

Valvetrain Optimisation Using Solidworks CAD And FEA With Excel Model From First Principals

Major Project

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Abstract

This paper shows the process of designing a new camshaft pulley with the aims to improve the performance for the valvetrain of a Ford Sigma 1.6. The report starts with the calculations undertaken to determine the forces acting upon the cam gear throughout the engine speeds. By the use Solidworks, simulations on the original cam gear were conducted to find the boundaries for which the new gear must be designed within. A final analysis was then made from the results which were achieved.

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Introduction

With an increasing awareness for the effect of fossil fuel powered vehicle emissions towards global warming, the internal combustion engine has become a topic of great controversy due to its inefficiency to convert burned fuel in to usable energy. It is estimated that transportation in the US contributes to almost thirty percent of its total global warming emissions. A vehicles efficiency is measured in thermal efficiency with most petrol cars in today's market ranging between 20 to 30 percent efficient. About thirty percent of the thermal energy is lost down the exhaust as heat and unburned fuel particles which means that the rest of the thermal energy is lost to friction and moving the components within the engine.

Every few years, new regulations which are ever tightening are placed upon the emissions produced by road vehicles. This has led to many car manufacturers exploring different ways to power their vehicles leading to the seemingly imminent extinction of road going internal combustion powered vehicles. A highly adopted method by car manufactures to increase the efficiency of their vehicles is to down-size and hybridise the powertrain by adding batteries which can be used to power the vehicle. However, the current most popular battery used in these vehicles is lithium-ion, which has come under great scrutiny as the mining of the mineral seems to violate ethical practices and the production from the raw materials into usable batteries generates high amounts off carbon emissions. Although new advancements are constantly being made to these batteries, the energy density of lithium-ion batteries is still significantly lower than that of gasolines, at 0.72 MJ/kg and 44 MJ/Kg respectively.

A relatively low-cost method to improve an engines efficiency would be to make the moving components lighter. With a lower mass, less force will be required to accelerate the components meaning more energy can be transferred to the driving force of the vehicle. This allows for great improvement in the vehicles performance. Benefits can include better handling, higher range for fuel, and faster accelerations.

A key section of the engine with a high number of moving components is the valvetrain. The engine for which this report is based on is the four-cylinder Ford Sigma 1.6 litre engine. The valvetrain in this engine is a dual overhead camshaft with sixteen valves driven by a cambelt which is directly connected to the crankshaft. A significant amount of force is needed to open the valves as they are accelerated at high speeds throughout the engine speed range.

To improve the efficiency of the engine, it was suggested that by reducing the mass of the valvetrain, less energy will be required to drive it. Attention was given to the cam pulley to reduce its mass by using alternative materials and designs. The cam pulley is subjected to the driving force from the cambelt and the resisting forces of the valve systems. Figure 1 shows the setup for the inlet valve train from the Ford engine.

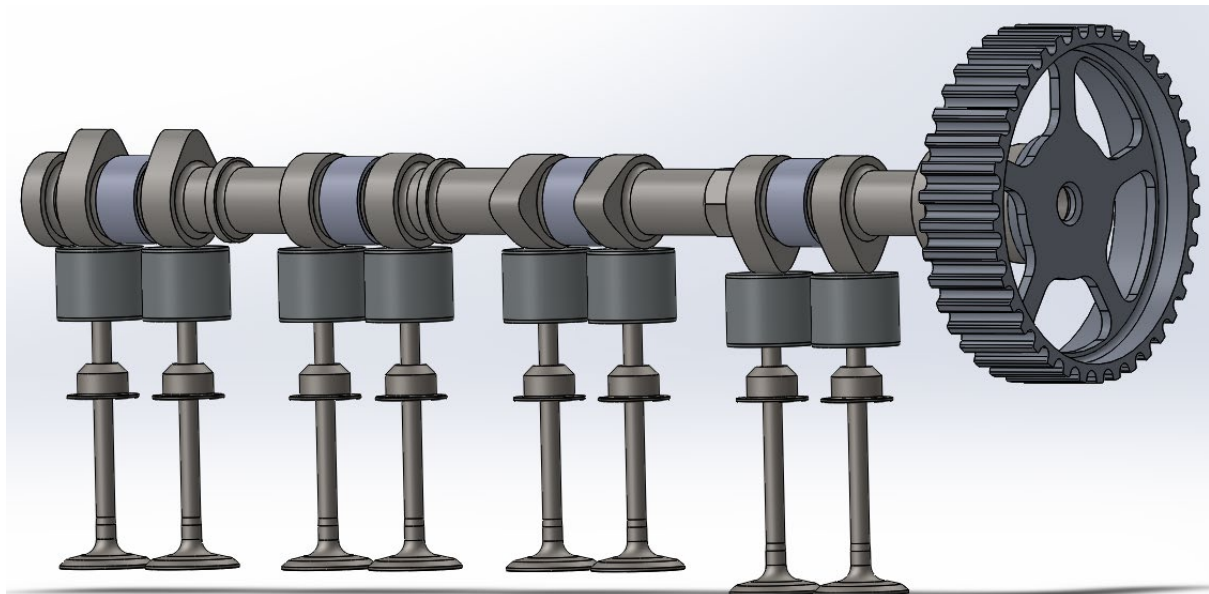


Figure 1 CAD model of Ford Sigma 1.6 litre inlet valvetrain

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

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Literature Review

Limitations of CAD

The main form of CAD used to analyse the structural properties for a component is Finite Element Analysis. FEA is a computational technique used to obtain approximate solutions of boundary value problems in engineering (Hutton D. V., 2004). From various tests conducted on the components of the valvetrain, it seemed that the meshing and the boundaries applied to the component played a significant role in the results which will be obtained. To understand how errors can be minimised in the results, research was conducted to find methods which can be applied to the work carried out for this report. The papers which were reviewed are listed below with the conclusions which were drawn afterwards.

1. J. H Underwood, *Applications and Limitations of Finite Element Analysis to Armament Components*. 1989

In this report, Underwood investigates the fracturing which occurs in the notches on the outside of cannon tubes caused by firing. He states in the report that the circumferential stresses which occur on the outside diameter are tensile due to the pressure forces which build up in the tube. By running a FEA simulations, he concluded that the results he obtained were accurate to 7% of the data from full scale testing. However, he does not mention the effect of temperature changes which would occur through the tube when firing. At the point of firing, the temperature of the inside diameter of the tube would initially be greater than that of the outside diameter. As metal expands when heated, the inside portion of the tube would expand before the outside portion would heat up and expand. This would lead to additional tensile forces on the outer diameter which may contribute to the fracturing of the outside diameter.

2. I. Ghionea, *Using FEA to study a mechanical part*. Design World, 2007

Ghionea clearly states that the development of a model is based on the intuition, experience and imagination of the user. To optimize a model, it should incorporate all available information about the analysed structure. He also mentions that the best result that can be obtained from a simulated model is an approximation. Meshing must be done so as to have a number of nodes large enough to achieve a satisfactory approximation for the boundary and loading conditions. Thus, the mesh of the structure has a major importance in the analysis of the model's mechanical behaviour. This paper briefly outlines the reasons as to why the results of the simulations conducted on the valvetrain components may vary depending on the finite element size which is applied. However, it does not give a suggestion on how to find the suitable mesh finite element size for a component.

3. R. Penner, *The Limitations of Finite Element Analysis*. Sparta Design Solutions, 2016

In this report, Penner investigates the issues which may occur from linear FEA. Most metals are described as ductile materials. This means that when a load is applied to it, the material will react in a linear way up to its yield strength. After this point, the material will behave non-linearly and will resist very little load before reaching its ultimate strength point and breaking. The non-linear region is known as the plastic zone and is very hard to describe mathematically and computationally. Linear FEA assumes that the material will constantly behave in a linear way even past the yield strength. What this means is that the results from a simulation under load will remain correct up to a certain point. Another common case where linear FEA is not accurate is in the case of stress concentrations in small areas. These stress concentrations will often be higher than the materials yield point. These are usually caused by sudden

changes in the geometry of the model. This may lead the technician to believe that more material would need to be applied in these areas, however this is not usually the case. This issue can be solved by undertaking experiments to find the true stress values in these areas and compare them to the computed results.

To truly understand how to apply a suitable mesh to the components for this report, a meeting was taken with Kelvin Lake, Advance stress analysis & FEA lecturer. It was advised, to get accurate results from the model the mesh size should be fine enough to ensure that a minimum of four nodes can be placed across the thinnest section of the component. This is because when a component such as a beam is bending, one side will see tensional stresses and the opposite side will see compressional stresses. Due to the way the finite elements calculate the average value around its nodes, if the mesh size is too large the value obtained could be an average of the tensile and compressive force which may cancel out each other leading to inaccurate results. By having enough nodes across the section, the values will show accurate tensile stress on one side and compressive stress on the other. This allows for a neutral axis between the two sides to separate them.

Proposed new materials for the cam pulley

As the aim for the new cam pulley design was to make it lighter than the original, research for possible alternative materials was undertaken. The materials which were to be chosen would need to be able to be modelled into intricate designs relatively easily. They would also need to be strong enough to withstand high forces and have a long life-span.

Plastic

It was found that Kawasaki have used a plastic material for a cam pulley in one of their production engines. This seemed to be a suitable alternative to aluminium as plastics are generally lighter and can be very strong. Further research found that a suitable plastic to use would be nylon/PA type 6. This material has a long life-span and good resistance to fatigue and corrosive substances which would benefit it in the engine as it may come in contact with oils. However, as it is a plastic it is much more flexible than aluminium which means the new design may require extra structural reinforcements to meet the stiffness of the aluminium. Production for plastic components is also relatively easy and cost effective compared to metals making it an attractive option for manufacturers. Although PA type 6 has a high thermal resistance, it may become brittle in cold conditions which may cause it shatter under impact loads.

Chopped carbon fibre

Carbon fibre has long been a popular material chosen to create light and strong components. Lamborghini has become one of the most prominent figures to revolutionise the possibilities which can be achieved with carbon fibre. Their Sesto Elemento model utilizes carbon fibre in a chopped and forged form rather than the traditional layered sheets. They have seen great benefits from this claiming that they have reduced their vehicle weight by 25% with an unprecedented carbon fibre content of 80% achieved through the intense use of carbon fibre composites (Quantum Composites, 2018). Instead of layering sheets of carbon fibre and then impregnating it with resin, carbon fibre composites uses a paste of chopped fibres (500,000 turbostratic fibres per square inch) mixed with resin which can be squeezed out to create almost any shape. Since the fibres are not oriented in any particular direction, the result is a strong overall part which is still very light. Lamborghini are currently working with Forged Composites to attempt development for a forged carbon fibre connecting rod which will greatly reduce the rotating mass of the engine allowing it to spin up more easily and sacrificing less power to internal drag.



Figure 2 Lamborghinis concept conrod made of forged carbon fibre (taken from digital trends)

Calculations

Camshaft Torque and Forces

In this chapter, the total torque on the inlet camshaft will be calculated from the cam lobe profile and the forces generated by the valve system. Throughout the entire speed range of the engine, the valve train carries high loads. During lower engine speeds, the loads acting on the valve train are primarily due to the spring forces. However, at higher engine speeds the inertia forces of the component masses dominate (Heywood, 1988). Figure 3 and 4 shows the comparison of the loads from the valve spring and the component masses at 1000 and 3000 camshaft RPMs.

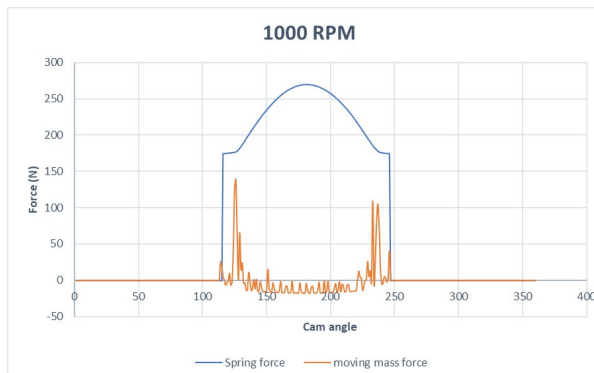


Figure 3 Spring forces vs moving mass forces 1000RPM

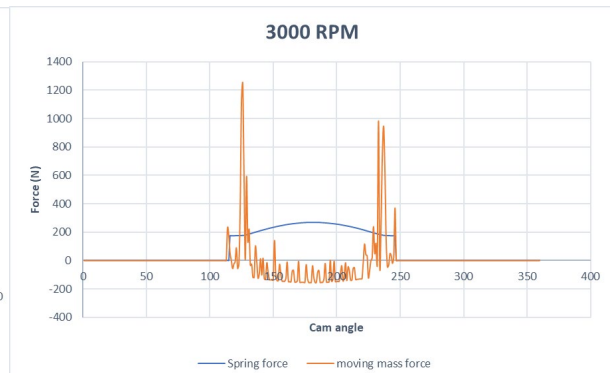


Figure 4 Spring forces vs moving mass forces 3000 RPM

The valve lift was calculated using measurements taken from the camshaft lobe profile. The camshaft lobe profile was measured using a DTI gauge placed on the lobe and a dividing head to rotate the camshaft a degree at a time. At each degree of camshaft rotation, the lift on the DTI gauge was read. Another DTI gauge was placed on the bearing section of the camshaft to ensure the camshaft was rotating along its centre axis to minimise the chances of an error in the readings. The measurements for the camshaft lobe profile was taken a total of three times and the results were averaged to eliminate possible anomalies. Figure 5 shows the setup used to carry out the measurements.

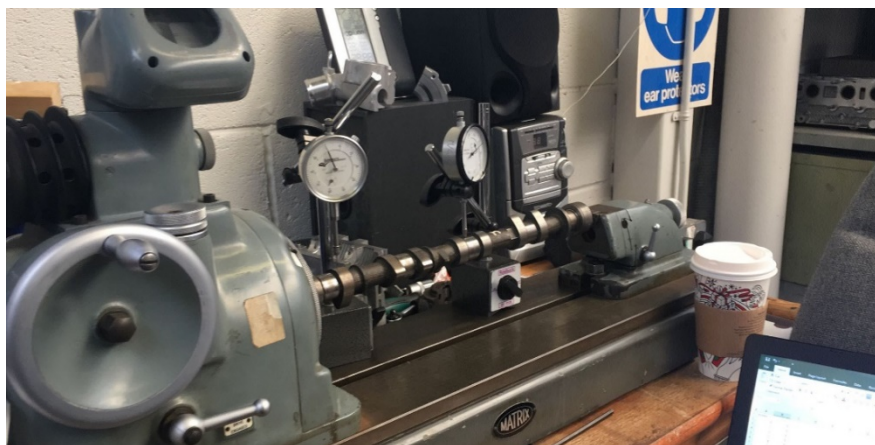


Figure 5 Cam lobe profiling setup

Due to tappet clearances and the design of the valve train, the valve lift does not follow the lift profile of the camshaft lobe. To calculate the valve lift, the tappet clearance and the base radius of the camshaft is required. These were measured to be 0.25mm and 15.05mm respectively. A value of 'OB' is calculated which represents the distance between the camshafts axis of rotation to the top of the valve bucket. The equation used is as shown below;

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

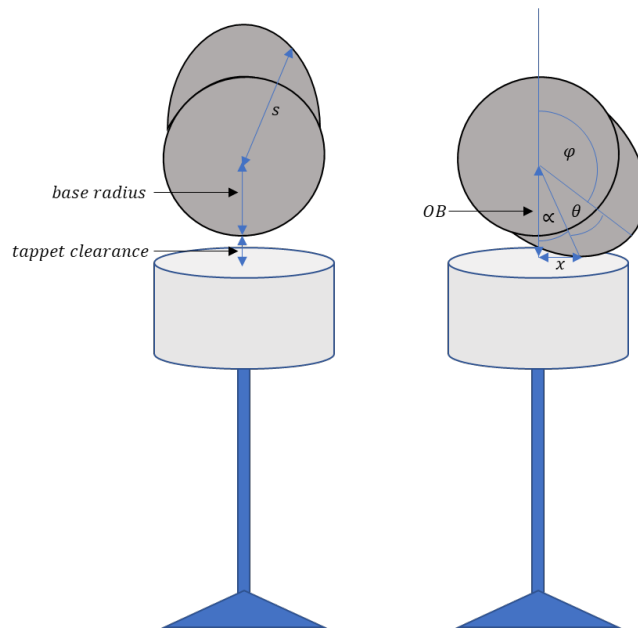


Figure 6 angles and measurements diagram

Where φ is the camshaft rotation angle from vertical and θ is the angle from the maximum lift in the cam lobe to the contact point on the bucket. 'S' is the sum of the cam base circle and the lift of the profile. This calculation is repeated for each angle of the camshaft rotation using θ values ranging from 0 to 360 degrees. This was carried out in Excel and the angles were converted into radians before used. The maximum value for OB at each degree of cam rotation is subtracted by the total sum of the base radius and the tappet clearance which gives the true valve lift value. Figure 7 shows the comparison between the cam lobe profile and the valve lift.

