the following general equation:

$$Lift = d + (e \times Alpha)$$

The flank portion is a septic function. Its general equation is the following:

$$Lift = (f \times Alpha^7) + (g \times Alpha^6) \cdots + (k \times Alpha^2) + (l \times Alpha) + M$$

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3.2 Profile Construction

It is apparent that there are 13 unknowns that need to be found to plot the profile. They are found through a series of input parameters. These are a mixture of variables that can change to optimise the performance, and variables that change to make sure the profile fits the engine. MATLAB was used to create the profile. MATLAB allows for easy recalculation of the profile if the input parameters change. It also allows the profile to be output easily and a graph to be plot to easily visualise the lift curve as well as its derivatives. Figure 3.1 Shows the input section of code from the MATLAB Script:

```
Lift =          8;  % Set for valve dimension + piston crown shape
Heel =          15; % Set for cam housing size

TIMINGMATRIX = [23,47;45,17];% Example timing  @ 1mm lift
Lift1mm =      (180-TIMINGMATRIX(1,1)-TIMINGMATRIX(1,2))/4;
% Change to    (180-TIMINGMATRIX(2,1)-TIMINGMATRIX(2,2))/4 for exhaust

FNKnot =       45;  % Angle where flank meets nose
RampLength =   8;   % Duration of ramp in degrees

FNHeight =     4;   % Height where flank meets nose
RFHeight =     0.1; % Height ramp reaches
BRHeight =     0;   % Lift of heel, always 0
MaxVelocity = 0.17;% mm/deg CAUTION! THIS REALLY MESSES THINGS UP!
%Crank shaft information to plot valve space diagram
w = 7.6;            %Valve Clearance
x = 60;             %Crankshaft displacement
y = 120;            %Conrod length (joint centre to joint centre)
z = 20;             %"piston rod" distance
VAngle = 80;        %Angle from vertical valve is angled N.B. exhaust-intake

graph = true;
export = false;
ClashError = false;
```

Figure 3.1: *Input Section of MATLAB Code. Each variable is labeled. The Full code is shown in Appendix A. MathWorks 2013.*

The arrangement of the script as seen in figure 3.1 is configured for the intake profile. Commented are the alterations required to generate an exhaust profile. The script found in Appendix A uses the variables to construct the splines between 0 and 90 degrees of the cam lobe. 90 degrees is the position of the nose. The splines are constructed in the following order:

1. The nose function is found completely. The sinusoidal function has three variables that need to be found. The known facts of the spline are:

- The position of the tip of the nose. The occurs at (90 deg, Lift).
- The velocity at the tip of the nose. This is 0 mm/deg.
- The velocity at the tail of the nose. This is the maximum velocity.
- The position of the flank-nose knot. This is set by the cam designer and is the only unconstrained value. its co-ordinates in the MATLAB script are (FNKnot, FNHeight).

The variables are found through a range of simultaneous equations and iterative means.

2. The ramp function is partially found. The gradient of the ramp function is found from the input variables. This is set to a standard gradient used almost concomitantly by cam designers. The position of the ramp along the x axis cannot be found before the flank function is found. The flank function cannot be found before the gradient of the ramp is found.

3. The flank function is the large septic function with many variables. To find the first 7 variables, an inverse matrix method is used. Differentiating the nose and ramp spline gave the values the derivatives had to be at either end of the spline. Knowing the point of 1mm lift (from the input variables) added another known value.

4. Upon finding the flank function, the ramp function could be shifted along the x axis to meet the flank.

5. The rest of the 90 degree distance (from the ramp beginning, to 0 degrees on the plot) was made into the heel with zero lift.

Throughout the script, there are areas to check if the input parameters are correct and if the script can create a “plausible” profile. Sometimes a velocity may be chosen that is physically too high given the other parameters, or the position of the flank-nose knot could be too far removed form an ideal location. The script also has a section that takes in the dimension of the piston and checks if the valves will clash with the piston crown.

After the script has created the profile, the values are output to Microsoft Excel. This is because the profile needs to mapped to a cam.

3.3 CATIA Modeling

It has been shown that the shape of a cam profile is governed by many factors and a broad depth of mathematical parameters. Even after following all these parameters, there is still the possibility of creating a profile that cannot be mapped onto cam lobe. Shown below in figure 3.2 is what can occur to a quarter lobe section when the profile cannot be mapped properly:

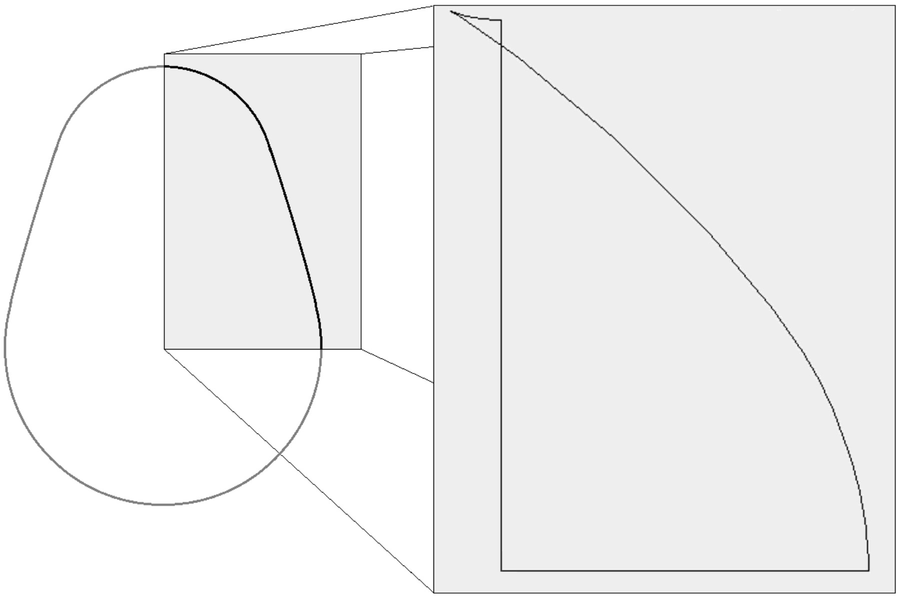

Figure 3.2: *An example of an "unplottable" cam profile. Shown on the left is an example lobe. On the right is the top right section of the lobe with erroneous overlap portion.*

Figure 3.2 is an example of what happens when the deceleration isn't high enough for the given heel and lift sizes. The surface of the profile cannot decelerate quickly enough for the velocity of the cam to reach 0 at the nose. This only happens on flat tappet followers as the contact point of the lobe and follower moves from side to side depending on the tappet velocity (figure 2.6 back on page 22).

Tabulated in Appendix B is a matrix output created from the MATLAB script, that is exported into a Microsoft Excel document. The important matrix output was the lift curve as this was used to drive a CATIA file. Each dimension was used to create a dynamically updating parameter table in CATIA. This updated as the Excel file was updated through running the MATLAB script. Each of the parameters was then related to a dimension on the CATIA sketch.

The first attempt at creating the cam on CATIA yielded the following result. (CATIA screen grabs have been altered to increase visual clarity):

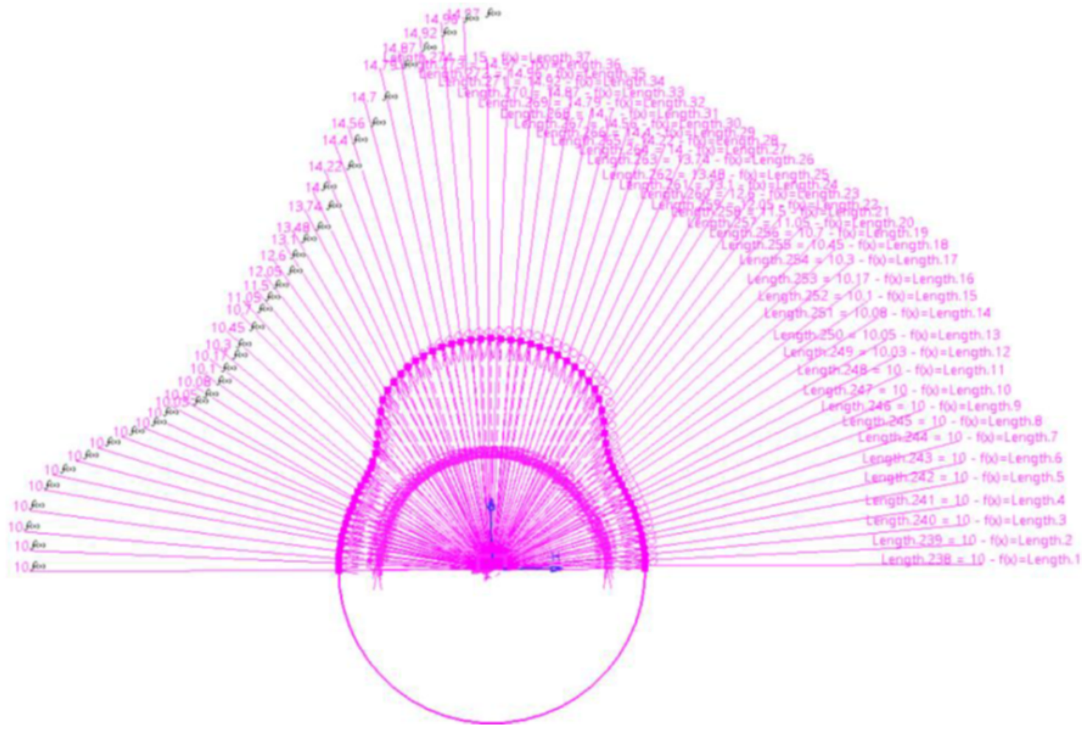

Figure 3.3: *The First Attempt at Cam Lobe Construction. Seen are the dimensions use to define the position of the outer surface. Dassault Systèmes 2010.*

Figure 3.3 is a screen grab of a CATIA sketch. it is composed of a semicircle at the bottom to occupy most of the heel portion and a spoked top section. There are 37 spokes, each linked to a parameter that maps the profile outside of CATIA. The tips of each spoke are joined using a spline to smooth out the gaps between each spoke. The labels on the right hand side show the in-CATIA formulas used to link the parameters. The left had labels are the dimensions in mm of each spoke.

Figure 3.3 is one of the first CATIA models created for this project. As a large amount of the theory was not known at this stage of the project, there are a few flaws with this method of cam profile mapping:

- The whole active portion of the cam lobe has been mapped and dimensioned. As the cam will be symmetrical, half of the constraints are redundant.
- The angular increments between the dimensioned sections are too coarse. The precise nature of the cam requires a finer resolution of detail otherwise the work carried out on reducing the magnitude of the higher derivatives goes to waste.

- A unique issue with flat tappet followers that has already been mentioned, is that the contact point of the lobe and tappet moves perpendicularly to the tappet motion. This means that the simple "spoke" model created in figure 3.3 will not work. Figure 3.4 shown below, is used to depict the issue:

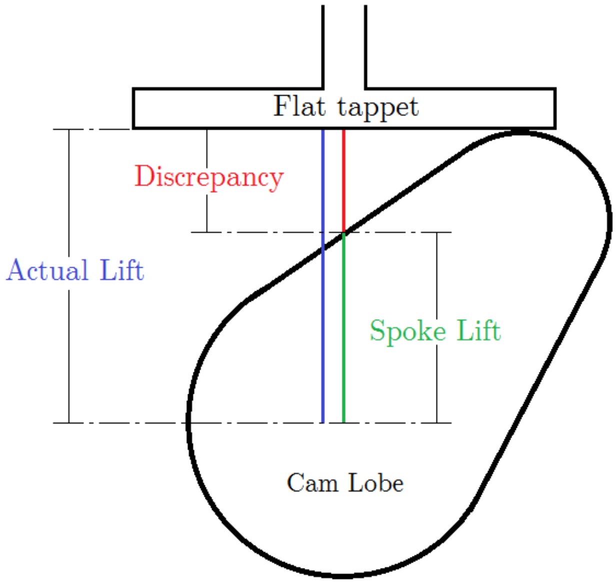

Figure 3.4: *Diagram Showing the Issue with Having a simple "Spoke" Model. The viewing position is the same as that depicted in figure 2.7.*

Figure 3.4 shows that due to the flat surface of the tappet, the contact point of the lobe isn't at the same point as the positioning of the spoke drawn on the first CATIA model. The tappet and lobe only make contact at the centre axis when the velocity of the tappet is zero.

