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Camshaft Design for an Inlet-Restricted FSAE Engine

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ABSTRACT

Restricting the flow rate of air to the intake manifold is a convenient and popular method used by several motor sport disciplines to regulate engine performance. This principle is applied in the Formula SAE and Formula Student competitions, the rules of which stipulate that all the air entering the engine must pass though a 20mm diameter orifice. The restriction acts as a partially closed throttle which generates a vacuum in the inlet plenum. During the valve overlap period of the cycle, which may be as much as 100 degrees crank angle in the motorcycle engines used by most FSAE competitors, this vacuum causes reverse flow of exhaust gas into the intake runners. This, in turn, reduces the amount of fresh air entering the cylinder during the subsequent intake stroke and therefore reduces the torque produced. This effect is particularly noticeable at medium engine speeds when the time available for reverse flow is greater than at the peak torque speed.

The objective of the study described in this paper was to mitigate the reverse flow effect by reducing the duration of the valve overlap period. A thermodynamic model of the Yamaha YZF R6 engine was developed for this purpose and validated using cycle-averaged and crank-angle-resolved test data. The resulting model was then used to find the optimum values of lift, duration and timing for both the intake and exhaust valves. The camshafts required to give these valve lift profiles were designed using valve train analysis software. This process included a consideration of the dynamic forces encountered by the valve train and ensured that the resulting stresses remained within safe limits.

The new camshafts increased the torque output by up to 30% at medium engine speeds, without reducing the high-speed torque, and therefore significantly improved the vehicle drivability.

INTRODUCTION

To be eligible to compete in the Formula SAE or Formula Student events, the rules [1] dictate the use of a four-stroke engine with a maximum displacement of 610 cm^3 and that all air supplied to the engine must pass through a single Ø20 mm restrictor for a gasoline fuelled car.

The engine of choice for the majority of teams is a stock 600 cm^3 motorcycle engine [2, 3]. The Queen's Formula Racing (QFR) team employs a 2003-2005 Yamaha YZF R6 engine, the specification of which is shown in Table 1. It was chosen as it has one of the longest piston strokes of this classification of engine and, in its stock specification, is designed to produce peak torque of 68.5 Nm at 12,000 rpm. This high performance engine employs the full benefit of exhaust system tuning by using a large valve overlap to scavenge the clearance volume of exhaust gas therefore increasing high-speed torque.

Table 1: Engine Specification

03-05 Yamaha R6	
Engine	In-line, 4-cyl, DOHC.
Displacement	600 cm^3
Bore	65.5 mm
Stroke	44.5 mm
Compression Ratio	13.5
Maximum Power	92 kW (123 hp) @ 13,000 rpm
Maximum Torque	68.5 Nm @ 12,000 rpm

However, in engines with a large valve overlap, the part-load operation is poor due to a reduced induction manifold pressure. This causes exhaust gases to travel back through the cylinder, past the intake valve and into the intake system when both valves are open [4, 5]. As the engine has to be restricted under the FSAE rules, it effectively operates at part load. The standard engine is therefore not fully optimized for use in a restricted format. This affects the torque produced by the engine and ultimately the drivability of the car. To develop the engine and give a desirable torque curve during competition conditions, where engine speed is typically in the range 3000-12000 rpm, modification of the intake and exhaust camshafts was undertaken. Previous research within the team concentrated on modification of the intake cam shaft only in an older model of the engine [6, 7]. To reduce the effects of the Ø20mm restrictor, yet maintain a naturally aspirated configuration, an approach of reducing the valve overlap was used.

STOCK LIFT PROFILES

The valve lift profiles of the stock engine were carefully measured at ambient conditions using the recommended clearances between the camshaft base circle and the bucket (0.15 – 0.2 mm). The resulting profiles are plotted in Figure 1 and their durations and peak lifts are listed in Table 2.

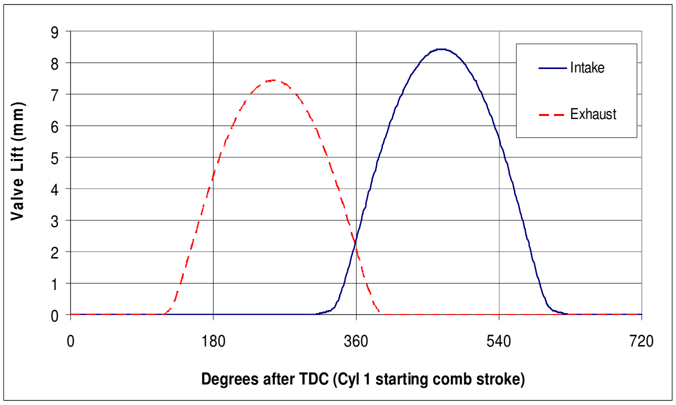


Figure 1 – Stock Valve Lift with Cold Lash Clearance

Table 2 – Stock Valvetrain Specifications

Parameter	Inlet	Exhaust
Valves per cylinder	2	2
Lift Duration at Cold Lash (degrees crank angle)	320	280
Peak Lift (degrees crank angles)	470	260
Valve Overlap at Cold Lash (degrees crank angle)	90	

As discussed earlier a high performance engine has a substantial valve overlap to increase volumetric efficiency. Due to the exhaust valve opening before BDC on the power stroke a compression wave is generated which travels along the exhaust pipe. When this wave meets an increase in area at the collector, an expansion wave is generated which travels back towards the exhaust port. If the inlet valve is open when this expansion wave arrives at the port then the wave draws particles from the intake system into the cylinder, due to a lower pressure in the cylinder than in the intake plenum. However, due to the use of a mandatory restrictor on the engine during the FSAE competition, the pressure in the intake plenum is lower than in the exhaust system and so, during the valve overlap period, there is a reverse flow of exhaust gas into the intake system.