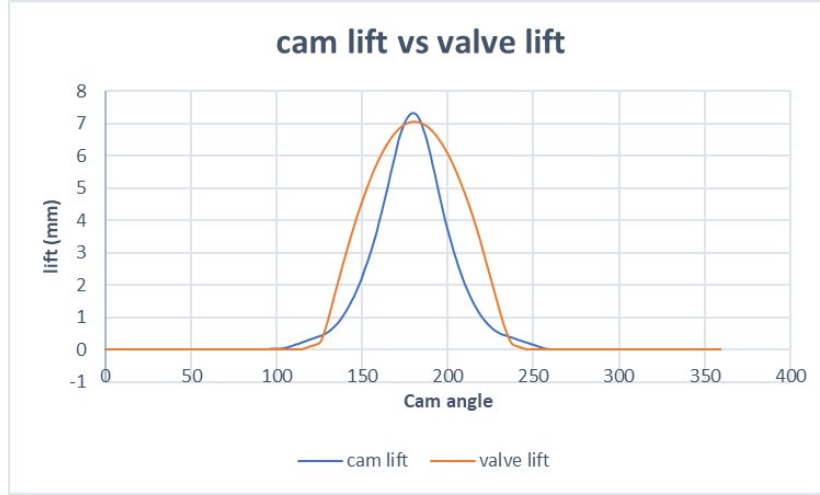


Figure 7 cam profile vs valve lift

From the valve lift values, the valve acceleration can be calculated through the following steps.

1. Calculate valve velocity in mm/ degree.

$$V(mm/^{\circ}) = Lift_2 - Lift_1$$

2. Calculate valve acceleration in mm/ degree/ degree

$$Acc(mm/^{\circ}/^{\circ}) = V_2 - V_1$$

3. Convert valve acceleration in mm/ degree/ degree to m/ seconds²

$$Acc(m/s/s) = \frac{Acc(mm/^{\circ}/^{\circ})}{1000} \times (6RPM)^2$$

The torque on the camshaft is generated by the friction force from the contact force and the contact force itself pushing up on the cam lobe at a horizontal distance from its rotational axis. This horizontal distance is denoted by 'x'

$$x = OB \times \sin \alpha$$

Where $\alpha = 180 - (\theta - \varphi)$

With the known acceleration of the valve lift, the contact force from the moving components can be calculated. The moving components in the system include the valve, collets, retainer and the bucket. The masses of each component were measured and summed up to acquire the total moving mass. The equation for the contact force is derived from Newtons Second Law of Motion, ($F = M \times A$), as shown below.

$$F_{contact} = M_{moving\ components} \times A_{valve}$$

As the cam lobe can only push down on the moving components, a spring is used to bring the moving components back up. When the spring is installed in the valvetrain, it has an initial compression to ensure that the valve will be fully closed when the cam lobe isn't in contact. The spring force opposes the contact force and therefore is added to the force needed to accelerate the moving components.

$$\therefore F_{contact} = (M_{moving\ components} \times A_{valve}) + F_{spring}$$

The spring force is dependent on the spring rate (k), initial compression length (Y_o), and the valve lift (Y). The spring rate is measured by the amount of force needed to compress the spring a certain distance. For the Ford Sigma 1.6 the spring rate was determined to be 13.508N/mm. to calculate the spring force the equation used was;

$$F_{spring} = k(Y_o + Y)$$

At higher engine speeds, after full lift the acceleration of the valve train components become negative. The inertia force of these components can be large enough and opposite to the spring force so that the total contact force on the lobe becomes negative. This point is known as the float speed and is the factor which dictates the maximum engine speed which can be achieved with this valve train.

The friction coefficient for this calculation was set at $\mu = 0.1$. To calculate the total torque produced by the friction, the friction force must first be calculated. This is done by multiplying the contact force at each degree with the friction coefficient. The torque on the camshaft from the friction force can then be calculated with the equation below.

$$T_{friction} = OB \times F_{friction}$$

The torque generated by the contact force can be calculated by multiplying the contact force with 'x', which was calculated in the earlier equation. As both torques on the camshaft are acting in the same direction when opening the valve, they add together to produce the total torque. When the valve is closing, the friction torque remains in the same direction, but the contact force torque is applied on the other side of the rotating axis which means it is going in the opposite direction from the friction torque. This torque becomes a negative and at lower engine speeds can help rotate the camshaft. At the float point this doesn't generate any torque on the cam shaft.

The Ford Sigma 1.6 is a 16-valve, 4-cylinder engine and therefore there are eight cam lobes per camshaft and two cam lobes per cylinder. The valves don't all open simultaneously but rather at 90-degree angles from each other. However, there is some overlap when one valve is closing, and another is opening. The valve opening sequence for this engine is cylinders 1,3,4, and 2, same as the firing order. This means as cylinder 1 valves are closing, cylinder 3 valves are opening and so on.

To find the total torque on the camshaft at each degree, a table on Excel was generated. The table consisted of the torques for each cylinder at each degree of rotation. The total torque on the camshaft will then be the sum of the torque from each cylinder at that cam angle.

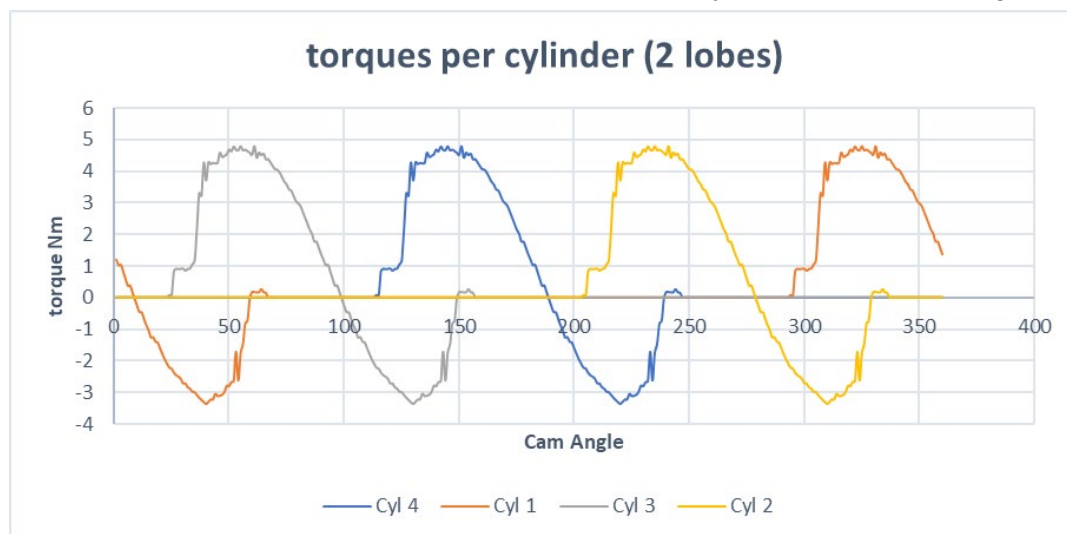


Figure 8 camshaft torques per cylinder

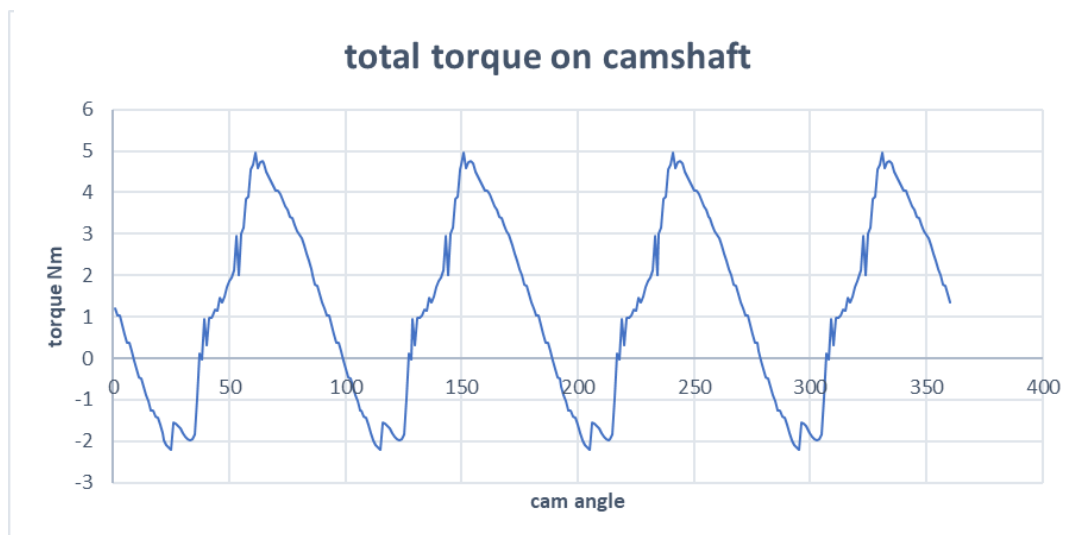


Figure 9 total camshaft torques

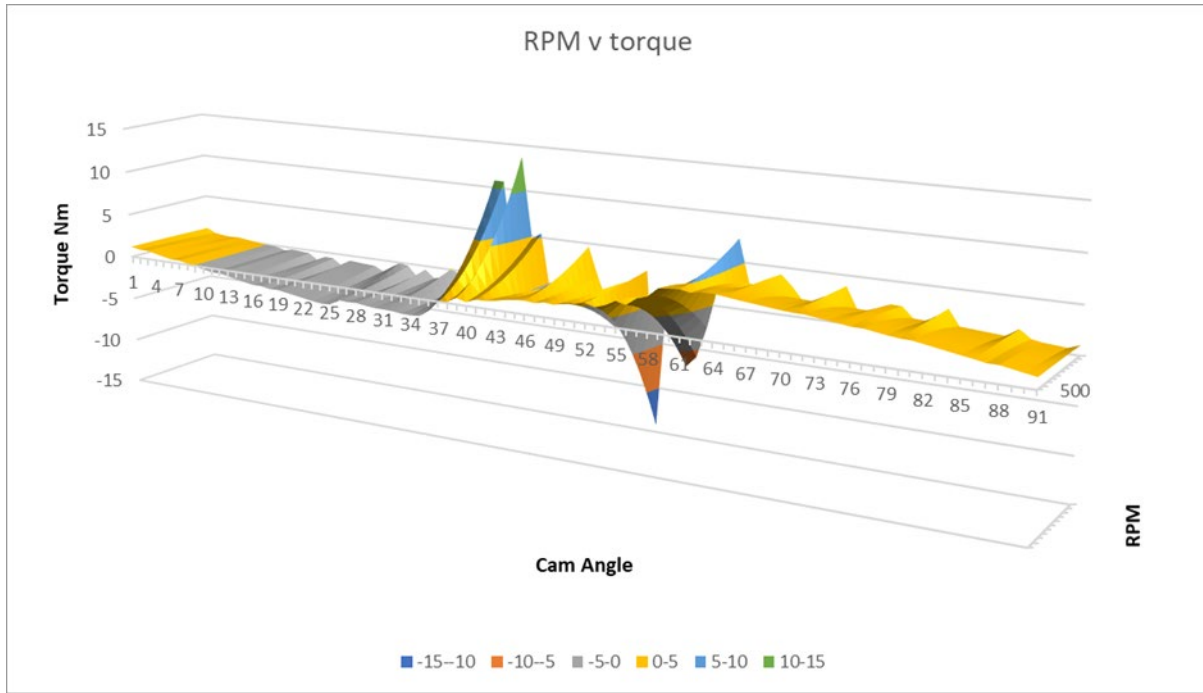


Figure 10 camshaft torque vs RPM

To calculate the Friction Mean Effective Pressure (FMEP), for the intake camshaft, the work done per degree must first be calculated. This can be done by multiplying the torque at each angel with $\frac{2\pi}{360}$. The work done per degree must then be summed up to get a total work done per cam lobe. Multiplying the work done per lobe by eight produces the total work done for the camshaft. To calculate the FMEP the following equation was used.

$$FMEP = \frac{\text{Work done}}{\text{engine displacement}}$$

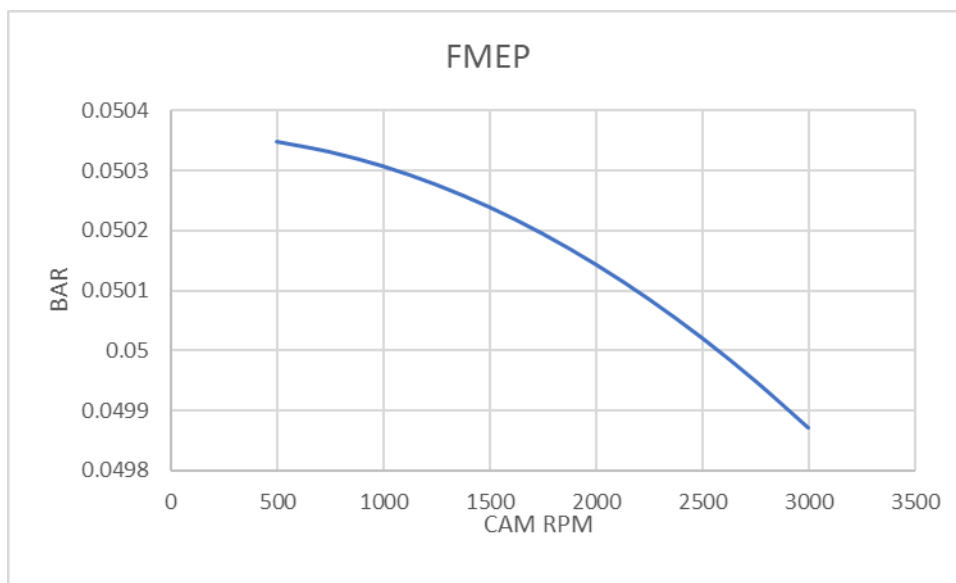


Figure 11 FMEP

Cambelt Forces

The cambelt synchronizes the rotation of the camshafts with the crankshaft so that the valves open and close at proper times during each cylinder's intake and exhaust strokes. The Ford Sigma 1.6 has an externally mounted rubber toothed belt. An advantage of this design is that the rubber damps out torsional vibration which may otherwise be troublesome (Stone, 1992). The cambelt must be stiff enough to resist the torque forces from the camshaft without flexing too much as this would off-set the valve timing from the crankshaft. A single belt drives both



Figure 12 front view of Ford engine

the intake and exhaust camshafts. In this study, for the exhaust camshaft torques, the same data will be used as the inlet camshaft torques.

To turn the camshafts, the crankshaft rotates clockwise and pulls on the belt from the intake cam pulley. This tensional force is denoted as F_1 . This in turn creates tension in the part of the belt connecting the intake cam pulley to the exhaust cam pulley. This tensional force will be denoted as F_2 . The part of the belt from the crankshaft to the

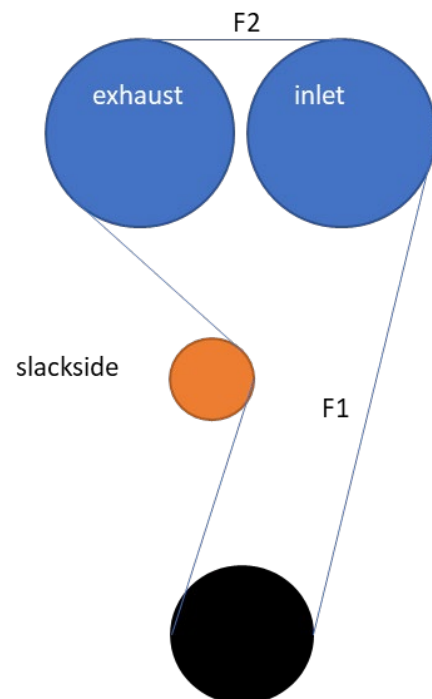


Figure 13 pulley and belt diagram

exhaust cam pulley is the slack side. The tension on this side is generated by the belt tensioner. This side does not see the torque forces generated by the camshafts.

