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Y Drindod Dewi Sant

UNIVERSITY OF WALES

Trinity Saint David

SWANSEA - ABERTAWE

Valve Train Friction Reduction Through the use of Materials and Coatings

Todd Farr

School of Engineering

BEng Advanced Engine Design

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

Signed:

Date:

Acknowledgements

First and foremost I would like to thank my supervisor Malcolm McDonald for the immense help, encouragement and support he provided me with throughout this study. Without being able to work under his supervision this study would not have been possible and it has been a great pleasure of mine.

Abstract

Automotive engines are consistently being developed to reduce friction and improve efficiency due to tougher emissions legislation consistently being introduced. This thesis looks into valve train friction specifically with the intention of decreasing the friction force in a Ford 1.6 GDTI EcoBoost engine through the use of coatings and materials.

The thesis includes calculating valve lift through the measurement of cam lift, this was then in turn used to find the accelerations the valve experiences, key figures needed for calculating stress and friction forces within the system. Finding the values of acceleration allowed a friction model to be made for the OEM system. Through the use of this model the research carried out was able to be confirmed, that valve train friction reduces with increased RPM.

FEA and SolidWorks allowed Fatigue safety factors to be calculated and new valves to be designed using the chosen Ti-6AL-4V titanium alloy. Using this alloy allowed the frictional forces to be reduced therefore increasing the efficiency of the engine. Along with a change of material for the valve, a DLC coating was mathematically modelled as being used on the valve bucket, allowing the friction within the engine to further be reduced and efficiency further increased. Research and the knowledge obtained from simulations and models then allowed a cost benefit analysis to be carried out determining whether either modification was viable for Ford to use within the future production of the engine.

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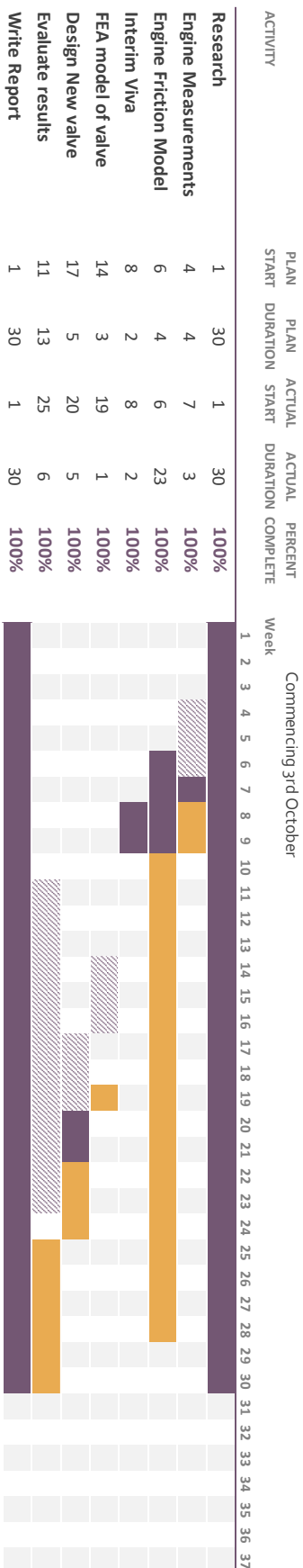
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Timing Plan

As can be seen from the picture to the left a timing plan had been made for this project. A timing plan aided in keeping on track with the project and not falling behind. As can be seen the engine friction modelling has a much longer duration than first specified, this was due to more modelling being carried out than originally thought. Another big change was the 3D modelling of the valve. This was started later but had a much shorter duration than originally thought due to being able to use a 3D scanner, this allowed a 3D model of the valve to be accurately made in a much shorter space of time. Overall the project didn't take longer than expected but some of the tasks were started later than planned, this however did not affect the project overall.

Project Planner



Chapter One: Introduction

1.1 Introduction

In the future of the automotive world, fuel consumption and engine emissions will become increasingly important due to stricter laws and legislation formed by governments to help prevent climate change. It is currently known (L28186 - EN - EUR-Lex, 2012), that European legislation (Euro 5 and 6) states that CO₂ levels of new passenger cars cannot exceed 130 g CO₂/km, this will also be reduced from 2020 to 95 g CO₂/km. These regulations were put in place due to 15% of the EU's total CO₂ emissions coming from light duty vehicles, i.e. cars and vans. These regulations mean that drive cycle measurements will further dominate the engineering of car sales. To obtain and improve the fuel consumption of an engine, friction reduction is key.

Figure 1 shows that of the total energy supplied to an engine, roughly 75% is lost due to engine inefficiencies, this means that only roughly 25% of the total energy supplied is actually used to drive the wheels of the vehicle.

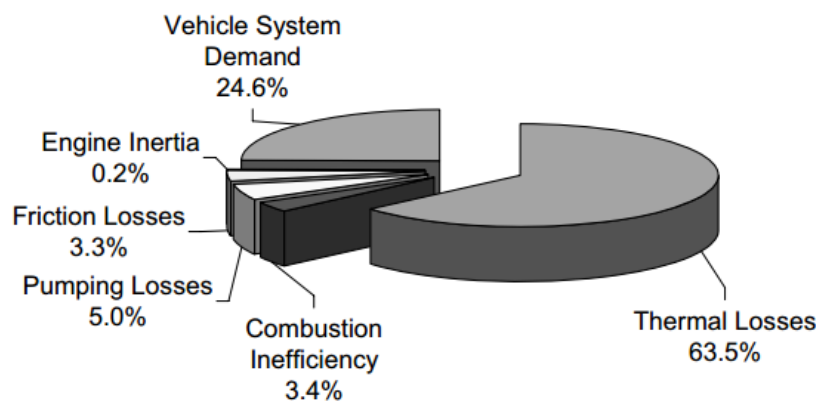


Figure 1, Pie Chart showing the Energy Losses in the Internal Combustion Engine
(Baglione et al., 2007)

The major causes of friction within the internal combustion engine are as follows:

- The piston and connecting rod assembly
- The valve train assembly
- The crankshaft bearings
- And engine ancillaries such as; the water pump, oil pump, air conditioning and power assisted steering.

It is generally understood that friction losses get worse with higher rpm, this is shown in the paper 'Prediction and Analysis of Engine Friction Power of a Diesel Engine Influenced by Engine Speed, Load, and Lubricant Viscosity' where it is stated that 'engine friction power increases approximately four to six times when engine speed increases from 1000 rpm to 2000 rpm at constant load of 50 Nm' (Singh et al., 2014).

This paper is going to look in depth into valve train friction, this will include looking into the causes of this friction, the effects that it has on the engine's power output and the possible improvements that could be made to improve friction reduction in the valve train. The paper will carry out the research on the Ford 1.6 GDTI (gasoline direct turbo injection) Ecoboost engine as this is the most readily available engine and one that the university currently has the most information on and specifications of.

1.2 Objectives

One of the major objectives of this paper is to research into the causes and effects of friction in the valve train of the internal combustion engine. This will allow the practical part of the paper to be carried out due to the research providing information on the measurements that will need to be carried out.

Another major objective is to calculate the friction in the valve train of the Ford engine. This will be carried out by taking measurements of the valve train parts in a workshop, the measurements will consist of dimensions and masses of the parts. By also finding the coefficient of friction within the system, the overall friction will be able to be calculated.

A further objective of this paper is to look into the stresses that occur on the valves within the engine. This objective will also look into the nominal fatigue safety factors of the valves to be able to carry out the next objective.

The final objective will be to design a new valve that conforms to the current nominal safety factors, the purpose of this objective is to improve on the current design with the goal of reducing friction within the valve train. Alongside this, a finite element Analysis of the valve will be carried out to determine where the major stresses will occur and to make sure the new design will not break when in use.

Chapter Two: Valve Train Friction

2.1 Introduction

Of the total friction losses in an engine, shown in figure 2 (Kamil, Rahman, and Bakar, 2014), piston friction and pumping losses are the largest factors making up 63% of the total engines friction losses. Valve train friction is the third largest contributor towards engine friction and is responsible for contributing 16%. Valve train friction is described as being made up from the friction of many different components, these include (Patton et al., 1989); camshaft hydrodynamic bearing friction, cam shaft seals, cam follower friction, oscillating hydrodynamic friction and the oscillating mixed lubrication friction. The continued research will look into these components that make up the total valve train friction so that calculations can be carried out to predict the friction caused by the valve train in the Ford 1.6L Ecoboost engine.

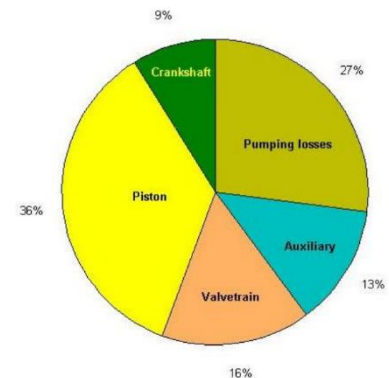


Figure 2, Pie chart showing the components of engine friction

Overall engine friction has already been stated as getting worse with increased rpm, the same however cannot be said about the individual component of valve train friction.

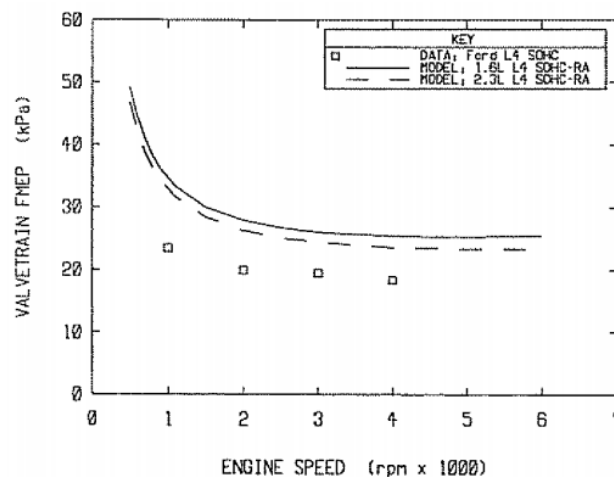


Figure 3, Valve FMEP against Engine Speed (Paton et al., 1989)

Figure 3 above shows the graph of valve train FMEP against engine rpm for both a ford 1.6L and 2.3L engine, it shows that valve train Friction has a unique characteristic in that it gets reduced when the rpm is increased. This is due to (Heywood, 1988), the valve train being under higher loads, because of spring forces, at low rpm whereas at higher rpm the inertia forces of the components masses take over leading to a lower FMEP at these rpm levels.

Chapter Three: Calculated Valve Train Friction

3.1 Introduction

This chapter will show the steps taken to obtain values of friction between the cam lobe and valve bucket. These steps are being carried out so that a base line friction value can be used when comparing the friction values of different valve types/materials later on in the paper.

3.2 Valve Train Measurements

To calculate the friction occurring between the valve bucket and the cam lobe a number of measurements first had to be taken. As can be seen from the figures below the mass of the moving parts within the valve train had to be found. This included (From right to left); valve spring, valve collets, valve, valve bucket and spring top cap.



Figure 6, Valve spring mass



Figure 5, Valve collet mass



Figure 4, Valve mass



Figure 9, Valve bucket mass



Figure 8, Spring cap mass



Figure 7, Spring stiffness

To then find the moving mass within the valve train the masses of the valve spring, valve collets, valve, valve bucket and spring top cap were added together; because approximately half of the spring is moving the weight of the spring was divided by 2. This is shown in figure 10 below, as can be seen in the table the total moving mass is 74.2 grams.

Inlet masses (g)					
Valve	bucket	collets	spring	spring cap	total moving mass
31.7	24	0.5	20.4	7.8	74.2

Figure 10, Moving mass within the valve train

As shown in figure 9 on the previous page the spring stiffness was tested by applying a mass, in pounds, to the spring and measuring the change in length that occurred, this gave the following table shown in figure 11. To find the Force in newton's applied to the spring, the mass in lbs was divided by 2.204 to convert to kg and was then multiplied by the acceleration due to gravity, 9.81. The stiffness of the spring was then calculated by dividing the change in force by the change in length from 1mm to 4mm which gave a spring stiffness of 15.13N/mm.

Inlet Spring			
x (mm)	mass (lbs)	force (N)	
0.5	0	0	
1	1.4	6.231397459	
1.5	3.6	16.02359347	
2	5	22.25499093	
2.5	6.6	29.37658802	
3	8	35.60798548	
3.5	9.8	43.61978221	K (N/mm)
4	11.6	51.63157895	15.13339383

Figure 11, Spring stiffness table

The spring was then measured to find the free length and the compressed length when fitted inside the head, this would allow the initial compression of the spring to be found, this will then be used later to find the spring force using the equation;

$$\text{Spring force} = K \cdot \Delta x$$

This shows that the spring force is the spring stiffness multiplied by the change in length of the spring, this can therefore be shown as the following;

$$\text{Spring force} = K \cdot (Y + Y_0)$$

Where K = the spring stiffness, Y = the maximum Valve lift and Y_0 = the initial compression.

The equation has been changed as such because the maximum valve lift plus the initial compression of the spring is the change in length from the spring's original free length. The measurements of the spring are shown in figure 12 below and the maximum valve lift will be calculated further on in the paper.

spring free length (mm)	spring compressed length (mm)	initial compression (Y0) (mm)
48.6	37.7	10.9

Figure 12, Valve spring initial compression

The next step was to measure the geometry of the cam lobe, this involved using a machine to turn the camshaft and a dial gauge. To find the shape of the cam lobe a measurement was taken every degree through all 360 degrees of rotation. The following table shown in figure 14 is the result of these measurements and gives a value of cam lift for every degree of camshaft rotation. From this table a graph of cam lift against camshaft angle could be made as shown below in figure 13.

cam angle	cam lift (mm)
0	0.03
1	0.03
2	0.04
3	0.04
4	0.05
5	0.05
6	0.06
7	0.07
8	0.08
9	0.08
10	0.09
11	0.1

Figure 14, Table of cam angle and cam lift

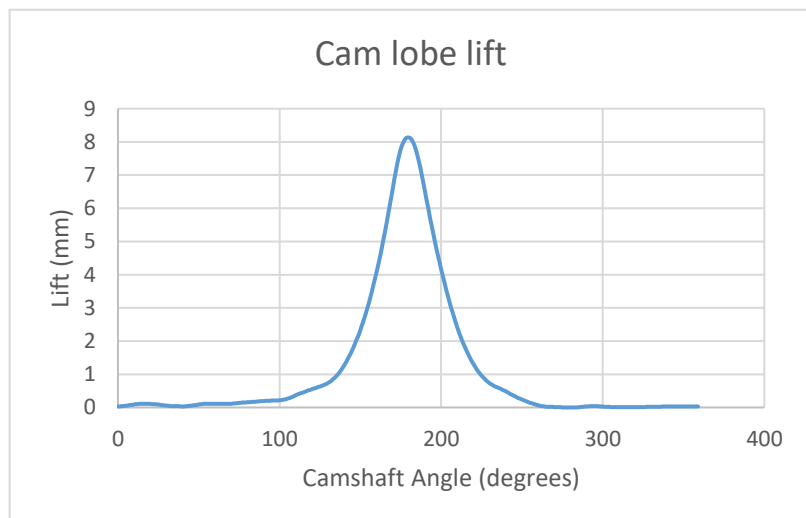


Figure 13, Cam lift versus Camshaft angle

3.3 Valve Train Calculations

By measuring the cam lobe lift, the actual valve lift could then be calculated using the base radius (R) of the cam lobe previously measured. Figure 15 below shows the method of calculating valve lift from cam lift, as can be seen OA is the base circle radius plus the cam lobe lift at every degree which gives a value 'S' for every degree of rotation. This then allows a value of OB to be calculated by knowing that:

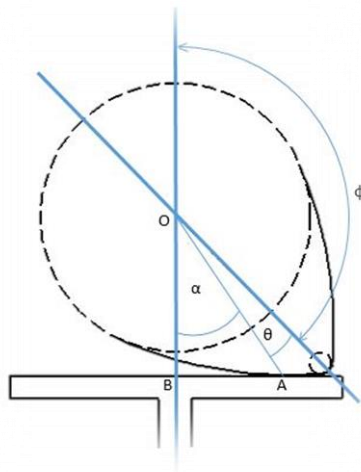


Figure 15, Calculating Valve lift from cam lift

$$OB = OA \cos \alpha$$

Which in terms of the known numbers and values gives the following equation;

$$OB = S \cos[180 - (\theta + \phi)]$$

This however gives 360 values of OB for every value of Cam lobe lift, to then find the actual Value of valve lift for every degree of camshaft rotation the following final equation is used;

$$\text{Valve lift} = OB_{\max} - R - \text{tappet clearance}$$

Cam lift (mm) starting at the nose	Base circle radius (mm)	θ (°)	S (mm)
8.13	15	0	23.13
8.11		1	23.11
8.04		2	23.04
7.95		3	22.95
7.82		4	22.82
7.65		5	22.65
7.46		6	22.46
7.25		7	22.25
7.01		8	22.01
6.76		9	21.76

Figure 16, Calculation of S

Figure 16 above shows the Excel table used to calculate 'S', this involved starting at theta equalling zero which occurs at the nose of the cam lobe (the point with the most lift). The base circle radius was then added on to each of the cam lift values to give the final value of 'S' for each degree of camshaft rotation.

Once the values of S had been calculated a 360 by 360 matrix could be made to calculate the values of OB for every degree of camshaft rotation (θ). This was done by using the previously stated calculation;

$$OB = S \cos[180 - (\theta + \phi)]$$

To do this the angles in degrees had to first be converted to radians to allow the following table, shown in figure 17 below, to be made.