This is demonstrated by a virtual model of the stock engine fitted with the FSAE competition intake system with the mandatory restrictor incorporated. This simulation was conducted using a computer simulation package, Virtual Engines (VE) [8]. This software bases its results on fundamental unsteady gas dynamics [9]. For reliance on the predicted data it is of utmost importance that the model is validated by empirical results [10, 11 and 12]. The engine model used was the subject of an extensive validation process described in detail by Walkingshaw [13]. Therefore the engine model could be used confidently for thermodynamic analysis. Figure 2 shows air purity just upstream of the inlet valve at 5000 rpm and wide open throttle (WOT), where 1 is fresh charge and 0 is exhaust gas. It can be seen during the overlap period between IO1 and EC1 that the gas purity drops, indicating the presence of exhaust gas in the intake manifold. Figure 3 shows mass flow rate and confirms that reverse flow occurs during the overlap period.

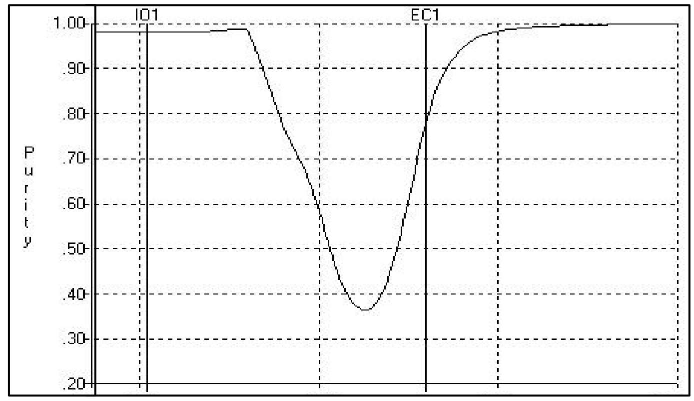


Figure 2 – Air purity at intake valve at 5000 rpm

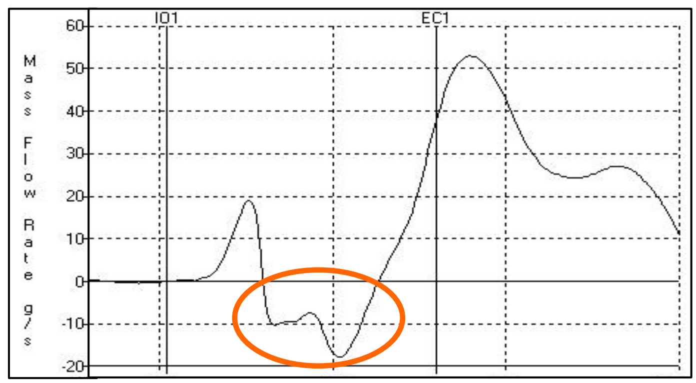


Figure 3 – Mass Flow Rate at intake valve at 5000 rpm

This analysis suggests that reducing the valve overlap and hence backflow, could result in increased torque across the engine's speed range (5000 – 12000 rpm).

In addition to the reverse flow during the overlap phase, the engine also experiences reverse flow as fresh charge escapes through the intake valve at the beginning of the compression stroke before the intake valve shuts. This is due to the standard engine being tuned for higher speeds to make use of a ramming wave to force fresh charge into the cylinder at this point in the cycle. However, at lower engine speeds, the ramming wave does not return to the port at the optimum time and so, with fixed valve timing, the piston pumps some air back through the inlet valve at the beginning of the compression stroke. This negative flow effect is highlighted by Figures 4 and 5 showing purity and mass flow rate between BDC and the intake valve closing IC1 at 5000rpm.