The tension in F_1 is the result of the torques from the exhaust and intake camshafts. To find the force in the belt at F_1 the sum of the two torques are divided by the cam pulley radius.

At F_2 , the belt is only pulling on the exhaust cam pulley and therefore the tensile force on this part is the torque from the exhaust camshaft divided by the cam pulley radius. The cam pulley radius for both the intake and exhaust are 57.5mm and for the crank shaft this will be halved. This gearing ratio is what allows the cam shafts to complete one full rotation for every two crank shaft rotations.

When the cambelt is installed in the engine, it has an initial tension which is added on to F1 and F2. This initial tension is set by the spring in the cam tensioner pulley. This prevents the cam belt from slacking when the engine is running which could lead to the belt slipping on the pulleys and disrupt the valve timing. A mistimed valve opening could lead to the valves colliding with the piston and snapping causing damage to the cylinders and piston itself.

By a rule of thumb, the initial cambelt tension must be tight enough so that it can only be twisted ninety degrees. To figure out how much force is needed to create this amount of tension, a make shift experiment was carried out to simulate the initial tensile force. By knowing the length of the belt at F1, the belt was clamped at one end and weights were added to it at another end. The other end was then clamped while maintaining the tension in the belt caused by the applied force. The belt was then twisted to see how much resistance there was. Once the belt could only be twisted up to ninety degrees, the mass applied was noted and from here the initial tensile force could be estimated. Figure 14 shows the setup of the experiment that was conducted.



Figure 14 initial cambelt tension experiment setup

The results from the experiment showed that at three kilograms of mass, the belt could only be twisted by ninety degrees. By converting this into newtons, the initial tensile force was calculated to be 29.43N.

To find the cam belt forces acting on the intake cam pulley, the forces from F1 and F2 need to be resolved for the horizontal and vertical vectors. The cam belt is in contact with more than a quarter of the intake cam pulley's circumference to prevent slippage from happening. The forces from F2 are to be considered on the horizontal vector, but the forces from F1 act in both the horizontal and vertical vector. The angle at which the F1 force acts is 10 degrees from the vertical and is denoted as α .

The following equations were used to calculate the resultant forces in the horizontal and vertical directions.

Horizontal resultant forces.

$$F_{horizontal} = F_2 + F_1 \sin \alpha + F_{initial}$$

Vertical resultant forces.

$$F_{vertical} = F_1 \cos \alpha + F_{initial}$$

With the horizontal and vertical resultant forces resolved, the overall resultant force can then be calculated using Pythagoras Theorem.

$$F_{resultant} = \sqrt{F_{horizontal}^2 + F_{vertical}^2}$$

Figure 15 resultant belt forces

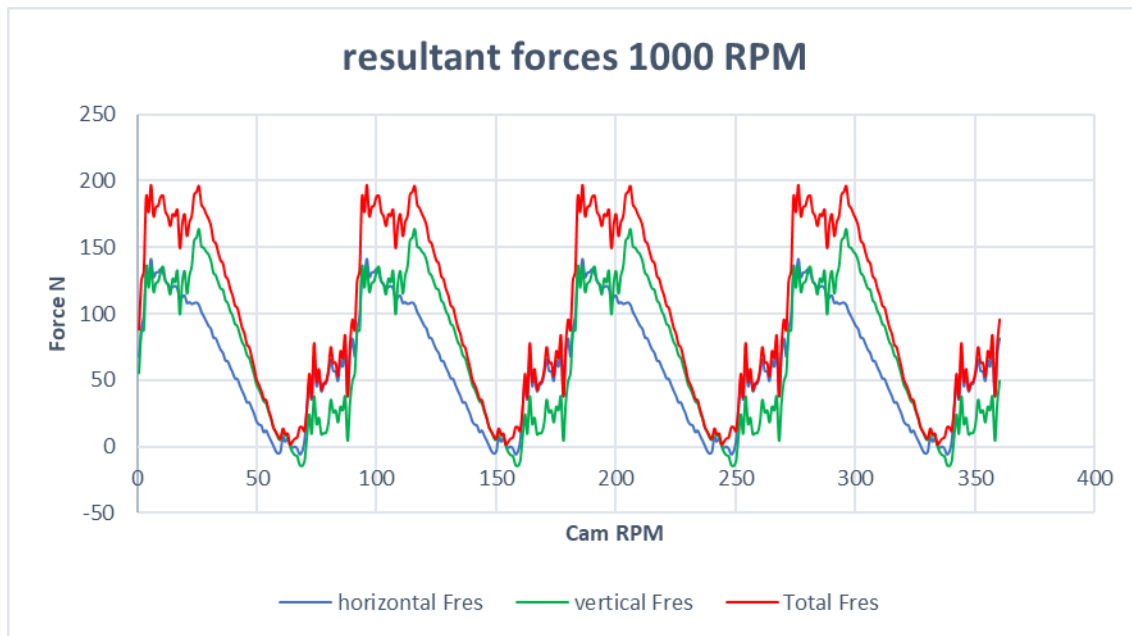
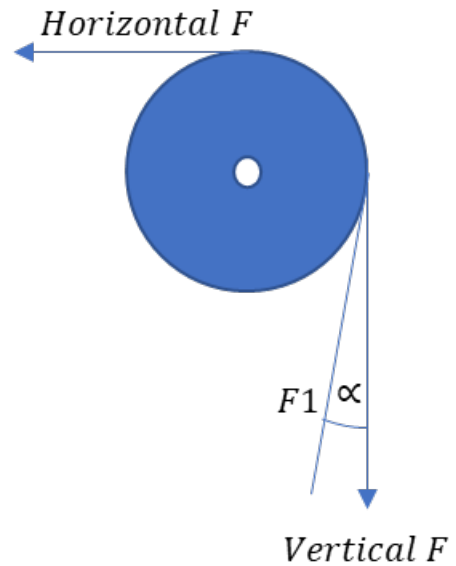


Figure 16 resultant forces at 1000 RPM

Figure 16 shows that the horizontal and vertical resultant force at points drop below zero. This is because the valve spring is pushing up on the camshaft helping it rotate. The force dropping below zero would imply that the belt is being pushed towards the camshaft, however in reality the belt would simply slack as compressive forces cannot be conveyed through a belt. To resolve this the values which are below zero are made to be zero in the excel spread sheet using an 'if statement'. Therefore, the total resultant force does not drop below zero. If the total resultant force dropped below zero, the belt would slip off from the cam pulley.

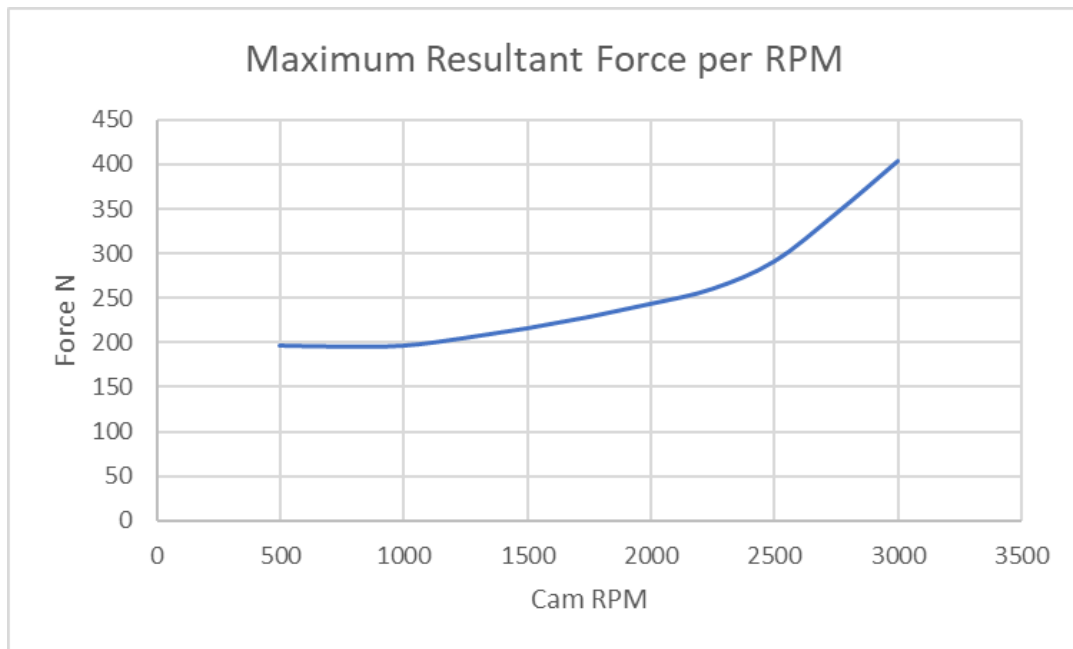


Figure 17 Maximum resultant vs RPM

CAD Simulation

Simulation Setup

The cam pulley was measured up and detailly modelled in Solidworks so that the forces which were calculated can be simulated in a static study. The simulations which were carried out were used to observe the stresses, deformation, and the minimum safety factor of which the cam pulley has.

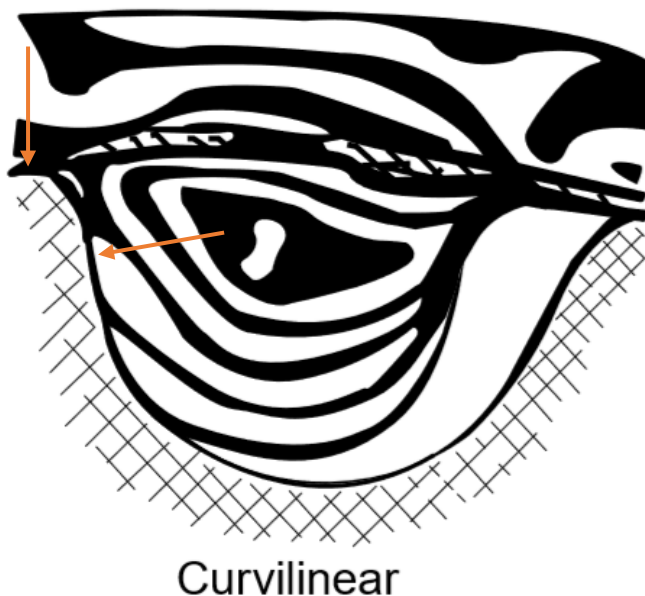


Figure 18 (sdp-si, 2018) shows how the forces from the belt is applied to the cam pulley. The number of teeth on the inlet cam pulley which are in contact with the belt were counted as these would be where the belt forces are spread across. For the inlet camshaft, thirteen teeth were in contact at any time.

Figure 18 belt forces on pulley teeth

The torsional force on the pulley was converted to a linear force which could then be applied to the Solidworks model. This was done by dividing the torque by the radius of the cam pulley. This linear force was then divided by the thirteen teeth. The total resultant force was also divided by the thirteen teeth and were directed towards the mounting point of the cam pulley.

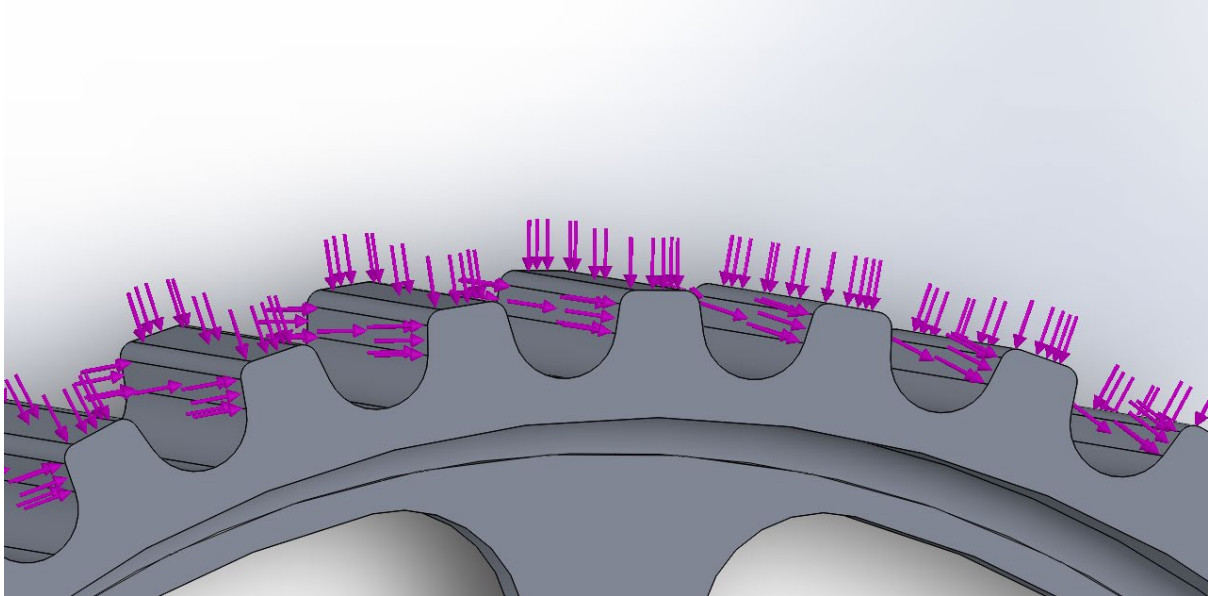


Figure 19 force application on cam pulley

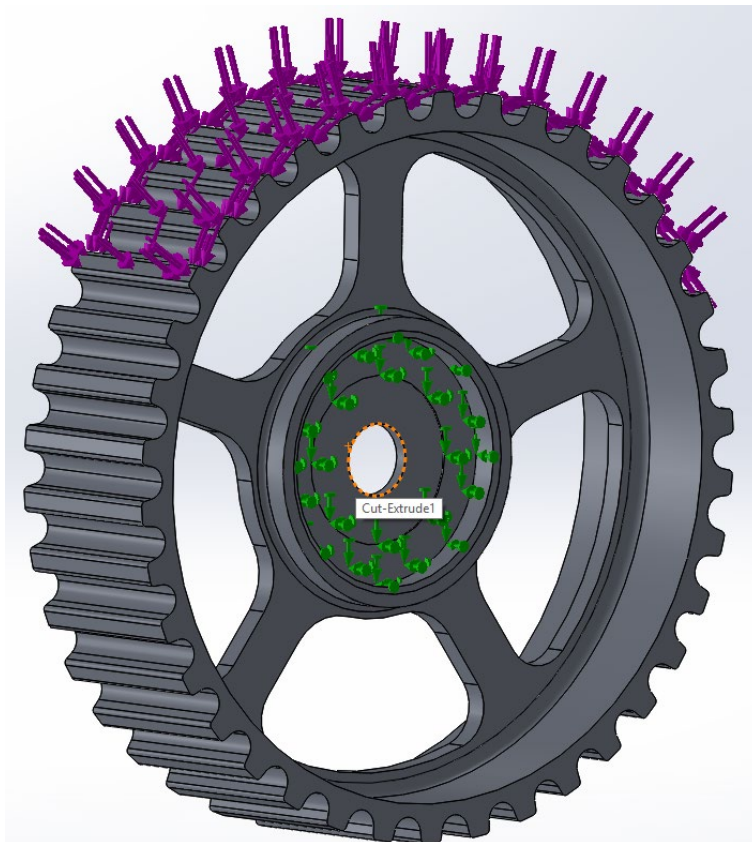


Figure 20 mounting and forces on cam pulley

Figure 19 shows how the forces were applied on the Solidworks simulation. The forces were spread equally among the thirteen teeth.

The cam pulley was then given a fixed mounting where it would be connected to the camshaft. This would allow the simulation to calculate the forces applied without the cam pulley spinning freely.

A standard mesh element of 1.91mm was applied and a refined mesh element of 1.2 mm was applied in the more critical areas where higher stresses occurred. These mesh element sizes were chosen to ensure that the depth of the cam pulley spokes had more than 4 nodes across it to give a proper reading. This is due to the way the results are calculated by Solidworks. The values produced at each element are an average reading of the nodes on that element. If the mesh element sizes are too large, the nodes would be too far apart, and the average reading would be taken from a too large area which could differ the results from an accurate reading.