		ϕ	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
S	θ	0	0.017453	0.034907	0.05236	0.069813	0.087267	0.10472	0.122173	0.139626	0.15708	0.174533	0.191986	0.20944	0.226893	0.244346	0.2618	0.279253	
23.13	0	0	-23.13	-23.1265	-23.1159	-23.0983	-23.0737	-23.042	-23.0033	-22.9576	-22.9049	-22.8452	-22.7786	-22.705	-22.6245	-22.5372	-22.4429	-22.3418	-22.234
23.11	1	0.017453	-23.1065	-23.0959	-23.0783	-23.0537	-23.0221	-22.9834	-22.9377	-22.8851	-22.8255	-22.7589	-22.6854	-22.605	-22.5177	-22.4235	-22.3225	-22.2147	-22.1002
23.04	2	0.034907	-23.026	-23.0084	-22.9839	-22.9523	-22.9138	-22.8683	-22.8158	-22.7563	-22.69	-22.6167	-22.5365	-22.4495	-22.3556	-22.2549	-22.1475	-22.0332	-21.9123
22.95	3	0.05236	-22.9185	-22.8941	-22.8627	-22.8243	-22.7789	-22.7266	-22.6674	-22.6013	-22.5283	-22.4485	-22.3618	-22.2683	-22.168	-22.0609	-21.9472	-21.8267	-21.6996
22.82	4	0.069813	-22.7644	-22.7332	-22.695	-22.6499	-22.5979	-22.539	-22.4733	-22.4007	-22.3213	-22.2351	-22.1421	-22.0424	-21.936	-21.8229	-21.7031	-21.5767	-21.4438
22.65	5	0.087267	-22.5638	-22.5259	-22.4812	-22.4296	-22.3711	-22.3059	-22.2338	-22.155	-22.0695	-21.9772	-21.8782	-21.7726	-21.6603	-21.5414	-21.416	-21.284	-21.1456
22.46	6	0.10472	-22.337	-22.2926	-22.2414	-22.1835	-22.1188	-22.0473	-21.9692	-21.8843	-21.7928	-21.6947	-21.5899	-21.4786	-21.3607	-21.2363	-21.1055	-20.9682	-20.8245
22.25	7	0.122173	-22.0841	-22.0335	-21.9761	-21.912	-21.8412	-21.7638	-21.6797	-21.5891	-21.4918	-21.3881	-21.2778	-21.161	-21.0378	-20.9081	-20.7721	-20.6298	-20.4812
22.01	8	0.139626	-21.7958	-21.739	-21.6756	-21.6056	-21.529	-21.4459	-21.3562	-21.26	-21.1574	-21.0482	-20.9327	-20.8108	-20.6826	-20.5481	-20.4073	-20.2603	-20.1071
21.76	9	0.15708	-21.4921	-21.4294	-21.3602	-21.2845	-21.2023	-21.1136	-21.0185	-20.917	-20.8092	-20.695	-20.5745	-20.4477	-20.3147	-20.1755	-20.0302	-19.8787	-19.7212
21.51	10	0.174533	-21.1832	-21.1148	-21.0399	-20.9587	-20.871	-20.777	-20.6767	-20.5701	-20.4572	-20.3381	-20.2128	-20.0813	-19.9437	-19.8	-19.6503	-19.4947	-19.333
21.24	11	0.191986	-20.8498	-20.7758	-20.6956	-20.6091	-20.5162	-20.4172	-20.3119	-20.2004	-20.0828	-19.9591	-19.8292	-19.6934	-19.5515	-19.4037	-19.25	-19.0904	-18.925
21.01	12	0.20944	-20.5509	-20.4715	-20.3859	-20.2941	-20.1961	-20.0919	-19.9817	-19.8653	-19.7429	-19.6145	-19.4801	-19.3398	-19.1936	-19.0415	-18.8836	-18.72	-18.5507
20.74	13	0.226893	-20.2084	-20.1239	-20.0333	-19.9366	-19.8337	-19.7249	-19.61	-19.4892	-19.3624	-19.2298	-19.0912	-18.9469	-18.7968	-18.641	-18.4794	-18.3123	-18.1396
20.51	14	0.244346	-19.9008	-19.8084	-19.7115	-19.6138	-19.5062	-19.3926	-19.2731	-19.1477	-19.0165	-18.8795	-18.7368	-18.5883	-18.4342	-18.2745	-18.1092	-17.9384	-17.7622
20.27	15	0.2618	-19.5793	-19.4848	-19.3843	-19.2779	-19.1656	-19.0475	-18.9237	-18.794	-18.6586	-18.5175	-18.3708	-18.2185	-18.0607	-17.8973	-17.7285	-17.5543	-17.3748
20.03	16	0.279253	-19.2541	-19.1548	-19.0496	-18.9387	-18.822	-18.6996	-18.5715	-18.4377	-18.2983	-18.1533	-18.0028	-17.8468	-17.6854	-17.5186	-17.3465	-17.169	-16.9864

Figure 17, 360 by 360 Matric to calculate Valve Lift

Once the table had been made the values of valve lift could be calculated using the again previously stated calculation;

$$\text{Valve lift} = OB_{\max} - R - \text{tappet clearance}$$

This was done by using the max function in excel within each column to find the maximum value of OB, then as stated by the calculation the base circle radius and tappet clearance were subtracted from this value to give a value of valve lift. Due to the nature of the equation and it not factoring in that there is a clearance at zero lift some of the valve lift figures were calculated to be negative, to combat this an IF statement function was used within excel to state that when a valve lift figure was less than zero, it would in fact equal zero. This gave the following table shown in figure 18 below which gives values of valve lift for every degree of camshaft rotation.

Max	15.03084	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03	15.03
Valve lift	-0.11916	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12	-0.12
actual lift	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

Figure 18, calculated valve lift (mm) from cam lobe lift

Figure 18 above showing the calculated valve lift allowed the following graph of valve lift and cam lift against camshaft angle to be made shown in figure 19 on the following page. The graph is as expected, showing that maximum valve lift is slightly less than maximum cam lift, this is due to the tappet clearance. This graph also allows maximum valve lift (Y) to be calculated by looking at the maximum lift, this shows that Y = 7.98mm.

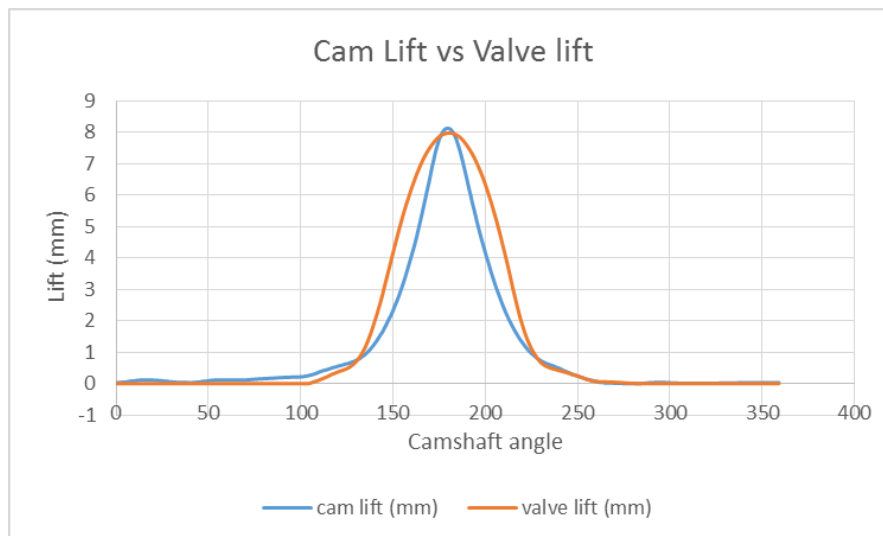


Figure 19, Graph of Cam Lift vs Valve Lift

Calculating the valve lift at every degree of camshaft rotation then allowed the velocity and therefore the acceleration occurring at every degree to be calculated, this was done using the following equations

$$Velocity (mm/^\circ) = L_2 - L_1$$

$$Acceleration (mm/^\circ/^\circ) = V_2 - V_1$$

The equations were used and the following table shown in figure 20 was made in excel to show the velocity and acceleration of the valve at every degree of rotation.

cam angle	cam lift (mm)	valve lift (mm)	velocity (mm/°)	acceleration (mm/°/°)
0	0.03	0		
1	0.03	0	0	
2	0.04	0	0	0
3	0.04	0	0	0
4	0.05	0	0	0
5	0.05	0	0	0
6	0.06	0	0	0
7	0.07	0	0	0
8	0.08	0	0	0
9	0.08	0	0	0
10	0.09	0	0	0

Figure 20, Table of valve lift, velocity and acceleration

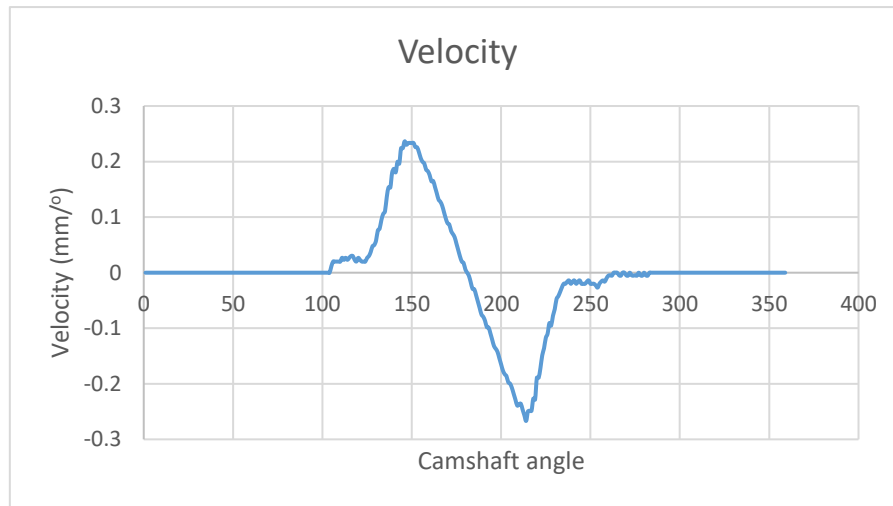


Figure 21, Valve velocity graph

Calculating the Velocity allowed the above graph shown in figure 21 of velocity against camshaft angle to be made, the graph is as expected as it shows the valve having no velocity at maximum lift and peak velocities occur when the valve is opening and closing. The graph shown below in figure 22 was also made from the table in figure 20 to show acceleration against camshaft angle. The graph has the expected shape but is very noisy, this is due to the cam lobe being measured every degree and using a dial gauge that only read to two decimal places. The graph of acceleration against cam lift would have been less noisy if a more accurate gauge had been used and measurements taken every half a degree of rotation.

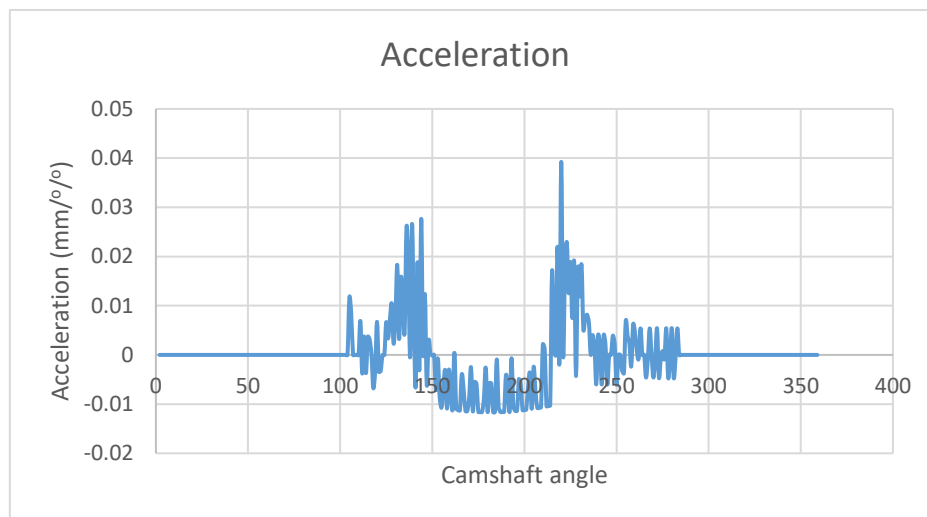


Figure 22, Graph of acceleration against camshaft angle

To Calculate Friction forces and maximum and minimum stress the maximum and minimum accelerations had to be used. Due to the noisy nature of the graph in figure 22, accurate figures could not be taken. This meant that the lift graph had to be split into three sections using polynomials to smooth out any tiny amounts of noise in the lift graph as these would be exaggerated when deriving the acceleration graph.

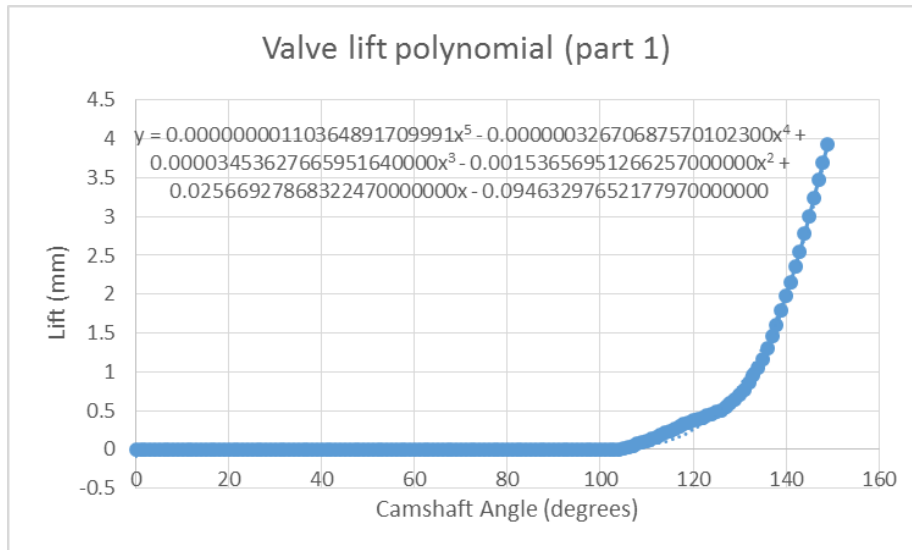


Figure 23, Lift graph and polynomial part 1

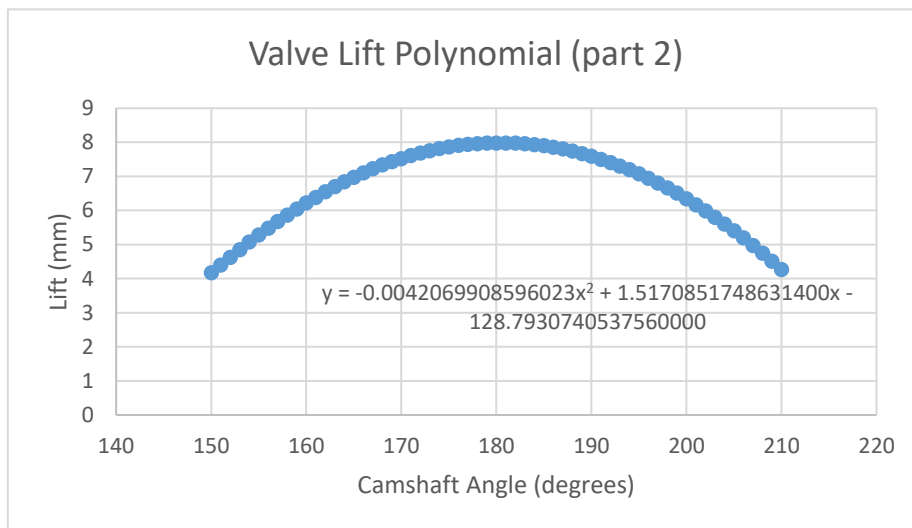


Figure 24, Lift graph and polynomial part 2

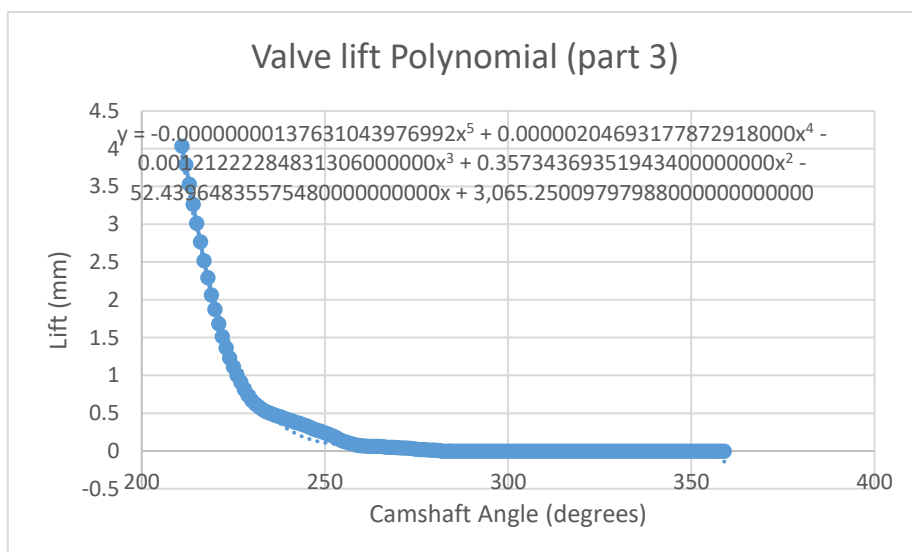


Figure 25, Lift graph and polynomial part 3

As can be seen on the previous page in figures 23, 24 and 25, the graphs have used a polynomial to create the equation of the lift line for all 3 parts of the entire lift graph. Using these equations allowed a new table to be made to derive a more accurate set of data for the acceleration of the valve at every degree of camshaft rotation. This table is shown below in figure 27. As can be seen in figure 27, there are some values of negative valve lift which are due to the nature of the polynomials, this occurred at each end of the valve lift graph which therefore does not affect the minimum and maximum acceleration values which occur in the middle of the valve lift graph.

cam angle	valve lift (mm)	valve lift polynomial	velocity (mm/°)	acceleration (mm/°/°)
0	0	-0.094632977		
1	0	-0.070466057	0.02416692	
2	0	-0.049169599	0.021296458	-0.00287046215216036
3	0	-0.030547982	0.018621617	-0.00267484039239091
4	0	-0.014413159	0.016134823	-0.00248679472189821
5	0	-0.000584529	0.01382863	-0.00230619270281217
6	0	0.011111199	0.011695728	-0.00213290189726277
7	0	0.020840137	0.009728938	-0.00196678986737986
8	0	0.028761351	0.007921214	-0.00180772417529360
9	0	0.035026992	0.006265642	-0.00165557238313366
10	0	0.039782432	0.004755439	-0.00151020205303018

Figure 26, Lift, velocity and acceleration calculated using polynomials

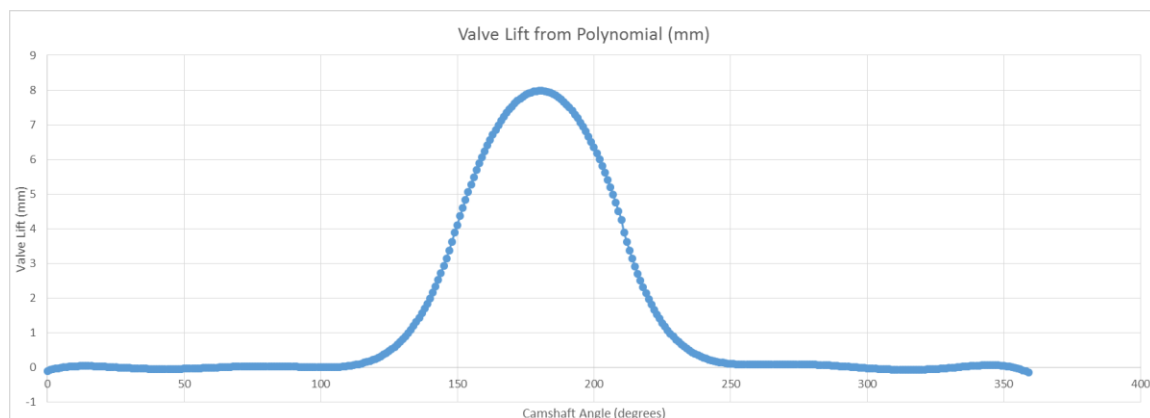


Figure 27, Valve lift graph made from polynomials

The valve lift graph made from three polynomials shown above in figure 26 does not appear any different to the original valve lift graph made from real data, this is as expected and its only use is to delete any minute amounts of noise within the original graph that occurred due to a lack of more accurate equipment being used. By smoothing out the graph as much as possible (through the use of polynomials), a more accurate acceleration graph can be calculated.