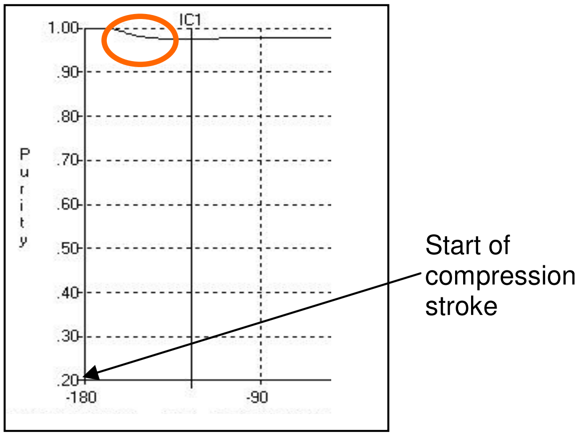


Figure 4 – Air purity at intake valve at 5000 rpm

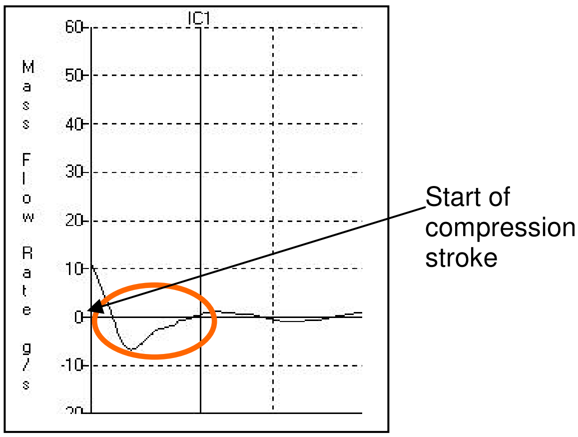


Figure 5 - Mass flow rate past intake valve at 5000 rpm

OPTIMISATION PROCESS

The validated R6 engine model was then optimized using the Automated Design (AD) software from Optimum Power Technology [8]. This software applies an extensive optimization process to the validated baseline model based on a design goal and strategy.

The design goal in this case was "maximize torque" within the specific usable speed range (5000-12000 rpm) and the design strategy was based on varying the target parameters shown in Table 3. This design space contains 2.03 x 10^{11} possible engine configurations. Obviously, for this optimization process to be reliable, the baseline engine model has to have been carefully validated.

An optimum engine design was found after analyzing 1230 engines, which took over 4.5 days on a Pentium 4 3GHz CPU with 1 Gb RAM. The optimum parameters can also be seen in Table 3.

Table 3 – Optimization Process Parameters

Parameter	Range	Increment	Optimum
Exhaust Pipe Primary Length(mm)	260-400	20	400
Intake Runner Lengths (mm)	100-300	20	260
Plenum Volume (litres)	2.5 - 6.0	0.50	4.0
Intake Cam: Peak Lift Angle(°)	458-518	5	460
Intake Cam: Lift Table Multiplier	0.8-1.0	0.01	1.0
Intake Cam: Lift Duration multiplier	0.8-1.0	0.01	0.81
Exhaust Cam: Peak Lift Angle(°)	241-331	5	250
Exhaust Cam: Lift Table Multiplier	0.8-1.0	0.01	1.0
Exhaust Cam: Lift Duration multiplier	0.8-1.0	0.01	0.86
Restrictor Diffuser Length(mm)	120-220	20	140

This automated design optimization converged on a "best engine" design that showed the greatest torque benefits in the low to mid speed range. The speed range chosen for such an optimization process is very important as it will have a considerable impact on the final optimal design. As discussed earlier, it was known that the standard valve durations and overlap were deficient at the lower speeds, so it was hoped that the new best design would improve performance in this region. At the higher speed the restrictor becomes the limiting factor on the performance.

After the optimum configuration was found, a sensitivity analysis was carried out to find which parameter had the most individual influence on engine performance. The same baseline engine model was used and the analysis showed that the intake and exhaust valve events provided the most substantial gains in performance. Figure 6, shows the simulated baseline engine torque, the torque predicted from this baseline engine with new intake and exhaust valve duration and timing events, and the predicted torque from the optimized baseline engine (table 3).

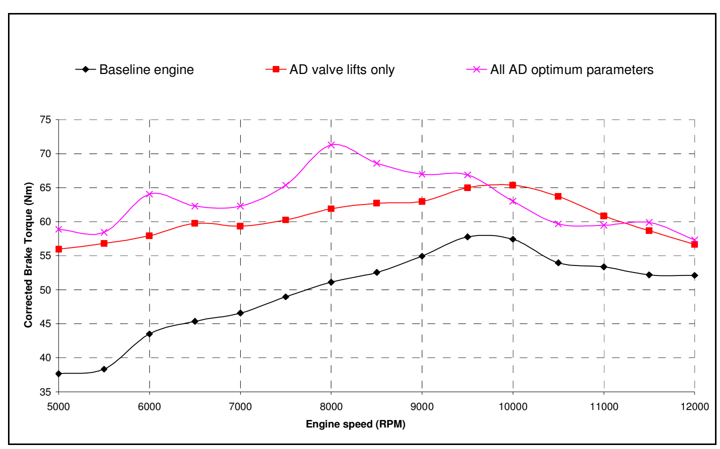


Figure 6 – Virtual Engine Simulations

Therefore, considering each of the optimum parameters in Table 3, the best torque gains could be achieved from modifying the valve lift profiles as predicted by the optimization process. Table 3 shows that this would mean keeping the valves peak lifts the same but reducing their duration.