To fix these issues many iterations of CAD models were created. First of all, the easy fixes were addressed; Increasing the angular resolution to 0.5° increments and only mapping half of the lift lobe. Figure 3.5 shown below, shows the next main CATIA model attempt. The model has be rotated 90° clockwise from the previous figure (figure 3.3):

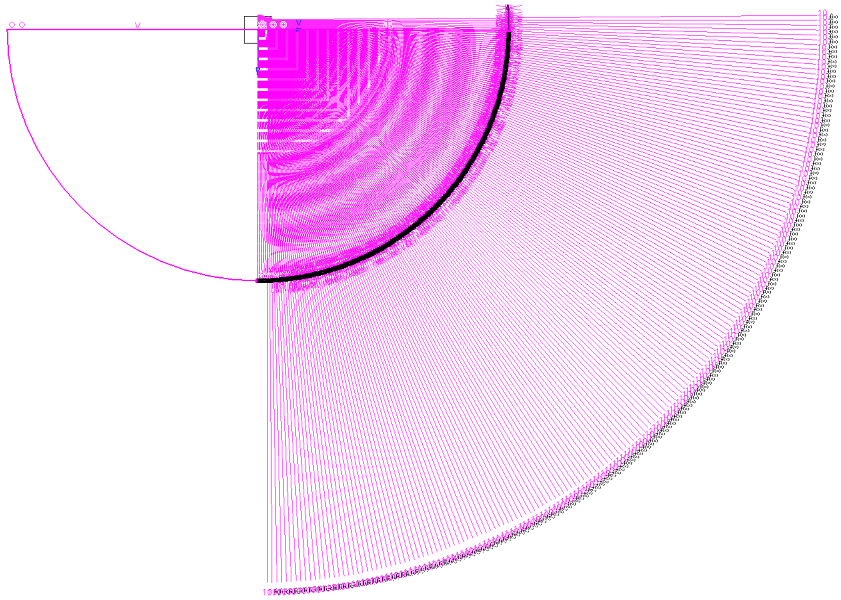

Figure 3.5: *The Improved CATIA Model. Showing how only half of the active lobe has been mapped. Dassault Systèmes 2010.*

As stated, the model in figure 3.5 improves on two of the main problems with the first model. Only half of the cam has been drawn as it will be mirrored outside of the sketch. The angular increments now make is hard to discern the constraints. In this model there are 181 relations for all the 181 parameters. In figure 3.5 the parameters have not been dimensioned so they only show a default circular shape. The fine degree of accuracy in the dimensioning lines creates an aliasing effect that makes it hard for one to distinguish the spokes.

The model in figure 3.5 still has yet to address the issue of the offset contact point. A complex alteration had to be made to the model that allowed the offset to be accounted for.

3.4 The Final CATIA Model

To create the final CATIA model that took into consideration the offset contact issue, a new method had to be employed. Figure 3.6 shown below. Is used to help explain the method used:

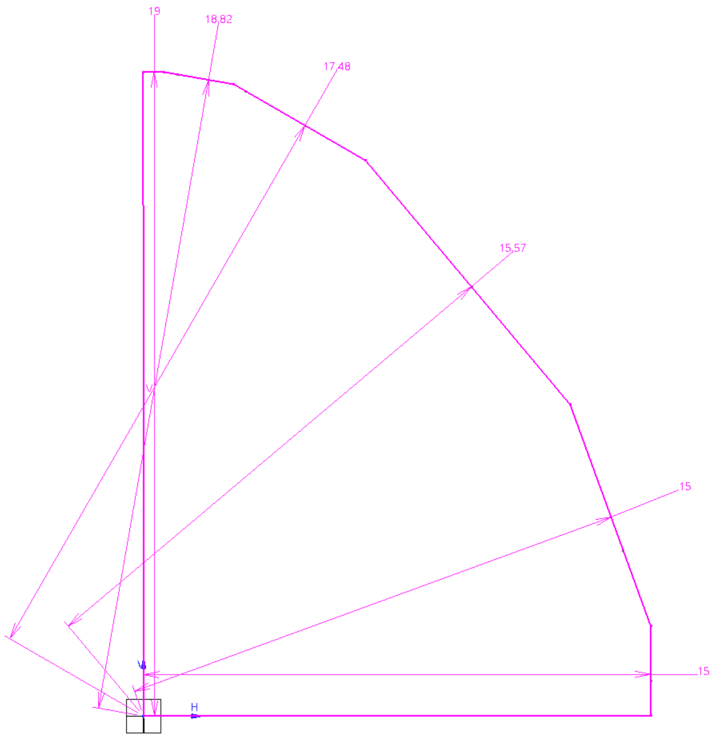

Figure 3.6: *An Example of the Method Used to Create the Final Model. Dassault Systèmes 2010.*

The example shown in figure 3.6 is a much lower resolution than required, this is to allow one to see what is going on. Each line seen is fixed to be 20° in advance of the next (apart from the last line that is only 10° due to 90 not being divisible by 20). The perpendicular distance between the centre of the cam and the surface is then constrained. This mean the "discrepancy" seen in figure 3.4 back on page 32 is no longer present. The "spoke lift" has changed to a perpendicular distance and is now equal to the "actual lift".

Reducing the angular increments from 20° down to 0.5° to get an appropriate level of resolution yielded some problems. The model became unstable when reducing the angle to low enough values. Troubleshooting found the model to become unstable when changing the angular value from 0.514° to 0.512°. Close, but not close enough, as the design required 0.5° increments.

After some trial and error methods, it was found that the "scale" in CATIA was not low enough. There is an option used to reduce the size of the working environment to better suit smaller projects. Changing this to "Small Scale" allowed the 0.5 degree increments to be used.

As .Part files' scale cannot be altered, this meant producing a new file. Creating it, and linking it to the Excel spreadsheet yielded the final model, shown in figure 3.7, below:

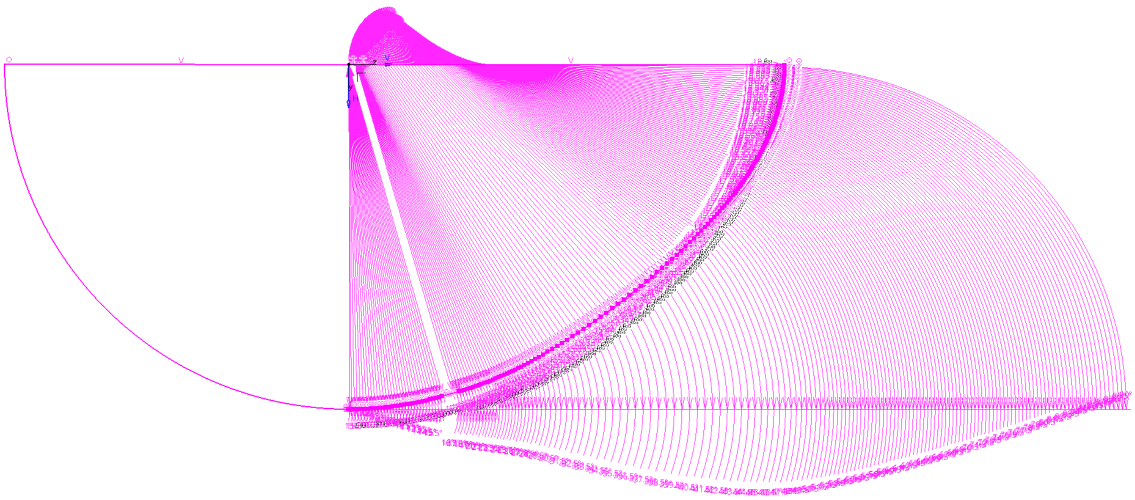

Figure 3.7: *The Final Working lobe Construction. Dassault Systèmes 2010.*

Not only does the final model allow one to check if the MATLAB profile generated is feasible, but it also provides some fascinating visual representation of the derivatives:

- A rudimentary visual for the velocity of the cam surface can be seen by the vertical displacement of the angular labels seen towards the bottom of figure 3.7. It is somewhat skewed due to the surface of the cam being curved, yet it gives one a rough idea.

- The acceleration can be seen in the spacing between the perpendicular measurement. Greater spacing equates to greater acceleration and narrower spacing represents greater deceleration.

AutoLibrary

Chapter 4

Finalising

4.1 Setting the Benchmark

All the design work that has been presented has been created to allow easy profile regeneration. Changing an aspect of the cam profile in the MATLAB input would change everything about the profile. Linking the MATLAB scrip to a Microsoft Excel document that drives a CATIA model allows for quick dynamic updating of the profile. This cuts down the time it takes to alter the profile drastically.

The profile created thus far, is a general cam profile and has not been altered to fit the CBR 600 engine. To fit the cam to the engine the current camshaft was used. In a process known as degreeing, the dimensions of one of the current camshaft were measured.

Degreeing is a method typically used to ensure the camshaft of an engine is correctly aligned or to check if the camshaft matches its specification table. The process can also be used to measure the rough shape of the cam profile. Due to the precise nature of the camshaft the exact shape of a cam cannot be measured, one can only measure a few points on the surface.

The process of degreeing calls upon the use of a degreeing wheel and a dial gauge. The degreeing wheel is attached to the crankshaft, and the dial gauge is positioned above the valve of the cam that is to be measured. The crank shaft is rotated by hand and the valve lift is measured at certain angular positions determined by the camshaft gear teeth (as can be seen in figure 1.1 back on page 9). Using this process, the rough shape of the cam can be determined. This will be used as a benchmark for improving engine performance.

Table 4.1 shown to the right, gives the output data taken from degreeing the intake cam of the stock CBR 600 camshaft:

Angle (°)	lift (mm)
0.000	0.000
10.588	0.000
21.176	0.000
31.765	0.000
42.353	0.000
52.941	0.000
63.529	0.000
74.118	0.000
84.706	0.000
95.294	0.000
105.882	0.000
116.471	0.000
127.059	0.105
137.647	1.991
148.235	3.734
158.824	5.093
169.412	6.007
180.000	6.401
190.588	6.401
201.176	6.007
211.765	5.093
222.353	3.734
232.941	1.991
243.529	0.105
254.118	0.000
264.706	0.000
275.294	0.000
285.882	0.000
296.471	0.000
307.059	0.000
317.647	0.000
328.235	0.000
338.824	0.000
349.412	0.000
360.000	0.000

Table 4.1: *Data Recorded Through Degreeing of Exhaust Cam. FST 2015.*

It should be noted that the data in table 4.1 was obtained by fourth year Mechanical Engineering students, who are also working on the formula student vehicle. One immediately visible attribute of the data is that most of the data points have a lift of 0 mm. This is expected at the valve remains closed for most of the engine cycle. This leaves only 12 data points of interest to replicate the active portion of the cam. It can also be seen that the lift distances are symmetrical between 180.000° and 190.588° This cuts the amount unique data point down to just 6 points.

These 6 points alone are not enough to replicate the cam profile. Drafting the profile by using a best fit line would produce a horrifically rudimentary shape, completely lacking in derivative definitions. The solution is to combine the MATLAB script with the data points to match the theoretical Triple Contour Curve method with the current design.