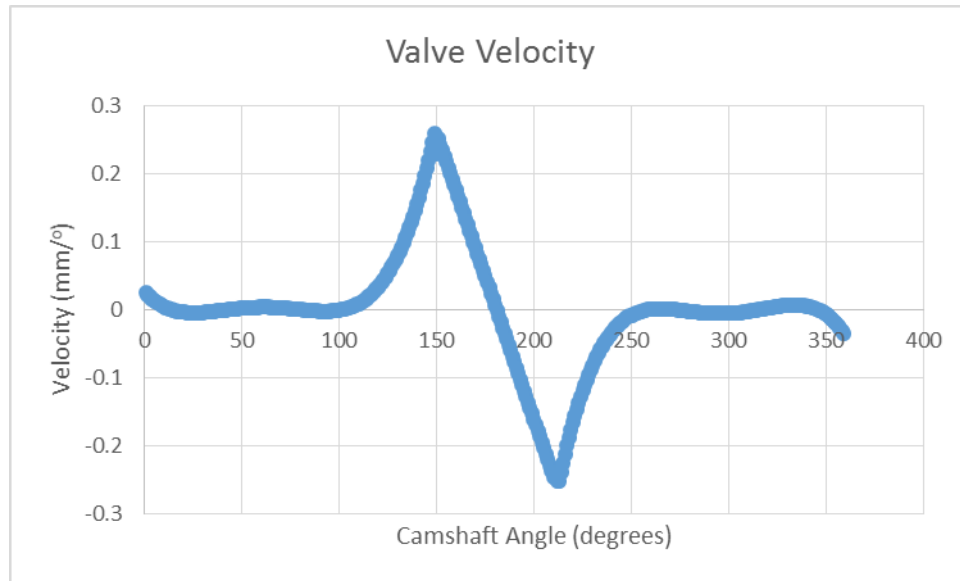


Figure 28, Valve velocity graph (polynomials)

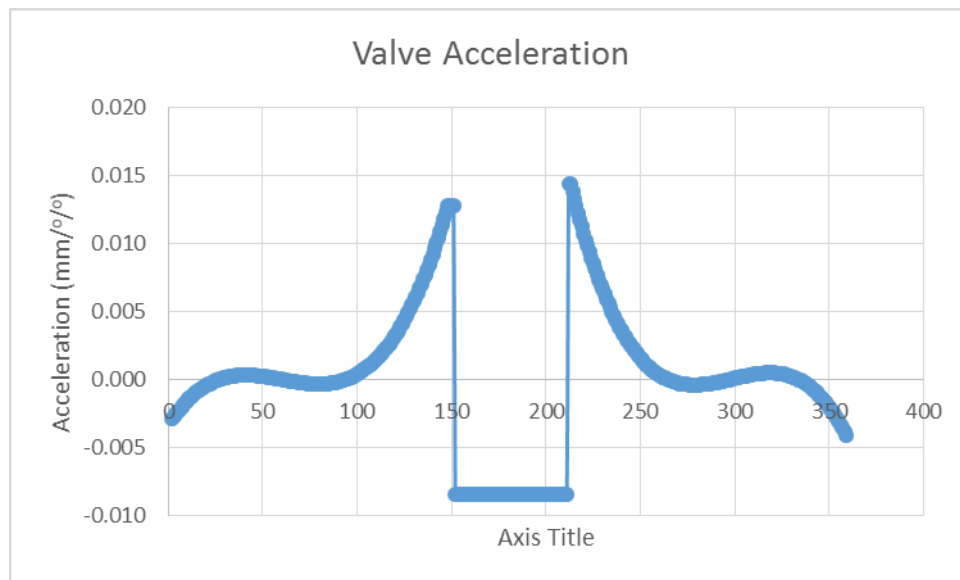


Figure 29, Valve acceleration graph (polynomials)

As can be seen from the above graphs in figure 28 and 29 the noise has been reduced considerably with almost none occurring at all. This allows much more accurate values for the maximum and minimum acceleration figures to be found. From this graph, and the table in figure 27, it can be seen that the maximum acceleration is $0.014435482 \text{ mm/}^\circ/\circ$ and the minimum acceleration is $-0.00841398171928631 \text{ mm/}^\circ/\circ$.

Calculating the maximum negative acceleration allows the contact force between the cam lobe and valve bucket to be calculated. The contact force is calculated using the following equation;

$$F_c = M \cdot a + \text{Spring force}$$

Where M = moving mass in kg and a = acceleration in m/s^2

The spring force can also now be calculated because every component of the equation is known.

$$Y_0 = 10.9\text{mm}, Y = 7.98\text{mm and } K = 15.13\text{N/mm}$$

$$\text{Spring force} = K \cdot (Y + Y_0)$$

$$\text{Spring force} = 15.13 \times (7.98 + 10.9)$$

$$\text{Spring force} = 285.65\text{N}$$

The mass of the moving parts is known to be 74.2 grams, this however needs to be converted to kg and therefore the mass of the moving parts is 0.0742Kg.

The acceleration must then be converted from the $mm/o/o$ that it is currently in to m/s^2 , to do this the following equation must be used;

$$\text{acc in } m/s^2 = \frac{\text{acc in } m/o/o}{1000} \times (6RPM)^2$$

$$\text{acc in } m/s^2 = \frac{\text{acc in } m/o/o \times 36}{1000} \times RPM^2$$

This means that for any given rpm (using the largest negative acceleration, mass and spring force already calculated) the contact force will be as follows

$$F_c = 0.0742 \times \left(\frac{-0.00841398171928631 \times 36}{1000} \times RPM^2 \right) + 285.65$$

The following table in figure 30 shows the resulting contact forces at varying Engine rpm, the previously stated calculation was used but was modified slightly. This is due to the fact the RPM of the camshaft is half of the RPM of the engine, this is because the camshaft rotates once for every two rotations of the crankshaft. To change the camshaft RPM to crankshaft RPM shown in figure 30 the RPM values were halved to make the equation as follows;

$$F_c = 0.0742 \times \left(\frac{-0.00841398171928631 \times 36}{1000} \times \left(\frac{RPM}{2} \right)^2 \right) + 285.65$$

RPM	F _c (N)
0	285.7185
50	285.7044
100	285.6623
150	285.5921
200	285.4937
250	285.3673
300	285.2128
350	285.0302
400	284.8195
450	284.5807
500	284.3138

Figure 31, Contact force at varying RPM

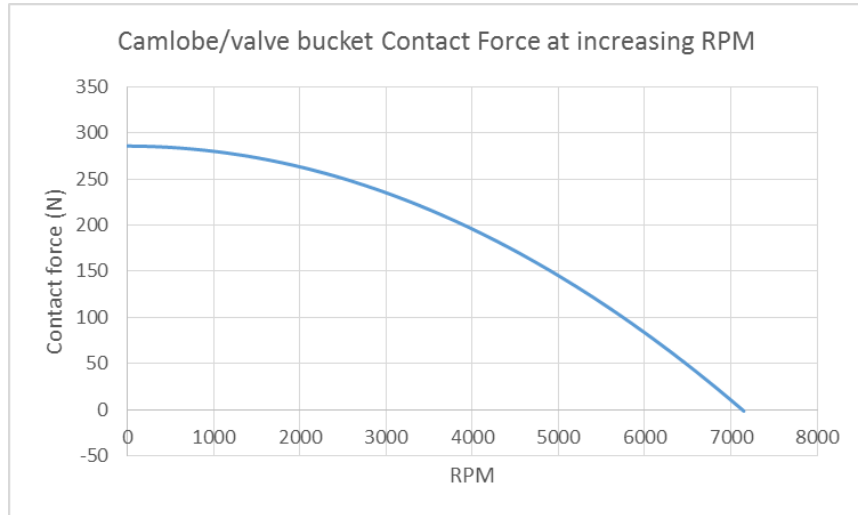


Figure 30, Graph of contact forces with increasing engine RPM

The Graph in figure 30 above shows how the contact force decreases with rpm. This confirms the research carried out that states valve train friction reduces with rpm due to higher spring forces at low rpm which changes due to the inertia of the moving parts taking over at higher rpm therefore lowering the contact force. The Ford 1.6GDTI EcoBoost engine also has a rev limiter which limits the speed of the engine to 7000rpm, this further supports the graph as the contact force reaches zero (when valve float occurs) after 7000rpm. This supports the graph as a rev limiter would not be placed after valve float occurs as this would decrease the performance of the engine.

To find the exact RPM at which valve float occurs the previously stated calculation shown as follows can be used;

$$F_c = 0.0742 \times \left(\frac{-0.00841398171928631 \times 36}{1000} \times \left(\frac{RPM}{2} \right)^2 \right) + 285.65$$

It is also known that when valve float occurs the contact force equals zero, which is represented in the following equation;

$$M \cdot a + \text{Spring force} = 0$$

This means that the first equation can be rearranged as follows;

$$\text{Valve float RPM} = 2 \times \sqrt{\frac{\text{spring force} \times 1000}{-mass \times acc \times 36}}$$

Using the previously found values this changes to;

$$\text{Valve float RPM} = 2 \times \sqrt{\frac{285.65 \times 1000}{-0.0742 \times -0.00841398171928631 \times 36}}$$

$$\text{Valve float RPM} = 7130$$

This shows that valve float in the Ford 1.6 GDTI EcoBoost engine occurs at 7130RPM

To calculate the friction force occurring throughout the rpm range the coefficient of friction must be used within the following equation;

$$F_{friction} = \mu \times N$$

Where μ = coefficient of friction and N = the normal force

The coefficient of friction between lubricated cast iron (camshaft) and mild steel (valve bucket) is roughly 0.17 (Engineering Tool Box, 2017), using this and the normal forces at each rpm already calculated, a table for friction forces has been made shown in figure 33 below.

RPM	Fc (N)	friction (N)
0	285.7185	48.572141
50	285.7044	48.569753
100	285.6623	48.562589
150	285.5921	48.550649
200	285.4937	48.533933
250	285.3673	48.51244
300	285.2128	48.486172
350	285.0302	48.455128
400	284.8195	48.419308
450	284.5807	48.378712
500	284.3138	48.333339

Figure 33, Section of Friction vs RPM table

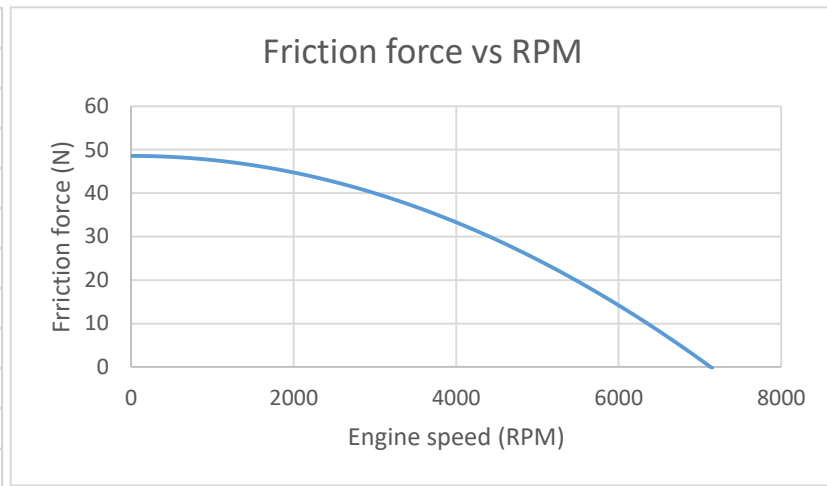


Figure 32, Graph of friction force vs RPM

As can be seen above in figure 32 and 33 the friction forces for increasing rpm have been calculated with the maximum friction force occurring at zero rpm and equalling 48.572N at each inlet valve. The graph shown above in figure 32 shows that the friction force also decreases at the same rate as the contact force, this is as expected and confirms the previous research that states valve train friction reduces with increased rpm. Calculating the friction occurring at each inlet valve as shown above gives a baseline friction value to work from. The chapters following will refer back to these friction values to compare alternative valve trains to OEM, this will allow any improvements made to be clearly seen.

Chapter 4: Valve modelling

4.1 Introduction

This section of the thesis will show the steps taken to build a 3D model of the Ford 1.6 EcoBoost inlet valve. Once this process has been carried out and an accurate model of the valve has been made, an FEA will be carried out on the valve using the accelerations previously calculated. This process will allow values of stress acting on the valve to be obtained and will also show the areas most affected by stress. Finding the values of stress will allow a fatigue safety factor to be calculated which will then be adhered to when designing a new valve with new dimensions or materials. The aim of this part of the paper is to decrease the weight of the valve to therefore decrease the amount of friction that occurs between the valve and cam lobe, by decreasing friction in this area, overall engine efficiency will increase.

4.2 Building the Valve 3D model

To build an accurate model of the valve, measurements alone would not suffice. To achieve maximum accuracy a 3D scan of the physical valve was undertaken along with measurements of the valve also being carried out. From the 3D scan two files were obtained, a mesh file and a cloud file of the valve.



Figure 34, Mesh model of valve from 3D scan

The above figure 34 shows the initial mesh file opened in SolidWorks, as can be seen the model is fairly noisy and the model in this form cannot be used as a part in a finite element analysis. From here the curve wizard tool in SolidWorks was used to give a 2 dimensional outline of the curves of the valve as shown on the following page.

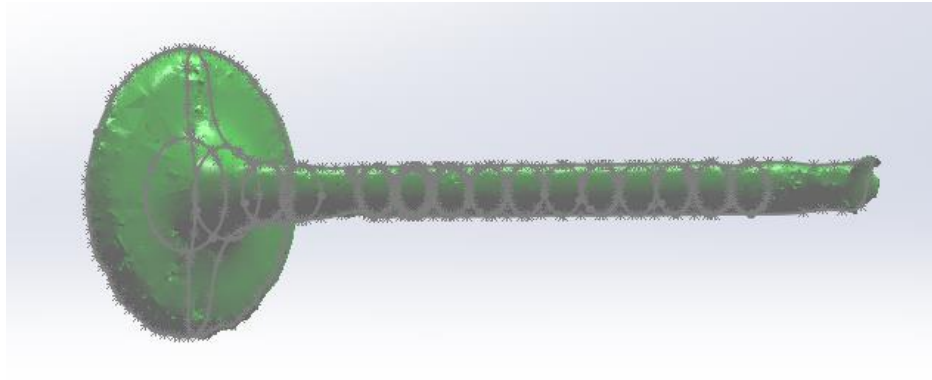


Figure 35, Curve wizard tool used to 2D model valve contours

Figure 35 above shows the 2D contours of the valve made using the curve wizard tool within SolidWorks. By doing this and using physical measurements of the valve previously taken, an accurate 3D model of the valve can be made. After using the curve wizard tool, a 2D sketch was made using the 2D contour lines and the measurements taken to alleviate any errors that may have occurred due to the noisy nature of the initial mesh model. This process is shown below in figure 36 below.

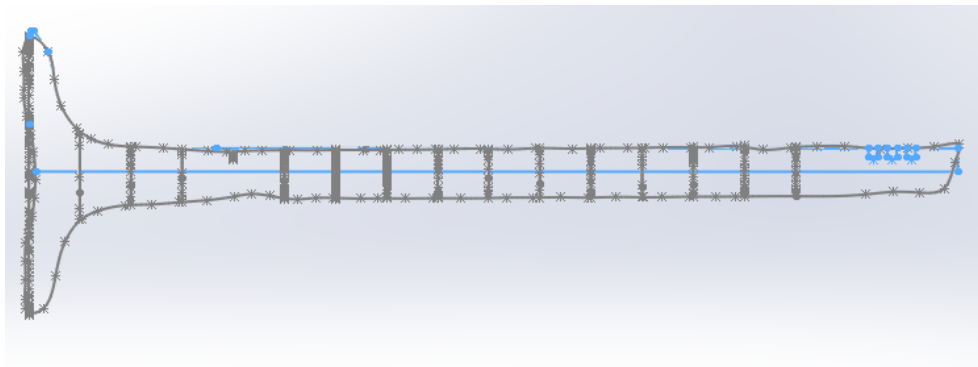


Figure 36, 2D sketch of the valve

As can be seen in figure 36 above only half of the valve has been sketched in the 2D model, this has been done using a combination of the contour lines made using the curve wizard and the measurements previously taken from the physical valve. Only half of the valve was sketched so that the 'Revolved Boss/Base' tool within SolidWorks could be used. This particular tool was used so that the valve would be perfectly symmetrical around the central axis. This allowed the following final model to be made shown on the following page.

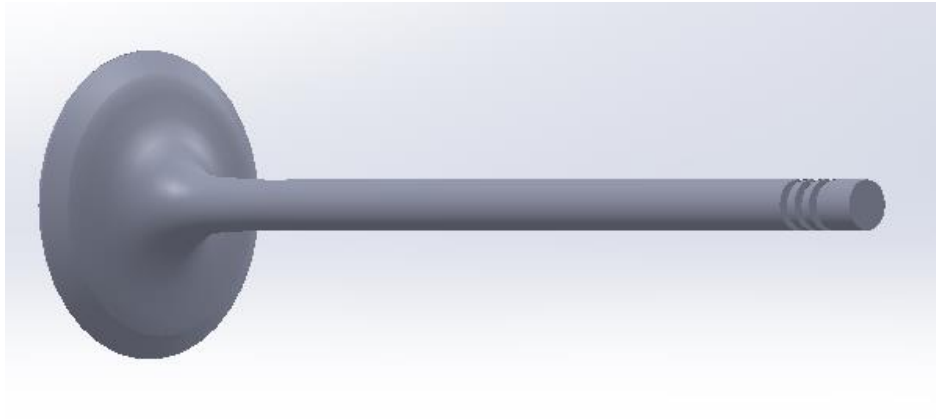


Figure 37, Ford 1.6GDTI EcoBoost inlet valve

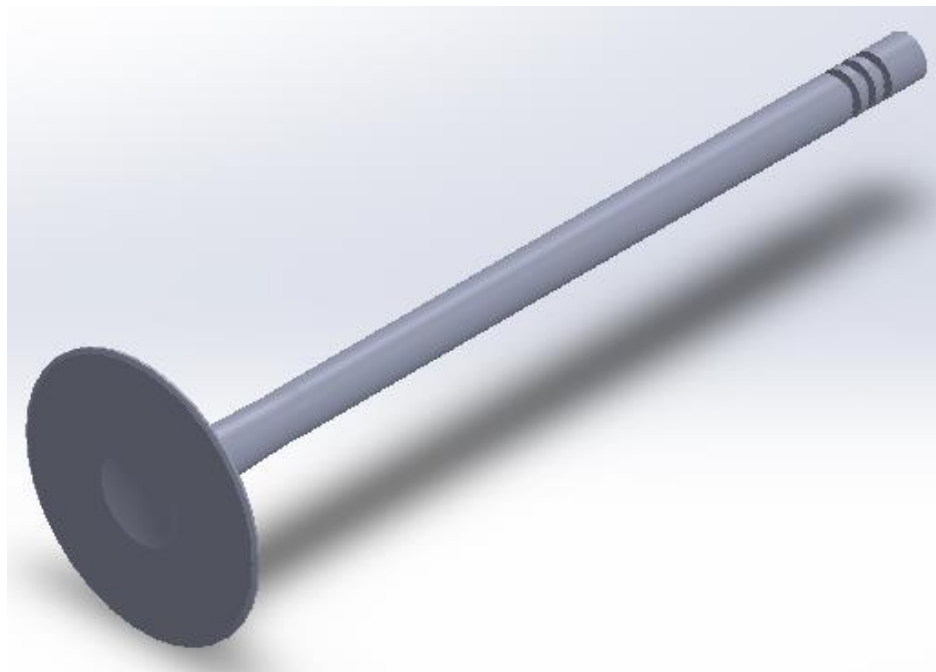


Figure 38, Ford 1.6GDTI EcoBoost inlet valve

The figures above show the 3D model of the inlet valve used within the 1.6 Ford EcoBoost engine, this model is an accurate representation of the physical valve and when combined with the calculated accelerations, will provide accurate values of stress that act on the valve.

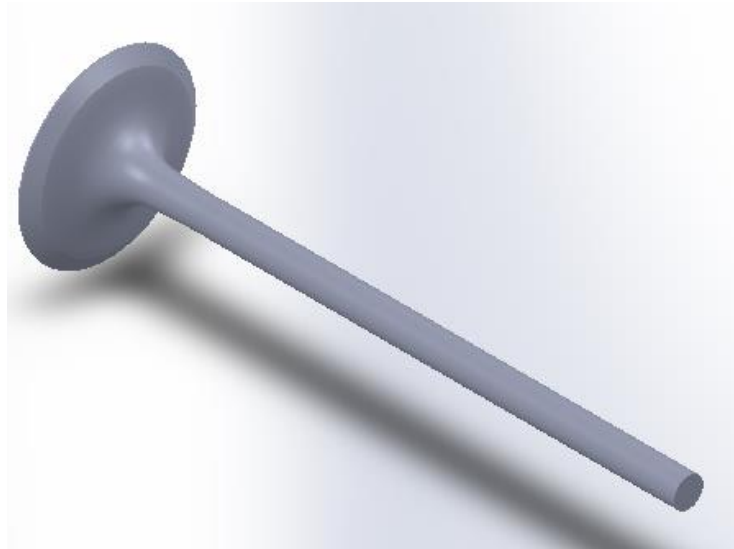


Figure 39, Valve model with collet grooves removed

As shown in figure 39 above the 3 grooves that the collets sit in have been removed, this has been done so that the valve doesn't experience extreme values of stress. If the grooves had not been removed the FEA would show extreme values of stress. This is due to the collets not being in place to fill the gaps resulting in a large force over a much smaller area. This would have given inaccurate results for stress and would therefore have given an inaccurate fatigue safety factor for the valve.