AUTOMATED DESIGN LIFT PROFILES

During the automated design (AD) process, the virtual model reduced the negative effects of reverse flow by reducing the valve overlap period. This was done by decreasing the intake and exhaust valve durations by 19% and 14% respectively. The automated design also found that both the intake and exhaust valve peak lift optimum positions were ten degrees earlier than the stock engine. The closing angle of the inlet valve is then also advanced by this ten degrees therefore reducing the reverse flow of fresh charge out of the cylinder during the initial phase of the compression stroke. The AD's optimized valve lift profiles are shown in Figure 7, in comparison to the stock engines valve lift profiles, and the reduction in valve overlap period along with the earlier closing of the intake valve can be clearly seen. Both are shown with no clearance between bucket and camshaft lobe.

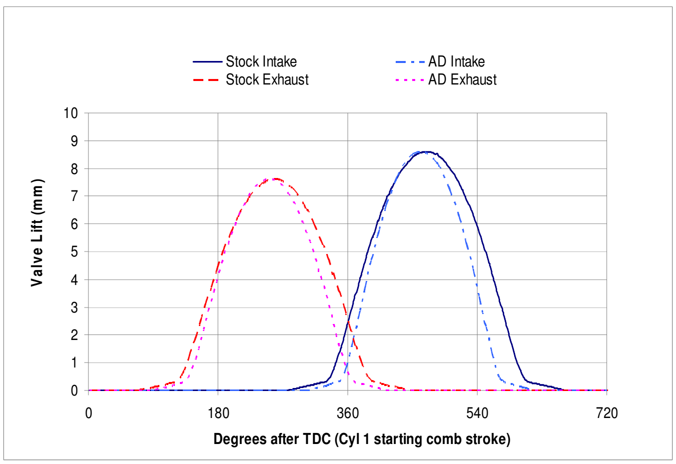


Figure 7. Automated Design and Stock Valve Lift Profiles

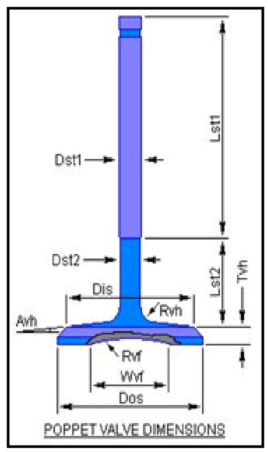
This thermodynamic optimization gives no consideration to the mechanical viability of these valve lift designs. Therefore it was necessary to check not only the mechanical feasibility of this new design, but also to determine whether it was possible to manufacture camshafts that would produce such lift profiles given the design constraints of cam base circle radius and bucket tappet diameter. There is a complex design relationship between both of these variables, the valve lift profiles and the manufacturability of the cam profiles.

MECHANICAL ANALYSIS

Changing a lift profile can impact the valve train dynamics and the durability of the components [4, 14]. To assess the mechanical implications of the proposed lift profile, a mechanical analysis was required using valvetrain mechanical analysis software called 4stHEAD [15, 16]. This software has been extensively used to analyze valve train dynamics and scrutinize the effects of particular lift profiles and components used [17, 18, 19 and 20].

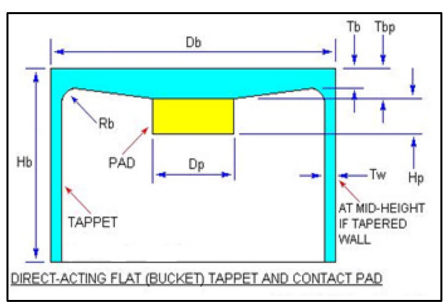
The software allows the designer to import or design lift profiles from which suitable cam profiles and all the associated valvetrain components can be analyzed and designed. The software can be provided with existing valvetrain geometry and valve lift profiles. These profiles can be manipulated such that the cam profiles meet the design criteria and fall within the limitations of the existing components and cylinder head layout. If the proposed lift profiles are more aggressive this can invoke effects such as valve float and bounce, generate higher stress, and have tribology and geometric implications.

Therefore to conduct this analysis, the valve-train and associated components within the engine needed to be measured and the data input to the software. The valves, buckets, springs and the masses for each were measured as shown in Tables 4-7 with the respective component images from the 4stHEAD software. The combustion chamber was also measured as the software can check for valve to piston interference.



Dimension (mm)	Intake	Exhaust
L_{st1}	78.8	82.2
L_{st2}	13.7	10.7
T_{vh}	2.1	2.1
D_{st1}	3.95	3.9
D_{st2}	4.55	4.6
D_{os}	25	22.05
D_{is}	23	20
A_{vh}	20	24
W_{vf}	15	10.5