The data points were added to the MATLAB script and plot as an overlay on top of the lift curve. Measuring the valve tappet radius allowed the maximum velocity to be altered correctly giving the splines a better fit to the data points. Figure 4.2 shows an intermittent stage in matching the theoretical model to the data points:

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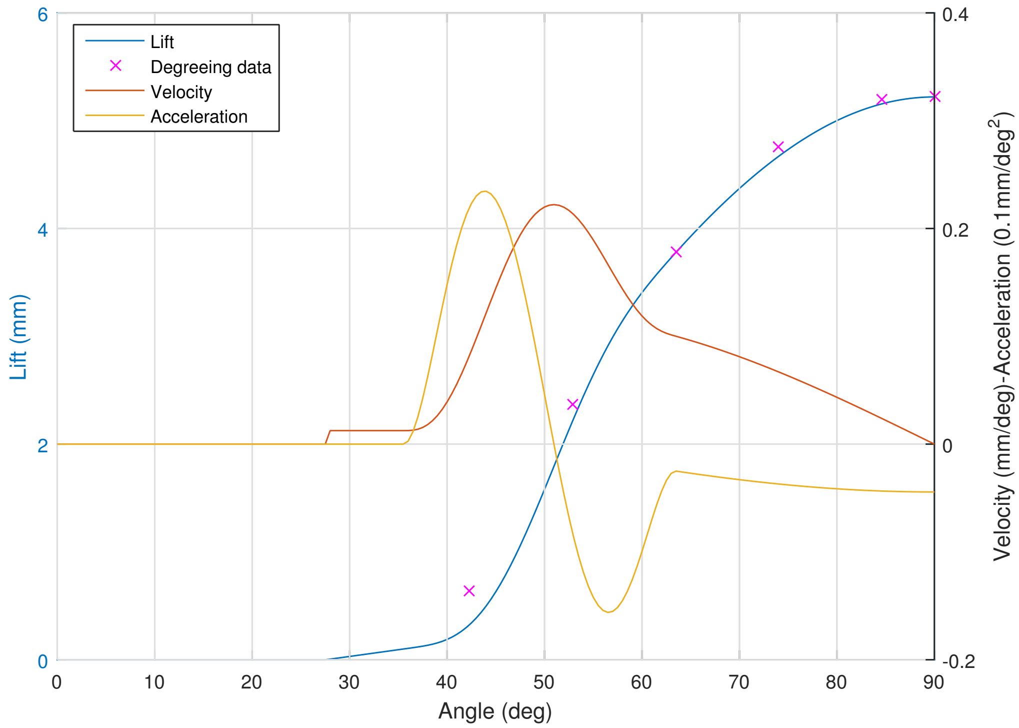


Figure 4.2: *Diagram to Show the Matching Process. It can be seen that the modeled lift profile doesn't quite yet match with the degreeing data. MathWorks 2013.*

Figure 4.2 shows that the profile has been partially matched. It has been made clear that the accuracy required from a cam profile is very high, the matching depicted thus far is not substantial enough. Updating the input section of the MATLAB script to try and match the data points took dozens of iterations. Yet there was always an issue that could be present, limiting the accuracy of the matching.

The method of profile construction that Honda used in the stock camshaft is highly unlikely to be the same as the method the MATLAB script employs. Although the principals behind all camshaft designs are the same (to maximise lift while adhering to limiting constraints) the details of a profile will change from construction method to construction method.

Many iterations of the script were ran. For each iteration, the input values were changed based on the output plot. After having worked with the profile plot for months, knowledge of how each input parameter affected the shape of the profile was obtained. Even with this knowledge, dozens of iterations were required. Parts of the script use iterative calculations to construct the splines. These took anywhere between less than a second, to several minute to calculate, depending on the starting state of the iteration (determined by the input variables).

Figure 4.3 shown below, shows the closet match achieved between the data points and the MATLAB generated profile:

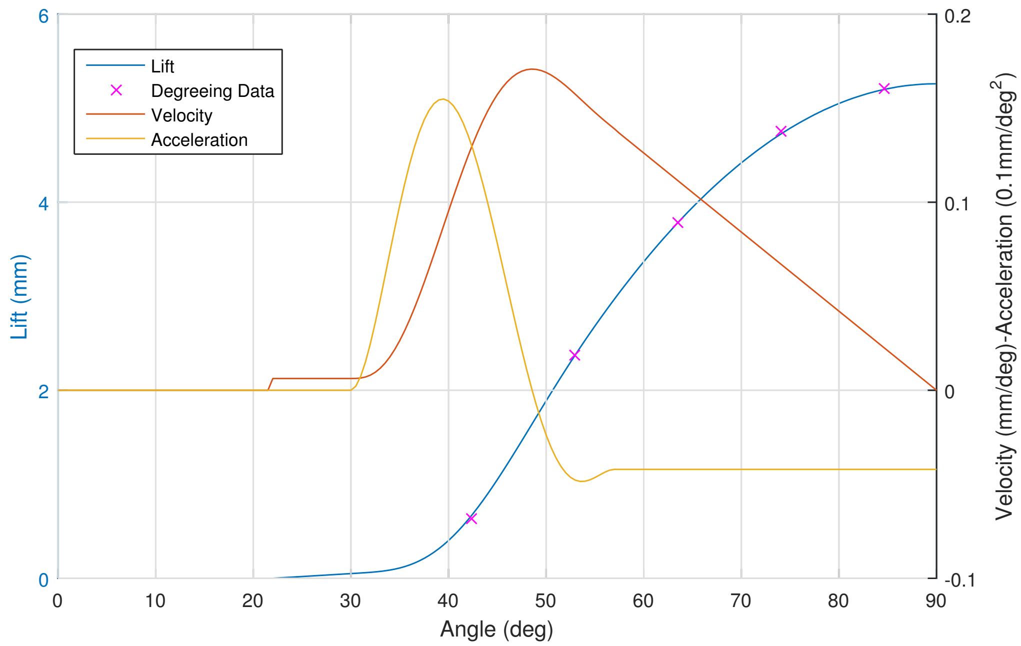


Figure 4.3: *Diagram to Show the Matched Plot. MathWorks 2013.*

An few interesting conclusion can be drawn from looking at the resultant shape of the derivatives in figure 4.3: The acceleration of the nose spline is seen to be close to constant. Acceleration is the second derivative of lift and so having a constant acceleration is attributed with the lift equation being a second order polynomial, I.E. a quadratic parabola. It is known that the script produces a sinusoidal function and that all derivatives of a sinusoidal function are also sinusoidal. The only explanation of how the acceleration is "constant" is that it is not. The acceleration is in fact a very shallow sinusoidal curve. In fact the equation of the nose in figure 4.3 is:

$$Lift = -7008659970554.33 + \cdots$$
$$\left(7008659970559.59 \times cos\Big((Alpha - 90) \times \pi \times 0.0000140353722731647/180\Big)\right)$$

A noticeably unique equation, with very extreme constants. Plotting the second derivative of this equation shows that the acceleration *does* have a sinusoidal shape to it, yet has been optimised to be extremely flat. Figure 4.4 shows this.

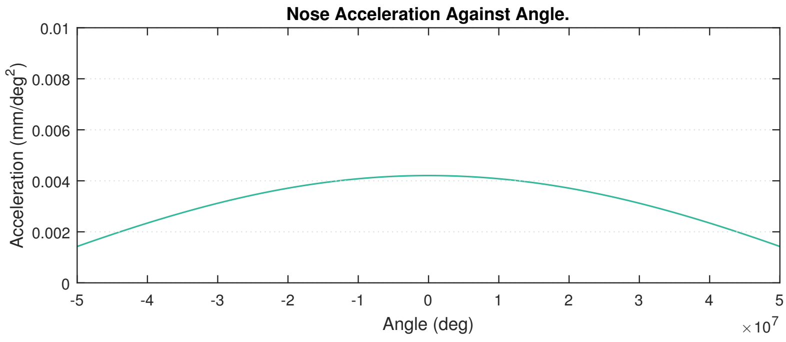


Figure 4.4: *Plot of the Nose Spline Acceleration. Showing how close to constant the acceleration is. Note the scale! MathWorks 2013.*

Having a constant acceleration nose is ideal as the nose portion requires the greatest deceleration to slow the valve to a velocity of 0. To do this in the shortest time, the cam must decelerate as much as is allowed by the valve springs.

The conclusion can be made that the stock camshaft used a quadratic nose equation. In attempting the match the nose shape as best as possible, the constants defining the sinusoidal equation have been pushed to the limits. The script has merely been trying to simulate a quadratic parabola. This matching process was repeated for both the intake and exhaust cam profiles.

The next step was to the apply the theoretical optimisations whilst using the stock profile as the base. The profile is improved to perform better under the altered engine conditioned encountered in the Formula Student racing vehicle.

4.2 Applying Theoretical Optimisations

After the stock cam profile for both the intake and exhaust profiles were found. The next task was to apply theoretical modifications to allow the peak engine performance to take place at the new conditions that the engine will be used in. To do so, the key differences in the engines working environment had to be established. These are tabulated below in table 4.5:

Table 4.5: *Table of Engine Requirement Changes*

Original Use in CBR 600 Motorcycle	**Modified for Formula Student Use**
Modal Working RPM of 3000-10000	Increased Modal RPM
Commercially limited fuel consumption	Not limited*
Commercially limited wear on engine	Not limited*
Power limited by fuel consumption	Not limited
Power limited by engine longevity	Not limited
Unrestricted intake air flow	Restricted intake

Table 4.5 summarises the key differences between the standard engine environment and the new engine environment. The starred items are to be taken with a heap of salt, as they are not quite "not limited". However, compared to the original specification they are something that should not have to be considered when altering the cam profiles to increase performance. The main point of consideration on table 4.5 is the last point. The air restrictor is the main reason as to why the profiles need to be altered. To accommodate these changes, the following will be applied to the cam profiles:

- The Duration of both the intake and exhaust valves will be increased. This done for one simple reason; to increase gas flow in and out of the combustion chamber. One may be mistaken to think that increase the duration will cause the gas flow to extend into the incorrect stroke sequence of the engine, however this is not the case. At higher engine RPMs the velocity of the gasses flowing though the valves increases. This, in turn, increase their momentum and so the gas can continue to flow against the motion of the piston. This means, on the intake stroke, more gas is packed into the chamber, and on the exhaust cycle, more gas is expelled from the chamber.

- The lobe overlap will be increased. The lobe overlap defines the angular duration for which both the intake and exhaust valve are simultaneously open. The overlap will be increase for the same reason the duration will be increased - due to increased gas velocity. In fact increasing the duration is somewhat ubiquitously linked to increasing the overlap. This is due to the closing and opening times of the exhaust and intake valves occurring later and earlier respectively.

- The total lift will be increased. To aid in gas flow, the total valve lift will be increased. This has to be carried out with caution as increasing the lift too much will cause the valves to clash with he piston crown. A section in the script has been included that can plot the position of the piston crown and check if the valve clashes.

Whilst performing all these alterations, all of the constraints discussed in this paper still have to be adhered to. This means the derivatives must remain below their respective threshold and the profile has to still be "poltable" on a circular cam lobe.

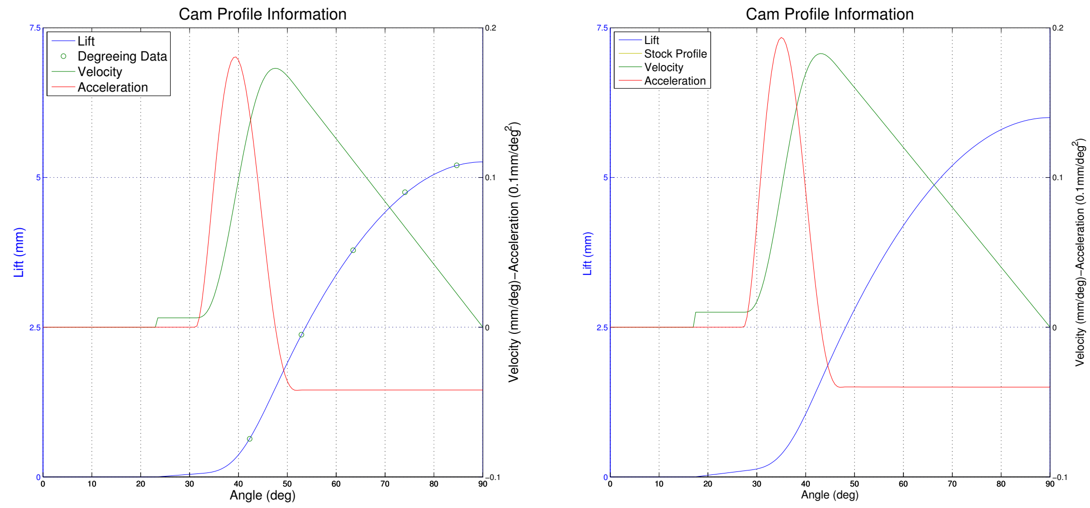
4.2.1 Creating the Optimised Profiles

Creating the optimised profiles was a long process. It was carried out in a similar way to the initial profile matching procedure, yet much more difficult do to as there were no guidelines. Initially the profiles were made to broadly match the items listed above. The profiles were then tweaked to ensure all the derivatives were in check. Finally the profiles were exported to the Excel spreadsheet and that in turn updated the CATIA file to check if the profiles were "plotable".