Chapter 5: Finite Element Analysis

5.1 Introduction

This chapter of the paper will perform various Finite element analyses on the valve. The aim is to first find the stresses acting on the OEM valve to calculate the fatigue safety factor currently being used, by doing this any changes in design and/or material can be done so adhering to the current safety factor. The purpose of this chapter is to find improvements that can be made to the current design of the valve to reduce the friction in the engine with the goal of increasing engine efficiency, this can potentially be done through different techniques such as hollow valve stems, sodium filled hollow valve stems and changes in valve material and geometry.

The accelerations of the valve at maximum rpm are the largest both minimum and maximum accelerations the valve will experience. The maximum engine speed of the ford 1.6GDTI EcoBoost engine is 7000RPM and the maximum and minimum accelerations in mm/s² are 0.014435482 and -0.00841398171928631 respectively. These must be converted to m/s² using the following equation;

$$acc \text{ in } m/s^2 = \frac{acc \text{ in } mm/s^2 \times 36}{1000} \times RPM^2$$

$$max \text{ acc in } m/s^2 = \frac{0.014435482 \times 36}{1000} \times 7000^2$$

$$max \text{ acc in } m/s^2 = 25464.19$$

$$min \text{ acc in } m/s^2 = \frac{-0.00841398171928631 \times 36}{1000} \times 7000^2$$

$$min \text{ acc in } m/s^2 = -14842.26$$

These accelerations are therefore the values that will be used within the simulations carried out.

5.2 Standard EV8 21-4N Stainless Steel

EV8 21-4N steel is a material used in OEM valve applications, this steel is used because of its ability to maintain its material properties at high temperatures and stresses. EV8 steel is an SAE classified valve steel and its properties are shown below in figures 40 and 41.

Yield $R_{p0.2}$ (MPa)	Tensile R_m (MPa)	Impact KV/Ku (J)
191 ($\geq$)	332 ($\geq$)	12

Figure 40, EV8 21-4N properties (Steel Grades, 2016)

Temperature (°C)	Modulus of elasticity (GPa)	Density (kg/dm³)	Poisson's coefficient, ν
14	-	-	-
758	935	-	-
452	-	333	441

Figure 41, EV8 21-4N properties (Steel Grades, 2016)

Ford currently uses a DLC coated valve, this allows the valve to be lighter due to the surface of the valve being under a compressive stress, DLC coatings put a residual surface compressive stress on the part of around 900N/mm² (Wang, J.S., et al., 1998). EV8 21-4N is often used for both inlet and exhaust valves but is much better suited for exhaust valves because of its ability to resist heat and corrosion. This gives a greater scope for finding a more suitable inlet valve material that does not need such good heat resistance properties due to it being cooled by fuel vapours.

Figure 42 below shows a 1.5mm curvature based mesh applied to the valve within SolidWorks simulation, this mesh has been chosen because the valve geometry is widely curved and this mesh type will allow more accurate values of stress to be found, the size of mesh has been chosen to allow at least three elements through the diameter of the valve. The figure also shows the acceleration in the positive direction and the previously calculated maximum acceleration of 25464.19m/s² at 7000RPM has been used.

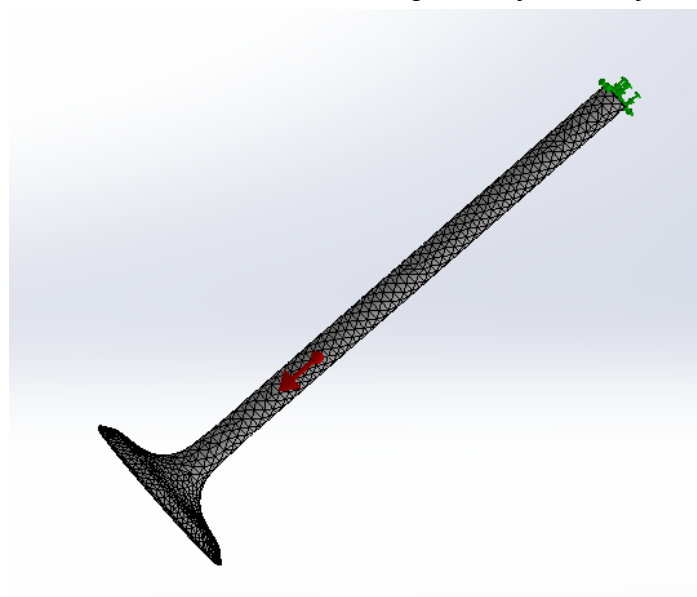


Figure 42, 1.5mm Curvature based mesh applied in SolidWorks simulation

Again shown in figure 42 on the previous page the top face of the valve has been fixed in 6 degrees of freedom so that it cannot move. This allows the acceleration, placed as a gravitational field, to 'pull' on the valve and give accurate values of maximum and minimum stress.

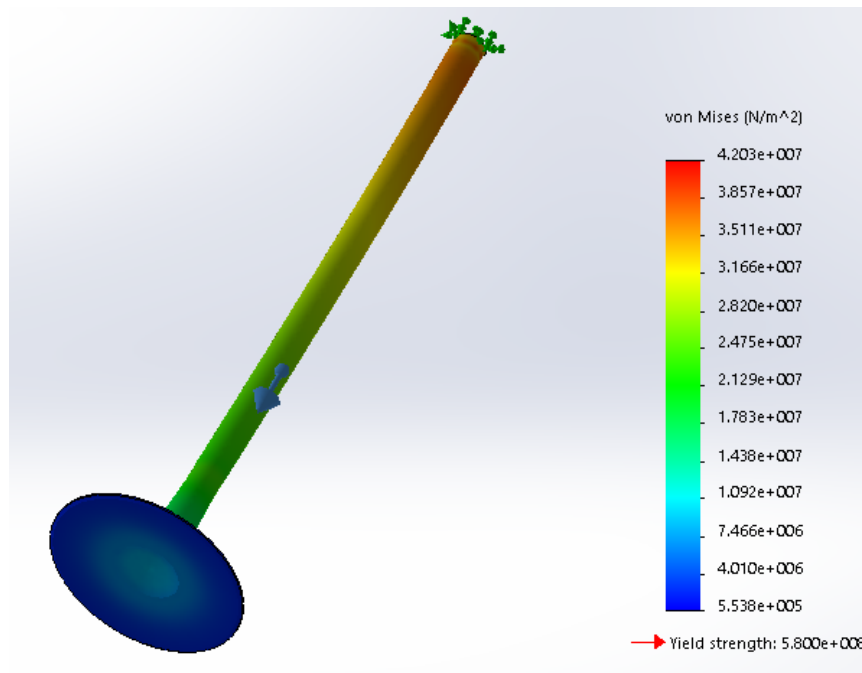


Figure 43, Maximum stress caused by acceleration at max RPM

Figure 43 above shows the results of running the study with the maximum positive acceleration acting on the valve. As can be seen from the picture it shows that the highest values of stress occur within the valve stem, this is because stress is equal to force divided by area and the valve stem has the smallest area therefore having the highest stress. As can be seen the highest positive stress value experienced within in the valve is 42.03MPa.

Figure 44 to the right shows the results of running the study with the maximum negative acceleration, this was simulated by reversing the direction of the acceleration and using the previously calculated negative acceleration value of 14842.26m/s² at 7000RPM. Figure 44 shows that the largest values of stress again occur within the stem of the valve, with the largest value of stress being 24.50MPa.

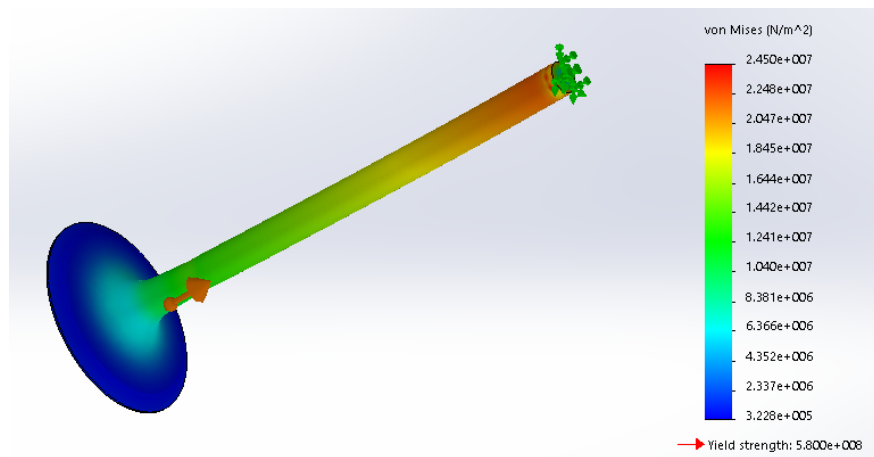


Figure 44, Largest minimum stress caused by max RPM

These simulations have showed that the maximum stress occurring within the valve is 42.03MPa and the minimum stress occurring is -24.50MPa. The surface of the valve in SolidWorks however does not have a surface compressive stress of 900N/mm² (Wang, J.S., et al., 1998) this then needs to be taken away from the stress figures found. These values will allow a Goodman diagram to be made and therefore a fatigue safety factor to be calculated for the valve. To create a Goodman diagram and calculate the fatigue safety factor the mean stress and range of stress must be calculated, this is shown as follows;

$$\sigma_{max} = 42.03 - 900N/mm^2$$

$$\sigma_{max} = -857.97N/mm^2$$

$$\sigma_{min} = -24.50 - 900N/mm^2$$

$$\sigma_{min} = -924.50N/mm^2$$

$$\sigma_{mean} = 0.5 \times (\sigma_{max} + \sigma_{min})$$

$$\sigma_{mean} = 0.5 \times (-857.97 + (-924.50))N/mm^2$$

$$\sigma_{mean} = -891.24N/mm^2$$

$$\sigma_{range} = \sigma_{max} - \sigma_{min}$$

$$\sigma_{range} = -857.97 - (-924.50)N/mm^2$$

$$\sigma_{range} = 66.53N/mm^2$$

Once the mean stress and range of stress have been calculated the endurance limit must be calculated, this is known as being approximately 0.45 of the UTS for a polished surface.

$$UTS = 332N/mm^2$$

$$polished\ endurance\ limit = 0.45 \times UTS$$

$$polished\ endurance\ limit = 0.45 \times 332N/mm^2$$

$$polished\ endurance\ limit = 149.4N/mm^2$$

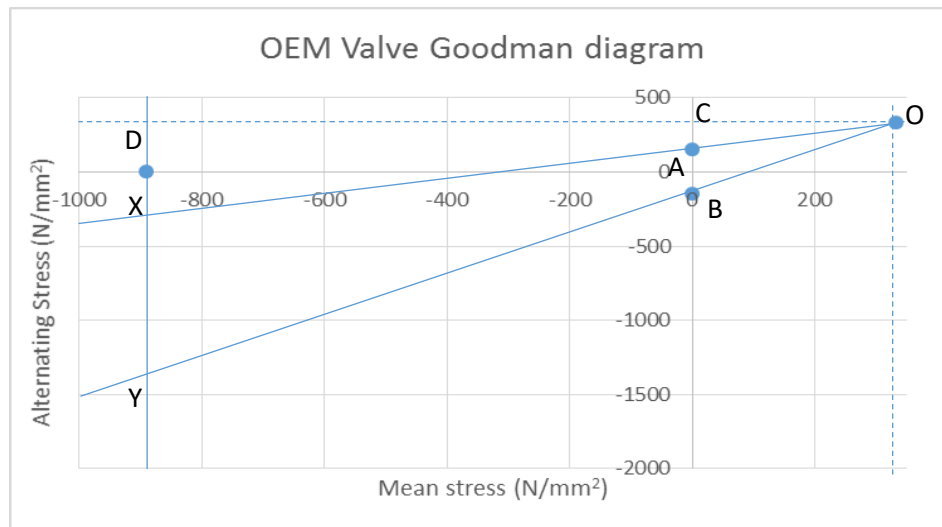


Figure 45, OEM valve Goodman diagram

XY in the Goodman diagram above is the allowable range at a mean stress of -891.24 N/mm^2

OXY and OAB are similar triangles, this means the following is true;

$$\frac{XY}{AB} = \frac{OX}{OA} \quad \text{Where } AB = 2 \times \text{Endurance limit, this can then be rearranged to show } XY$$

$$XY = AB \times \frac{OX}{OA}$$

OAC and OXD are also similar triangles, which again means the following is true;

$$\frac{OX}{OA} = \frac{OD}{OC} = \frac{UTS - \sigma_{mean}}{UTS}$$

$$XY = AB \times \frac{UTS - \sigma_{mean}}{UTS}$$

This can then be used to calculate the allowable range for the valve.

$$XY = 2 \times 149.4 \times \left(\frac{332 - (-891.24)}{332} \right) \text{ N/mm}^2$$

$$\text{Allowable range} = 1100.92 \text{ N/mm}^2$$

The allowable range and actual range then allow the fatigue safety factor to be calculated using the following calculation;

$$FSF = \frac{\text{allowable range}}{\text{actual range}}$$

$$FSF = \frac{1100.92}{66.53}$$

$$FSF = 16.55$$

Calculating the fatigue safety factor that Ford have used allows a standard to be used across all valve variations that will be tested within this paper. Therefore any valve design to be made will adhere to a fatigue safety factor of around 16.55 or higher to avoid a short fatigue life of the valve iteration.

5.3 Ti-6AL-4V Titanium Alloy

Titanium is often used in high performance applications for the inlet valve, this is due to it being lighter than steel alloys which allows an engine to rev higher which in turn can give an increase in power. In this application the lighter titanium valve will decrease friction due to the contact force between the cam lobe and valve bucket decreasing. The contact force would decrease because $Force = mass \times acceleration$, by decreasing the mass the force will also therefore decrease. The properties of the Ti-6AL-4V alloy are shown below in figure 46.

Tensile Strength, Ultimate	<u>950 MPa</u>
Tensile Strength, Yield	<u>880 MPa</u>
Elongation at Break	<u>14 %</u>
Reduction of Area	<u>36 %</u>
Modulus of Elasticity	<u>113.8 GPa</u>

Figure 46, Ti-6AL-4V Titanium alloy properties (ASM, 2017)

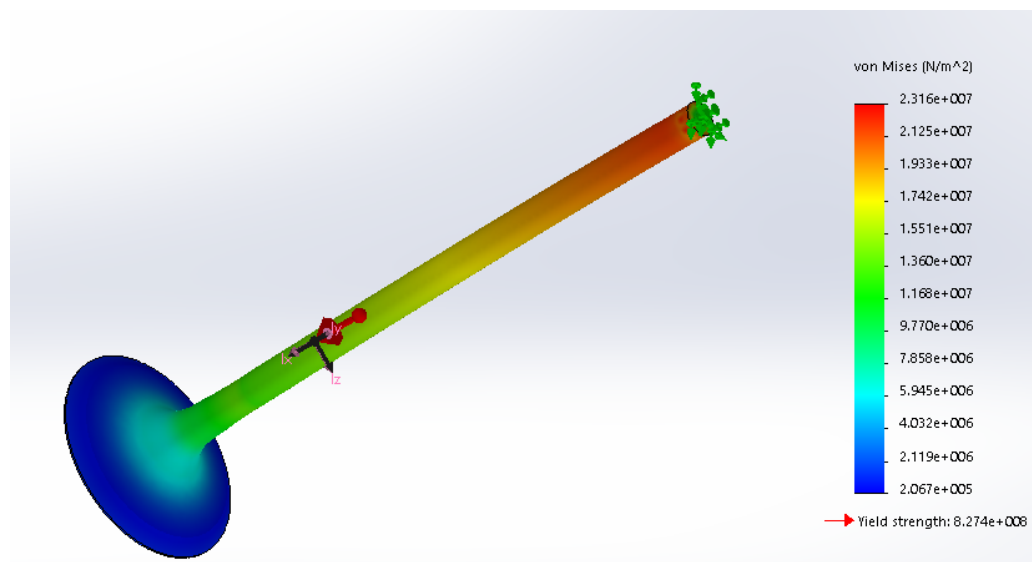


Figure 47, Ti-6AL-4V valve maximum positive stress at max RPM

The Titanium valve weighs much less than the OEM counterpart, 16.67 grams compared to 31.7 grams. This means that the valve experiences lower values of stress due to the force over each area being smaller. This is shown in the results from the simulation shown in figure 47 on the previous page, the maximum stress experienced by the valve is 23.16MPa, this is almost half the maximum stress experienced by the OEM valve.

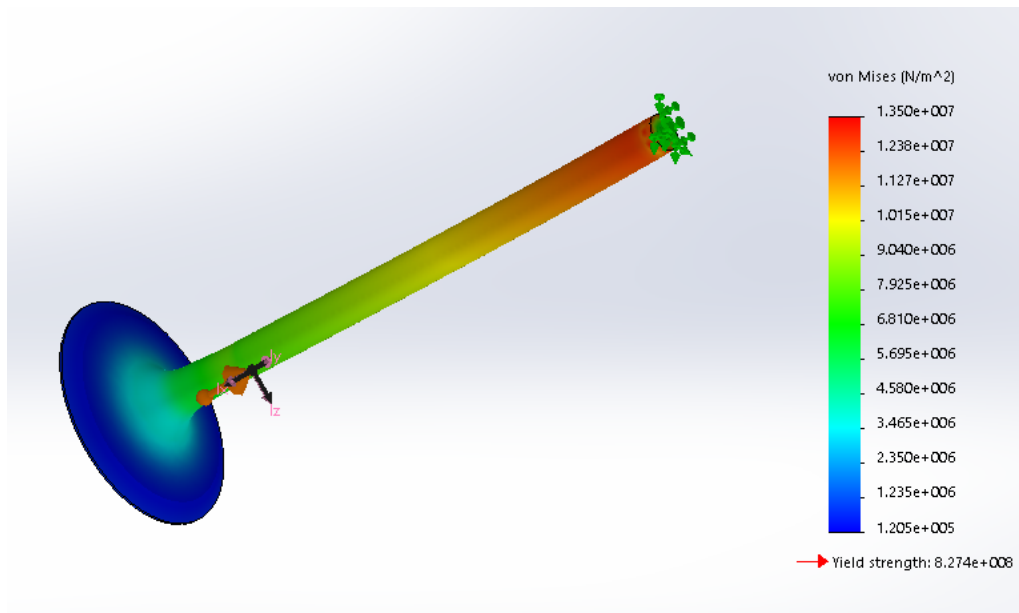


Figure 48, Ti-6AL-4V maximum negative stress at max RPM

Figure 48 above shows the results from the largest negative acceleration, again the stress experienced by the valve has been almost halved with the valve experiencing a maximum negative stress of 13.50MPa. To find out whether this valve would be suitable for production its fatigue safety factor must be calculated and be equal to roughly 16.55, again a Goodman diagram will be made for the valve to aid the FSF calculation.

$$\sigma_{max} = 23.15 \text{ N/mm}^2$$

$$\sigma_{min} = -13.50 \text{ N/mm}^2$$

$$\sigma_{mean} = 0.5 \times (\sigma_{max} + \sigma_{min})$$

$$\sigma_{mean} = 0.5 \times (23.15 + (-13.50)) \text{ N/mm}^2$$

$$\sigma_{mean} = 4.83 \text{ N/mm}^2$$

$$\sigma_{range} = \sigma_{max} - \sigma_{min}$$

$$\sigma_{range} = 23.15 - (-13.50) \text{ N/mm}^2$$

$$\sigma_{range} = 36.65 \text{ N/mm}^2$$

Again because the mean stress and range of stress have been calculated the endurance limit can be calculated.

$$UTS = 950 \text{ N/mm}^2$$

$$\text{polished endurance limit} = 0.45 \times UTS$$

$$\text{polished endurance limit} = 0.45 \times 950 \text{ N/mm}^2$$

$$\text{polished endurance limit} = 427.5 \text{ N/mm}^2$$

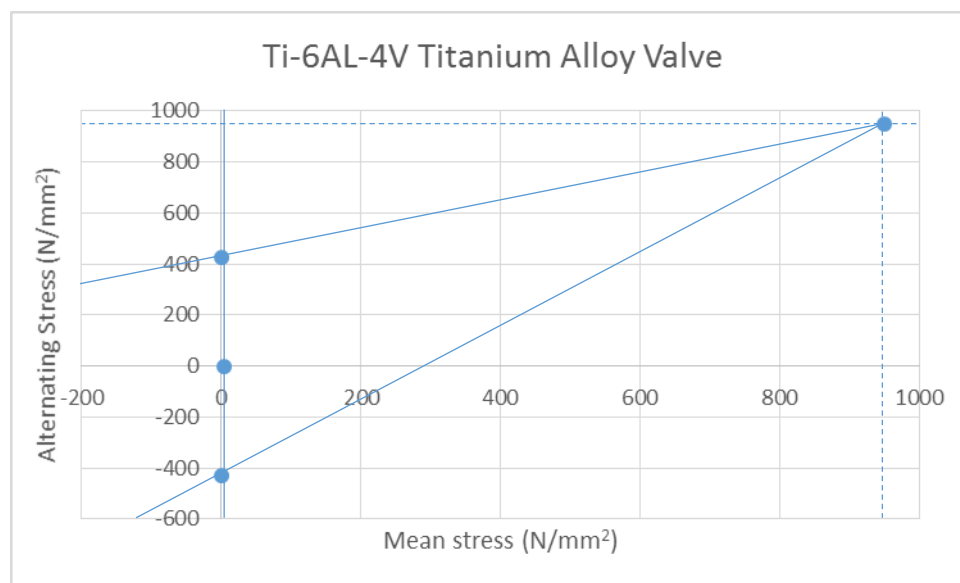


Figure 49, Ti-6AL-4V alloy valve Goodman diagram

$$\text{Allowable range} = 2 \times 427.5 \times \left(\frac{950 - 4.83}{950} \right) = 850.65 \text{ N/mm}^2$$

The allowable range and actual range then allow the fatigue safety factor to be calculated using the following calculation;

$$FSF = \frac{\text{allowable range}}{\text{actual range}}$$

$$FSF = \frac{850.65}{36.65}$$

$$FSF = 23.21$$

This shows that the titanium Valve has a fatigue safety factor of 23.21 which is much larger than the FSF of the OEM valve. This shows that the Titanium alloy would be safe to use in the production engine and would have a large fatigue life.