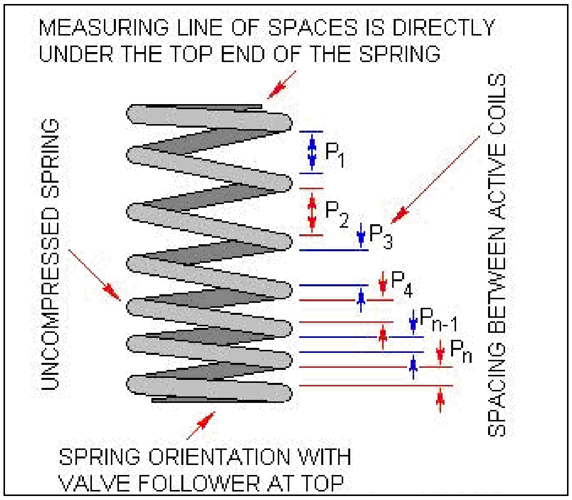
Table 4 – Valve Dimensions



Dimension (mm)	
D_b	24.44
H_b	17
T_b	2
T_{bp}	2.3
T_w	0.6
D_p	7.46

Table 5 – Bucket Follower Dimensions

Table 6. Valve Springs - Progressive Coil Spacing Dimensions



Coil Spacing (mm)	Intake Inner Spring	Intake Outer Spring	Exhaust Spring
Space 1	2.4	3.9	2.75
Space 2	2.4	4.34	3.05
Space 3	2.38	3.9	3
Space 4	2.35	2.5	2.4
Space 5	2.3	1.15	1.3
Space 6	1.45	1.3	0.75
Space 7	0.6	-	0.85
Space 8	0.75	-	-

Table 7 – Mass of Valvetrain Components

Component	Mass (g)
Intake Spring Inner	11.1
Intake Spring Outer	22.7
Intake Spring Retainer	6.4
2 Intake Collets	0.4
Exhaust Spring	35.1
Exhaust Spring Retainer	5.9
2 Exhaust Collets	0.5
Bucket	12.5

The narrower valve lift profiles produced by the optimization process were input to the 4stHEAD software. These profiles were matched precisely within the software, but the ensuing cam profiles could not be directly produced for two reasons. Firstly, the original shortened valve lifts had maximum velocities that were not compatible with the bucket follower diameters in the R6 cylinder head. The simple fact is that bucket diameter is a function of valve-lift velocity (i.e. the first derivative of the valve lift with respect to cam angle) – the higher the velocity the larger the bucket diameter to ensure contact between the cam and bucket. Secondly, there is a complex relationship between the radius of curvature of the ensuing cam profile, the base circle radius of the cam and the valve-lift acceleration (i.e. the second derivative of the valve lift with respect to cam angle) which also affects manufacturability of the cam profile. The bucket diameters and the base circle radii of the cams could not be changed and therefore the new narrower valve lift profiles had to be manipulated within 4stHEAD to ensure that feasible cam profiles could be manufactured based on the aforementioned constraints. Other factors that are constantly checked in this iterative process are the cam to follower Hertz stresses and oil film thicknesses to ensure that they remain within recommended limits. Figure 8 shows the finalized intake valves lift profiles with its associated velocity, acceleration and jerk characteristics. The key points to note here are the shape and smoothness of the acceleration profile as it is this that is manipulated to achieve the goals described previously.

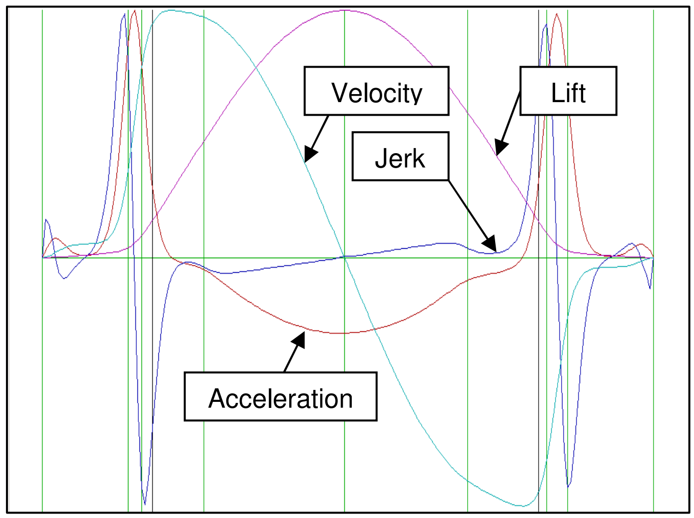


Figure 8 – Finalized Intake Valve Lift Characteristics

The output of this mechanical analysis is a set of camshaft profiles as well as new valve lift profiles. The new valve lift profiles are shown in Figure 9 compared to the original shortened profiles from the optimization process. This shows that the valve lift profiles had to be slightly more aggressive than the original shortened profiles with larger area envelopes. It was therefore necessary to run a new 1-D simulation with these new valve lift profiles to ensure they would not detrimentally affect the predicted torque. Figure 10 shows the torque curve for the original shortened profiles from the optimization process compared to the 4stHEAD profiles.

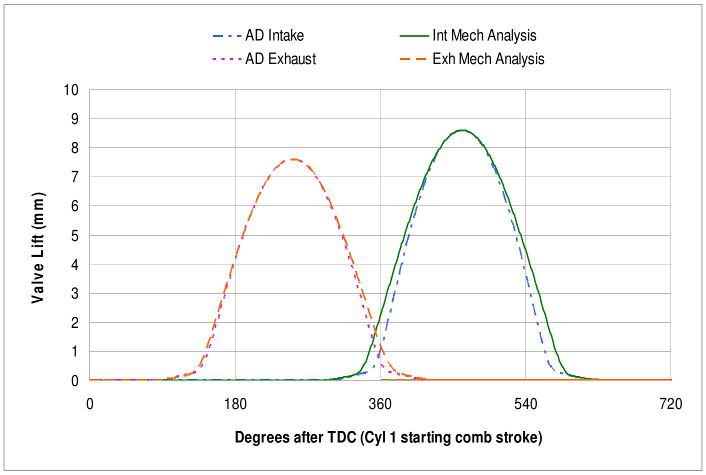


Figure 9 – AD and Mechanical Analysis Profiles

It can be seen that there is not as great a gain in predicted torque with the mechanical analysis valve lifts but still a substantial improvement upon the baseline.