Six new profiles were generated, three for each valve. Each profile had a varying degree of additional lift and duration. Two of the most significantly altered profiles are discussed next.

4.2.2 Exhaust Profile Optimisation

Shown below in figure 4.6 are the stock exhaust profile and one of the optimised exhaust profiles:



(a) *Stock Exhaust Profile.* **(b)** *Optimised Exhaust Profile.*

Figure 4.6: *Stock Profile and One of the Optimised Exhaust Profiles. MathWorks 2013.*

Figure 4.6 gives a rough visual overview to an optimised exhaust profile, showing how the derivatives have not changed too strongly between sub-figure 4.6a and 4.6b. Despite the total lift being greater, the magnitude of deceleration during the nose portion is actually less than that of the stock vehicle. The maximum jerk has remained similar. The velocity has increased yet is still below the maximum velocity imposed by the tappet radius.

To give a better visual aid all 3 of the optimised exhaust profiles have been superimposed onto the same plot along with the stock profile in Figure 4.7, shown below:

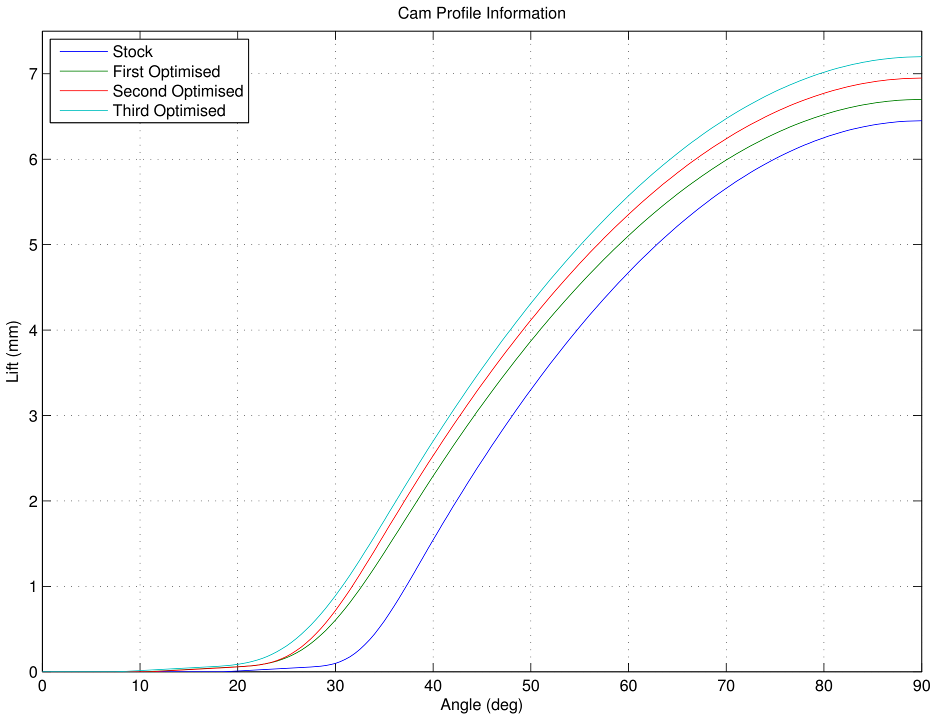
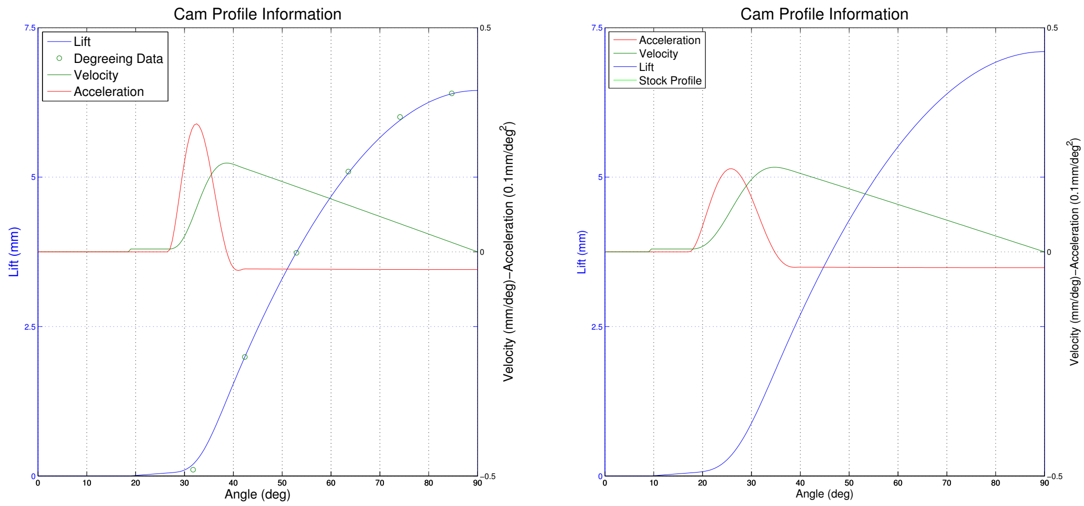


Figure 4.7: *Plot of Stock and Optimised Profile Superimposed on one Another. MathWorks 2013.*

It can be seen that the duration of all of the optimised exhaust profiles has been increased and that the lift is greater. These attributes have been obtained at the cost of the lift derivatives being closer to their thresholds. Due to there not being an exhaust restriction, it was theorised that the exhaust profile does not need to be as greatly altered as the intake profile.

4.2.3 Intake Profile Optimisation

Shown below in figure 4.8 are the stock exhaust profile and the optimised exhaust profile:



(a) *Stock Intake Profile.* **(b)** *Optimised Intake Profile.*

Figure 4.8: *Stock and Optimised Intake Profiles. MathWorks 2013.*

Figure 4.8 gives a rough visual overview of the improvements made to the intake profile. It also shows how the derivatives have not changed too strongly between sub-figure 4.8a and 4.8b, with the exception of the acceleration that has decreased. A consequence of the decreased acceleration is the jerk in the flank portion has been decreased. This, as stated earlier, should decrease the chance of vibrations occurring in the valve train. The peak velocity has remained the same as it was originally close to the limit imposed by the tappet radius.

In theory, the increased duration and lift should counteract the effect the intake restrictor has on the engine. This is more likely to affect engine performance at higher RPMs as a greater negative pressure is created in a to “draw” the gas in.

The 4 lift plots for the intake valve have been superimposed onto the same plot in Figure 4.9, shown below:

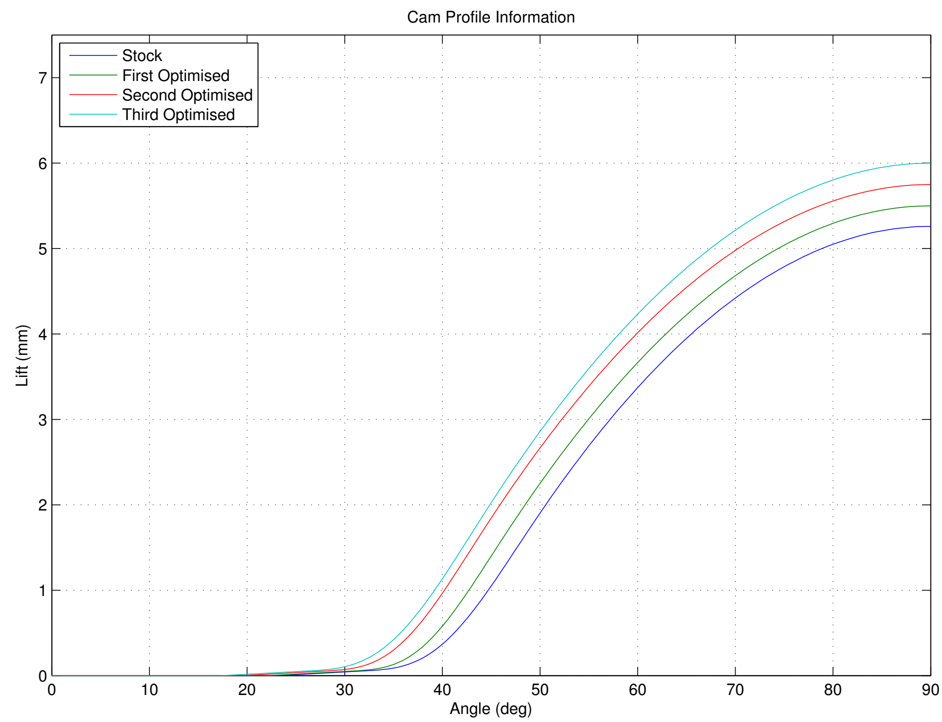


Figure 4.9: *Plot of Stock and Optimised Profile Superimposed on one Another. MathWorks 2013.*

As is the case with the exhaust valve, it can be seen that the duration of the optimised intake cycle has been increased and that the lift is greater. This again is in line with the outlined specifications set to counteract the restrictor.

The optimisations carried out have have seen the lobe separation altered minimally. This is because the increased valve durations have created a larger overlap, without the need to decrease the lobe separation angle.

4.3 Simulating the Optimisations

The Improved profiles were then converted from cam angle to crank angle to be used in GT Power, a sub program associated with GT Suite. Figure 4.10 shown below, is a screen grab of the main window:

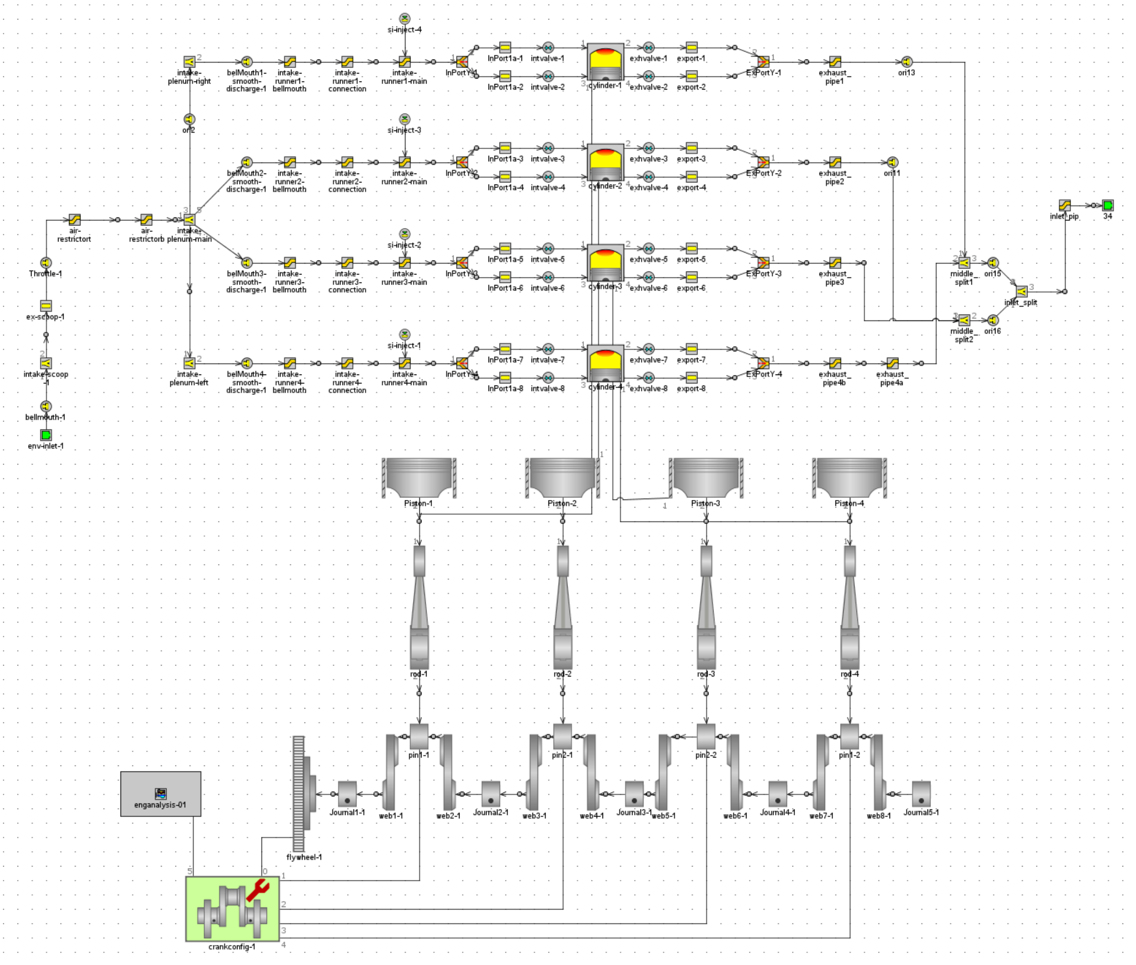


Figure 4.10: *Screen Grab of GT Power. Showing the engine model and all the component information. Model created by FST 2015. Gamma Technologies, Inc. 2014.*