To make sure the Ti-6AL-4V alloy valve would not break under fatigue a fatigue simulation within SolidWorks will be carried out. This will help to confirm that the valve can withstand 10^7 cycles without failing due to fatigue. The simulations were run as zero based and the S-N curve was derived from the materials elastic modulus. The results of the fatigue testing are shown below in figure 50.

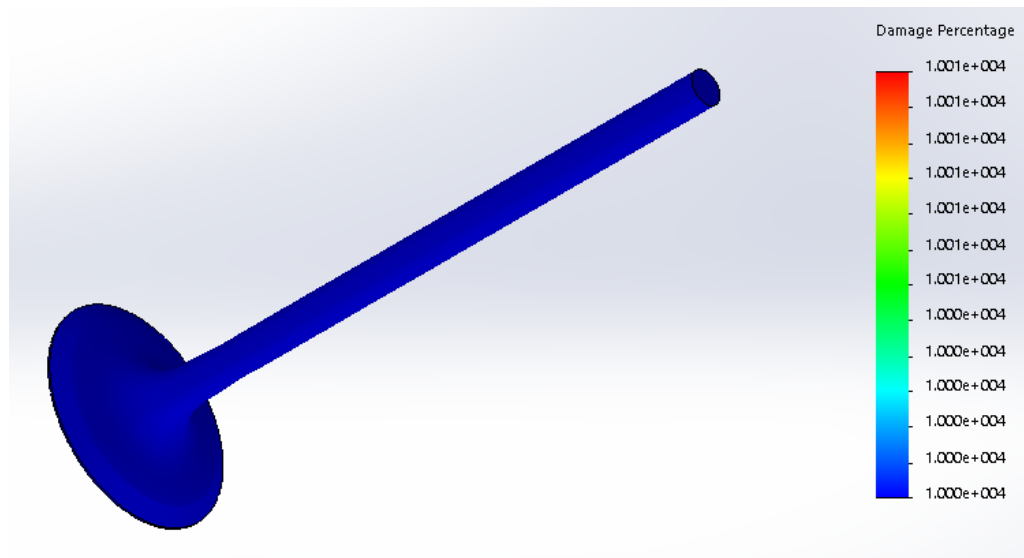


Figure 50, Fatigue simulation for 10^7 cycles under maximum positive acceleration

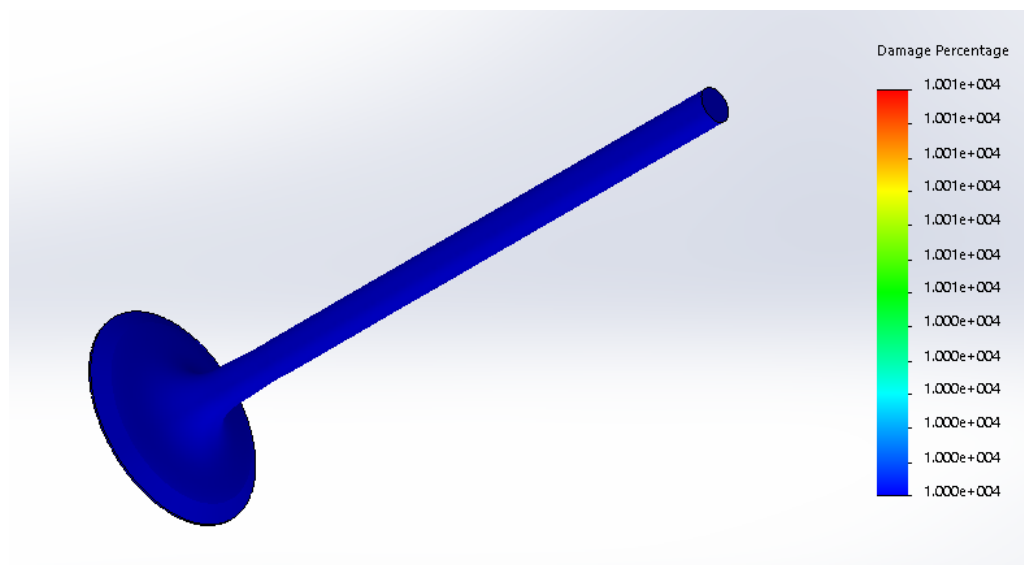


Figure 51, Fatigue simulation for 10^7 cycles under maximum negative acceleration

As can be seen from both figures 50 and 51, the simulations resulted in no damage percentage to the valve, this shows that the valve has a theoretical infinite fatigue life and shouldn't break in the lifetime of the engine. This means the Ti-6AL-4V alloy would be a direct replacement for the OEM valve and should be equally reliable.

5.4 Hollow Stem Ti-6AL-4V

To further decrease weight in the valve a hollow stem can be introduced, this allows the valve to keep the same geometry and size whilst decreasing in volume and therefore mass. Figure 52 below shows the sketch of the Ford 1.6GDTI EcoBoost inlet valve with a hollow stem introduced. The hollow cylinder within the stem measures 40mm long with a 2mm diameter and is situated 2mm beneath the top of the valve.

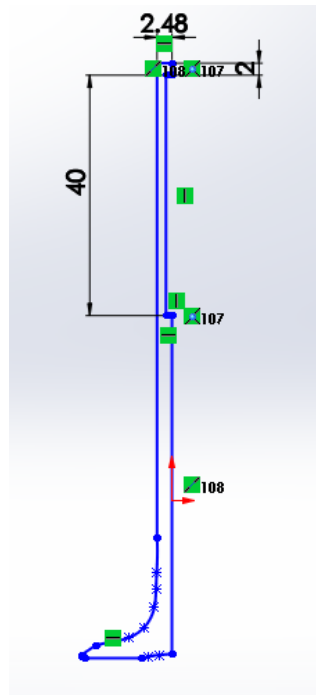


Figure 52, Inlet valve sketch with hollow stem

The sketch was then made into a part by revolving it through 360 degrees to make the final hollow stem valve shown below in figure 53. As can be seen the hollow valve stem looks identical to the normal solid stem valve, the main difference is found in the weight. With the hollow stem valve weighing 16.12 grams compared to 16.67 grams for the solid stem valve.

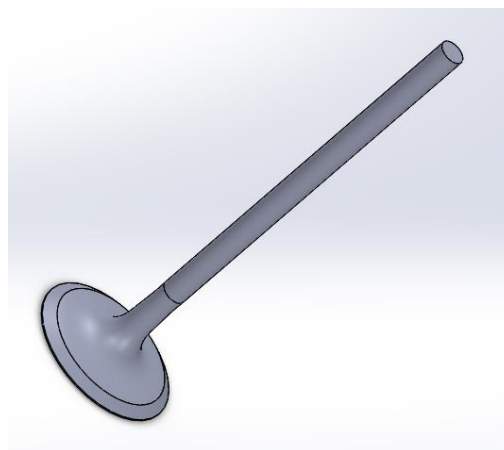


Figure 53, Hollow stem valve

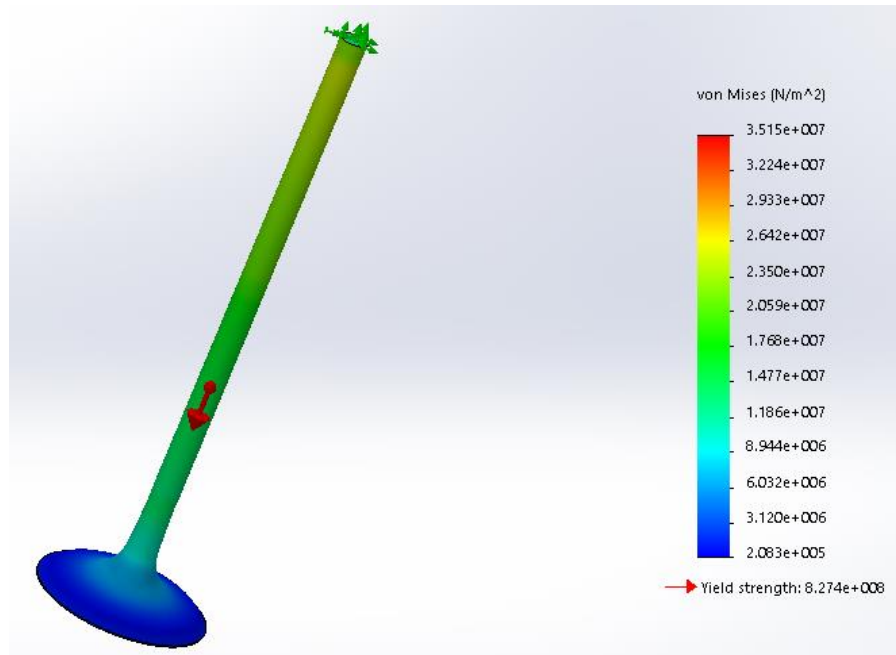


Figure 54, Stress simulation for maximum positive acceleration

Figure 54 above shows the simulation carried out on the hollow stem valve for maximum acceleration, it shows a maximum stress occurring of 35.15MPa which is larger than the maximum stress, 23.16MPa, occurring on the solid stem valve. This is expected due to the stress being equal to the force divided by the area, this means the smaller area of the hollow stem will therefore increase the stress. The stress occurring on the hollow stem valve was then simulated for the maximum negative acceleration, this was again done by reversing the direction of the gravitational field. As can be seen in figure 55 below the maximum stress occurring on the valve is 20.49MPa, this as expected is again larger than the stress experienced by the solid stem valve.

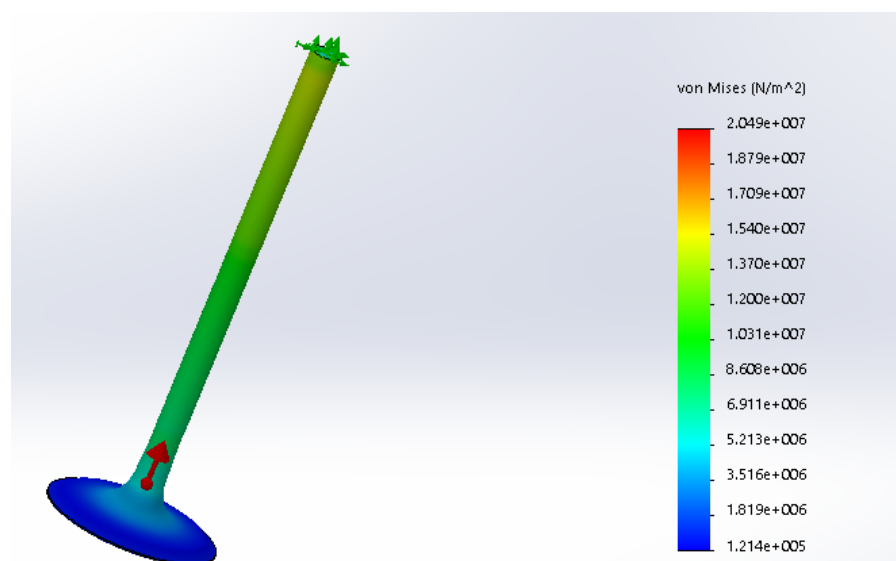


Figure 55, Stress simulation for maximum negative acceleration

A fatigue safety factor will now be calculated for the hollow stem valve to see if it meets the standard of 16.55 set by the OEM Ford valve.

$$\sigma_{max} = 35.15 \text{ N/mm}^2$$

$$\sigma_{min} = -20.49 \text{ N/mm}^2$$

$$\sigma_{mean} = 0.5 \times (35.15 + (-20.49)) \text{ N/mm}^2$$

$$\sigma_{mean} = 7.33 \text{ N/mm}^2$$

$$\sigma_{range} = 35.15 - (-20.49) \text{ N/mm}^2$$

$$\sigma_{range} = 55.64 \text{ N/mm}^2$$

$$UTS = 950 \text{ N/mm}^2$$

$$\text{polished endurance limit} = 0.45 \times 950 = 427.5 \text{ N/mm}^2$$

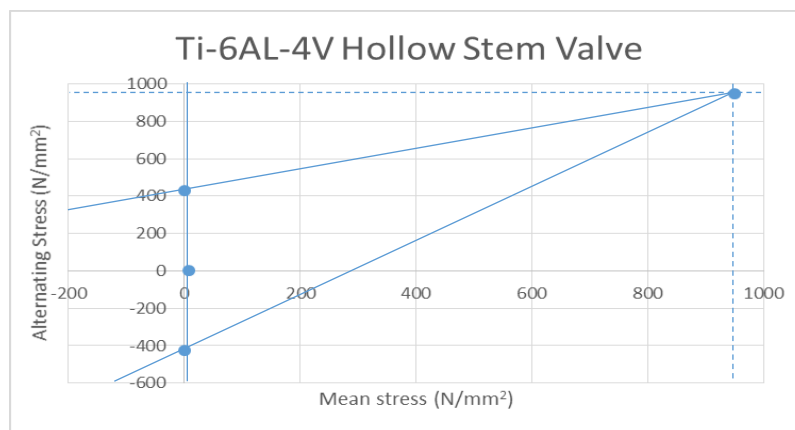


Figure 56, Ti-6AL-4V hollow stem valve Goodman diagram

$$\text{Allowable range} = 2 \times 427.5 \times \left(\frac{950 - 7.33}{950} \right) = 848.40 \text{ N/mm}^2$$

The allowable range and actual range then allow the fatigue safety factor to be calculated using the following calculation;

$$FSF = \frac{848.40}{55.64}$$

$$FSF = 15.25$$

This shows the hollow stem valve has a lower FSF than the standard set by Ford's OEM valve, this means the valve would be unsuitable to fit in a production car as its Fatigue life is not as long as Ford dictates it should be. The hollow stem valve would be suitable for a performance application but not for a production application where efficiency is to be increased and reliability also needs to be kept high.

Chapter Six: Alternate Valve train Friction Calculations

6.1 Introduction

This section of the paper will further look into alternate options for coatings and materials to be used within the valve train with the intention of decreasing friction and increasing efficiency. This chapter will mathematically calculate the friction force that occurs when using these different materials and coatings and will compare them to the stock values obtained. Furthermore this chapter will combine the different options mathematically to find the maximum friction decreases possible with the methods and options used.

6.2 DLC Coating

DLC stands for diamond like carbon and is a coating that can be applied to the surface of other materials such as Ford has done to its stainless steel inlet valves. DLC coatings are often used to decrease wear and friction between two materials due its more 'slippery' properties. DLC coatings reduce friction and also reduce the wear rate of the part (C Donnet, A Erdemir, 2008) thus increasing the life span and increasing reliability in moving systems. Previously mentioned, DLC also puts the surface of a part into a compressive stress, thus allowing the part to cope with higher stresses than usual without breaking.

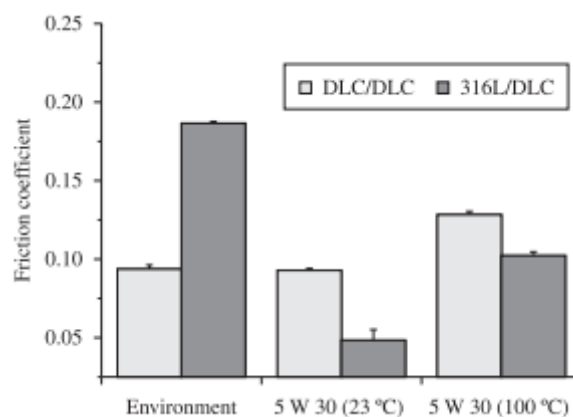


Figure 57, DLC friction coefficient (R.P de Castro Costa et al., 2011)

The graph above in figure 57 shows the friction coefficients of DLC at different temperatures and with the use of a 5W30 oil, it shows that at a working temperature of 100°C there is friction coefficient of 0.10 between DLC and 316L stainless steel.

To further lower the friction within the inlet valve system the coefficient of friction between the cam lobe and valve bucket must be lowered from 0.17. To achieve this the valve bucket will be simulated as having a DLC coating, this will reduce the coefficient of friction to 0.10 (R.P de Castro Costa et al., 2011) and will also reduce the wear rate between the cam lobe and valve bucket therefore giving a longer working life. The results of using the DLC coated valve bucket with the OEM valve and spring are shown below in figure 59.

RPM	F _c (N)	friction (N)
0	285.7185	28.571848
50	285.7044	28.570443
100	285.6623	28.566229
150	285.5921	28.559205
200	285.4937	28.549372
250	285.3673	28.53673
300	285.2128	28.521278
350	285.0302	28.503017
400	284.8195	28.481946
450	284.5807	28.458066
500	284.3138	28.431376

Figure 59, Friction and contact force using DLC coated Valve bucket

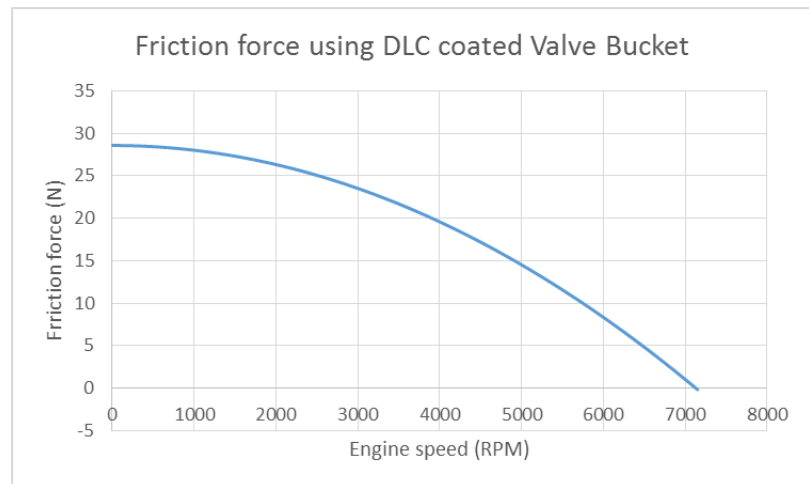


Figure 58, Graph of friction force vs RPM for inlet system using DLC coated valve bucket

As can be seen from figures 58 and 59 above, reducing the coefficient of friction between the cam lobe and valve bucket significantly reduces the friction force occurring at each rpm. The significance of this is that no other figures are affected such as engine rpm, this means that friction force can be reduced without affecting valve float rpm, a figure that would be affected when changing the mass of the valve. Figure 59 shows a maximum friction force at zero rpm of 28.572N compared to the stock engine value of 48.572N. This shows that by simply applying a DLC coating to the valve bucket a 41.2% decrease in friction force between the cam lobe and valve bucket can be achieved.