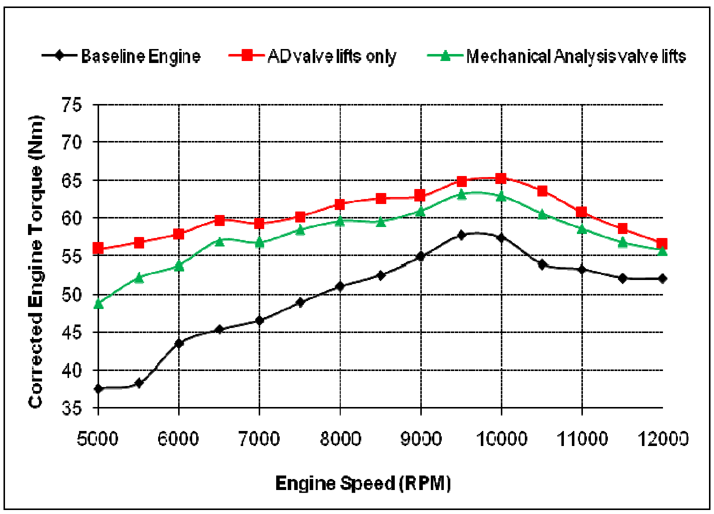


Figure 10 – AD and Mechanical Analysis Torque

The outputs from the mechanical analysis software also include the camshaft profiles, shown in comparison to the stock camshaft profiles in Figures 11 and 12.

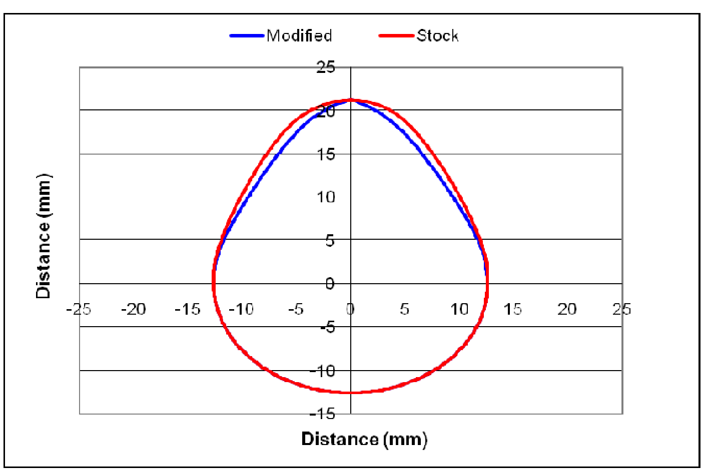


Figure 11 - Intake Camshaft Profiles

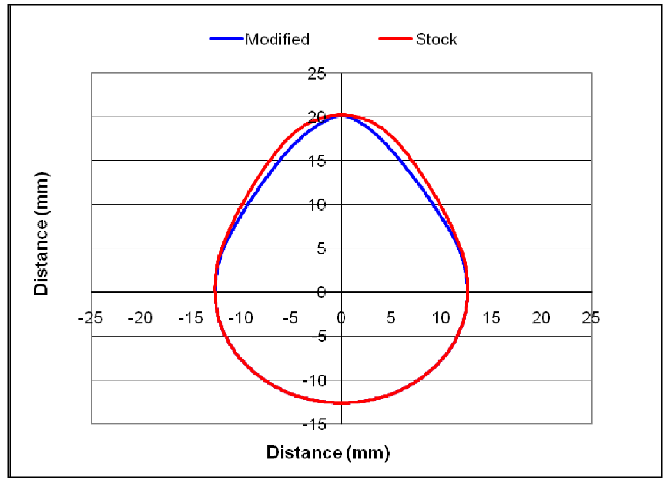


Figure 12 - Exhaust Camshaft Profiles

The 4stHEAD software was also used to check that the new valve lifts and their respective timing events would not cause any interference with the piston. In addition, it was necessary to check that the new valve lifts would not cause any problems to the dynamics of the valvetrain. First of all the dynamics of the valvetrain with the original valve lift profiles were modeled and these were then compared to the dynamics of the valvetrain with the new valve lift profiles. There were no anomalies with float, bounce or separation of the valvetrain components in the relevant speed range and the maximum stresses were within acceptable limits.

The modified camshaft profiles could be manufactured by regrinding the stock camshafts. The modified camshafts were then placed in the engine and the valvetrain was re-shimmed due to slight changes in the camshafts base circle diameters due to the manufacturing process. The valve lift profiles of the modified camshafts were measured and compared to the lift profiles that were provided by the mechanical analysis software to ensure that the manufacturing process was successful. The modified camshaft valve lifts are shown in Figure 13 in comparison to the stock profiles. The reduction in valve overlap and the earlier IVC timing are clearly visible. Stock and modified inlet camshafts are shown in Figure 14.