GT Power models the engine as a whole, it is a very powerful piece of software that dynamically simulates the engine performance based on nearly every real-life input possible. Figure 4.10 shows that every part of the engine is represented by a "node" and within each node there are a huge number of factors that can be changed to model the engine. The combustion chambers (shown as yellow and red in a silver box) are surrounded by 2 intake and 2 exhaust valves each. These are the nodes that will be changed to use the improved cam profiles.

4.4 The Final Result

GT Power was initially run with the stock profiles that have been modeled to give a performance benchmark. The intake and exhaust valves were then changed to simulated different combination of the optimised profiles and they were also simulated. The lobe separation angles were also slightly changed for some of the simulations. The performance of each simulation was recorded in a graphical format presenting the Brake Horsepower and Brake Torque over a range of engine RPMs. Shown below, in figure 4.11, is the benchmark run and overleaf, in figure 4.12 is the best optimised results:

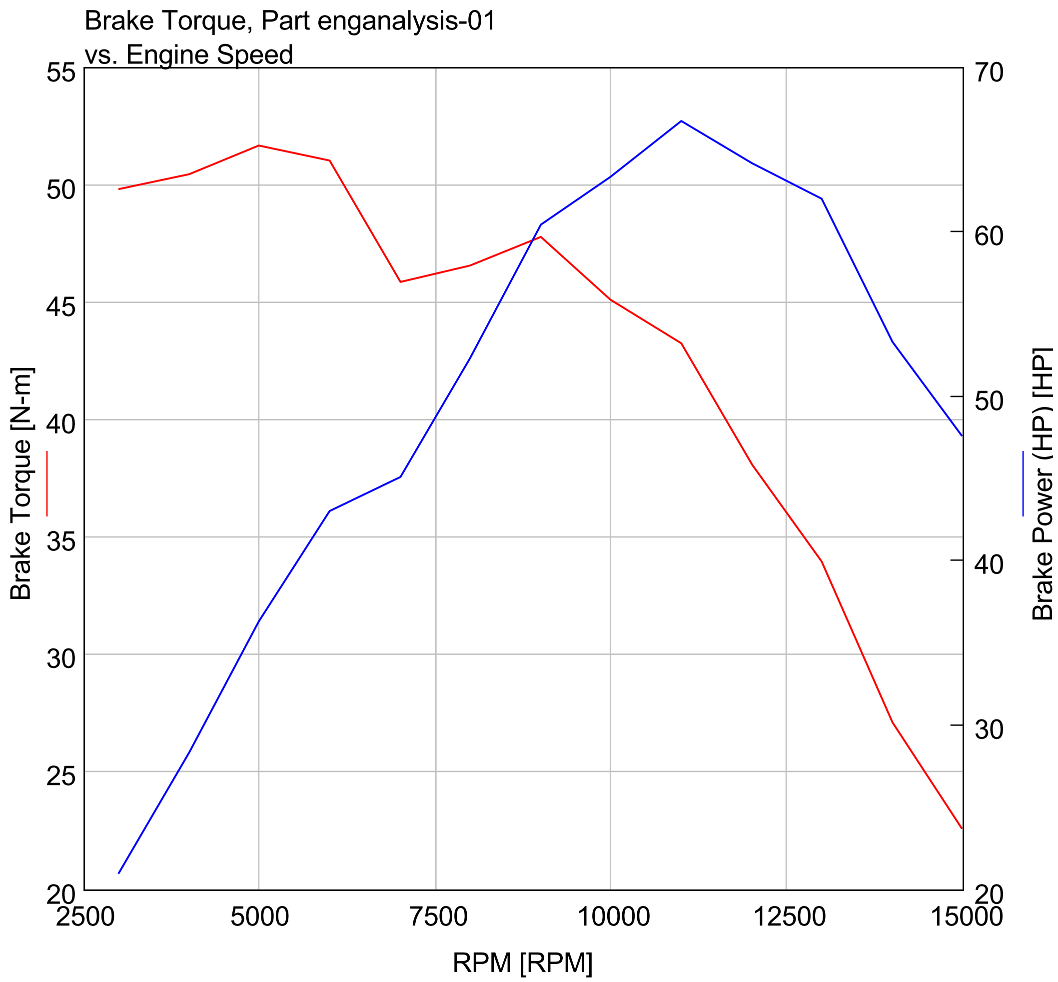


Figure 4.11: *Results from GT Power Simulation, Stock Cam. Gamma Technologies, Inc. 2014.*

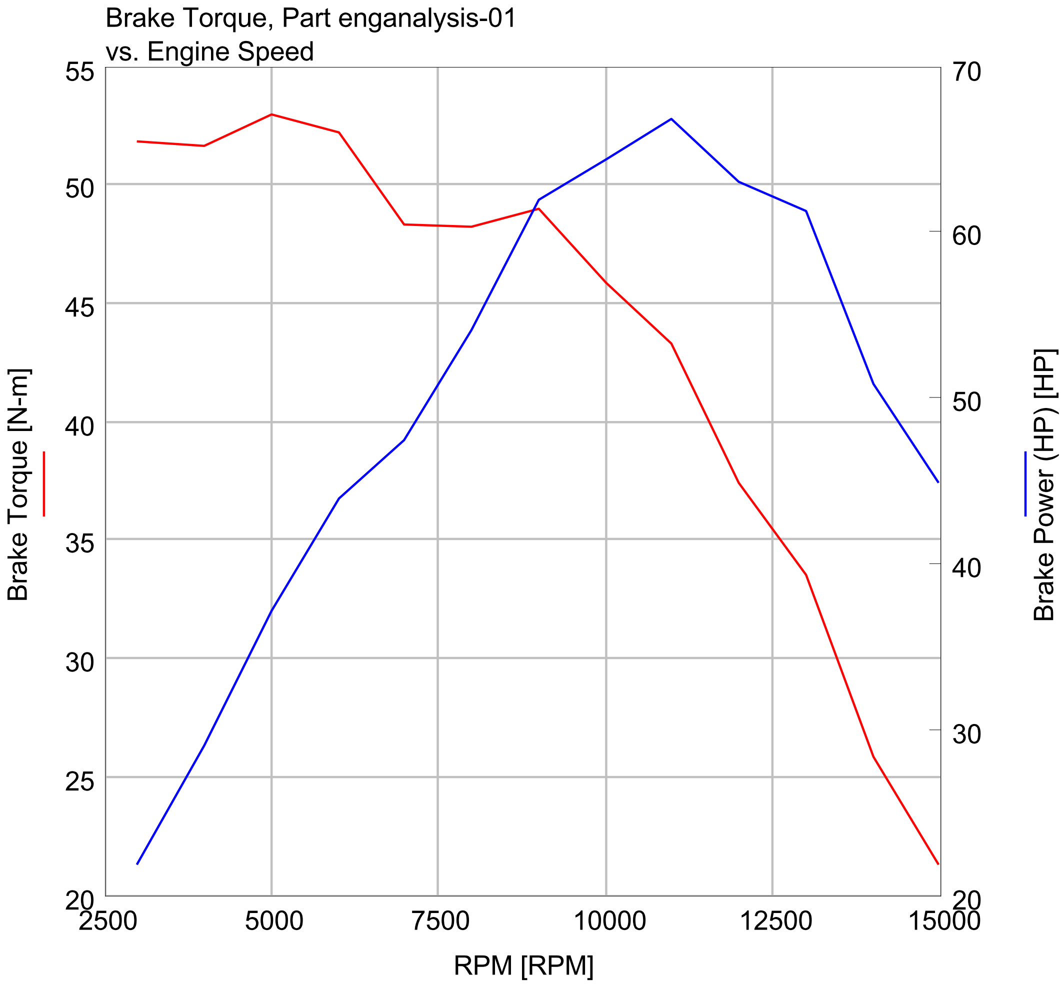


Figure 4.12: *Results from GT Power Simulation, Optimised Cam. Gamma Technologies, Inc. 2014.*

Figures 4.11 and 4.12 show the engine torque and engine power at different engine RPMs. The stock profile shows typical power and torque characteristics, yet severely reduced from the standard performance due to the intake restrictor.

To better compare the two plots, they have been superimposed on one another and are presented in figure 4.13.

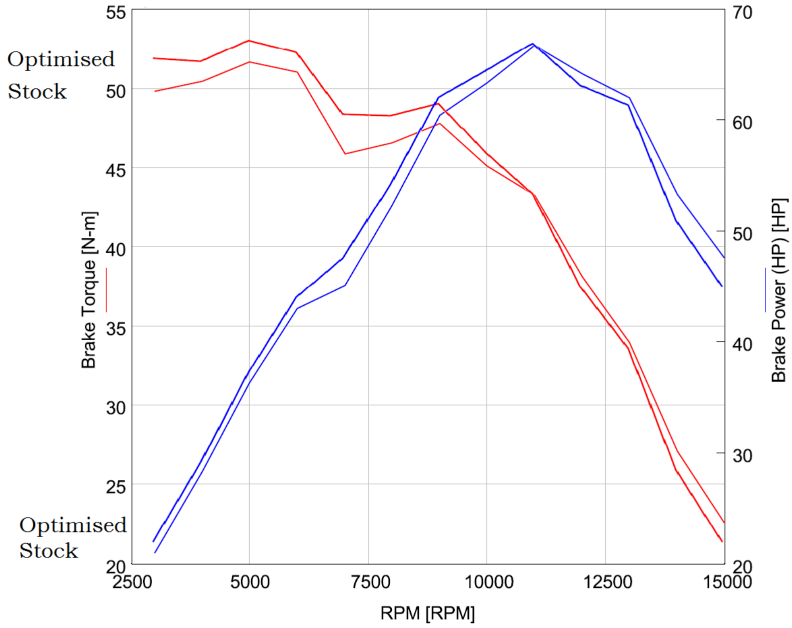


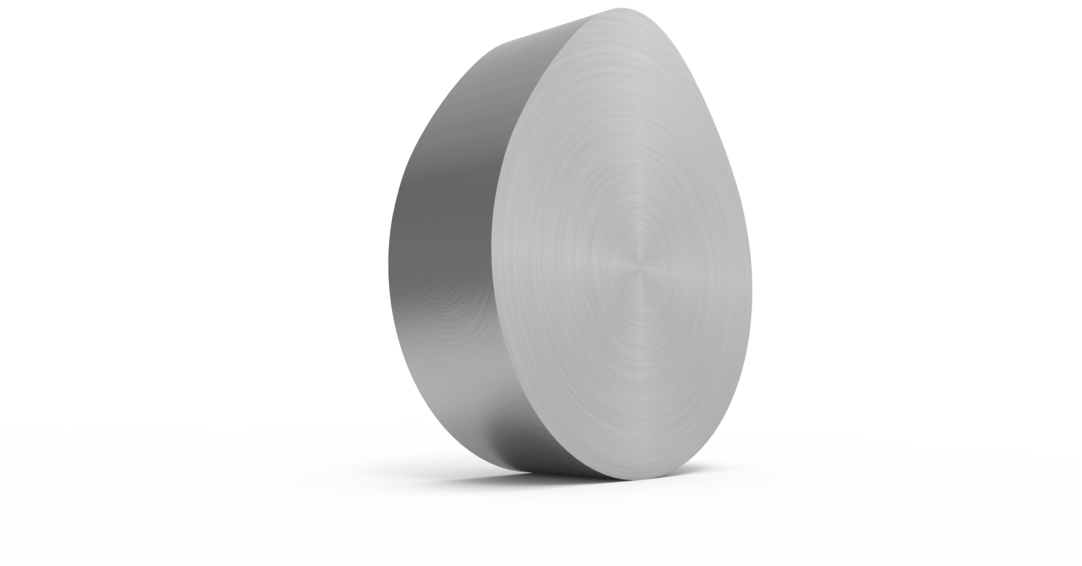
Figure 4.13: *Comparison of Optimised and Stock Simulations. The optimised profile is seen to outperform the stock profile at lower speeds. Gamma Technologies, Inc. 2014.*