6.3 Ti-6AL-4V Valve

The friction in the inlet valve train can now be calculated using the improved titanium alloy valve. The friction is expected to decrease because of the lighter mass within the moving system. The friction will be calculated in the same manner as previously done by first calculating the contact force between the cam lobe and cam bucket using the following equation previously used;

$$F_c = M \cdot a + \text{Spring force}$$

the mass of the Ti-6AL-4V valve is known to be 16.67 grams and from previous measurements the moving mass excluding the valve is 42.5 grams, this gives a total moving mass of 59.17 grams, therefore the contact force can be calculated as follows;

$$F_c = 0.05917 \times \left(\frac{-0.00841398171928631 \times 36}{1000} \times \left(\frac{RPM}{2} \right)^2 \right) + 285.65$$

The friction force is then calculated using the equation;

$$F_{friction} = \mu \times N$$

Using the two previous equations for varying crankshaft rpm gives the following table shown below in figure 61 and graph shown in figure 60.

RPM	F _c (N)	friction (N)
0	285.7185	48.572141
50	285.7073	48.570237
100	285.6737	48.564524
150	285.6177	48.555002
200	285.5392	48.541672
250	285.4384	48.524533
300	285.3152	48.503586
350	285.1696	48.47883
400	285.0016	48.450266
450	284.8111	48.417893
500	284.5983	48.381711

Figure 61, Friction force (N) created using Ti-6AL-4V valve

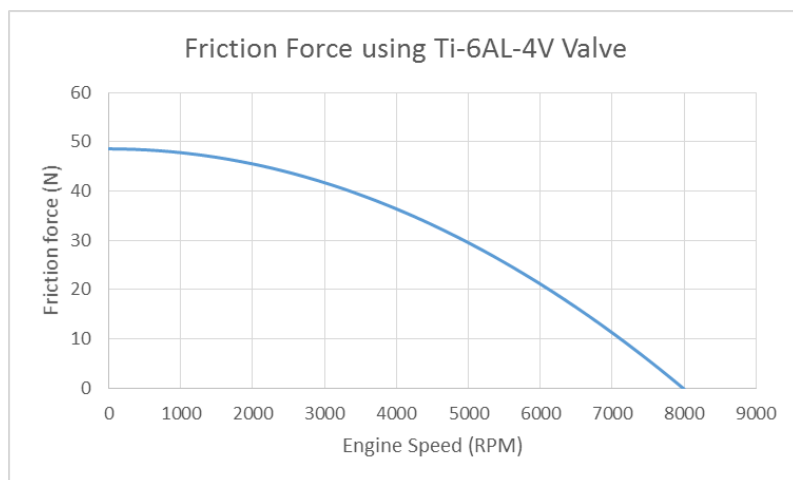


Figure 60, Graph of friction force versus RPM for Ti-6AL-4V valve

Comparing the table and graph for the titanium alloy valve compared to the OEM DLC coated stainless steel valve shows that both valves experience the same maximum friction force at zero rpm, this is because the same spring stiffness K has been used for both valves. At zero rpm the contact force is equal to the spring force, this means that by using an identical spring the friction forces will also be the same. The friction force of the titanium alloy valve however decreases at a lower rate compared to the OEM valve therefore increasing the overall friction inside of the valve train which is the opposite of the desired outcome. The benefit of this is that due to the lighter weight the engine is able to rev higher before valve float occurs, allowing more power to be gained from the engine. However the objective of this project is to decrease the friction force within the engine to increase the efficiency. To do this a new spring with a lower spring stiffness, K, will be used in conjunction with the new valve to lower the valve float rpm to 7130RPM that has previously been calculated for the OEM engine. Doing this in conjunction with the new titanium alloy valve will decrease the initial spring force at zero rpm and will therefore decrease the friction force throughout the

entire rpm range. This can be done by rearranging the calculation previously used to work out the valve float rpm;

$$\text{Valve float RPM} = 2 \times \sqrt{\frac{(k \cdot (Y + Y_0)) \times 1000}{-mass \times acc \times 36}}$$

This rearranges to;

$$K = \frac{-mass \times \left(\frac{acc \times 36}{1000} \times \left(\frac{RPM}{2} \right)^2 \right)}{(Y + Y_0)}$$

$$K = \frac{-0.05917 \times \left(\frac{-0.00841398171928631 \times 36}{1000} \times \left(\frac{7130}{2} \right)^2 \right)}{18.88}$$

$$K = 12.06 \text{ N/mm}$$

This gives a new spring stiffness of 12.06N/mm, using a spring with this stiffness along with the new titanium alloy valve would allow the engine to retain its valve float at 7130RPM. The benefit of this new spring rate is that the Contact force at each rpm will be lower, therefore reducing the friction force within the inlet valve train system. Figure 63 below was made using the same previous calculations for contact force and friction force and shows the values obtainable through the use of the new titanium alloy valve and the smaller spring rate spring.

RPM	Fc (N)	friction (N)
0	227.6928	38.707776
50	227.6816	38.705872
100	227.648	38.700159
150	227.592	38.690637
200	227.5136	38.677307
250	227.4128	38.660169
300	227.2895	38.639221
350	227.1439	38.614465
400	226.9759	38.585901
450	226.7855	38.553528
500	226.5726	38.517346

Figure 63, Friction force (N) created using Ti-6AL-4V valve and spring with $K=12.06\text{N/mm}$

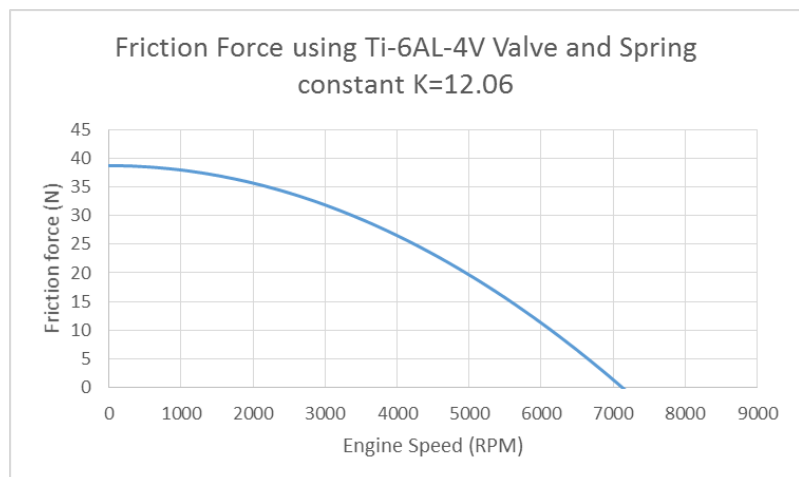


Figure 62, Graph of friction force versus RPM for Ti-6AL-4V valve and spring with $k=12.06\text{N/mm}$

As can be seen in the table and graph in figures 62 and 63 above, the friction force has clearly been reduced through the use of a spring with less stiffness and a lighter valve. The maximum friction force experienced with the OEM valve and spring was calculated to be 48.572N, compared to the new maximum friction force of 38.708N. This means through the use of a Ti-6AL-4V inlet valve and a different inlet spring with a spring constant, k , equal to

12.06N/mm, a 20.3% reduction in friction force can be achieved compared to the OEM DLC coated stainless steel valve.

6.4 Ti-6AL-4V Valve with DLC Coated Valve Bucket

Using a Ti-6AL-4V titanium alloy valve along with a spring with stiffness $k=12.06\text{N/mm}$ has been proven to reduce friction force by 20.3%. The most effective way found this far however has been to use the stock valve and spring along with a DLC coated valve bucket, this technique resulted in a friction force reduction of 41.2% over the stock engine system. To get even further friction force reductions the two proven improvements will be mathematically modelled together to find the maximum friction reduction possible.

RPM	F_c (N)	friction (N)
0	227.6928	22.76928
50	227.6816	22.76816
100	227.648	22.764799
150	227.592	22.759198
200	227.5136	22.751357
250	227.4128	22.741276
300	227.2895	22.728954
350	227.1439	22.714391
400	226.9759	22.697589
450	226.7855	22.678546
500	226.5726	22.657263

Figure 65, Ti-6AL-4V valve, spring $K=12.06\text{N/mm}$ and DLC valve bucket friction force (N)

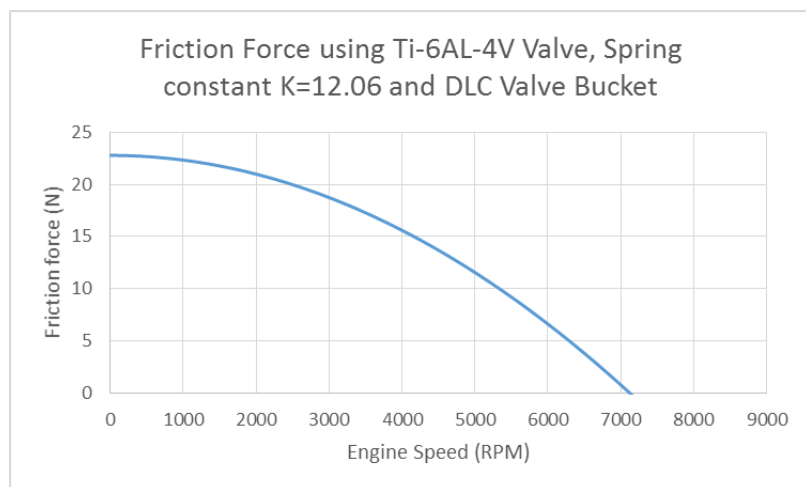


Figure 64, Graph of friction force using a Ti-6AL-4V valve with spring $k=12.06\text{N/mm}$ and DLC valve bucket

As can be seen in figures 64 and 65 above the maximum friction force at zero rpm is 22.769N, this is immediately known to be much smaller than the friction force in the OEM system of 48.572N. This was expected due to both methods previously showing improvements in friction force over the OEM valve, spring and bucket, however it can immediately be seen that the combination used in this model has decreased friction by over 50%, a huge improvement over the OEM inlet valve train. The exact reduction in friction force is shown below;

$$100 - \left(\frac{22.769}{48.572} \times 100\% \right) = 53.12\%$$

This shows that through a combination of different materials and coatings, a serious reduction in friction force within the inlet valve train can be achieved.

Chapter Seven: Conclusions and Further Work

7.1 Conclusions

Many steps were taken in this project starting off with measuring the masses of the moving components within the valve train of the Ford 1.6GDTI EcoBoost engine to allow them to be used in future calculations. The cam lift was also measured which allowed, through numerous calculations, the actual valve lift to be found. Finding Valve lift was the main starting point for the project as it allowed the accelerations to be found which were major contributing factors for calculating both friction forces and stresses that occur within the valve train. Finding the accelerations allowed the maximum friction force of the OEM valve train system to be found, a force equalling 48.572N. Calculating the Frictional forces within the OEM system also proved the research carried out was correct. This therefore confirmed the statement that valve train friction decreases as RPM increases, this is unique as it is normally known that friction increases with increased speed. Calculating the friction forces within the engine also allowed a comparison to be made between any potential improvements and the original OEM components.

The FEA component of this project allowed a detail analysis of the valve to be carried out, by doing this a fatigue safety factor of 16.55 was calculated which allowed an understanding of the safety factor that Ford uses within their OEM valves. Calculating this fatigue safety factor also allowed the new designs to conform to this standard or higher. The Ti-6AL-4V hollow stem valve iteration however did not meet this fatigue safety factor which means, from an OEM point of view, that it is not suitable for the application as Ford would not risk using it due to a potentially shorter fatigue life. The hollow valve stem however was lighter than all other iterations and would have further reduced friction within the system, this means that for a performance application, where longevity is not always needed, the hollow valve stem would be a viable option. The solid stem Ti-6AL-4V valve however met and surpassed the fatigue safety factor set by the OEM valve, meaning that it would be a viable option for the production vehicle from a fatigue life point of view.

The friction of the newly designed components was then calculated. The Ti-6AL-4V valve when paired with a lower spring rate valve spring reduced the friction force within the system by 20.3%, a respectable improvement over the OEM system. This was achievable due to the lower mass within the moving system, with a lower mass a lower spring rate valve spring could be used which meant that the contact forces were lower and therefore a lower friction force was achievable. The DLC coated valve bucket also saw great reductions in friction, even more so than the titanium alloy valve, with a friction reduction of 41.2%. This

was achieved because the coefficient of friction was reduced from 0.17 to 0.10, a figure that made an immense difference to the friction force experienced within the modified valve train system. Next the two improvements were paired together mathematically to model the 1.6GDTI valve train as consisting of a DLC coated valve bucket with a Ti-6AL-4V valve and a spring with stiffness $K=12.06\text{N/mm}$. This yielded the greatest reduction in friction, showing that the friction force was over halved from the OEM value, a reduction of 53.12%. This is a major improvement and would greatly increase the efficiency of the inlet Valve train, and throughout the entire valve train would decrease the friction force by 25%. This would mean a 25% increase in efficiency would be achievable through the use of a titanium alloy valve, paired spring and DLC coated valve bucket.

7.2 Evaluation

The measurements of the masses of the components within the Engine were carried out accurately and gave good results that matched the research carried out, this meant that calculations using these results could also be considered accurate. The measurements that led to some inconsistency was the measurement of the cam lobe itself, this was due to only having access to a dial gauge accurate to two decimal places. This led to some errors that are most noticeable in figure 22 showing the acceleration of the valve, which shows a lot of noise due to the measurements not being as accurate as they could have been. To improve this measurement a more accurate digital dial gauge, accurate to three decimal places, should have been used, this would have shown smaller changes in cam lift which would give more accurate changes in velocity and therefore more accurate changes in acceleration.

The FEA segment of this paper was very successful and very little error occurred due to the 3D scanned valve being used. Using the 3D scanner, along with measurements of the valve, allowed a very accurate model of the valve to be created. Using an accurate model along with the calculated accelerations and a fine mesh also allowed accurate values of stress to be calculated. Having these accurate stress values also allowed a good fatigue safety factor for each valve to be calculated and compared, this allowed the modified valves to be compared to the OEM standard.

A major Friction reduction has been achievable mathematically through the use of coatings and materials within the Ford 1.6GDTI EcoBoost engine. The cost for Ford however to implement these changes would be greater than the current OEM system costs, this is due to the cost of Ti-6AL-4V titanium alloy. The titanium alloy costs approximately £11.57/kg (Metal Prices, 2016) compared to the approximate cost of £4.51/kg for EV8 21-N stainless steel (Zaubas, 2016), this shows that the cost per inlet valve would be almost three times

more than the standard stainless steel currently being used. The DLC coated valve bucket returned the largest decrease in friction force and would simply mean coating the currently produced valve buckets in a diamond like carbon coating. This option would be viable for Ford as the current Inlet valves already have a DLC coating, this shows that Ford has the technology and the systems in place to apply these coatings. There would be a slight cost increase over the OEM system for Ford as it would include applying DLC to both the inlet valve and the valve bucket rather than just the valve as currently done. This slight increase in cost however would be beneficial to increasing the efficiency of the engine to help Ford reach the 95g CO₂/km that has to be adhered to by 2020. After carrying out a cost/benefit analysis of the systems put in place to reduce friction force it can be clearly seen that the titanium alloy valve costs too much for the potential increase in efficiency it gives, the titanium valve would be better suited in a performance application where increased rpm and power is desired. The DLC coated valve bucket however gives a much larger decrease in friction force due to the high contact forces between the cam lobe and valve bucket, decreasing the coefficient of friction in this area greatly reduces the friction forces experienced. Due to Ford having the ability and means to already carry out DLC coating, DLC coating would be the best option to carry out as the cost per product would not greatly increase but a big increase in efficiency could be obtained.

7.3 Further work

There are many more areas of valve train friction that could be researched with the goal of reducing friction force. If this project were continued to be carried out, the same calculations and FEA simulations would be carried out for the exhaust side of the valve train as well. This would allow a complete friction model for the cam lobe and valve bucket to be made and a friction force for the entire valve train could be obtained. Carrying out these extra steps would also allow a greater friction force reduction to be calculated through the use of the modified parts already simulated within this project.

Another step that would also be taken would be to further research the use of coatings to decrease friction and wear and to increase the compressive surface stress that these coatings apply. By researching these coatings in a more detailed manner, even lighter parts could be made and simulated that could cope with higher stresses and also reduce the friction force experienced within the valve train. Materials and designs would also be another topic that further research would be carried out on, this is because the hollow stem valve is a viable option for decreasing weight as long as a high enough fatigue safety factor can be achieved. To achieve the standard of fatigue safety that Ford have set, the hollow stem valve would either need a coating to cope with the increased stress or it would need to be made

out of a different material with a larger ultimate tensile stress. Increased research into materials and coatings could allow the hollow stem valve to be used in future production of the Ford 1.6GDTI engine, but from the simulations carried out is not currently a viable option.

Another step to be carried out in further work would be to calculate friction in the camshaft journal bearings. This would allow a more accurate valve train friction to be produced due to the current friction calculation only taking the cam lobe and valve bucket friction into consideration. If this friction were to be introduced then a friction model for the entire valve train could be produced. This would allow, again through the use of materials and coatings, friction reduction throughout the entire valve train to be researched and simulated. By simulating the entire valve train friction force, and using modifications over the standard system, the entire valve train friction could be decreased, further increasing the efficiency of the engine.