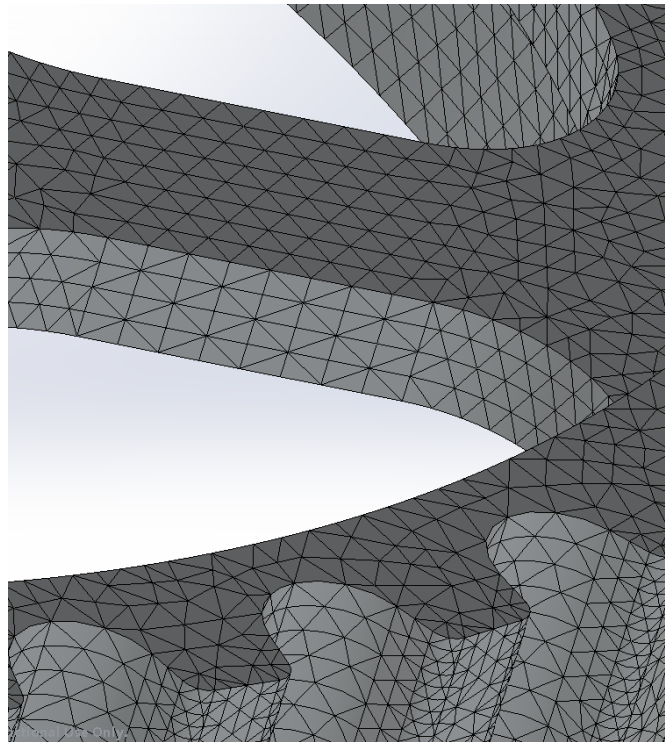


Figure 21 mesh applied on cam pulley

To obtain the minimum safety factor used by Ford for the intake cam pulley, the maximum resultant forces which were calculated were applied to the model. From figure 17, it is observable that the maximum resultant force occurs at 3000 cam RPM. The maximum resultant force at 3000 cam RPM was calculated to be 403.362N and occurs four times in a full camshaft rotation. This force is at the same angle as when the valve acceleration is at its highest. The valves for each cylinder are opening at ninety degrees cam rotation from each other, and with four cylinders, therefore the maximum resultant force occurs four times. This is the force which is applied on the top of the cam pulleys pointing towards the mounting point.

From the known angle of the maximum resultant force, the torsional force at that angle can be found. This was 13.74Nm which when calculated to find the linear force was 238.94N on the teeth. This is the total force which was applied to the side of the pulley teeth.

As the cam pulleys are externally mounted, they would not see the hot internal temperatures of the engine. Therefore, the operating temperatures of the cam pulleys were not considered in the simulations.

Simulation Results

Figure 22 shows the stresses which occur in the cam pulley. The maximum stress in the cam pulley is $3.65 \times 10^7 \text{ N/m}^2$ around the base of the cam pulley spoke. It can be visibly seen that most of the stress from the applied force is happening on the cam spokes. From this an assumption was made that the forces are dissipated among the spokes.

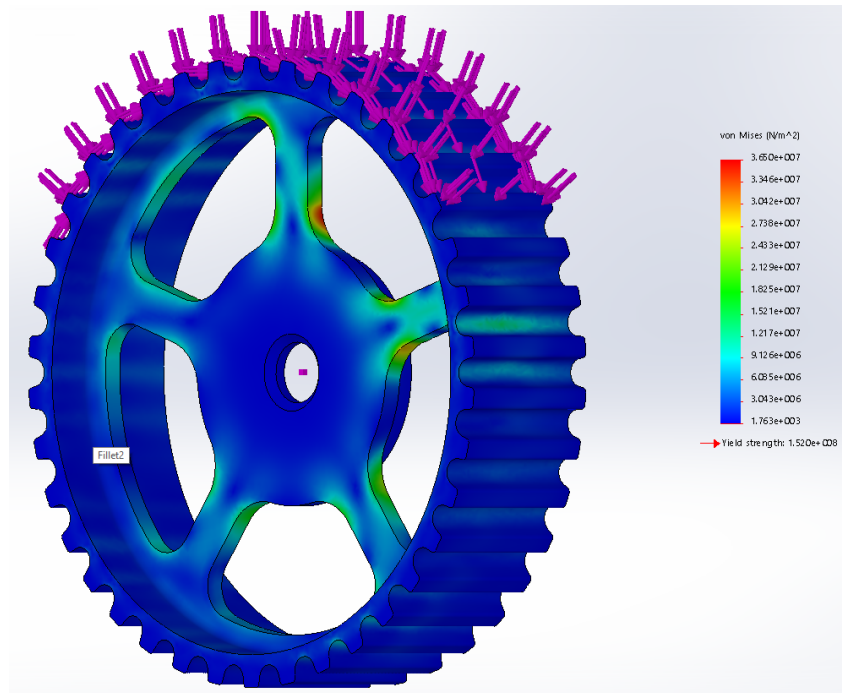


Figure 22 cam pulley stress plot

Figure 23 shows the safety factor of the component. The minimum safety factor found was 4.164. The safety factor of a component is calculated by dividing the ultimate stress value of the components material by the actual stress which it sees. Car manufacturers generally use a minimum safety factor 3 as at this point the components fatigue life is assumed to be infinite.

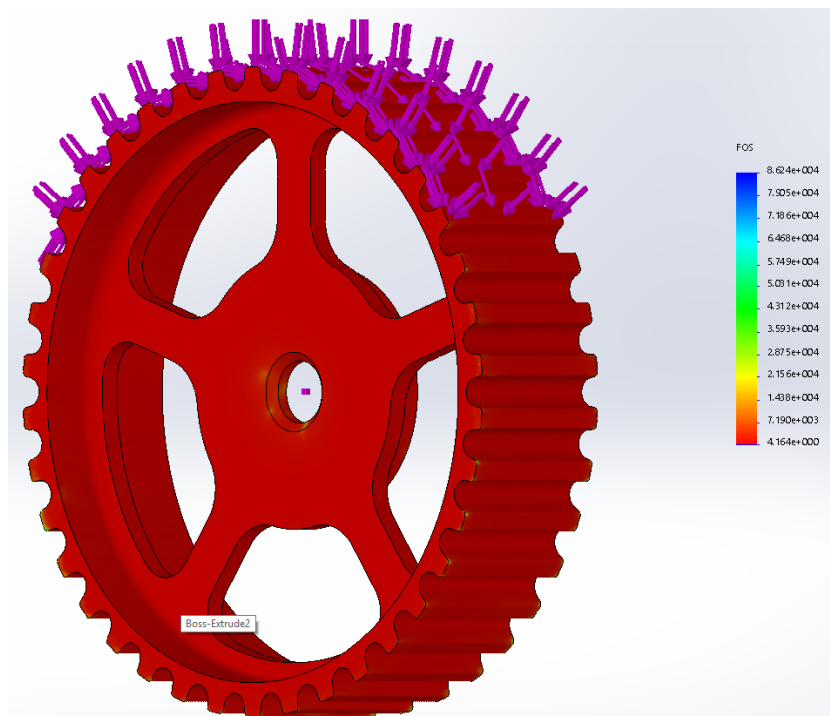


Figure 23 cam pulley safety factor plot

By applying the minimum forces acting on the cam pulley in a separate static study, a fatigue simulation in Solidworks was carried out. The test was set to run at a standard ten million cycles with the maximum applied force to the minimum applied force. The test results came back as a failed run as the forces were not high enough to break the component. This is the desired result as it shows that the results from the calculations are not too high.

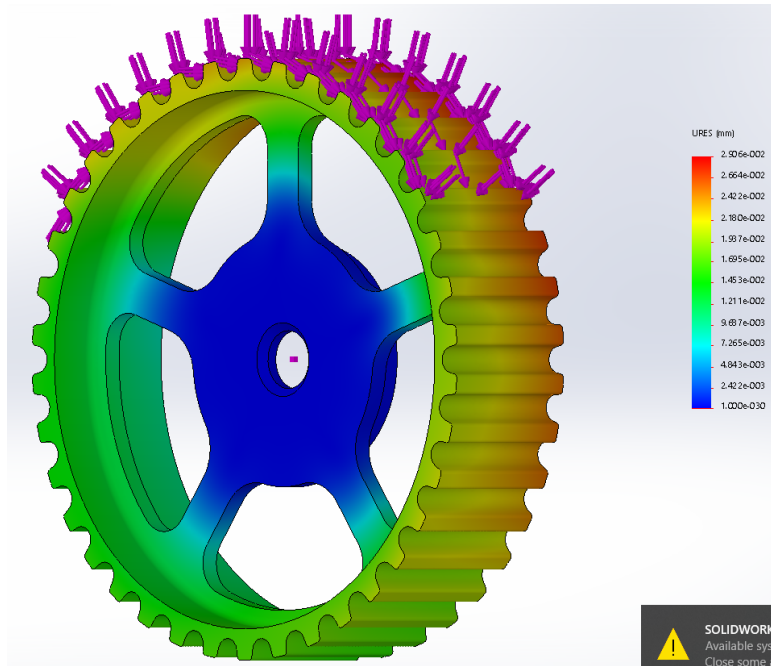


Figure 24 cam pulley deformation plot

Figure 24 shows the deformation scale of the cam pulley under maximum resultant force. The maximum deformation occurs around the cam pulley teeth where the forces are applied. The maximum deformation displacement was determined to be 0.029mm.

When redesigning the cam pulley with the different materials, the main goal was to ensure that the safety factor for the new component was not

lower, and that its deformation was not larger than the original. If the new designs deformation was larger, it could offset the valve timing slightly. The safety factor cannot be lower as the component may fail when tested on the engine.

Validating the Simulation Results

To ensure that the simulation results were valid, and that the setup of the mounting place and forces applied were simulated as it would be in real life, an experiment was carried out. The experiment conducted was used to find the strain in the spokes of the cam pulley when known forces were applied to it. The cam pulley was mounted on to a fixed cam shaft and the known forces of F1 and F2 were applied to a belt which was placed on the cam pulley. Two strain gauges were then placed on the cam pulley spokes, one along the length of the cam pulley spoke and one across the spoke. The strain gauges can only measure the strain in one direction which is why they were set up in this way. The values from the two would later be combined using Pythagoras Theorem to find the resultant force on the spoke. The stress values from the Solidworks simulation are also the resultant stresses.

The measuring instrument used would only allow for one strain gauge measurement to be taken at a time. Therefore, the cambelt was placed on the cam pulley across the thirteen teeth which had a spoke in the middle of them. The first test was ran with the strain gauge along the spoke, then the pulley was rotated so that the strain gauge that lies across the spoke takes the position of the spoke with the strain gauge along it. Figure 25 shows how the strain gauges were placed on the cam pulley. The test was conducted twenty times for each strain gauge and an average result was taken.



Figure 25 strain gauge setup on pulley spokes

Figure 26 shows the setup for the experiment. To simulate the forces from F1, weights were tied onto the cam belt and were let to hang from the cam pulley. The force from F2 was simulated using a spring balance. This was set so that the belt would be pulled at 10 degrees from the horizontal as F1 now has become vertical.

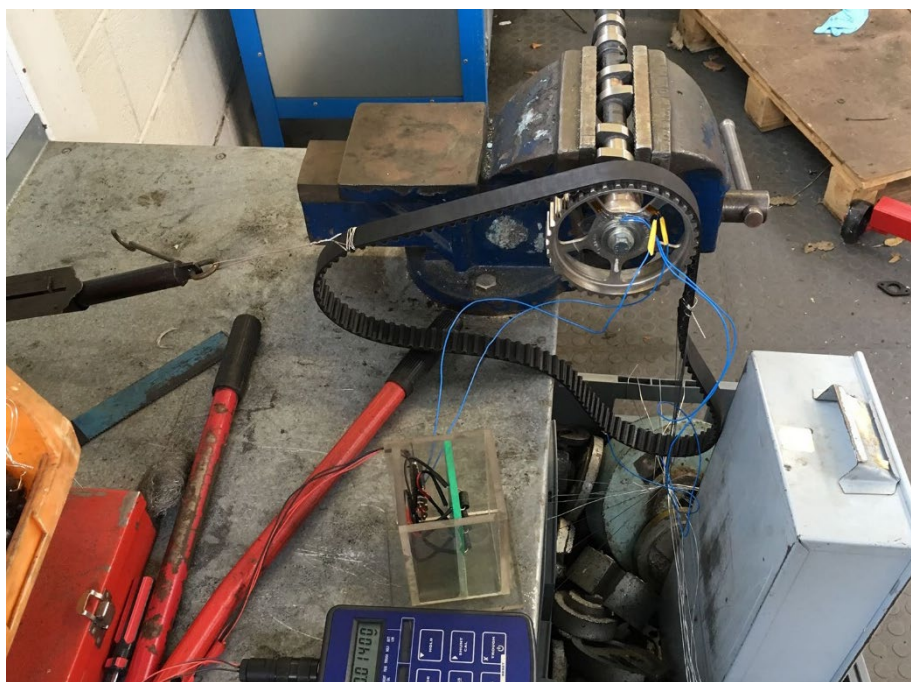


Figure 26 setup for strain gauge test

The cam belt was taken off, and the strain gauge readings were reset to zero between each test. The readings which were produced were in millivolt/ Volt. These values had to be converted into strain values before calculating stress.

Strain is the amount of deformation of a body and is defined as $\varepsilon = \frac{\Delta L}{L}$, where ΔL is the change in length and L is the original length.

To convert the readings into strain, the gauge factor of the PSD is required. This was determined to be 2.14. The formula to convert the read values is:

$$\frac{V_o}{V_{ex}} = -\frac{GF \times \varepsilon}{4} \left(\frac{1}{1 + GF \times \frac{\varepsilon}{2}} \right)$$

Where $\frac{V_o}{V_{ex}}$ is the reading value in volts/volts and GF is the gauge factor.

To rearrange this formula for stress, the part of the equation which is in the bracket was to be considered as 1. This was to simplify the equation and was able to be done because the value of strain will be so small that it would be neglectable and the value of the bracket will be very close to one. The final form of the formula therefore can be written as shown below.

$$\varepsilon = \frac{4 \times \frac{V_o}{V_{ex}}}{GF}$$

The average strain values calculated from the tests were 3.34×10^{-5} for the strain gauge along the spoke and 6.66×10^{-6} for the strain gauge across the spoke. To calculate the stress from these values, they were multiplied by the materials Youngs modulus.

$$\sigma = \varepsilon \times E$$

Where σ is stress and E is the Youngs modulus.

Using Pythagoras Theorem from the average for the two strain gauges, the resultant stress value was then determined as $7.78 \times 10^6 N/m^2$.

To find the stress value of the Solidworks simulation in the same area, plot nodes were placed on the Solidworks model. Figure 27 shows the location of the nodes and figure 28 shows the results.

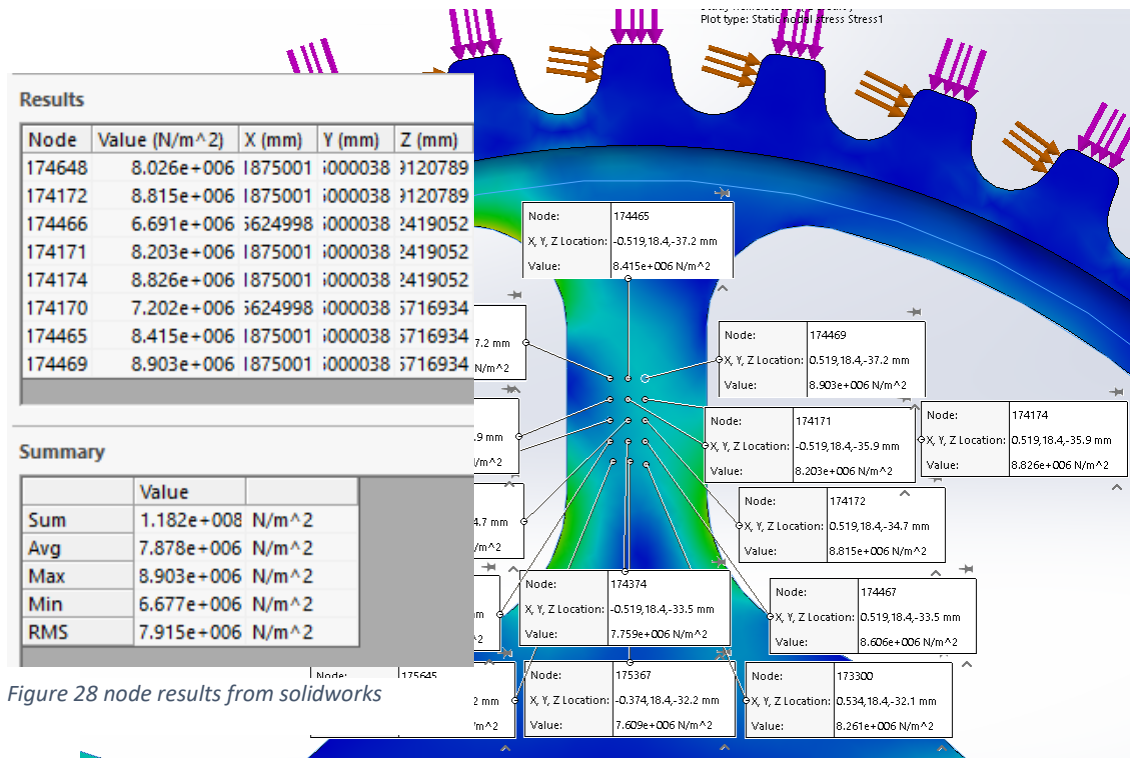


Figure 28 node results from solidworks

Figure 27 nodes for stresses on cam pulley spoke

The average stress from the Solidworks simulation nodes was $7.878 \times 10^6 \text{ N/m}^2$. This shows that the results from the simulation are only about 1.2% different from that of the real values. Therefore, the setup of the simulation can be deemed suitable for further testing of the new designs.