The optimised profile performance is seen to outperform the stock profile in both power and torque, predominantly at the lower end of the engine speed spectrum. Most of the optimised profiles aided in performance at the high end of the revolutions yet had a slight detrimental performance at the lower end. The lowered performance at the lower revolution speed is not a desired trait.

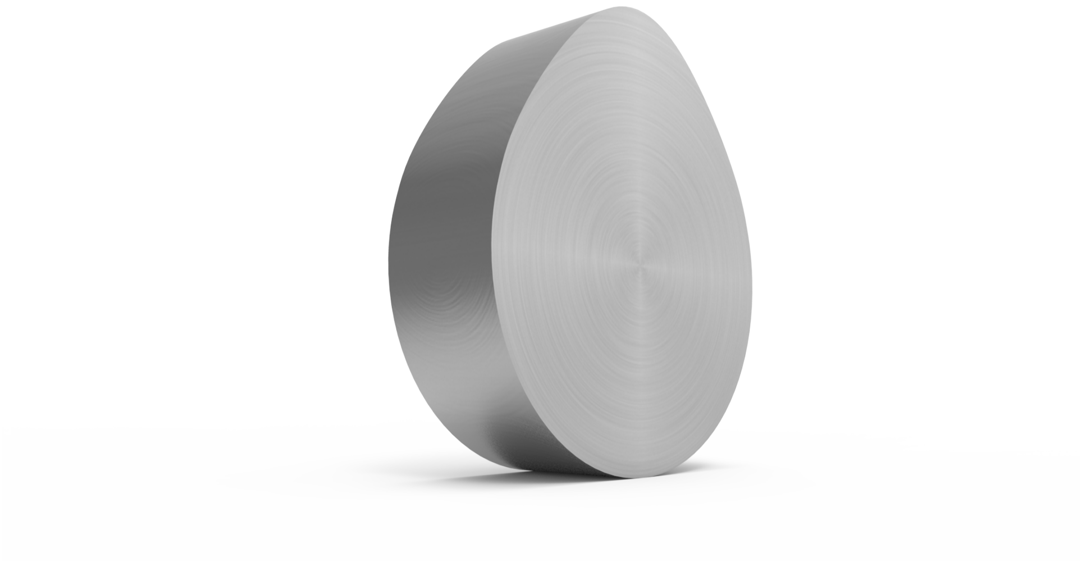
The greatest power increase occurred at 7000 RPM and is seen to be an increase of 2.5 BHP. Considering the target power increase was 10 BHP for all parts of the engine alterations. FST 2015. The results are excellent considering a quarter of this has been achieved with just altering the cams.

The profiles that gave the result shown in 4.12 were the 1st optimised intake profile and the 1st optimised exhaust profile, and a raised lobe separation Angle of 1°. Both profiles are shown in appendix C.

The work of this project amounts to two nonidentical cams, shown below:



(a) *Render of the Optimised Intake Cam*



(b) *Render of the Optimised Exhaust Cam*

Figure 4.14: *Render of the Optimised Intake and Exhaust Cams. Renders created in KeyShot 5. Luxion, Inc. 2015*

Chapter 5

Conclusion

Throughout this project, a large amount of knowledge has been gained in the niche, yet very assiduous field of cam profile design. The seemingly trivial camshaft has revealed itself to posses many traits that have a huge affect on engine performance. The importance of the lift curve and its derivatives have also been studied showing how the derivatives, that on the surface seem esoteric, are in fact important. The depth one can go into in cam profile design has been emphasised by this report. Not venturing into valve train dynamics to derive limiting factors to the high level derivatives, still left a large amount of detail to be analysed.

The optimised intake and exhaust profile are tabulated in appendix C in terms of crank angle. From theory and simulations, manufacturing these profiles and putting them in the engine should increase performance under the Formula Student racing conditions.

5.0.1 Possible Improvements

As more knowledge was gained whilst working on the project, earlier work carried out was sometimes looked back as being rudimentary and of course, in hindsight could have been done differently. The initial argument for using a modified version of the Triple Contour Curve method was that it required less experience from the designer. Now more knowledge has been gained, a better profile could be created using a more advanced method that requires more knowledge from the designer. Perhaps this is work that can be continually improved. . .

Chapter 6

Appendices

Appendix A

Profile Analysis Script

The aim of this script is to calculate the profile of the cam from the input parameters below combined with the Triple Curve Contour method.

Contents

Inputting Parameters

Units in mm and degrees unless specified otherwise

```
Lift =          8;  % Set for valve dimension + piston crown shape
Heel =          15; % Set for cam housing size

TIMINGMATRIX = [23,47;45,17];% Example timing  @ 1mm lift
Lift1mm =      (180-TIMINGMATRIX(1,1)-TIMINGMATRIX(1,2))/4;
% Change to    (180-TIMINGMATRIX(2,1)-TIMINGMATRIX(2,2))/4 for exhaust

FNKnot =       45;  % Angle where flank meets nose
RampLength =   8;   % Duration of ramp in degrees

FNHeight =     4;   % Height where flank meets nose
RFHeight =     0.1; % Height ramp reaches
BRHeight =     0;   % Lift of heel, always 0
MaxVelocity = 0.17;% mm/deg CAUTION! THIS REALLY MESSES THINGS UP!
%Crank shaft information to plot valve space diagram
```

```
w = 7.6;            %Valve Clearance
x = 60;             %Crankshaft displacement
y = 120;            %Conrod length (joint centre to joint centre)
z = 20;             %"piston rod" distance
VAngle = 80;        %Angle from vertical valve is angled N.B. exhaust-intake

graph = true;
export = false;
ClashError = false;
```

Finding Nose Constants

```
syms Alpha
```

C

```
c = 0.5;
cOld = 1;
while abs(c - cOld) > 0.000000001
    cOld = c;
    c = -(acosd(((-cOld*pi*(FNHeight-Lift)*sind((FNKnot-90)*cOld))/(MaxVelocity*180))+1)):...
/(FNKnot-90);
end
if c < 0.0000001
    errordlg('!!!PROFILE ANALYSIS TERMINATED. SIN FUNCTION ERROR!!!','***Error***')
    return
elseif c >= 180/(90-FNKnot)
    errordlg('!!!ERROR, NOSE GRADIENT INCORRECT!!!','***Error***')
end
```

B

```
b =(FNHeight-Lift)/(cosd((FNKnot-90)*c)-1);
```

A

```
a = Lift-b;
```

NOSE FORMULEA

```
NOSE = a+(b*cos((Alpha-90)*pi*c/180));
RadiiNOSE           = NOSE;
VelocitiesNOSE      = diff(NOSE, Alpha, 1);
AccelerationsNOSE   = diff(NOSE, Alpha, 2);
JerksNOSE           = diff(NOSE, Alpha, 3);
CracklesNOSE        = diff(NOSE, Alpha, 4);
```

FINDING VALUES USED FOR FINDING FLANK EQUATION

```
Alpha = FNKnot;
FNAcceleration = eval(AccelerationsNOSE);
FNJerk         = eval(JerksNOSE);
```

Finding Ramp Gradient

D

```
d = (RFHeight-BRHeight)/RampLength;
```

FINDING VALUES USED FOR FINDING FLANK EQUATION

```
RFVelocity     = d;
RFAcceleration = 0;
RFJerk         = 0;
```

Finding Flank Constants

```
syms Alpha RFKnot FLANK
```

F,G,H,I,J,K,L,M,N

```
FlankMatrix = [ FNKnot^7     FNKnot^6     FNKnot^5    FNKnot^4    FNKnot^3    FNKnot^2 FNKnot 1;...
                RFKnot^7     RFKnot^6     RFKnot^5    RFKnot^4    RFKnot^3    RFKnot^2 RFKnot 1;...
                7*FNKnot^6   6*FNKnot^5   5*FNKnot^4  4*FNKnot^3  3*FNKnot^2 2*FNKnot 1      0;...
                7*RFKnot^6   6*RFKnot^5   5*RFKnot^4  4*RFKnot^3  3*RFKnot^2 2*RFKnot 1      0;...
                42*FNKnot^5  30*FNKnot^4  20*FNKnot^3 12*FNKnot^2 6*FNKnot   2        0      0;...
                42*RFKnot^5  30*RFKnot^4  20*RFKnot^3 12*RFKnot^2 6*RFKnot   2        0      0;...
                210*FNKnot^4 120*FNKnot^3 60*FNKnot^2 24*FNKnot   6          0        0      0;...
                210*RFKnot^4 120*RFKnot^3 60*RFKnot^2 24*RFKnot   6          0        0      0];
AnswersMatrix = [ FNHeight;...
                  RFHeight;...
                  MaxVelocity;...
                  RFVelocity;...
                  FNAcceleration;...
                  RFAcceleration;...
                  FNJerk;...
                  RFJerk ];
Constants = FlankMatrix\AnswersMatrix;
FLANK =  (Constants(1)*(Alpha.^7))...
        +(Constants(2)*(Alpha.^6))...
        +(Constants(3)*(Alpha.^5))...
        +(Constants(4)*(Alpha.^4))...
        +(Constants(5)*(Alpha.^3))...
        +(Constants(6)*(Alpha.^2))...
        +(Constants(7)*(Alpha.^1))...
        +(Constants(8)*(Alpha.^0));
Alpha = Lift1mm;
for RFKnot = (0:0.01:40);
    if abs(eval(FLANK)-1) < 0.05
break
    end
end
BRKnot = RFKnot-RampLength;
syms Alpha
FLANK = eval(FLANK);
```

FLANK FORMULEA

```
RadiiFLANK         = FLANK;
VelocitiesFLANK    = diff(FLANK, Alpha, 1);
AccelerationsFLANK = diff(FLANK, Alpha, 2);
JerksFLANK         = diff(FLANK, Alpha, 3);
CracklesFLANK      = diff(FLANK, Alpha, 4);
```

Finding Ramp Equation

```
syms Alpha
```

E

```
e = RFHeight-(d*RFKnot);
```

RAMP FORMULEA

```
RAMP = (d*Alpha)+e;
RadiiRAMP          = RAMP;
VelocitiesRAMP     = diff(RAMP, Alpha, 1);
AccelerationsRAMP  = diff(RAMP, Alpha, 2);
```

```
JerksRAMP          = diff(RAMP, Alpha, 3);
CracklesRAMP       = diff(RAMP, Alpha, 4);
```