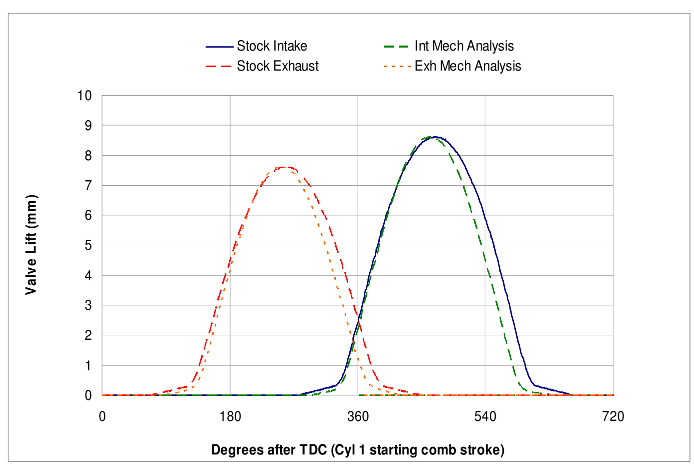


Figure 13 – Modified and Stock Valve Lifts

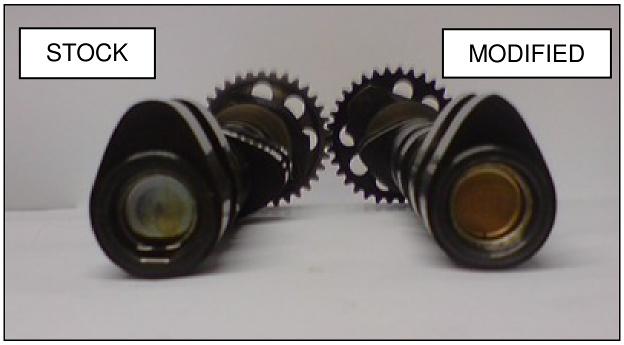


Figure 14 – Intake Stock and Modified Camshafts

The camshafts when then fitted with vernier pulley wheels so that their timing could be adjusted to the desired settings.

Using the VE model of the baseline engine with the 4stHEAD camshafts, a simulation was conducted to examine the changes the camshafts had made on the reverse flow effects. A comparison of air purity and the mass flow rate for the stock and modified camshaft engines at 5000 rpm are shown in Figures 15 and 16. These show an increase in charge purity and a decrease in the reverse flow using the modified camshafts.

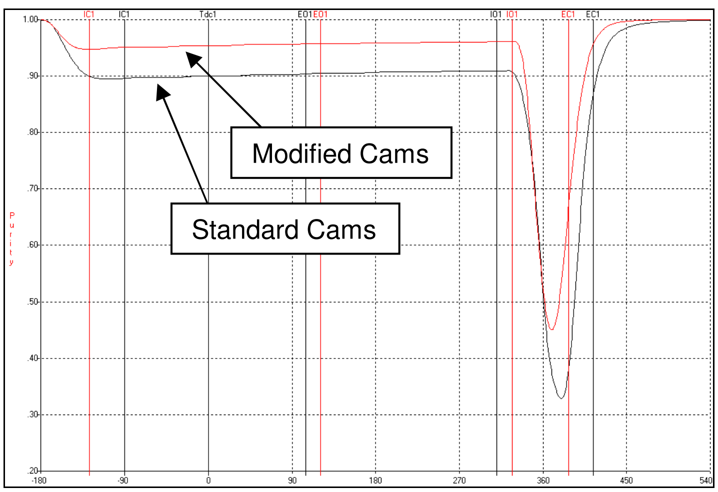


Figure 15 – Comparison of Air Purity at Intake Valve

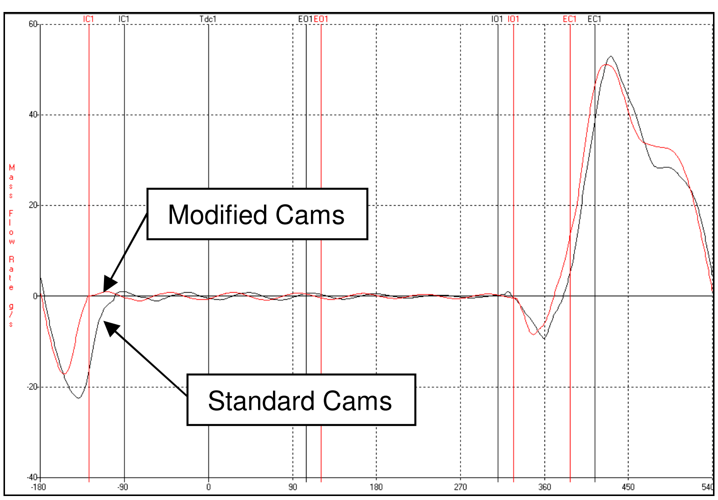


Figure 16 – Comparison of Mass Flow Rate past intake valve

EXPERIMENTAL VERIFICATION

The modified camshafts were tested on a 145kW Schenck dynamometer with the driveshaft connected to the gearbox output shaft. A Horiba MEXA-7170DEGR Exhaust Emissions Analyzer was used to measure AFR. The engine is fitted with a custom fuel injection and ignition system which are controlled by a DTA ECU. The engine was mapped at full load throughout the speed range between 3000 to 12500 rpm with fuel adjusted to leanest for best torque and ignition timing adjusted to minimum advance for best torque for both the standard camshaft and the modified camshaft configurations. Figure 17 shows the full load corrected brake torque curve comparison between the two tests.