One factor which was noticeable during the strain gauge testing was that when the force was applied faster onto the cam pulley, the readings would be significantly larger. This is because the weights carried momentum as they were being applied and therefore generated a larger force on the cam pulley. To make sure the force was applied the same way each time, an office chair with a damper was used to lower the weights gradually on to the cam pulley.

Designing the Modified Cam pulley

The new designs were generated in Solidworks with the original cam pulley as a base. The aim for the new designs was so that it could be replaced onto the engine without any modification towards the cam belt or other components. The restricting factors therefore were

the pulley teeth and the mounting point had to remain in the same dimensions as the original cam pulley. The modifications made were hence mainly towards the spokes of the pulley.

Plastic Cam pulley

The material chosen for the first design was nylon, specifically Polyamide type 6 (PA type 6). The numeral nomenclature is derived from the number of carbon atoms in the diamine and dibasic acid monomers used to manufacture it (RTP, 2018). This material was chosen as it is a popular material used to make plastic pulley due to its toughness and high fatigue resistance (AZO, 2018). PA type 6 has a high rate of water absorbance compared to several other nylons, but as the cam pulley is externally mounted in a dry area, this was not a concern. The density for PA type 6 is 1120 Kg/m³ which is almost sixty percent less of the original aluminium which was 2680 Kg/m³. Figure 29 shows PA type 6's technical specifications. This nylon is

Property	Value	Units
Elastic Modulus	72400	N/mm ²
Poisson's Ratio	0.33	N/A
Tensile Strength	228	N/mm ²
Yield Strength	152	N/mm ²
Tangent Modulus		N/mm ²
Thermal Expansion Coefficient	2.1e-005	/K
Mass Density	2680	kg/m ³
Hardening Factor	0.85	N/A

also easy to process as components can be made by reaction injection moulding instead of casting.

Figure 29 PA type 6 technical specifications

The new design included two extra spokes compared to the original. This is because PA type 6 is much more flexible than aluminium which caused the rim of the pulley between the spokes to bend in too much under the same load as there was not enough support from the spokes beneath it. Because the PA type 6 is much lighter than the aluminium, more material was able to be used in the design and still manage keep it lighter.

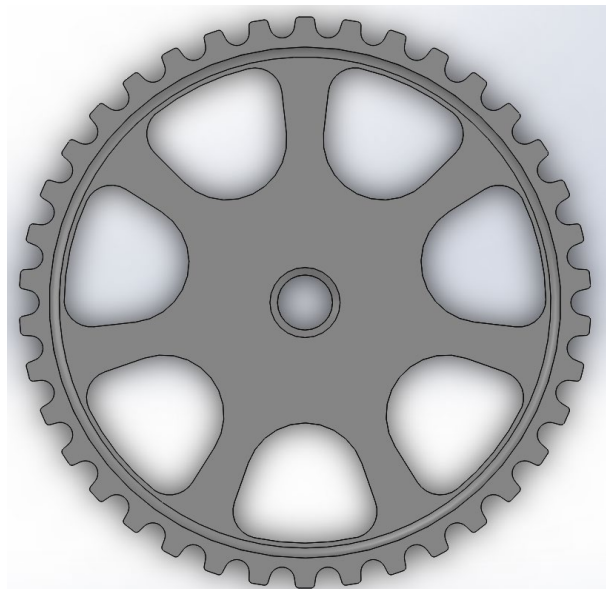


Figure 30 Plastic cam pulley design

As the force from the belt is applied across the cam pulley, the part or the rim which was not directly above the cam pulley would bend inwards. For the frontal part, the deformation was not as critical as it doesn't protrude as far out from the spokes as the rear side. To combat this, ribs were placed along the back of the spokes from the mounting point to the inside of the rim. This can be seen in figure 31.

Further ribs were applied from these ribs to the rim to prevent those areas from bending inwards.

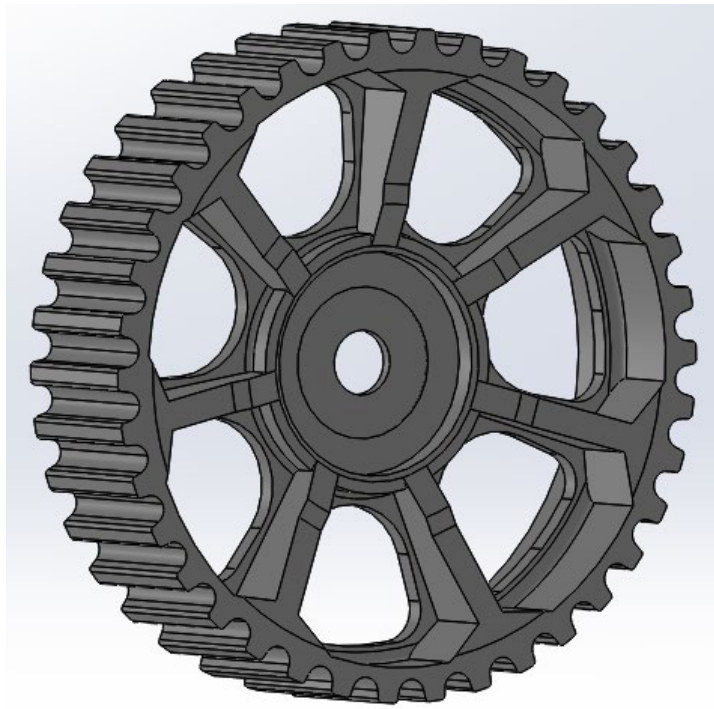


Figure 31 Rear of plastic cam pulley

The new model was simulated in the same static study as the original cam pulley was.

It is visible that the stress in this cam pulley is spread across more of the cam pulley spokes compared to the original. The highest stress value from the simulation was $1.625 \times 10^7 \text{ N/m}^2$ which is much lower than the originals.

The minimum safety factor for the new design was determined to be 6.4. this is higher than the original as the PA Type 6 can flex more without breaking compared to the aluminium. However, the cam pulley needs to maintain stiffness so that any deformation doesn't affect the valve timing.

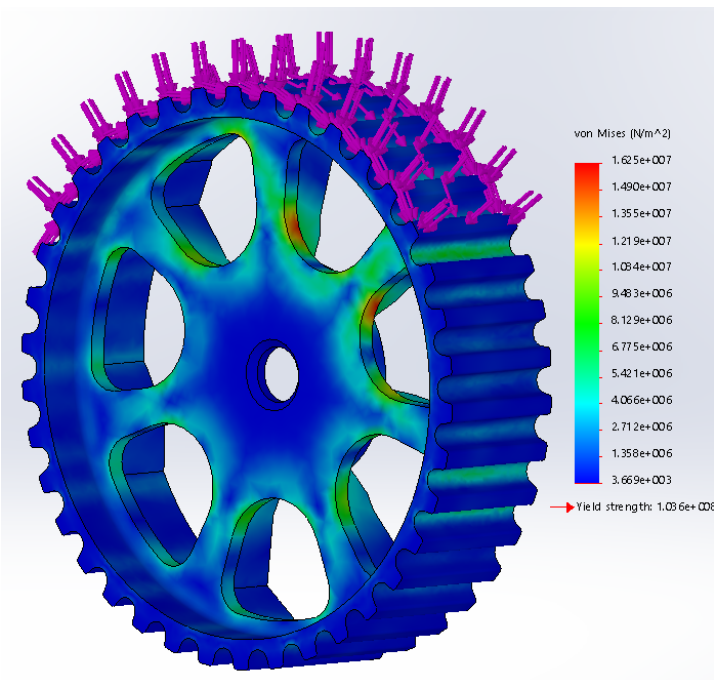


Figure 32 stress plot of plastic cam pulley

Figure 33 shows the deformation of the pulley at a deformation scale of 37.5.

Most of the deformation occurs along the pulleys teeth and this is at a maximum of 0.328mm. this is over eleven times higher than the original. This shows that although the spokes of the new design with PA type 6 can withstand the forces from the belt, the teeth which are a restricting factor to the design cannot take the load without shifting to much. This deformation may be enough to off-set the valve timing.

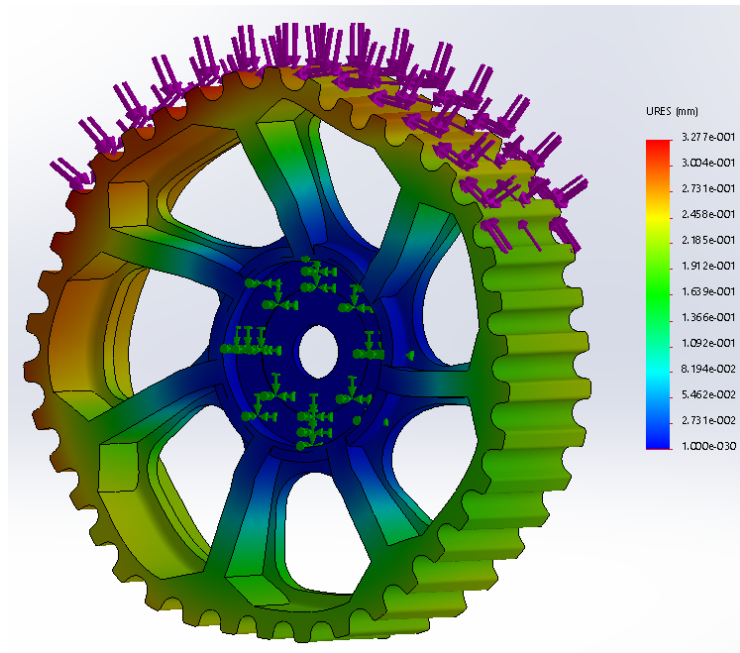


Figure 33 deformation plot of plastic cam pulley

Carbon Fibre Cam pulley

The second cam pulley model was to be made of chopped carbon fibre with technical specifications taken from Quantum Composites AMC 8593. This is the same chopped carbon fibre specification as used in the suspension arm and the single monocoque as well as several other components for the Lamborghini Sesto Elemento. This material was chosen as it can easily be forged into intricate dimensions unlike carbon fibre sheets (Quantum Composites, 2018).

Carbon fibre has a very high tensile strength and is very durable while maintaining a very low mass density, however it is more flexible than aluminium (Bino, Retnam, Ramachandran, & Sivapragash, 2015). The technical specification of the chopped carbon fibre AMC 8593 is shown below.

Property	Value	Units
Elastic Modulus	44800	N/mm ²
Poisson's Ratio	0.32	N/A
Shear Modulus		N/mm ²
Mass Density	1470	kg/m ³
Tensile Strength	421	N/mm ²
Compressive Strength		N/mm ²
Yield Strength	378.9	N/mm ²
Thermal Expansion Coefficient		/K

Figure 34 AMC 8593 technical specifications

From the results of the PA type 6 cam pulley, it was suggested that by increasing the number of spokes, the stress on the component was distributed more across the spokes implying that there wasn't as much stress concentration on the spoke under the load applied. Using this knowledge, the carbon fibre cam pulley was also designed with seven spokes. When running the model in the same simulation as the previous two models, the carbon fibre cam pulley seemed to retain its shape very well while maintaining a high safety factor. The next step was then to see how much less material could be used to make the design lighter.

By considering the spokes as cantilever beams, the formula for the deflection at the end of the beam is as shown below.

$$\text{Max Deflection} = \frac{FL^3}{3EI}$$

Where: F = force applied

L = length of the beam

E = Youngs modulus

I = second moment of area

This formula shows that as the second moment of area increases, the deflection at the end of the beam will decrease. The value for the second moment of area (I) is completely dependable on the breadth (b) and depth (d) of the beam and is calculated using the equation below.

$$I = \frac{bd^3}{12}$$

This shows that the depth of the beam has more effect on the second moment of area compared to the breadth. This is because the depth value is cubed, and if it is doubled for example, the value of the second moment of area will go up by eight times.

Using this logic, to maintain minimum deformation, the thickness of the spokes would initially be reduced. This proved to be an efficient means to reduce the mass of the cam pulley as the total deformation was still higher than the original cam pulleys deformation.

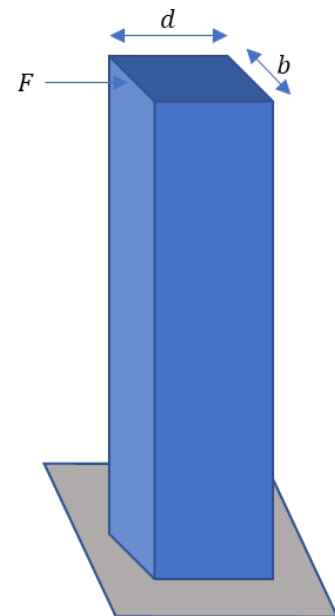


Figure 35 bending beam diagram

The next modification made was to the width of the spokes. These were made thinner by increasing the area between the spokes until the maximum allowable deformation of the spokes was obtained while maintaining a high enough safety factor.

With the maximum amount of allowable material removed from the spokes, attention was given to the rim of the cam pulley. By reducing the amount of material on the rim, this will significantly reduce the components moment of inertia. As the outer part of the rim, the teeth, is a restricted factor, the modifications were made to the inside of the rim. The rim's thickness was reduced in a tapered design so that it would gradually get thinner away from the spokes. This was method was chosen as would keep the maximum deformation on the far sides to a minimum.

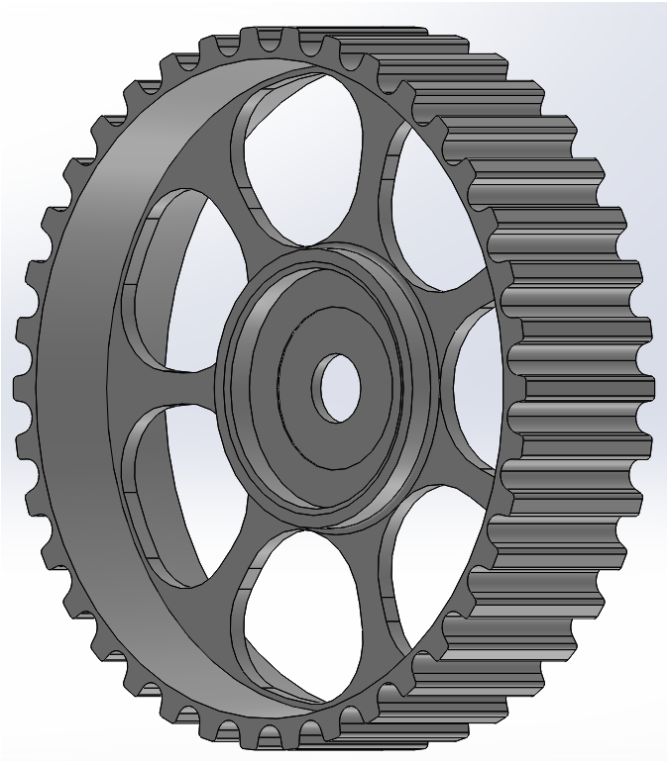


Figure 36 carbon fibre cam pulley design

Like the plastic cam pulley design, the stress caused by the applied forces have been distributed more across the spokes. However, the maximum stress on the component at 4.503 N/m^2 is higher than that of the original cam pulley. Because the chopped carbon fibre has a higher yield strength, it is capable of withstanding higher stresses than the aluminium.