Dimensioning Variables

```
Radii = zeros(1,181);
Velocities = zeros(1,181);
Accelerations = zeros(1,181);
Jerks = zeros(1,181);
Crackles = zeros(1,181);
```

Finding Values

```
for Alpha = (0:0.5:90);
    if Alpha <= BRKnot
        Radii(1+(Alpha*2))         = 0;
        Velocities(1+(Alpha*2))    = 0;
        Accelerations(1+(Alpha*2)) = 0;
        Jerks(1+(Alpha*2))         = 0;
        Crackles(1+(Alpha*2))      = 0;
    elseif Alpha <= RFKnot
        Radii(1+(Alpha*2))         = eval(RadiiRAMP);
        Velocities(1+(Alpha*2))    = eval(VelocitiesRAMP);
        Accelerations(1+(Alpha*2)) = eval(AccelerationsRAMP);
        Jerks(1+(Alpha*2))         = eval(JerksRAMP);
        Crackles(1+(Alpha*2))      = eval(CracklesRAMP);
    elseif Alpha <= FNKnot
        Radii(1+(Alpha*2))         = eval(RadiiFLANK);
        Velocities(1+(Alpha*2))    = eval(VelocitiesFLANK);
        Accelerations(1+(Alpha*2)) = eval(AccelerationsFLANK);
        Jerks(1+(Alpha*2))         = eval(JerksFLANK);
        Crackles(1+(Alpha*2))      = eval(CracklesFLANK);
    else
        Radii(1+(Alpha*2))         = eval(RadiiNOSE);
        Velocities(1+(Alpha*2))    = eval(VelocitiesNOSE);
        Accelerations(1+(Alpha*2)) = eval(AccelerationsNOSE);
        Jerks(1+(Alpha*2))         = eval(JerksNOSE);
        Crackles(1+(Alpha*2))      = eval(CracklesNOSE);
    end
end
```

Piston Crown Position

```
for Alpha = (0:0.5:90);
    CrankAlpha(1+(Alpha*2)) = (Alpha/2)-TIMINGMATRIX(1,1)-(Lift1mm/2);
    xPrime = x*cosd(CrankAlpha);
    yPrime = (y.^2 - (sind(CrankAlpha)*x).^2).^0.5;
    VValveSpace = w + x + y + z -(xPrime+yPrime+z); % Vertical valve space
    ValveSpace = VValveSpace/sind(VAngle); %Actual valve space, angle considered
    if (ValveSpace(1+(Alpha*2)) < Radii(1+(Alpha*2)))
        ClashError = true;
    end
end
Alpha = (0:0.5:90);
```

Transposing Matrices

```
Radii = transpose(Radii);
```

```
Velocities = transpose(Velocities);
Accelerations = transpose(Accelerations);
Jerks = transpose(Jerks);
Crackles = transpose(Crackles);
ValveSpace = transpose(ValveSpace);
Alpha = transpose(Alpha);
```

Creating Graphs

```
x = Alpha;
y1=Radii; y2=Velocities; y3=Accelerations; y4=Jerks;
ylabels{1} = 'Lift (mm)';
ylabels{2} = 'Velocity (mm/deg)';
ylabels{3} = 'Acceleration (mm/deg^2)';
ylabels{4} = 'Jerk (mm/deg^3)';
[ax,hlines] = ploty4(x,y1,x,y2,x,y3,x,y4,ylabels);
leghandle = legend(hlines, 'Lift','Velocity','Acceleration','Jerk',2);
ploty4
```

Displaying Dialog

```
if ClashError == true
errordlg('The current configuration will cause the valve;...
to clash with the crown','Valve - Crown Clash')
end
```

Writing Data to Excel

```
if export == true
filename = 'PROFILE DIMENSIONING.xls';
Excel = [(Radii+Heel)*1000 , Velocities , Accelerations , Jerks , Crackles];
sheet = 2;
xlRange = 'B1';
xlswrite(filename,Excel,sheet,xlRange)
disp('Exported to Excel')
end
```

Appendix B

Table 6.1: *MATLAB Script Output For an Example Intake Profile.*

Lift (microns)	Velocity (mm/deg)	Acceleration (mm/deg^2)	Jerk (mm/deg^3)	Crackle (mm/deg^4)
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15000	0.0000	0.0000	0.0000	0.0000
15006	0.0000	0.0000	0.0000	0.0000
15011	0.0000	0.0000	0.0000	0.0000
15017	0.0000	0.0000	0.0000	0.0000
15022	0.0000	0.0000	0.0000	0.0000
15028	0.0000	0.0000	0.0000	0.0000
15033	0.0000	0.0000	0.0000	0.0000
15039	0.0000	0.0000	0.0000	0.0000
15044	0.0000	0.0000	0.0000	0.0000
15050	0.0000	0.0000	0.0000	0.0000
15056	0.0000	0.0000	0.0000	0.0000
15061	0.0000	0.0000	0.0000	0.0000
15067	0.0000	0.0000	0.0000	0.0000
15072	0.0000	0.0000	0.0000	0.0000
15078	0.0000	0.0000	0.0000	0.0000
15083	0.0000	0.0000	0.0000	0.0000
15089	0.0000	0.0000	0.0000	0.0000

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Table 6.1 – continued from previous page

Lift (microns)	Velocity (mm/deg)	Acceleration (mm/deg^2)	Jerk (mm/deg^3)	Crackle (mm/deg^4)
15094	0.0000	0.0000	0.0000	0.0000
15100	0.0000	0.0000	0.0000	0.0000
15106	1.11E-02	4.39E-05	1.72E-04	3.18E-04
15111	1.12E-02	1.67E-04	3.19E-04	2.71E-04
15117	1.13E-02	3.59E-04	4.43E-04	2.27E-04
15122	1.15E-02	6.07E-04	5.46E-04	1.86E-04
15128	1.19E-02	9.02E-04	6.29E-04	1.48E-04
15134	1.24E-02	1.23E-03	6.94E-04	1.12E-04
15141	1.32E-02	1.59E-03	7.42E-04	7.94E-05
15148	1.40E-02	1.97E-03	7.74E-04	4.93E-05
15155	1.51E-02	2.37E-03	7.92E-04	2.18E-05
15163	1.64E-02	2.76E-03	7.96E-04	-3.33E-06
15171	1.79E-02	3.16E-03	7.89E-04	-2.61E-05
15181	1.96E-02	3.55E-03	7.71E-04	-4.65E-05
15191	2.14E-02	3.93E-03	7.43E-04	-6.47E-05
15202	2.35E-02	4.29E-03	7.06E-04	-8.07E-05
15214	2.57E-02	4.63E-03	6.62E-04	-9.48E-05
15228	2.81E-02	4.95E-03	6.12E-04	-1.07E-04
15243	3.07E-02	5.24E-03	5.56E-04	-1.17E-04
15259	3.34E-02	5.51E-03	4.95E-04	-1.25E-04
15276	3.62E-02	5.74E-03	4.31E-04	-1.32E-04
15295	3.91E-02	5.94E-03	3.64E-04	-1.37E-04
15315	4.21E-02	6.10E-03	2.94E-04	-1.40E-04
15337	4.52E-02	6.23E-03	2.24E-04	-1.42E-04
15360	4.83E-02	6.33E-03	1.52E-04	-1.43E-04
15385	5.15E-02	6.38E-03	8.09E-05	-1.42E-04
15412	5.47E-02	6.41E-03	1.03E-05	-1.40E-04
15440	5.79E-02	6.39E-03	-5.90E-05	-1.37E-04
15470	6.11E-02	6.35E-03	-1.26E-04	-1.33E-04
15501	6.43E-02	6.27E-03	-1.92E-04	-1.28E-04
15534	6.74E-02	6.16E-03	-2.54E-04	-1.22E-04
15568	7.04E-02	6.02E-03	-3.13E-04	-1.15E-04
15604	7.34E-02	5.84E-03	-3.68E-04	-1.07E-04
15642	7.62E-02	5.65E-03	-4.20E-04	-9.86E-05
15680	7.90E-02	5.43E-03	-4.67E-04	-8.97E-05
15721	8.17E-02	5.18E-03	-5.09E-04	-8.04E-05
15762	8.42E-02	4.92E-03	-5.47E-04	-7.06E-05
15805	8.66E-02	4.64E-03	-5.80E-04	-6.05E-05
15849	8.88E-02	4.34E-03	-6.08E-04	-5.02E-05
15894	9.09E-02	4.03E-03	-6.30E-04	-3.97E-05
15940	9.29E-02	3.71E-03	-6.47E-04	-2.91E-05
15986	9.46E-02	3.38E-03	-6.59E-04	-1.85E-05
16034	9.62E-02	3.05E-03	-6.66E-04	-7.99E-06
16083	9.77E-02	2.72E-03	-6.67E-04	2.34E-06
16132	9.90E-02	2.38E-03	-6.63E-04	1.24E-05
16182	1.00E-01	2.05E-03	-6.55E-04	2.22E-05
16232	1.01E-01	1.73E-03	-6.41E-04	3.15E-05
16283	1.02E-01	1.41E-03	-6.23E-04	4.03E-05
16334	1.02E-01	1.11E-03	-6.01E-04	4.86E-05
16385	1.03E-01	8.14E-04	-5.75E-04	5.62E-05
16436	1.03E-01	5.33E-04	-5.45E-04	6.31E-05
16488	1.03E-01	2.69E-04	-5.12E-04	6.91E-05

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Table 6.1 – continued from previous page

Lift (microns)	Velocity (mm/deg)	Acceleration (mm/deg^2)	Jerk (mm/deg^3)	Crackle (mm/deg^4)
16540	1.04E-01	2.20E-05	-4.76E-04	7.42E-05
16592	1.04E-01	-2.07E-04	-4.38E-04	7.83E-05
16643	1.03E-01	-4.16E-04	-3.98E-04	8.14E-05
16695	1.03E-01	-6.04E-04	-3.57E-04	8.33E-05
16746	1.03E-01	-7.72E-04	-3.15E-04	8.40E-05
16798	1.02E-01	-9.19E-04	-2.73E-04	8.34E-05
16849	1.02E-01	-1.05E-03	-2.32E-04	8.13E-05
16899	1.01E-01	-1.15E-03	-1.92E-04	7.78E-05
16950	1.01E-01	-1.24E-03	-1.54E-04	7.27E-05
17000	1.00E-01	-1.31E-03	-1.19E-04	6.60E-05
17050	9.93E-02	-1.37E-03	-1.19E-04	1.63E-06
17099	9.86E-02	-1.42E-03	-1.18E-04	1.70E-06
17148	9.79E-02	-1.48E-03	-1.17E-04	1.77E-06
17197	9.72E-02	-1.54E-03	-1.16E-04	1.84E-06
17246	9.64E-02	-1.60E-03	-1.15E-04	1.91E-06
17294	9.56E-02	-1.66E-03	-1.14E-04	1.98E-06
17341	9.47E-02	-1.71E-03	-1.13E-04	2.05E-06
17388	9.38E-02	-1.77E-03	-1.12E-04	2.11E-06
17435	9.29E-02	-1.83E-03	-1.11E-04	2.18E-06
17481	9.20E-02	-1.88E-03	-1.10E-04	2.25E-06
17527	9.11E-02	-1.94E-03	-1.09E-04	2.31E-06
17572	9.01E-02	-1.99E-03	-1.08E-04	2.38E-06
17617	8.91E-02	-2.04E-03	-1.06E-04	2.44E-06
17661	8.80E-02	-2.10E-03	-1.05E-04	2.50E-06
17705	8.70E-02	-2.15E-03	-1.04E-04	2.57E-06
17748	8.59E-02	-2.20E-03	-1.03E-04	2.63E-06
17791	8.48E-02	-2.25E-03	-1.01E-04	2.69E-06
17833	8.36E-02	-2.30E-03	-9.99E-05	2.75E-06
17875	8.25E-02	-2.35E-03	-9.85E-05	2.81E-06
17916	8.13E-02	-2.40E-03	-9.71E-05	2.87E-06
17956	8.01E-02	-2.45E-03	-9.56E-05	2.92E-06
17996	7.88E-02	-2.50E-03	-9.42E-05	2.98E-06
18035	7.76E-02	-2.54E-03	-9.27E-05	3.04E-06
18073	7.63E-02	-2.59E-03	-9.11E-05	3.09E-06
18111	7.50E-02	-2.63E-03	-8.96E-05	3.14E-06
18148	7.37E-02	-2.68E-03	-8.80E-05	3.20E-06
18185	7.23E-02	-2.72E-03	-8.64E-05	3.25E-06
18221	7.10E-02	-2.76E-03	-8.47E-05	3.30E-06
18256	6.96E-02	-2.81E-03	-8.31E-05	3.35E-06
18290	6.81E-02	-2.85E-03	-8.14E-05	3.40E-06
18324	6.67E-02	-2.89E-03	-7.97E-05	3.45E-06
18357	6.53E-02	-2.93E-03	-7.79E-05	3.50E-06
18389	6.38E-02	-2.97E-03	-7.62E-05	3.54E-06
18421	6.23E-02	-3.00E-03	-7.44E-05	3.59E-06
18451	6.08E-02	-3.04E-03	-7.26E-05	3.63E-06
18481	5.93E-02	-3.08E-03	-7.08E-05	3.67E-06
18511	5.77E-02	-3.11E-03	-6.89E-05	3.71E-06
18539	5.61E-02	-3.14E-03	-6.71E-05	3.76E-06
18567	5.46E-02	-3.18E-03	-6.52E-05	3.79E-06
18594	5.30E-02	-3.21E-03	-6.33E-05	3.83E-06
18620	5.14E-02	-3.24E-03	-6.13E-05	3.87E-06
18645	4.97E-02	-3.27E-03	-5.94E-05	3.91E-06