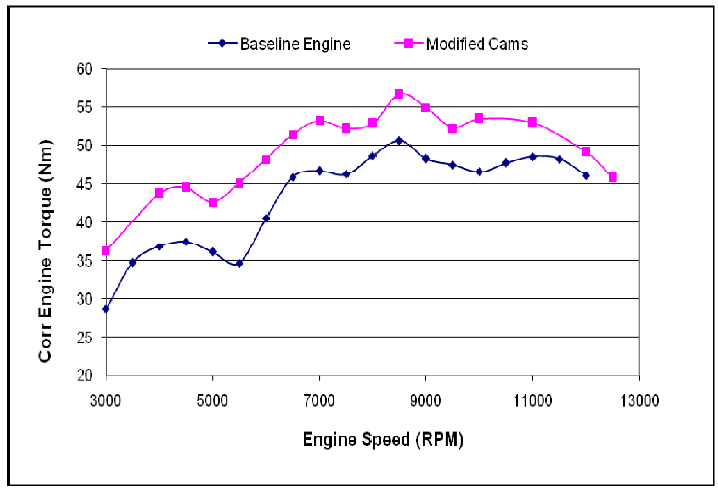


Figure 17 – Comparison of Engine Torque

It can be seen that there are large torque gains in the lower speed range with a maximum of 30% gain at 5500 rpm. The peak torque increased by around 12% at 8500 rpm and an average torque increase across the speed range of 15.3% was achieved. There are smaller gains at speeds greater than 11000 rpm due to the the airflow becoming increasingly choked through the restrictor.

During the optimization process, the Automated Design feature found that an optimum runner length would be 260 mm. It was decided to investigate this factor by increasing the length in the direction indicated by the analysis but not to the extent recommended due to difficulties in packaging such a long runner length within the FSAE car. Therefore an extension of 50mm in runner length was implemented and the engine remapped at full load. A comparison of the engine with the extended runners and modified camshafts is shown compared to the standard runners and modified camshafts in Figure 18.

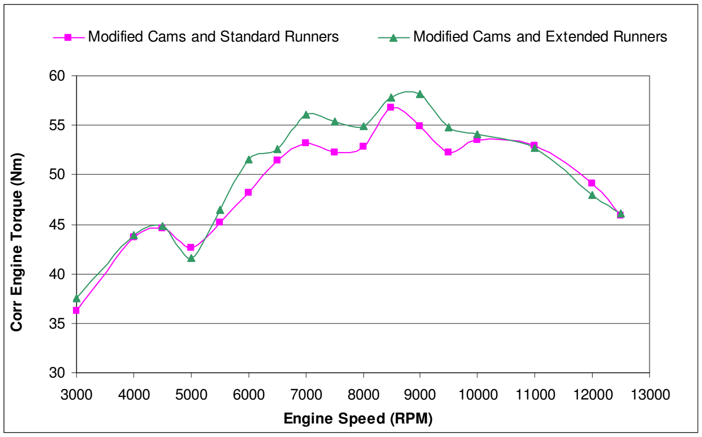


Figure 18 – Comparison of Runner Length Changes

As expected mid-range torque was increased but the torque started to decrease slightly at higher engine speeds. A maximum gain of 6.9% at 6000 rpm was achieved using the extended runner lengths.

CONCLUSIONS

The modified camshafts increased the low to mid-range torque output by:

1. Reducing the valve overlap period and therefore the reverse flow.

2. Advancing the intake valve closing point thereby reducing the amount of reverse flow occurring at the beginning of the compression stroke.

These changes to the valve timing increased the torque output by a maximum of 30% at 5500 rpm in the restricted FSAE engine. However if these camshafts were used in the unrestricted stock engine then at high speeds the valve timing settings would greatly reduce its performance. When the modified camshafts are used in conjunction with the restrictor, the restrictor masks the negative effects of the camshafts at the higher engine speeds as it becomes the limiting factor on the engine's performance.

ACKNOWLEDGMENTS

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NOMENCLATURE

FSAE	Formula SAE
SAE	Society of Automotive Engineers
QFR	Queen’s Formula Racing
RPM	Revolutions per minute
TDC	Top dead centre
BDC	Bottom dead centre
Cyl	Cylinder
VE	Virtual Engines
IO1 / IVO	Inlet valve open
IC1 / IVC	Inlet valve closed
EO1 / EVO	Exhaust valve open
EC1 / EVC	Exhaust valve closed
AD	Automated Design