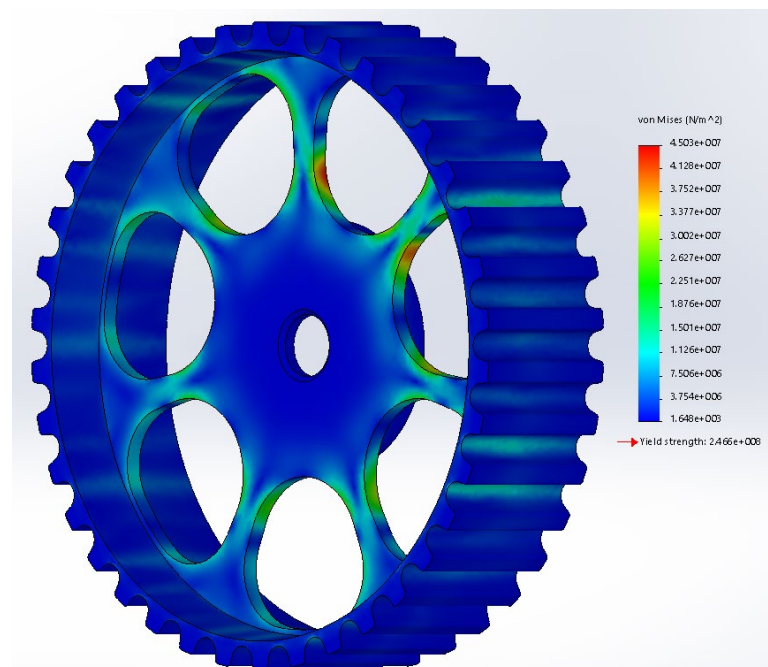


Figure 37 carbon fibre cam pulley stress plot

Figures 38 and 39 show the safety factor plot and deformation plot at a deformation scale of 1145.05. The maximum deformation in the cam pulley is 0.012mm at the rim and the minimum safety factor was determined to be 5.48. The maximum deformation is less than half of the original cam pulley, and the safety factor is 1.3 times higher.

This was the optimal design which the author was able to achieve. When more material was attempted to be reduced from the spokes, the deformation became larger than the original cam pulley, but the safety factor was still high enough. This is because the spokes became too thin and would allow too much rotational movement of the rim. When the rim was made thinner, the deformation was close to that of the original cam pulley, but the safety factor dropped below 4. This is because although the rim was still stiff enough, the stress in the rim was increased resulting in a lower factor of safety.

However, less deformation and a higher safety factor is not necessarily a disadvantage as the cam pulley should still perform in the same way as the original, if not better. Having a higher safety factor and a lower deformation also means that the cam pulley would be able to withstand higher forces for example if the engine was tuned to reach higher RPMs.

Full dimensions of the cam gears can be found in the appendix.

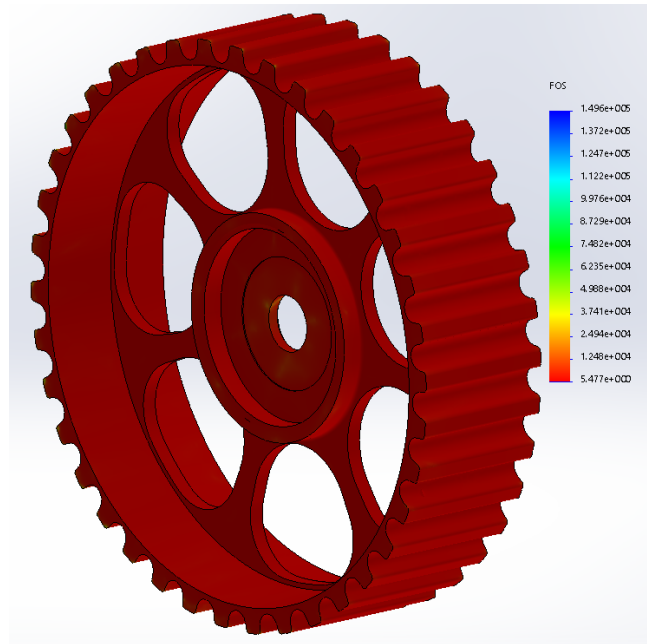


Figure 38 carbon fibre cam pulley safety factor plot

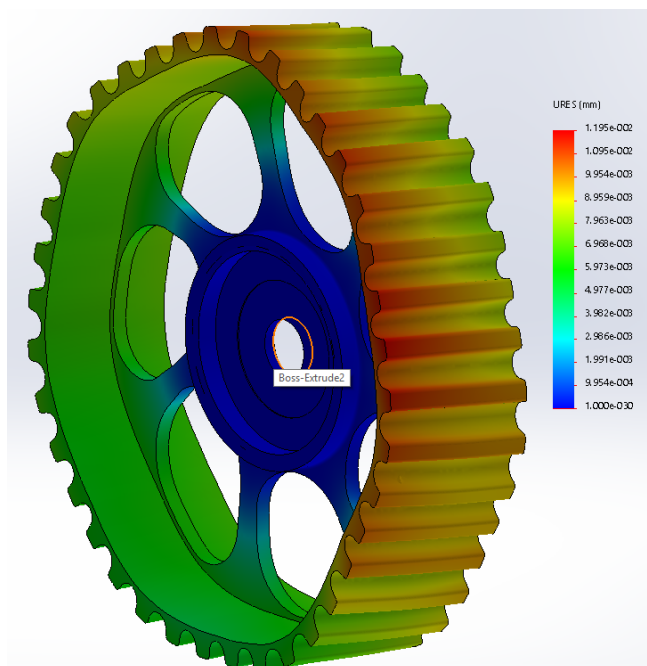


Figure 39 carbon fibre cam pulley deformation plot

Analysis

The plastic cam pulley was unable to meet the performance level of the original cam pulley as the deformation in the teeth was too large. This cam pulley was therefore deemed unsuitable as a replacement cam pulley for the Ford engine. However, for other pulleys or gears which are externally mounted with a lighter load, such as the fan and alternator pulleys, a pulley made of PA type 6 may be a suitable alternative. PA type 6 can also have useful stiffness improvements with the inclusion of fibre glass (AZO, 2018).

The chopped carbon fibre pulley was able to meet the performance of the aluminium cam pulley. This shows that it would be a suitable option as a modified cam pulley for the Ford sigma 1.6. The mass properties of the original and the chopped carbon fibre pulley are shown below.

	Carbon fibre gear	Aluminium gear
Mass (grams)	69.99	155.77
Volume (m^3)	4.76E-05	5.56E-05
Moment of Inertia ($Kg.m^2$)	1.66E-04	3.64E-04

Figure 40 mass properties of original pulley vs carbon fibre pulley

From figure 40, the mass properties of the chopped carbon fibre cam pulley show great improvement over the original aluminium cam pulley. The mass of the new pulley is 45% of the original pulley and the moment of inertia is $1.98 \times 10^{-4} Kg.m^2$ less. A reduced moment of inertia means that less torque is needed to accelerate it.

Using data from another engine of similar size, the engine acceleration for the Ford Sigma 1.6 was assumed to be 3200 RPM/second. As the crankshaft rotates twice for every cam shaft revolution, the cam shaft acceleration will therefore be 1600 RPM/second. By multiplying the camshaft acceleration with the moment of inertia, the extra torque needed to accelerate the cam pulley can be determined. For the original cam pulley, the extra torque needed was calculated to be $6.10 \times 10^{-2} Nm$ and for the new cam pulley this was $2.78 \times 10^{-2} Nm$. Although this is a significant improvement, it won't have much effect on the engines performance as these values are miniscule in the grand scheme of the engines torque production.

Despite the chopped carbon fibre cam pulley not having a significant improvement to the engines performance, the principals used to design it can be applied to other pulleys and gears. As carbon fibre has a high thermal resistance and doesn't corrode in oil, it can be used for internal engine components such as drive gears where it may significantly improve an engines performance.

Currently carbon fibre costs are prohibitively high for many of the industry sectors at £6.38/KG compare to that of aluminium at £1.36/KG (Meredith, Bilson, Powe, Collings, & Kirwan, 2015). This price difference will make it cost ineffective to mass produce using carbon fibre. However, with further developments and advances in the carbon fibre industry, this may eventually become a cost-effective method. For the moment, applications of carbon fibre are mainly used for high performance or luxury cars as costs may not be a limitation to the market for these vehicles.

A proven result of using forged composite processes to produce chopped carbon fibre components in a matter of minutes is the monocoque of the Lamborghini Sesto Elemento. The traditional way of producing carbon fibre components is out of an autoclave which would take hours. The forged composite process has been proven to have a high-volume parts output of advanced carbon fibre structures in commercial production (Quantum Composites, 2018).

To obtain better measurements of the cam shaft lobe profile, it would be recommended to use a CNC machine with a laser measuring system. When measuring with DTI gauges, the contact point is sometimes shifted by the friction from the cam lobe to the DTI pointer tip. This results in minute errors in the cam profile measurements, but when calculated for valve acceleration, these minute errors create a significant amount of noise to the graph. This noise is then translated further into the calculation of the forces. By averaging the measurement results, the noise can be reduced but not entirely eliminated.

Further Work

To further develop the chopped carbon fibre cam pulley, the next step would be to produce a prototype. To do this a high-quality 3D print of the component should first be made to ensure that the dimensions are suitable and correct. Using the 3D printed component, a cast can be made to produce the component out of the chopped carbon fibre. Close inspections of the component should be made to ensure that no imperfections have occurred during the production process.

By testing the component in a simulated environment by applying known forces will allow the technician to assess the components performance and limitations. The technician can then decide if the component is suitable to be tested on an engine. This process will reduce the chances of the component failing when tested on the engine, which may become troublesome and costly because other components such as valves or pistons may be ruined if the component fails.

Once the component has been thoroughly tested, the most efficient and cost-effective process must be designed to produce the component at a larger scale whilst maintaining the production quality. From here the component will be ready for sale.

Conclusion

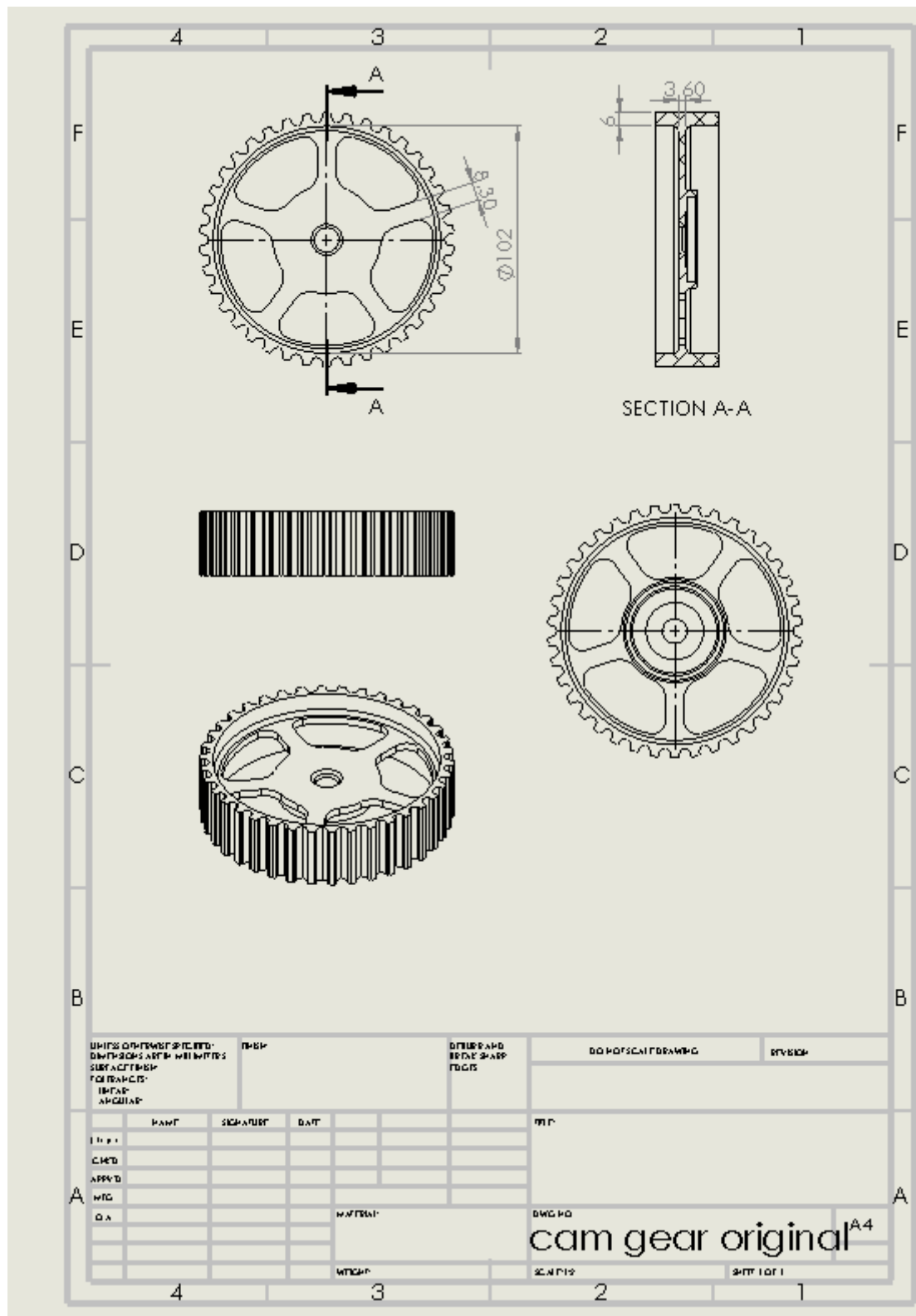
The results obtained for the forces acting on the cam pulley by calculating from first principals show that this is a suitable method to find accurate values. By validating the results which were acquired from Solidworks, the new models can be designed with confidence that their results will be accurate. The chopped carbon fibre cam pulley has been proven to improve the performance of the original cam pulley by a significant amount and the practices which were used in this report can be applied to other models which will have a significant impact to the engines performance. With further advancements being made in the carbon fibre industry, the useful possibilities for its application will continue to grow.