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Table 6.1 – continued from previous page

Lift (microns)	Velocity (mm/deg)	Acceleration (mm/deg^2)	Jerk (mm/deg^3)	Crackle (mm/deg^4)
18669	4.81E-02	-3.30E-03	-5.74E-05	3.94E-06
18693	4.64E-02	-3.33E-03	-5.55E-05	3.97E-06
18716	4.48E-02	-3.36E-03	-5.35E-05	4.01E-06
18738	4.31E-02	-3.38E-03	-5.14E-05	4.04E-06
18759	4.14E-02	-3.41E-03	-4.94E-05	4.07E-06
18779	3.97E-02	-3.43E-03	-4.74E-05	4.10E-06
18799	3.79E-02	-3.45E-03	-4.53E-05	4.13E-06
18817	3.62E-02	-3.48E-03	-4.33E-05	4.15E-06
18835	3.45E-02	-3.50E-03	-4.12E-05	4.18E-06
18852	3.27E-02	-3.52E-03	-3.91E-05	4.20E-06
18867	3.10E-02	-3.54E-03	-3.70E-05	4.22E-06
18883	2.92E-02	-3.55E-03	-3.49E-05	4.25E-06
18897	2.74E-02	-3.57E-03	-3.27E-05	4.27E-06
18910	2.56E-02	-3.59E-03	-3.06E-05	4.28E-06
18922	2.38E-02	-3.60E-03	-2.84E-05	4.30E-06
18934	2.20E-02	-3.62E-03	-2.63E-05	4.32E-06
18944	2.02E-02	-3.63E-03	-2.41E-05	4.33E-06
18954	1.84E-02	-3.64E-03	-2.20E-05	4.35E-06
18963	1.66E-02	-3.65E-03	-1.98E-05	4.36E-06
18970	1.47E-02	-3.66E-03	-1.76E-05	4.37E-06
18977	1.29E-02	-3.67E-03	-1.54E-05	4.38E-06
18983	1.11E-02	-3.67E-03	-1.32E-05	4.39E-06
18988	9.22E-03	-3.68E-03	-1.10E-05	4.40E-06
18993	7.38E-03	-3.69E-03	-8.82E-06	4.40E-06
18996	5.54E-03	-3.69E-03	-6.62E-06	4.41E-06
18998	3.69E-03	-3.69E-03	-4.41E-06	4.41E-06
19000	1.85E-03	-3.69E-03	-2.21E-06	4.41E-06
19000	0.00E+00	-3.69E-03	0.00E+00	4.41E-06

Appendix C

Table 6.2: *Final Optimised Exhaust Profile. Shown in terms of crank angle in 1 degree increments. All dimensions given in mm of lift.*

Data point number							
1-45	**46-90**	**91-135**	**136-180**	**181-225**	**226-270**	**271-315**	**316-361**
First point occurring at 72.5°				**Last point occurring at 432.5°**			
0.000	0.034	1.203	4.266	5.301	4.266	1.203	0.034
0.000	0.038	1.282	4.313	5.301	4.219	1.125	0.031
0.000	0.041	1.363	4.355	5.297	4.172	1.050	0.028
0.000	0.044	1.445	4.398	5.297	4.125	0.976	0.025
0.000	0.047	1.528	4.441	5.293	4.074	0.906	0.022
0.000	0.050	1.611	4.484	5.289	4.023	0.838	0.019
0.000	0.053	1.695	4.523	5.285	3.973	0.773	0.016
0.000	0.056	1.779	4.563	5.273	3.918	0.711	0.013
0.000	0.059	1.863	4.602	5.270	3.867	0.652	0.009
0.000	0.063	1.946	4.637	5.262	3.813	0.596	0.006
0.000	0.066	2.028	4.676	5.250	3.754	0.544	0.003
0.000	0.069	2.109	4.711	5.238	3.695	0.495	0.000
0.000	0.073	2.190	4.742	5.227	3.641	0.449	0.000
0.000	0.077	2.270	4.777	5.215	3.578	0.407	0.000
0.000	0.081	2.348	4.809	5.199	3.520	0.368	0.000
0.000	0.086	2.426	4.840	5.188	3.461	0.332	0.000
0.000	0.092	2.504	4.871	5.168	3.398	0.300	0.000
0.000	0.098	2.578	4.902	5.152	3.336	0.270	0.000
0.000	0.105	2.652	4.930	5.137	3.270	0.243	0.000
0.000	0.114	2.727	4.953	5.113	3.207	0.219	0.000
0.000	0.123	2.797	4.980	5.094	3.141	0.198	0.000
0.000	0.135	2.867	5.008	5.074	3.074	0.179	0.000
0.000	0.147	2.938	5.027	5.051	3.004	0.162	0.000
0.000	0.162	3.004	5.051	5.027	2.938	0.147	0.000
0.000	0.179	3.074	5.074	5.008	2.867	0.135	0.000
0.000	0.198	3.141	5.094	4.980	2.797	0.123	0.000
0.000	0.219	3.207	5.113	4.953	2.727	0.114	0.000
0.000	0.243	3.270	5.137	4.930	2.652	0.105	0.000
0.000	0.270	3.336	5.152	4.902	2.578	0.098	0.000
0.000	0.300	3.398	5.168	4.871	2.504	0.092	0.000
0.000	0.332	3.461	5.188	4.840	2.426	0.086	0.000
0.000	0.368	3.520	5.199	4.809	2.348	0.081	0.000
0.000	0.407	3.578	5.215	4.777	2.270	0.077	0.000
0.000	0.449	3.641	5.227	4.742	2.190	0.073	0.000
0.000	0.495	3.695	5.238	4.711	2.109	0.069	0.000
0.003	0.544	3.754	5.250	4.676	2.028	0.066	0.000
0.006	0.596	3.813	5.262	4.637	1.946	0.063	0.000
0.009	0.652	3.867	5.270	4.602	1.863	0.059	0.000
0.013	0.711	3.918	5.273	4.563	1.779	0.056	0.000
0.016	0.773	3.973	5.285	4.523	1.695	0.053	0.000
0.019	0.838	4.023	5.289	4.484	1.611	0.050	0.000
0.022	0.906	4.074	5.293	4.441	1.528	0.047	0.000
0.025	0.976	4.125	5.297	4.398	1.445	0.044	0.000
0.028	1.050	4.172	5.297	4.355	1.363	0.041	0.000
0.031	1.125	4.219	5.301	4.313	1.282	0.038	0.000
							0.000

Table 6.3: *Final Optimised Intake Profile. Shown in terms of crank angle in 1 degree increments. All dimensions given in mm of lift*

Data point number							
1-45	**46-90**	**91-135**	**136-180**	**181-225**	**226-270**	**271-315**	**316-361**
First point occurring at 282°					**Last point occurring at 642°**		
0.000	0.000	1.974	5.147	6.250	5.147	1.974	0.000
0.000	0.000	2.065	5.195	6.249	5.098	1.883	0.000
0.000	0.000	2.155	5.242	6.248	5.048	1.791	0.000
0.000	0.000	2.243	5.288	6.245	4.997	1.698	0.000
0.000	0.002	2.332	5.333	6.241	4.944	1.604	0.000
0.000	0.005	2.419	5.376	6.236	4.891	1.509	0.000
0.000	0.008	2.505	5.419	6.230	4.837	1.413	0.000
0.000	0.011	2.590	5.461	6.223	4.782	1.317	0.000
0.000	0.014	2.675	5.502	6.215	4.726	1.220	0.000
0.000	0.017	2.759	5.541	6.205	4.669	1.122	0.000
0.000	0.021	2.842	5.580	6.195	4.610	1.023	0.000
0.000	0.024	2.923	5.617	6.183	4.551	0.922	0.000
0.000	0.027	3.005	5.654	6.171	4.491	0.819	0.000
0.000	0.030	3.085	5.689	6.157	4.430	0.715	0.000
0.000	0.033	3.164	5.724	6.142	4.368	0.611	0.000
0.000	0.036	3.242	5.757	6.126	4.305	0.510	0.000
0.000	0.039	3.320	5.789	6.109	4.240	0.414	0.000
0.000	0.042	3.396	5.820	6.091	4.175	0.326	0.000
0.000	0.046	3.472	5.850	6.072	4.109	0.249	0.000
0.000	0.049	3.546	5.879	6.052	4.042	0.185	0.000
0.000	0.052	3.620	5.907	6.030	3.974	0.135	0.000
0.000	0.056	3.693	5.934	6.008	3.905	0.099	0.000
0.000	0.062	3.765	5.960	5.984	3.835	0.076	0.000
0.000	0.076	3.835	5.984	5.960	3.765	0.062	0.000
0.000	0.099	3.905	6.008	5.934	3.693	0.056	0.000
0.000	0.135	3.974	6.030	5.907	3.620	0.052	0.000
0.000	0.185	4.042	6.052	5.879	3.546	0.049	0.000
0.000	0.249	4.109	6.072	5.850	3.472	0.046	0.000
0.000	0.326	4.175	6.091	5.820	3.396	0.042	0.000
0.000	0.414	4.240	6.109	5.789	3.320	0.039	0.000
0.000	0.510	4.305	6.126	5.757	3.242	0.036	0.000
0.000	0.611	4.368	6.142	5.724	3.164	0.033	0.000
0.000	0.715	4.430	6.157	5.689	3.085	0.030	0.000
0.000	0.819	4.491	6.171	5.654	3.005	0.027	0.000
0.000	0.922	4.551	6.183	5.617	2.923	0.024	0.000
0.000	1.023	4.610	6.195	5.580	2.842	0.021	0.000
0.000	1.122	4.669	6.205	5.541	2.759	0.017	0.000
0.000	1.220	4.726	6.215	5.502	2.675	0.014	0.000
0.000	1.317	4.782	6.223	5.461	2.590	0.011	0.000
0.000	1.413	4.837	6.230	5.419	2.505	0.008	0.000
0.000	1.509	4.891	6.236	5.376	2.419	0.005	0.000
0.000	1.604	4.944	6.241	5.333	2.332	0.002	0.000
0.000	1.698	4.997	6.245	5.288	2.243	0.000	0.000
0.000	1.791	5.048	6.248	5.242	2.155	0.000	0.000
0.000	1.883	5.098	6.249	5.195	2.065	0.000	0.000
							0.000

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