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Appendix

Camshaft angle	Cam lobe lift (mm)
0	0.03
1	0.03
2	0.04
3	0.04
4	0.05
5	0.05
6	0.06
7	0.07
8	0.08
9	0.08
10	0.09
11	0.1
12	0.1
13	0.11
14	0.11
15	0.11
16	0.11
17	0.11
18	0.11
19	0.11
20	0.11
21	0.1
22	0.1
23	0.1
24	0.09
25	0.09
26	0.08
27	0.07
28	0.07
29	0.06
30	0.06
31	0.05
32	0.05
33	0.04
34	0.04
35	0.04
36	0.04
37	0.04
38	0.04
39	0.03
40	0.03

Figure 67, Cam lobe lift part 1

41	0.03
42	0.04
43	0.04
44	0.05
45	0.05
46	0.06
47	0.07
48	0.07
49	0.08
50	0.09
51	0.1
52	0.1
53	0.11
54	0.11
55	0.11
56	0.11
57	0.11
58	0.11
59	0.11
60	0.11
61	0.11
62	0.11
63	0.11
64	0.11
65	0.11
66	0.11
67	0.11
68	0.11
69	0.11
70	0.11
71	0.12
72	0.12
73	0.13
74	0.13
75	0.14
76	0.14
77	0.15
78	0.15
79	0.15
80	0.16

Figure 66, Cam lobe lift part 2

81	0.16
82	0.16
83	0.17
84	0.17
85	0.17
86	0.18
87	0.18
88	0.19
89	0.19
90	0.19
91	0.2
92	0.2
93	0.2
94	0.2
95	0.21
96	0.21
97	0.21
98	0.21
99	0.21
100	0.22
101	0.22
102	0.23
103	0.24
104	0.25
105	0.27
106	0.28
107	0.3
108	0.32
109	0.34
110	0.37
111	0.39
112	0.41
113	0.43
114	0.44
115	0.46
116	0.48
117	0.5
118	0.52
119	0.53
120	0.55

Figure 69, Cam lobe lift part 3

121	0.57
122	0.58
123	0.6
124	0.62
125	0.63
126	0.65
127	0.67
128	0.69
129	0.71
130	0.74
131	0.77
132	0.8
133	0.85
134	0.89
135	0.94
136	0.99
137	1.05
138	1.12
139	1.19
140	1.27
141	1.35
142	1.44
143	1.54
144	1.62
145	1.73
146	1.84
147	1.96
148	2.08
149	2.19
150	2.33
151	2.48
152	2.62
153	2.77
154	2.94
155	3.1
156	3.28
157	3.47
158	3.67
159	3.87
160	4.07

Figure 68, Cam lobe lift part 4

161	4.28
162	4.5
163	4.73
164	4.99
165	5.23
166	5.49
167	5.76
168	6.03
169	6.3
170	6.57
171	6.87
172	7.12
173	7.38
174	7.59
175	7.78
176	7.92
177	8.02
178	8.1
179	8.13
180	8.13
181	8.11
182	8.04
183	7.95
184	7.82
185	7.65
186	7.46
187	7.25
188	7.01
189	6.76
190	6.51
191	6.24
192	6.01
193	5.74
194	5.51
195	5.27
196	5.03
197	4.8
198	4.58
199	4.38
200	4.16

Figure 71, Cam lobe lift part 5

201	3.97
202	3.78
203	3.57
204	3.39
205	3.21
206	3.03
207	2.88
208	2.72
209	2.56
210	2.41
211	2.27
212	2.14
213	2.01
214	1.9
215	1.79
216	1.68
217	1.58
218	1.48
219	1.39
220	1.31
221	1.23
222	1.15
223	1.08
224	1.02
225	0.96
226	0.91
227	0.86
228	0.81
229	0.77
230	0.73
231	0.7
232	0.67
233	0.64
234	0.62
235	0.6
236	0.58
237	0.56
238	0.53
239	0.51
240	0.49

Figure 70, Cam lobe lift part 6

241	0.46
242	0.43
243	0.4
244	0.38
245	0.35
246	0.33
247	0.3
248	0.28
249	0.26
250	0.24
251	0.22
252	0.2
253	0.18
254	0.16
255	0.14
256	0.13
257	0.11
258	0.1
259	0.08
260	0.07
261	0.05
262	0.04
263	0.04
264	0.03
265	0.02
266	0.02
267	0.02
268	0.02
269	0.02
270	0.01
271	0.01
272	0.01
273	0.01
274	0.01
275	0
276	0
277	0
278	0
279	0
280	0

Figure 73, Cam lobe lift part 7

281	0
282	0
283	0
284	0
285	0.01
286	0.01
287	0.02
288	0.02
289	0.03
290	0.03
291	0.03
292	0.04
293	0.04
294	0.04
295	0.04
296	0.04
297	0.04
298	0.03
299	0.03
300	0.03
301	0.02
302	0.02
303	0.02
304	0.02
305	0.01
306	0.01
307	0.01
308	0.01
309	0.01
310	0.01
311	0.01
312	0.01
313	0.01
314	0.01
315	0.01
316	0.01
317	0.01
318	0.01
319	0.01
320	0.01

Figure 72, Cam lobe lift part 8

321	0.01
322	0.01
323	0.01
324	0.01
325	0.01
326	0.01
327	0.02
328	0.02
329	0.02
330	0.02
331	0.02
332	0.02
333	0.02
334	0.02
335	0.02
336	0.02
337	0.03
338	0.03
339	0.03
340	0.03
341	0.03
342	0.03
343	0.03
344	0.03
345	0.03
346	0.03
347	0.03
348	0.03
349	0.03
350	0.03
351	0.03
352	0.03
353	0.03
354	0.03
355	0.03
356	0.03
357	0.03
358	0.03
359	0.03

Figure 74, Cam lobe lift part 9

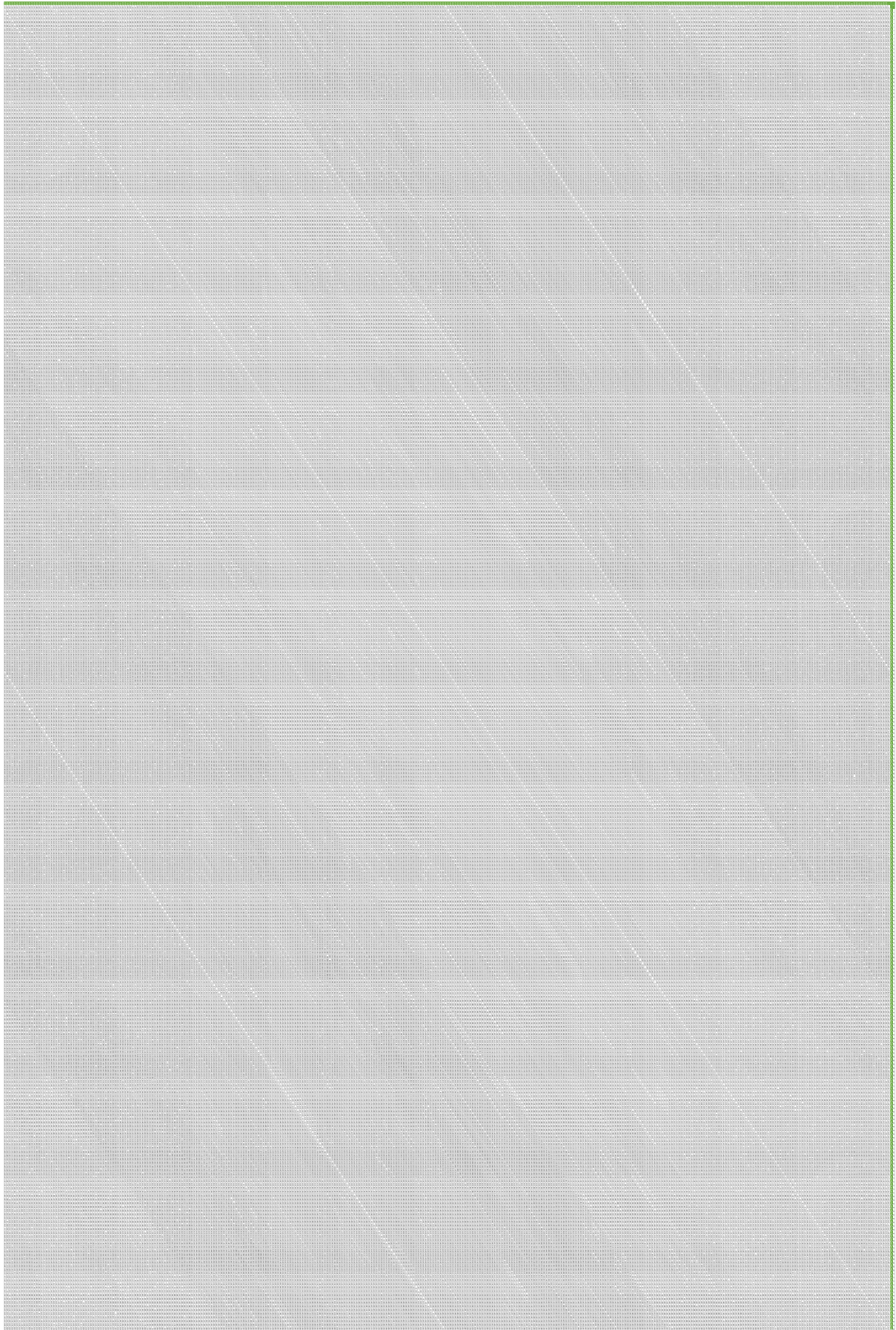


Figure 75, 360 by 360 matrix to calculate valve lift

cam angle	cam lift (mm)	valve lift (mm)
0	0.03	0
1	0.03	0
2	0.04	0
3	0.04	0
4	0.05	0
5	0.05	0
6	0.06	0
7	0.07	0
8	0.08	0
9	0.08	0
10	0.09	0
11	0.1	0
12	0.1	0
13	0.11	0
14	0.11	0
15	0.11	0
16	0.11	0
17	0.11	0
18	0.11	0
19	0.11	0
20	0.11	0
21	0.1	0
22	0.1	0
23	0.1	0
24	0.09	0
25	0.09	0
26	0.08	0
27	0.07	0
28	0.07	0
29	0.06	0
30	0.06	0
31	0.05	0
32	0.05	0
33	0.04	0
34	0.04	0
35	0.04	0
36	0.04	0
37	0.04	0
38	0.04	0
39	0.03	0
40	0.03	0

Figure 77, Valve lift part 1

41	0.03	0
42	0.04	0
43	0.04	0
44	0.05	0
45	0.05	0
46	0.06	0
47	0.07	0
48	0.07	0
49	0.08	0
50	0.09	0
51	0.1	0
52	0.1	0
53	0.11	0
54	0.11	0
55	0.11	0
56	0.11	0
57	0.11	0
58	0.11	0
59	0.11	0
60	0.11	0
61	0.11	0
62	0.11	0
63	0.11	0
64	0.11	0
65	0.11	0
66	0.11	0
67	0.11	0
68	0.11	0
69	0.11	0
70	0.11	0
71	0.12	0
72	0.12	0
73	0.13	0
74	0.13	0
75	0.14	0
76	0.14	0
77	0.15	0
78	0.15	0
79	0.15	0
80	0.16	0

Figure 76, Valve lift part 2

81	0.16	0
82	0.16	0
83	0.17	0
84	0.17	0
85	0.17	0
86	0.18	0
87	0.18	0
88	0.19	0
89	0.19	0
90	0.19	0
91	0.2	0
92	0.2	0
93	0.2	0
94	0.2	0
95	0.21	0
96	0.21	0
97	0.21	0
98	0.21	0
99	0.21	0
100	0.22	0
101	0.22	0
102	0.23	0
103	0.24	0
104	0.25	0
105	0.27	0.011619357
106	0.28	0.031607176
107	0.3	0.051594995
108	0.32	0.071582814
109	0.34	0.091570634
110	0.37	0.111558453
111	0.39	0.138440103
112	0.41	0.161528001
113	0.43	0.18837159
114	0.44	0.21149755
115	0.46	0.238303077
116	0.48	0.268261969
117	0.5	0.298220862
118	0.52	0.321430556
119	0.53	0.341418375
120	0.55	0.368124944

Figure 79, Valve lift part 3

121	0.57	0.391387924
122	0.58	0.411375743
123	0.6	0.431363562
124	0.62	0.451351382
125	0.63	0.477974215
126	0.65	0.507933108
127	0.67	0.545054348
128	0.69	0.592579041
129	0.71	0.642388793
130	0.74	0.700082018
131	0.77	0.776088912
132	0.8	0.855310401
133	0.85	0.950250365
134	0.89	1.055951926
135	0.94	1.166282099
136	0.99	1.302517687
137	1.05	1.456319729
138	1.12	1.609674253
139	1.19	1.789659911
140	1.27	1.976729469
141	1.35	2.157430434
142	1.44	2.35697181
143	1.54	2.553370583
144	1.62	2.777430215
145	1.73	3.001489848
146	1.84	3.237942341
147	1.96	3.46828075
148	2.08	3.701534987
149	2.19	3.935071652
150	2.33	4.168467103
151	2.48	4.401862554
152	2.62	4.628072634
153	2.77	4.853599179
154	2.94	5.070613727
155	3.1	5.276899725
156	3.28	5.476621845
157	3.47	5.673218574
158	3.67	5.858903195
159	3.87	6.041543121
160	4.07	6.217145244

Figure 78, Valve lift part 4

161	4.28	6.381669574
162	4.5	6.546620007
163	4.73	6.700709053
164	4.99	6.843573007
165	5.23	6.97516835
166	5.49	7.102779981
167	5.76	7.223480093
168	6.03	7.332795872
169	6.3	7.430694019
170	6.57	7.519818194
171	6.87	7.606473909
172	7.12	7.681628626
173	7.38	7.751245188
174	7.59	7.814993106
175	7.78	7.867176509
176	7.92	7.907779503
177	8.02	7.936789718
178	8.1	7.956777537
179	8.13	7.974195273
180	8.13	7.98
181	8.11	7.979999999
182	8.04	7.974188157
183	7.95	7.956763307
184	7.82	7.927730759
185	7.65	7.897771883
186	7.46	7.857172447
187	7.25	7.804995648
188	7.01	7.74125738
189	6.76	7.665977057
190	6.51	7.586573431
191	6.24	7.499956242
192	6.01	7.401870502
193	5.74	7.303101773
194	5.51	7.19386543
195	5.27	7.073253774
196	5.03	6.941303545
197	4.8	6.804363078
198	4.58	6.66164334
199	4.38	6.50771041
200	4.16	6.342611176

Figure 81, Valve lift part 5

201	3.97	6.166395929
202	3.78	5.983756564
203	3.57	5.797420822
204	3.39	5.600135139
205	3.21	5.400413714
206	3.03	5.193401176
207	2.88	4.975622686
208	2.72	4.747144582
209	2.56	4.50803646
210	2.41	4.271019816
211	2.27	4.035174005
212	2.14	3.788915049
213	2.01	3.532317962
214	1.9	3.265460906
215	1.79	3.015693225
216	1.68	2.767157639
217	1.58	2.518622053
218	1.48	2.292073188
219	1.39	2.063742056
220	1.31	1.874638786
221	1.23	1.685535516
222	1.15	1.513898409
223	1.08	1.365151776
224	1.02	1.228929792
225	0.96	1.111552249
226	0.91	1.001656908
227	0.86	0.910799464
228	0.81	0.815653122
229	0.77	0.738010657
230	0.73	0.67224581
231	0.7	0.624836483
232	0.67	0.58258834
233	0.64	0.547861094
234	0.62	0.521297296
235	0.6	0.501309484
236	0.58	0.481321672
237	0.56	0.465332844
238	0.53	0.451339955
239	0.51	0.431352143
240	0.49	0.415340466

Figure 80, Valve lift part 6

241	0.46	0.401370425
242	0.43	0.381382613
243	0.4	0.365348087
244	0.38	0.351400895
245	0.35	0.331413083
246	0.33	0.311425272
247	0.3	0.29143746
248	0.28	0.275361805
249	0.26	0.261455742
250	0.24	0.24146793
251	0.22	0.221480118
252	0.2	0.201492306
253	0.18	0.178368296
254	0.16	0.151522777
255	0.14	0.131534965
256	0.13	0.115386192
257	0.11	0.101553247
258	0.1	0.085390765
259	0.08	0.075392289
260	0.07	0.069999999
261	0.05	0.065393813
262	0.04	0.059999999
263	0.04	0.059999999
264	0.03	0.059999999
265	0.02	0.059999999
266	0.02	0.055395337
267	0.02	0.049999999
268	0.02	0.049999999
269	0.02	0.049999999
270	0.01	0.045396862
271	0.01	0.039999999
272	0.01	0.039999999
273	0.01	0.035398386
274	0.01	0.029999999
275	0	0.02539991
276	0	0.019999999
277	0	0.019999999
278	0	0.015401434
279	0	0.009999999
280	0	0.009999999

Figure 83, Valve lift part 7

281	0	0.005402958
282	0	0
283	0	0
284	0	0
285	0.01	0
286	0.01	0
287	0.02	0
288	0.02	0
289	0.03	0
290	0.03	0
291	0.03	0
292	0.04	0
293	0.04	0
294	0.04	0
295	0.04	0
296	0.04	0
297	0.04	0
298	0.03	0
299	0.03	0
300	0.03	0
301	0.02	0
302	0.02	0
303	0.02	0
304	0.02	0
305	0.01	0
306	0.01	0
307	0.01	0
308	0.01	0
309	0.01	0
310	0.01	0
311	0.01	0
312	0.01	0
313	0.01	0
314	0.01	0
315	0.01	0
316	0.01	0
317	0.01	0
318	0.01	0
319	0.01	0
320	0.01	0

Figure 82, Valve lift part 8

321	0.01	0
322	0.01	0
323	0.01	0
324	0.01	0
325	0.01	0
326	0.01	0
327	0.02	0
328	0.02	0
329	0.02	0
330	0.02	0
331	0.02	0
332	0.02	0
333	0.02	0
334	0.02	0
335	0.02	0
336	0.02	0
337	0.03	0
338	0.03	0
339	0.03	0
340	0.03	0
341	0.03	0
342	0.03	0
343	0.03	0
344	0.03	0
345	0.03	0
346	0.03	0
347	0.03	0
348	0.03	0
349	0.03	0
350	0.03	0
351	0.03	0
352	0.03	0
353	0.03	0
354	0.03	0
355	0.03	0
356	0.03	0
357	0.03	0
358	0.03	0
359	0.03	0

Figure 84, Valve lift part 9

[illegible]

Figure 86, Velocity and acceleration part 1

[illegible]

Figure 85, Velocity and acceleration part 2

[illegible]

Figure 88, Velocity and acceleration part 3

0.02326298	-0.003443588
0.019987819	-0.003275161
0.019987819	0
0.019987819	0
0.026622834	0.006635014
0.029958892	0.003336059
0.03712124	0.007162348
0.047524693	0.010403453
0.049809752	0.002285059
0.057693225	0.007883473
0.076006894	0.018313668
0.079221489	0.003214596
0.094939964	0.015718474
0.105701561	0.010761597
0.110330173	0.004628612
0.136235588	0.025905415
0.153802042	0.017566454
0.153354524	-0.000447519
0.179985658	0.026631134
0.187069558	0.0070839
0.180700965	-0.006368593
0.199541376	0.018840411
0.196398772	-0.003142604
0.224059633	0.027660861
0.224059633	0
0.236452493	0.01239286
0.230338409	-0.006114084
0.233254237	0.002915828
0.233536665	0.000282428
0.233395451	-0.000141214
0.233395451	-6.66134E-15
0.22621008	-0.007185371
0.225526545	-0.000683535
0.217014548	-0.008511997
0.206285999	-0.010728549
0.199722119	-0.006563879
0.19659673	-0.00312539
0.185684621	-0.010912109
0.182639925	-0.003044696
0.175602124	-0.007037802

Figure 87, Velocity and acceleration part 4

0.164524329	-0.011077794
0.164950434	0.000426104
0.154089046	-0.010861388
0.142863953	-0.011225093
0.131595343	-0.01126861
0.127611631	-0.003983712
0.120700113	-0.006911518
0.109315779	-0.011384334
0.097898146	-0.011417633
0.089124176	-0.008773971
0.086655715	-0.002468461
0.075154717	-0.011500998
0.069616562	-0.005538155
0.063747918	-0.005868644
0.052183403	-0.011564515
0.040602993	-0.01158041
0.029010215	-0.011592778
0.019987819	-0.009022396
0.017417736	-0.002570083
0.005804727	-0.011613009
-5.48951E-10	-0.005804727
-0.005811842	-0.005811842
-0.017424849	-0.011613007
-0.029032549	-0.011607699
-0.029958875	-0.000926327
-0.040599436	-0.010640561
-0.052176799	-0.011577363
-0.063738268	-0.011561469
-0.075280322	-0.011542054
-0.079403626	-0.004123304
-0.086617189	-0.007213563
-0.09808574	-0.011468551
-0.098768728	-0.000682988
-0.109236343	-0.010467615
-0.120611656	-0.011375313
-0.131950229	-0.011338573
-0.136940467	-0.004990237
-0.142719738	-0.005779271
-0.153932931	-0.011213193
-0.165099234	-0.011166303