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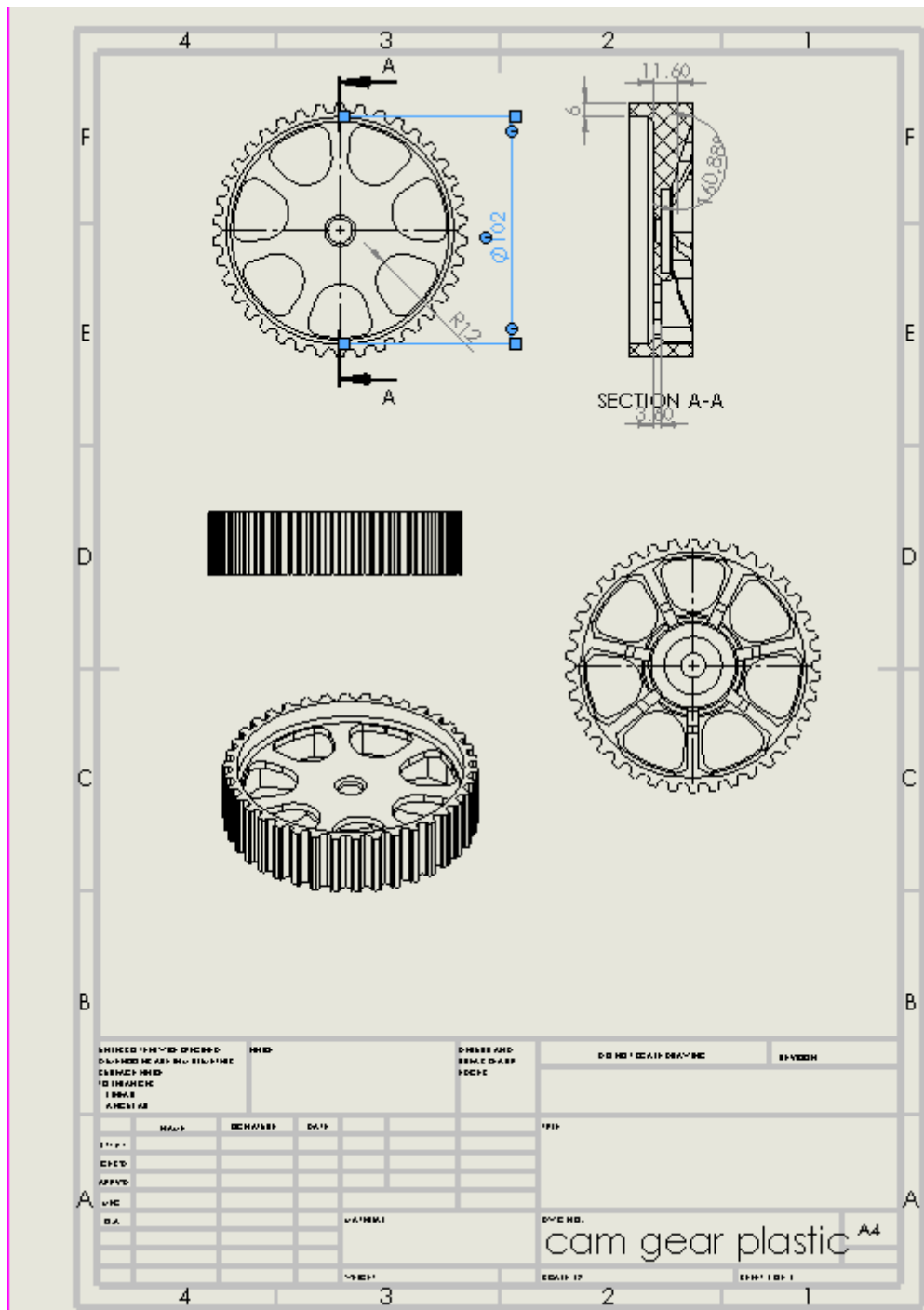
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Appendix

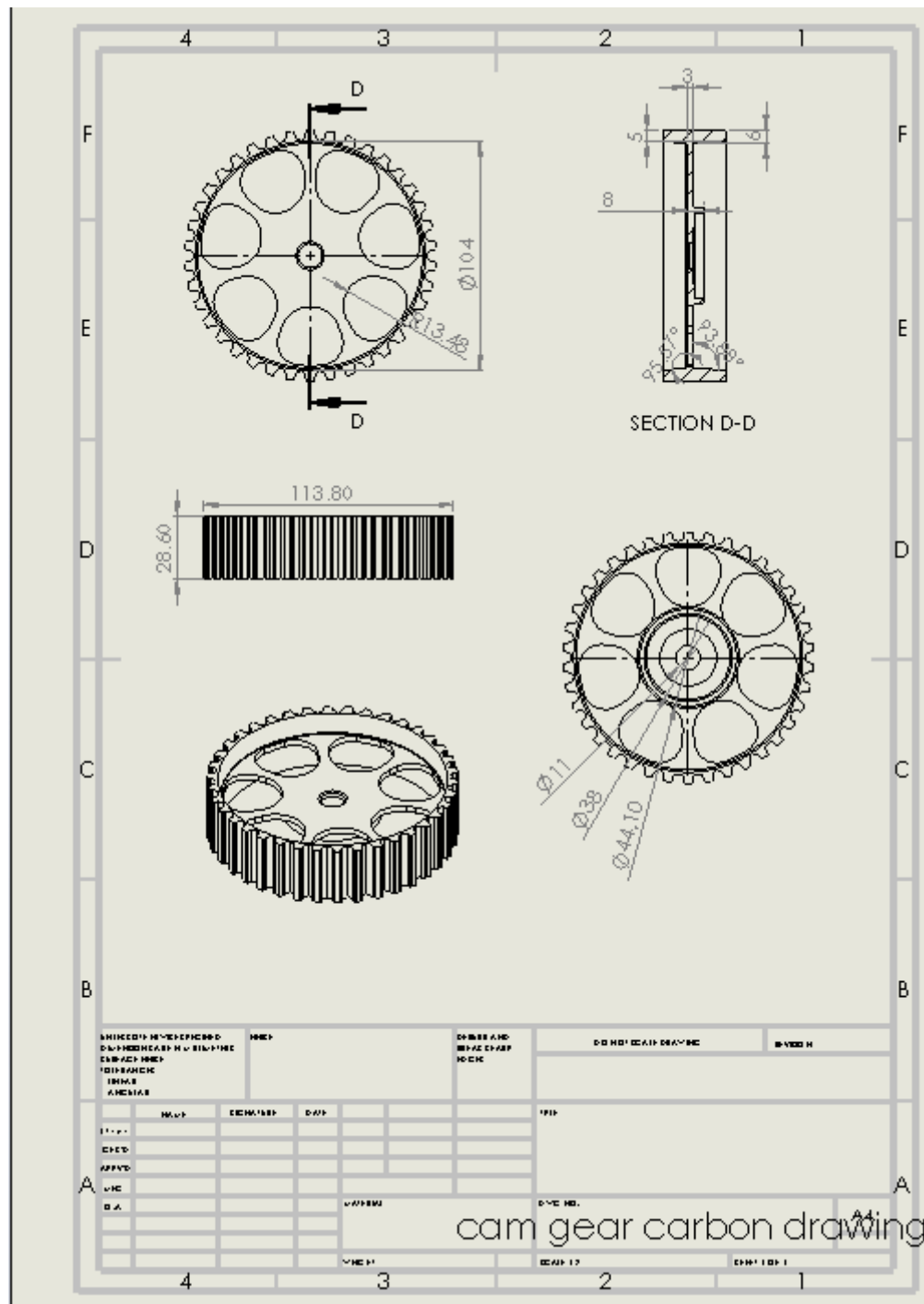
original cam pulley



Plastic cam pulley

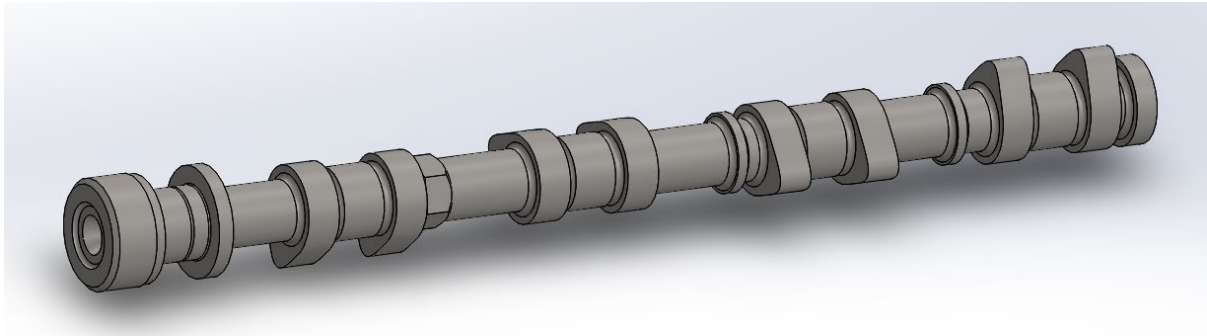


Chopped Carbon Fibre cam pulley

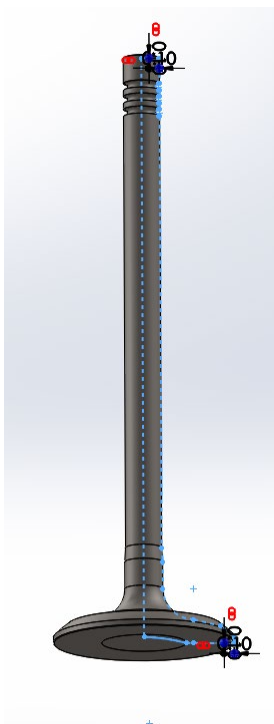


Valvetrain Components

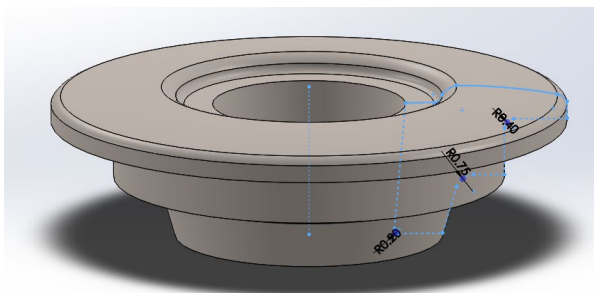
Camshaft



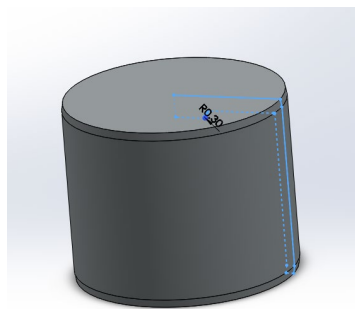
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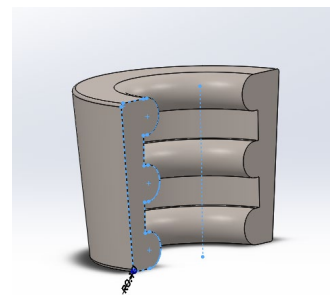
Spring retainer



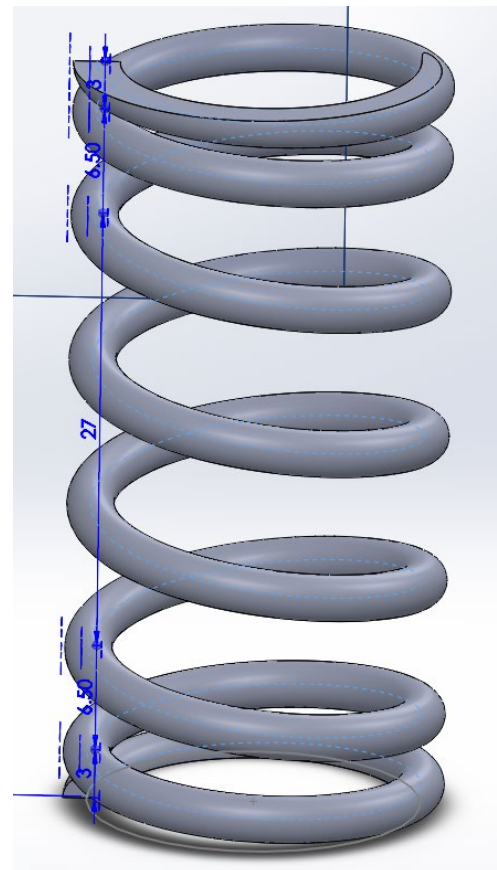
Buckets`



Collets

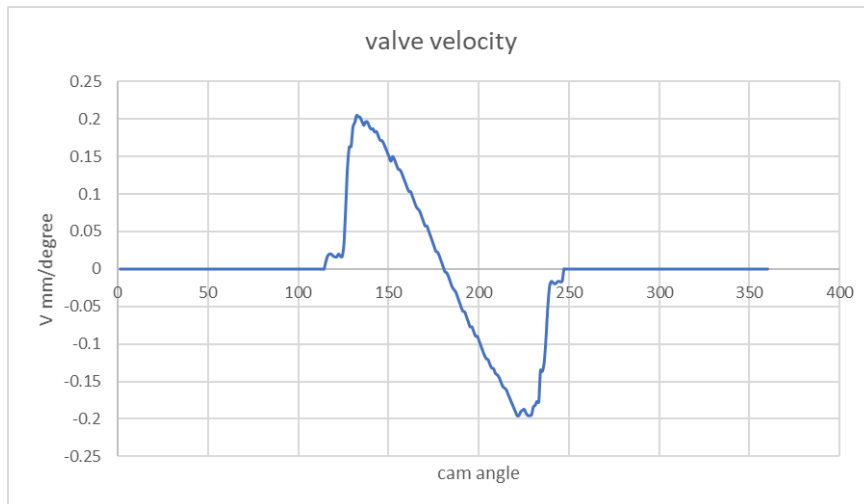


Valve Spring

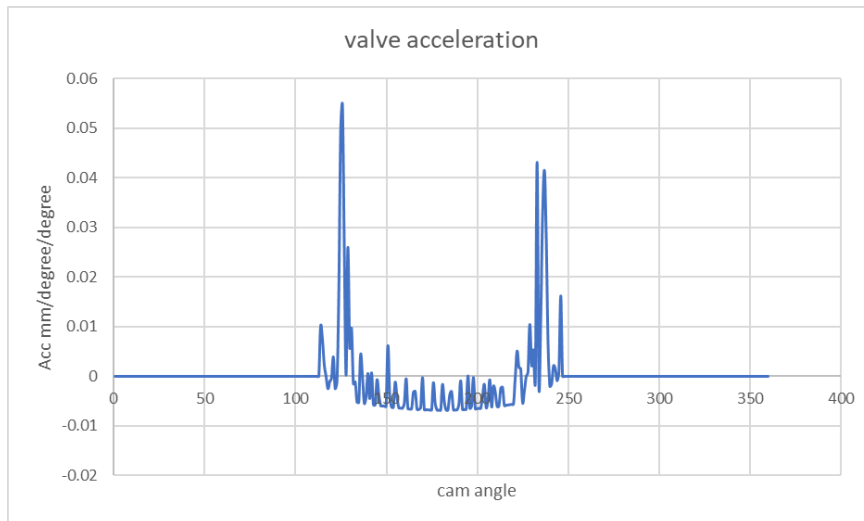


Excel Graphs

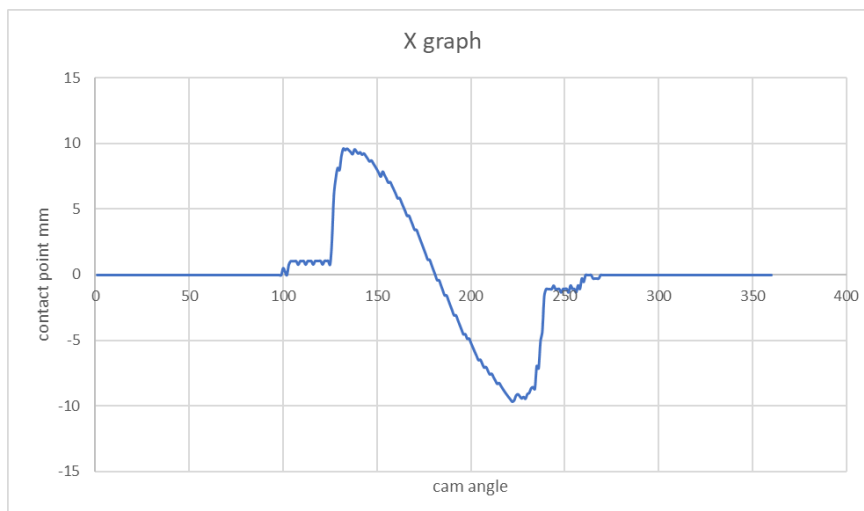
Valve velocity



Valve acceleration



X graph



Calculating torque from first principles

Input		0.25 mm		0.5 mm		1 mm		2 mm		3 mm		4 mm		5 mm		6 mm		7 mm		8 mm		9 mm		10 mm		11 mm		12 mm		13 mm		14 mm		15 mm		16 mm		17 mm		18 mm		19 mm		20 mm		21 mm		22 mm		23 mm		24 mm		25 mm		26 mm		27 mm		28 mm		29 mm		30 mm		31 mm		32 mm		33 mm		34 mm		35 mm		36 mm		37 mm		38 mm		39 mm		40 mm		41 mm		42 mm		43 mm		44 mm		45 mm		46 mm		47 mm		48 mm		49 mm		50 mm		51 mm		52 mm		53 mm		54 mm		55 mm		56 mm		57 mm		58 mm		59 mm		60 mm		61 mm		62 mm		63 mm		64 mm		65 mm		66 mm		67 mm		68 mm		69 mm		70 mm		71 mm		72 mm		73 mm		74 mm		75 mm		76 mm		77 mm		78 mm		79 mm		80 mm		81 mm		82 mm		83 mm		84 mm		85 mm		86 mm		87 mm		88 mm		89 mm		90 mm		91 mm		92 mm		93 mm		94 mm		95 mm		96 mm		97 mm		98 mm		99 mm		100 mm		101 mm		102 mm		103 mm		104 mm		105 mm		106 mm		107 mm		108 mm		109 mm		110 mm		111 mm		112 mm		113 mm		114 mm		115 mm		116 mm		117 mm		118 mm		119 mm		120 mm		121 mm		122 mm		123 mm		124 mm		125 mm		126 mm		127 mm		128 mm		129 mm		130 mm		131 mm		132 mm		133 mm		134 mm		135 mm		136 mm		137 mm		138 mm		139 mm		140 mm		141 mm		142 mm		143 mm		144 mm		145 mm		146 mm		147 mm		148 mm		149 mm		150 mm		151 mm		152 mm		153 mm		154 mm		155 mm		156 mm		157 mm		158 mm		159 mm		160 mm		161 mm		162 mm		163 mm		164 mm		165 mm		166 mm		167 mm		168 mm		169 mm		170 mm		171 mm		172 mm		173 mm		174 mm		175 mm		176 mm		177 mm		178 mm		179 mm		180 mm		181 mm		182 mm		183 mm		184 mm		185 mm		186 mm		187 mm		188 mm		189 mm		190 mm		191 mm		192 mm		193 mm		194 mm		195 mm		196 mm		197 mm		198 mm		199 mm		200 mm		201 mm		202 mm		203 mm		204 mm		205 mm		206 mm		207 mm		208 mm		209 mm		210 mm		211 mm		212 mm		213 mm		214 mm		215 mm		216 mm		217 mm		218 mm		219 mm		220 mm		221 mm		222 mm		223 mm		224 mm		225 mm		226 mm		227 mm		228 mm		229 mm		230 mm		231 mm		232 mm		233 mm		234 mm		235 mm		236 mm		237 mm		238 mm		239 mm		240 mm		241 mm		242 mm		243 mm		244 mm		245 mm		246 mm		247 mm		248 mm		249 mm		250 mm		251 mm		252 mm		253 mm		254 mm		255 mm		256 mm		257 mm		258 mm		259 mm		260 mm		261 mm		262 mm		263 mm		264 mm		265 mm		266 mm		267 mm		268 mm		269 mm		270 mm		271 mm		272 mm		273 mm		274 mm		275 mm		276 mm		277 mm		278 mm		279 mm		280 mm		281 mm		282 mm		283 mm		284 mm		285 mm		286 mm		287 mm		288 mm		289 mm		290 mm		291 mm		292 mm		293 mm		294 mm		295 mm		296 mm		297 mm		298 mm		299 mm		300 mm		301 mm		302 mm		303 mm		304 mm		305 mm		306 mm		307 mm		308 mm		309 mm		310 mm		311 mm		312 mm		313 mm		314 mm		315 mm		316 mm		317 mm		318 mm		319 mm		320 mm		321 mm		322 mm		323 mm		324 mm		325 mm		326 mm		327 mm		328 mm		329 mm		330 mm		331 mm		332 mm		333 mm		334 mm		335 mm		336 mm		337 mm		338 mm		339 mm		340 mm		341 mm		342 mm		343 mm		344 mm		345 mm		346 mm		347 mm		348 mm		349 mm		350 mm		351 mm		352 mm		353 mm		354 mm		355 mm		356 mm		357 mm		358 mm		359 mm		360 mm		361 mm		362 mm		363 mm		364 mm		365 mm		366 mm		367 mm		368 mm		369 mm		370 mm		371 mm		372 mm		373 mm		374 mm		375 mm		376 mm		377 mm		378 mm		379 mm		380 mm		381 mm		382 mm		383 mm		384 mm		385 mm		386 mm		387 mm		388 mm		389 mm		390 mm		391 mm		392 mm		393 mm		394 mm		395 mm		396 mm		397 mm		398 mm		399 mm		400 mm		401 mm		402 mm		403 mm		404 mm		405 mm		406 mm		407 mm		408 mm		409 mm		410 mm		411 mm		412 mm		413 mm		414 mm		415 mm		416 mm		417 mm		418 mm		419 mm		420 mm		421 mm		422 mm		423 mm		424 mm		425 mm		426 mm		427 mm		428 mm		429 mm		430 mm		431 mm		432 mm		433 mm		434 mm		435 mm		436 mm		437 mm		438 mm		439 mm		440 mm		441 mm		442 mm		443 mm		444 mm		445 mm		446 mm		447 mm		448 mm		449 mm		450 mm		451 mm		452 mm		453 mm		454 mm		455 mm		456 mm		457 mm		458 mm		459 mm		460 mm		461 mm		462 mm		463 mm		464 mm		465 mm		466 mm		467 mm		468 mm		469 mm		470 mm		471 mm		472 mm		473 mm		474 mm		475 mm		476 mm		477 mm		478 mm		479 mm		480 mm		481 mm		482 mm		483 mm		484 mm		485 mm		486 mm		487 mm		488 mm		489 mm		490 mm		491 mm		492 mm		493 mm		494 mm		495 mm		496 mm		497 mm		498 mm		499 mm		500 mm		501 mm		502 mm		503 mm		504 mm		505 mm		506 mm		507 mm		508 mm		509 mm		510 mm		511 mm		512 mm		513 mm		514 mm		515 mm		516 mm		517 mm		518 mm		519 mm		520 mm		521 mm		522 mm		523 mm		524 mm		525 mm		526 mm		527 mm		528 mm		529 mm		530 mm		531 mm		532 mm		533 mm		534 mm		535 mm		536 mm		537 mm		538 mm		539 mm		540 mm		541 mm		542 mm		543 mm		544 mm		545 mm		546 mm		547 mm		548 mm		549 mm		550 mm		551 mm		552 mm		553 mm		554 mm		555 mm		556 mm		557 mm		558 mm		559 mm		560 mm		561 mm		562 mm		563 mm		564 mm		565 mm		566 mm		567 mm		568 mm		569 mm		570 mm		571 mm		572 mm		573 mm		574 mm		575 mm		576 mm		577 mm		578 mm		579 mm		580 mm		581 mm		582 mm		583 mm		584 mm		585 mm		586 mm		587 mm		588 mm		589 mm		590 mm		591 mm		592 mm		593 mm		594 mm		595 mm		596 mm		597 mm		598 mm		599 mm		600 mm		601 mm		602 mm		603 mm		604 mm		605 mm		606 mm		607 mm		608 mm		609 mm		610 mm		611 mm		612 mm		613 mm		614 mm		615 mm		616 mm		617 mm		618 mm		619 mm		620 mm		621 mm		622 mm		623 mm		624 mm		625 mm		626 mm		627 mm		628 mm		629 mm		630 mm		631 mm		632 mm		633 mm		634 mm		635 mm		636 mm		637 mm		638 mm		639 mm		640 mm		641 mm		642 mm		643 mm		644 mm		645 mm		646 mm		647 mm		648 mm		649 mm		650 mm		651 mm		652 mm		653 mm		654 mm		655 mm		656 mm		657 mm		658 mm		659 mm		660 mm		661 mm		662 mm		663 mm		664 mm		665 mm		666 mm		667 mm		668 mm		669 mm		670 mm		671 mm		672 mm		673 mm		674 mm		675 mm		676 mm		677 mm		678 mm		679 mm		680 mm		681 mm		682 mm		683 mm		684 mm		685 mm		686 mm		687 mm		688 mm		689 mm		690 mm		691 mm		692 mm		693 mm		694 mm		695 mm		696 mm		697 mm		698 mm		699 mm		700 mm		701 mm		702 mm		703 mm		704 mm		705 mm		706 mm		707 mm		708 mm		709 mm		710 mm		711 mm		712 mm		713 mm		714 mm		715 mm		716 mm		717 mm		718 mm		719 mm		720 mm		721 mm		722 mm		723 mm		724 mm		725 mm		726 mm		727 mm		728 mm		729 mm		730 mm		731 mm		732 mm		733 mm		734 mm		735 mm		736 mm		737 mm		738 mm		739 mm		740 mm		741 mm		742 mm		743 mm		744 mm		745 mm		746 mm		747 mm		748 mm		749 mm		750 mm		751 mm		752 mm		753 mm		754 mm		755 mm		756 mm		757 mm		758 mm		759 mm		760 mm		761 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mm		887 mm		888 mm		889 mm		890 mm		891 mm		892 mm		893 mm		894 mm		895 mm		896 mm		897 mm		898 mm		899 mm		900 mm		901 mm		902 mm		903 mm		904 mm		905 mm		906 mm		907 mm		908 mm		909 mm		910 mm		911 mm		912 mm		913 mm		914 mm		915 mm		916 mm		917 mm		918 mm		919 mm		920 mm		921 mm		922 mm		923 mm		924 mm		925 mm		926 mm		927 mm		928 mm		929 mm		930 mm		931 mm		932 mm		933 mm		934 mm		935 mm		936 mm		937 mm		938 mm		939 mm		940 mm		941 mm		942 mm		943 mm		944 mm		945 mm		946 mm		947 mm		948 mm		949 mm		950 mm		951 mm		952 mm		953 mm		954 mm		955 mm		956 mm		957 mm		958 mm		959 mm		960 mm		961 mm		962 mm		963 mm		964 mm		965 mm		966 mm		967 mm		968 mm		969 mm		970 mm		971 mm		972 mm		973 mm		974 mm		975 mm		976 mm		977 mm		978 mm		979 mm		980 mm		981 mm		982 mm		983 mm		984 mm		985 mm		986 mm		987 mm		988 mm		989 mm		990 mm		991 mm		992 mm		993 mm		994 mm		995 mm		996 mm		997 mm		998 mm		999 mm		1000 mm		1001 mm		1002 mm		1003 mm		1004 mm		1005 mm		1006 mm		1007 mm		1008 mm		1009 mm		1010 mm		1011 mm		1012 mm		1013 mm		1014 mm		1015 mm		1016 mm		1017 mm		1018 mm		1019 mm		1020 mm		1021 mm		1022 mm		1023 mm		1024 mm		1025 mm		1026 mm		1027 mm		1028 mm		1029 mm		1030 mm		1031 mm		1032 mm		1033 mm		1034 mm		1035 mm</	
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										MIN torque	-3.71481 Nm
										MAX torque	13.01641 Nm
		cam 1-2		cam 3-4		cam 5-6		cam 7-8			
torque/lobe	angle	torque	angle	torque	angle	torque	angle	torque	total		
0	0	0	90	0	270	0	180	1.040316	1.040316229		
0	1	0	91	0	271	0	181	1.428737	1.428736605		
0	2	0	92	0	272	0	182	1.562195	1.562194976		
0	3	0	93	0	273	0	183	1.230708	1.23070757		
0	4	0	94	0	274	0	184	0.748389	0.748388837		
0	5	0	95	0	275	0	185	0.456434	0.456433897		
0	6	0	96	0	276	0	186	0.567996	0.567995923		
0	7	0	97	0	277	0	187	0.240511	0.240510912		
0	8	0	98	0	278	0	188	-0.08449	-0.084491426		
0	9	0	99	0	279	0	189	-0.38316	-0.383158855		
0	10	0	100	0	280	0	190	-0.49573	-0.495732536		
0	11	0	101	0	281	0	191	-0.72041	-0.72040565		
0	12	0	102	0	282	0	192	-1.03212	-1.032123003		
0	13	0	103	0	283	0	193	-1.33864	-1.338642766		
0	14	0	104	0	284	0	194	-1.03424	-1.034235284		
0	15	0	105	0	285	0	195	-1.91282	-1.91281967		
0	16	0	106	0	286	0	196	-1.85422	-1.854224858		
0	17	0	107	0	287	0	197	-1.42888	-1.428883263		
0	18	0	108	0	288	0	198	-2.18528	-2.185280978		
0	19	0	109	0	289	0	199	-2.46027	-2.460267339		
0	20	0	110	0	290	0	200	-2.72714	-2.727141433		
0	21	0	111	0	291	0	201	-2.98537	-2.985368454		
0	22	0	112	0	292	0	202	-2.80349	-2.803494342		
0	23	0	113	1.184372	293	0	203	-2.53096	-1.346587515		
0	24	0	114	0.893265	294	0	204	-3.4327	-2.539437596		
0	25	0	115	1.054769	295	0	205	-3.32577	-2.271001113		
0	26	0	116	0.904386	296	0	206	-2.61798	-1.713589582		
0	27	0	117	0.61194	297	0	207	-3.8238	-3.211862366		
0	28	0	118	0.781834	298	0	208	-3.03113	-2.249294546		
0	29	0	119	0.871793	299	0	209	-3.46677	-2.594974739		
0	30	0	120	1.236653	300	0	210	-4.1568	-2.920145518		
0	31	0	121	0.619654	301	0	211	-4.33446	-3.714808419		

Calculating belt and resultant forces

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y
1		max lift																							
2	inlet	110 ATDC																	Fres hor'	Fres vert		Fres angle	Fres int	F res exh	
3	exhaust	110 BTDC																	-77.1424	92.8028		-50.2649	120.679	75.6158	
4				crank angle	0		inlet			exhaust			F1		F2		W/D/deg								
5	D	115 mm	TDC				125	10.7951		235	-7.06882		64.8045		-122.936		0.06504								
6	R	57.5 mm					126	10.7765		236	-5.89253		84.9386		-102.479		0.08524								
7	RPM						127	1.98877		237	-2.51583		-9.1662		-43.7536		-0.0092								
8	θ intake*	80 degrees					128	13.7391		238	1.9057		272.083		33.1427		0.27305								
9	θ exhaust*	45 degrees					129	4.77288		239	2.04996		118.658		35.6516		0.11908								
10							130	5.43631		240	7.49424		224.879		130.335		0.22568								
11	Initial tension*	29.43 N					131	-1.66409		241	2.97936		22.8742		51.8149		0.02296								
12							132	-0.17202		242	2.16258		34.6183		37.61		0.03474								
13	Work Done	10.4742 J					133	-1.84198		243	2.16235		5.57158		37.6061		0.00559								
14							134	0.50894		244	4.2615		82.9642		74.113		0.08326								
15	FMEP	0.06546					135	4.6435		245	3.20748		136.539		55.7823		0.13703								
16							136	1.32614		246	1.89055		55.9425		32.8791		0.05614								
17	max F2	238.94 N					137	-1.4477		247	1.84722		6.9482		32.1256		0.00697								
18	max F1	272.083 N					138	-4.1518		248	1.8003		-40.8918		31.3096		-0.04104								
19	min F1	-271.457 N					139	1.35468		249	2.11848		60.4028		36.8432		0.06062								
20							140	-2.0816		250	3.87693		31.2231		67.4248		0.03133								
21							141	2.84766		251	1.71525		79.355		29.8304		0.07964								
22							142	-14.2299		252	1.65752		-218.651		28.8265		-0.21943								
23	max F res int	0 N					143	-0.02948		253	1.59663		27.2549		21.7676		0.02735								
24	min F res int	0 N					144	-0.2924		254	2.60535		40.2252		45.3105		0.04037								
25							145	-7.06862		255	2.54662		-78.647		44.2891		-0.07893								
26	max F res exh	289.928 N					146	-5.89253		256	1.47843		-76.7669		25.7118		-0.07704								
27	min F res exh	21.7977 N					147	-2.51583		257	1.40832		-19.2611		24.4925		-0.01933								
28							148	1.9057		258	1.44735		58.3139		25.1712		0.05862								
29	max torque	13.7391 Nm					149	2.04996		259	2.91555		86.3368		50.7053		0.08666								
30	Force on gears	238.94 N					150	7.49424		260	1.26813		152.389		22.0544		0.15293								
31							151	2.97936		261	1.19034		72.5165		20.7016		0.07278								
32	Min torque	-14.2299 Nm					152	2.16258		262	1.1105		56.9231		19.313		0.05713								
33	Force on gears	-247.477 N					153	2.16235		263	1.02789		55.4981		17.8921		0.0557								
34		F1	F2				154	4.2615		264	0.94542		90.5551		16.4421		0.00908								
35		272.083	33.1427				155	3.20748		265	1.80564		87.1848		31.4024		0.0875								
36		27.7353	3.37846				156	1.89055		266	0.97701		49.8706		16.9915		0.05005								
37							157	1.84722		267	0.77625		45.6256		13.5		0.04579								
38							158	1.8003		268	0.68888		43.2901		11.9805		0.04344								
39							159	2.11848		269	0.60066		47.2894		10.4462		0.04746								
40							160	3.87693		270	1.04032		85.5172		18.0925		0.08582								
41							161	1.71525		271	0.55668		39.5118		9.68145		0.03965								
42							162	1.65752		272	0.42228		36.1705		7.34405		0.0363								
43							163	1.59663		273	0.33282		33.5557		5.78817		0.03368								