Figure 90, Velocity and acceleration part 5

-0.176215246	-0.011116012
-0.182639365	-0.006424119
-0.186335741	-0.003696376
-0.197285683	-0.010949942
-0.199721425	-0.002435742
-0.207012538	-0.007291113
-0.21777849	-0.010765952
-0.228478104	-0.010699614
-0.239108121	-0.010630017
-0.237016644	0.002091477
-0.235845812	0.001170833
-0.246258956	-0.010413144
-0.256597087	-0.010338131
-0.266857056	-0.010259969
-0.249767681	0.017089375
-0.248535586	0.001232095
-0.248535586	0
-0.226548865	0.021986721
-0.228331131	-0.001782266
-0.18910327	0.039227861
-0.18910327	-1.77636E-14
-0.171637106	0.017466164
-0.148746633	0.022890473
-0.136221984	0.012524649
-0.117377543	0.018844441
-0.109895342	0.007482201
-0.090857444	0.019037898
-0.095146342	-0.004288899
-0.077642465	0.017503877
-0.065764847	0.011877618
-0.047409327	0.01835552
-0.042248143	0.005161184
-0.034727245	0.007520898
-0.026563798	0.008163447
-0.019987812	0.006575986
-0.019987812	0
-0.015988828	0.003998984
-0.01399289	0.001995938
-0.019987812	-0.005994922
-0.016011677	0.003976135

Figure 89, Velocity and acceleration part 6

-0.013970041	0.002041637
-0.019987812	-0.006017771
-0.016034526	0.003953285
-0.013947191	0.002087335
-0.019987812	-0.006040621
-0.019987812	0
-0.019987812	3.55271E-15
-0.016075655	0.003912157
-0.013906063	0.002169592
-0.019987812	-0.006081749
-0.019987812	-3.58047E-15
-0.019987812	0
-0.02312401	-0.003136198
-0.02684552	-0.00372151
-0.019987812	0.006857708
-0.016148773	0.003839039
-0.013832945	0.002315828
-0.016162482	-0.002329537
-0.009998476	0.006164007
-0.00539229	0.004606186
-0.004606186	0.000786103
-0.005393814	-0.000787628
0	0.005393814
0	0
0	0
-0.004604662	-0.004604662
-0.005395338	-0.000790676
0	0.005395338
0	0
-0.004603138	-0.004603138
-0.005396862	-0.000793724
0	0.005396862
-0.004601614	-0.004601614
-0.005398386	-0.000796773
-0.004600089	0.000798297
-0.005399911	-0.000799821
0	0.005399911
-0.004598565	-0.004598565
-0.005401435	-0.00080287
0	0.005401435

Figure 92, Velocity and acceleration part 7

[illegible]

Figure 91, Velocity and acceleration part 8

[illegible]

Figure 93, Velocity and acceleration part 9

RPM	Fc (N)	friction (N)
0	285.7185	48.572141
50	285.7044	48.569753
100	285.6623	48.562589
150	285.5921	48.550649
200	285.4937	48.533933
250	285.3673	48.51244
300	285.2128	48.486172
350	285.0302	48.455128
400	284.8195	48.419308
450	284.5807	48.378712
500	284.3138	48.333339
550	284.0188	48.283191
600	283.6957	48.228267
650	283.3445	48.168566
700	282.9652	48.10409
750	282.5579	48.034838
800	282.1224	47.960809
850	281.6589	47.882005
900	281.1672	47.798424
950	280.6475	47.710068
1000	280.0996	47.616935
1050	279.5237	47.519027
1100	278.9197	47.416342
1150	278.2875	47.308881
1200	277.6273	47.196645
1250	276.939	47.079632
1300	276.2226	46.957843
1350	275.4781	46.831278
1400	274.7055	46.699938
1450	273.9048	46.563821
1500	273.076	46.422928
1550	272.2192	46.277259
1600	271.3342	46.126814
1650	270.4211	45.971593
1700	269.48	45.811596
1750	268.5107	45.646823

Figure 95, OEM system contact and friction forces part 1

1800	267.5134	45.477274
1850	266.4879	45.302949
1900	265.4344	45.123848
1950	264.3528	44.939971
2000	263.243	44.751318
2050	262.1052	44.557889
2100	260.9393	44.359684
2150	259.7453	44.156703
2200	258.5232	43.948945
2250	257.273	43.736412
2300	255.9947	43.519103
2350	254.6883	43.297017
2400	253.3539	43.070156
2450	251.9913	42.838519
2500	250.6006	42.602105
2550	249.1819	42.360916
2600	247.735	42.11495
2650	246.2601	41.864209
2700	244.757	41.608691
2750	243.2259	41.348398
2800	241.6666	41.083328
2850	240.0793	40.813483
2900	238.4639	40.538861
2950	236.8204	40.259463
3000	235.1488	39.97529
3050	233.4491	39.68634
3100	231.7213	39.392614
3150	229.9654	39.094112
3200	228.1814	38.790835
3250	226.3693	38.482781
3300	224.5291	38.169951
3350	222.6609	37.852345
3400	220.7645	37.529963
3450	218.84	37.202805
3500	216.8875	36.870871
3550	214.9068	36.534161
3600	212.8981	36.192675

Figure 94, OEM system contact and friction forces part 2

3650	210.8613	35.846413
3700	208.7963	35.495375
3750	206.7033	35.139561
3800	204.5822	34.778971
3850	202.433	34.413605
3900	200.2557	34.043462
3950	198.0503	33.668544
4000	195.8168	33.28885
4050	193.5552	32.90438
4100	191.2655	32.515133
4150	188.9477	32.121111
4200	186.6018	31.722312
4250	184.2279	31.318738
4300	181.8258	30.910388
4350	179.3957	30.497261
4400	176.9374	30.079359
4450	174.4511	29.65668
4500	171.9366	29.229226
4550	169.3941	28.796995
4600	166.8235	28.359988
4650	164.2247	27.918206
4700	161.5979	27.471647
4750	158.943	27.020312
4800	156.26	26.564202
4850	153.5489	26.103315
4900	150.8097	25.637652
4950	148.0424	25.167213
5000	145.2471	24.691999
5050	142.4236	24.212008
5100	139.572	23.727241
5150	136.6923	23.237698
5200	133.7846	22.743379
5250	130.8487	22.244284
5300	127.8848	21.740413
5350	124.8927	21.231766
5400	121.8726	20.718343
5450	118.8244	20.200144

Figure 97, OEM system contact and friction forces part 3

5500	115.7481	19.677169
5550	112.6436	19.149418
5600	109.5111	18.61689
5650	106.3505	18.079587
5700	103.1618	17.537508
5750	99.94502	16.990653
5800	96.70013	16.439021
5850	93.42714	15.882614
5900	90.12606	15.321431
5950	86.79689	14.755471
6000	83.43962	14.184736
6050	80.05426	13.609225
6100	76.64081	13.028937
6150	73.19926	12.443874
6200	69.72961	11.854034
6250	66.23187	11.259419
6300	62.70604	10.660027
6350	59.15211	10.055859
6400	55.57009	9.4469158
6450	51.95998	8.8331962
6500	48.32177	8.2147005
6550	44.65546	7.5914288
6600	40.96106	6.963381
6650	37.23857	6.3305573
6700	33.48799	5.6929575
6750	29.7093	5.0505816
6800	25.90253	4.4034298
6850	22.06766	3.7515019
6900	18.20469	3.094798
6950	14.31364	2.4333181
7000	10.39448	1.7670621
7050	6.447236	1.0960301
7100	2.471895	0.4202221
7150	-1.53154	-0.260362

Figure 96, OEM system contact and friction forces part 4

RPM	Fc (N)	friction (N)
0	285.7185	28.571848
50	285.7044	28.570443
100	285.6623	28.566229
150	285.5921	28.559205
200	285.4937	28.549372
250	285.3673	28.53673
300	285.2128	28.521278
350	285.0302	28.503017
400	284.8195	28.481946
450	284.5807	28.458066
500	284.3138	28.431376
550	284.0188	28.401877
600	283.6957	28.369569
650	283.3445	28.334451
700	282.9652	28.296524
750	282.5579	28.255787
800	282.1224	28.212241
850	281.6589	28.165885
900	281.1672	28.11672
950	280.6475	28.064746
1000	280.0996	28.009962
1050	279.5237	27.952369
1100	278.9197	27.891966
1150	278.2875	27.828754
1200	277.6273	27.762732
1250	276.939	27.693901
1300	276.2226	27.622261
1350	275.4781	27.547811
1400	274.7055	27.470552
1450	273.9048	27.390483
1500	273.076	27.307605
1550	272.2192	27.221917
1600	271.3342	27.13342
1650	270.4211	27.042114
1700	269.48	26.947998
1750	268.5107	26.851073

Figure 99, OEM system with DLC valve bucket part 1

1800	267.5134	26.751338
1850	266.4879	26.648794
1900	265.4344	26.54344
1950	264.3528	26.435277
2000	263.243	26.324305
2050	262.1052	26.210523
2100	260.9393	26.093932
2150	259.7453	25.974531
2200	258.5232	25.852321
2250	257.273	25.727301
2300	255.9947	25.599472
2350	254.6883	25.468834
2400	253.3539	25.335386
2450	251.9913	25.199129
2500	250.6006	25.060062
2550	249.1819	24.918186
2600	247.735	24.7735
2650	246.2601	24.626005
2700	244.757	24.475701
2750	243.2259	24.322587
2800	241.6666	24.166664
2850	240.0793	24.007931
2900	238.4639	23.846389
2950	236.8204	23.682037
3000	235.1488	23.514876
3050	233.4491	23.344906
3100	231.7213	23.172126
3150	229.9654	22.996537
3200	228.1814	22.818138
3250	226.3693	22.63693
3300	224.5291	22.452912
3350	222.6609	22.266085
3400	220.7645	22.076449
3450	218.84	21.884003
3500	216.8875	21.688748
3550	214.9068	21.490683
3600	212.8981	21.289809

Figure 98, OEM system with DLC valve bucket part 2

3650	210.8613	21.086125
3700	208.7963	20.879632
3750	206.7033	20.67033
3800	204.5822	20.458218
3850	202.433	20.243297
3900	200.2557	20.025566
3950	198.0503	19.805026
4000	195.8168	19.581676
4050	193.5552	19.355517
4100	191.2655	19.126549
4150	188.9477	18.894771
4200	186.6018	18.660184
4250	184.2279	18.422787
4300	181.8258	18.182581
4350	179.3957	17.939565
4400	176.9374	17.69374
4450	174.4511	17.445106
4500	171.9366	17.193662
4550	169.3941	16.939409
4600	166.8235	16.682346
4650	164.2247	16.422474
4700	161.5979	16.159792
4750	158.943	15.894301
4800	156.26	15.626001
4850	153.5489	15.354891
4900	150.8097	15.080972
4950	148.0424	14.804243
5000	145.2471	14.524705
5050	142.4236	14.242358
5100	139.572	13.957201
5150	136.6923	13.669234
5200	133.7846	13.378458
5250	130.8487	13.084873
5300	127.8848	12.788478
5350	124.8927	12.489274
5400	121.8726	12.187261
5450	118.8244	11.882438

Figure 101, OEM system with DLC
valve bucket part 3

5500	115.7481	11.574805
5550	112.6436	11.264363
5600	109.5111	10.951112
5650	106.3505	10.635051
5700	103.1618	10.316181
5750	99.94502	9.9945016
5800	96.70013	9.6700126
5850	93.42714	9.3427142
5900	90.12606	9.0126064
5950	86.79689	8.6796891
6000	83.43962	8.3439624
6050	80.05426	8.0054262
6100	76.64081	7.6640807
6150	73.19926	7.3199257
6200	69.72961	6.9729613
6250	66.23187	6.6231874
6300	62.70604	6.2706041
6350	59.15211	5.9152114
6400	55.57009	5.5570093
6450	51.95998	5.1959977
6500	48.32177	4.8321768
6550	44.65546	4.4655463
6600	40.96106	4.0961065
6650	37.23857	3.7238572
6700	33.48799	3.3487985
6750	29.7093	2.9709304
6800	25.90253	2.5902528
6850	22.06766	2.2067658
6900	18.20469	1.8204694
6950	14.31364	1.4313636
7000	10.39448	1.0394483
7050	6.447236	0.6447236
7100	2.471895	0.2471895
7150	-1.53154	-0.153154

Figure 100, OEM system with DLC
valve bucket part 4

RPM	Fc (N)	friction (N)
0	227.6928	38.707776
50	227.6816	38.705872
100	227.648	38.700159
150	227.592	38.690637
200	227.5136	38.677307
250	227.4128	38.660169
300	227.2895	38.639221
350	227.1439	38.614465
400	226.9759	38.585901
450	226.7855	38.553528
500	226.5726	38.517346
550	226.3374	38.477356
600	226.0797	38.433557
650	225.7997	38.38595
700	225.4973	38.334534
750	225.1724	38.279309
800	224.8252	38.220276
850	224.4555	38.157434
900	224.0634	38.090784
950	223.649	38.020325
1000	223.2121	37.946057
1050	222.7528	37.867981
1100	222.2712	37.786096
1150	221.7671	37.700403
1200	221.2406	37.610901
1250	220.6917	37.517591
1300	220.1204	37.420472
1350	219.5267	37.319544
1400	218.9106	37.214808
1450	218.2721	37.106263
1500	217.6112	36.993909
1550	216.9279	36.877747
1600	216.2222	36.757776
1650	215.4941	36.633997
1700	214.7436	36.506409
1750	213.9707	36.375013

Figure 103, Ti-6AL-4V valve and spring K=12.06 contact and friction force part 1

1800	213.1753	36.239808
1850	212.3576	36.100794
1900	211.5175	35.957972
1950	210.6549	35.811341
2000	209.77	35.660902
2050	208.8627	35.506654
2100	207.9329	35.348597
2150	206.9808	35.186732
2200	206.0062	35.021058
2250	205.0093	34.851576
2300	203.9899	34.678285
2350	202.9481	34.501185
2400	201.884	34.320277
2450	200.7974	34.13556
2500	199.6884	33.947035
2550	198.5571	33.754701
2600	197.4033	33.558558
2650	196.2271	33.358607
2700	195.0285	33.154847
2750	193.8075	32.947279
2800	192.5641	32.735902
2850	191.2983	32.520717
2900	190.0101	32.301723
2950	188.6995	32.07892
3000	187.3665	31.852309
3050	186.0111	31.621889
3100	184.6333	31.38766
3150	183.2331	31.149623
3200	181.8105	30.907777
3250	180.3654	30.662123
3300	178.898	30.41266
3350	177.4082	30.159389
3400	175.8959	29.902309
3450	174.3613	29.64142
3500	172.8043	29.376723
3550	171.2248	29.108217
3600	169.623	28.835903

Figure 102, Ti-6AL-4V valve and spring K=12.06 contact and friction force part 2

3650	167.9987	28.55978
3700	166.352	28.279848
3750	164.683	27.996108
3800	162.9915	27.708559
3850	161.2777	27.417202
3900	159.5414	27.122036
3950	157.7827	26.823061
4000	156.0016	26.520278
4050	154.1982	26.213687
4100	152.3723	25.903286
4150	150.524	25.589077
4200	148.6533	25.27106
4250	146.7602	24.949234
4300	144.8447	24.623599
4350	142.9068	24.294156
4400	140.9465	23.960904
4450	138.9638	23.623843
4500	136.9587	23.282974
4550	134.9312	22.938297
4600	132.8812	22.58981
4650	130.8089	22.237515
4700	128.7142	21.881412
4750	126.5971	21.5215
4800	124.4575	21.157779
4850	122.2956	20.79025
4900	120.1112	20.418912
4950	117.9045	20.043766
5000	115.6754	19.664811
5050	113.4238	19.282047
5100	111.1499	18.895475
5150	108.8535	18.505094
5200	106.5347	18.110905
5250	104.1936	17.712907
5300	101.83	17.3111
5350	99.44403	16.905485
5400	97.03566	16.496061
5450	94.60488	16.082829

Figure 105, Ti-6AL-4V valve and spring $K=12.06$ contact and friction force part 3

5500	92.1517	15.665788
5550	89.67611	15.244939
5600	87.17812	14.820281
5650	84.65773	14.391814
5700	82.11493	13.959538
5750	79.54973	13.523455
5800	76.96213	13.083562
5850	74.35212	12.639861
5900	71.71971	12.192351
5950	69.0649	11.741033
6000	66.38768	11.285906
6050	63.68806	10.826971
6100	60.96604	10.364227
6150	58.22161	9.897674
6200	55.45478	9.4273128
6250	52.66555	8.9531429
6300	49.85391	8.4751645
6350	47.01987	7.9933775
6400	44.16342	7.5077819
6450	41.28457	7.0183777
6500	38.38332	6.5251649
6550	35.45967	6.0281435
6600	32.51361	5.5273135
6650	29.54515	5.0226749
6700	26.55428	4.5142278
6750	23.54101	4.001972
6800	20.50534	3.4859076
6850	17.44726	2.9660347
6900	14.36678	2.4423531
6950	11.2639	1.914863
7000	8.138613	1.3835643
7050	4.990923	0.848457
7100	1.82083	0.309541
7150	-1.37167	-0.233183

Figure 104, Ti-6AL-4V valve and spring $K=12.06$ contact and friction force part 4

RPM	Fc (N)	friction (N)
0	227.6928	22.76928
50	227.6816	22.76816
100	227.648	22.764799
150	227.592	22.759198
200	227.5136	22.751357
250	227.4128	22.741276
300	227.2895	22.728954
350	227.1439	22.714391
400	226.9759	22.697589
450	226.7855	22.678546
500	226.5726	22.657263
550	226.3374	22.633739
600	226.0797	22.607975
650	225.7997	22.579971
700	225.4973	22.549726
750	225.1724	22.517241
800	224.8252	22.482515
850	224.4555	22.44555
900	224.0634	22.406343
950	223.649	22.364897
1000	223.2121	22.32121
1050	222.7528	22.275283
1100	222.2712	22.227116
1150	221.7671	22.176708
1200	221.2406	22.12406
1250	220.6917	22.069171
1300	220.1204	22.012042
1350	219.5267	21.952673
1400	218.9106	21.891063
1450	218.2721	21.827213
1500	217.6112	21.761123
1550	216.9279	21.692792
1600	216.2222	21.622221
1650	215.4941	21.54941
1700	214.7436	21.474358
1750	213.9707	21.397066

Figure 107, Ti-6AL-4V valve, spring $k=12.06$ and DLC valve bucket contact and friction forces part 1

1800	213.1753	21.317534
1850	212.3576	21.235761
1900	211.5175	21.151748
1950	210.6549	21.065495
2000	209.77	20.977001
2050	208.8627	20.886267
2100	207.9329	20.793292
2150	206.9808	20.698077
2200	206.0062	20.600622
2250	205.0093	20.500927
2300	203.9899	20.398991
2350	202.9481	20.294815
2400	201.884	20.188398
2450	200.7974	20.079741
2500	199.6884	19.968844
2550	198.5571	19.855706
2600	197.4033	19.740328
2650	196.2271	19.62271
2700	195.0285	19.502851
2750	193.8075	19.380752
2800	192.5641	19.256413
2850	191.2983	19.129833
2900	190.0101	19.001013
2950	188.6995	18.869953
3000	187.3665	18.736652
3050	186.0111	18.601111
3100	184.6333	18.46333
3150	183.2331	18.323308
3200	181.8105	18.181046
3250	180.3654	18.036543
3300	178.898	17.8898
3350	177.4082	17.740817
3400	175.8959	17.589593
3450	174.3613	17.43613
3500	172.8043	17.280425
3550	171.2248	17.122481
3600	169.623	16.962296

Figure 106, Ti-6AL-4V valve, spring $k=12.06$ and DLC valve bucket contact and friction forces part 2

3650	167.9987	16.799871
3700	166.352	16.635205
3750	164.683	16.468299
3800	162.9915	16.299153
3850	161.2777	16.127766
3900	159.5414	15.954139
3950	157.7827	15.778271
4000	156.0016	15.600164
4050	154.1982	15.419816
4100	152.3723	15.237227
4150	150.524	15.052398
4200	148.6533	14.865329
4250	146.7602	14.67602
4300	144.8447	14.48447
4350	142.9068	14.29068
4400	140.9465	14.094649
4450	138.9638	13.896378
4500	136.9587	13.695867
4550	134.9312	13.493116
4600	132.8812	13.288124
4650	130.8089	13.080891
4700	128.7142	12.871419
4750	126.5971	12.659706
4800	124.4575	12.445753
4850	122.2956	12.229559
4900	120.1112	12.011125
4950	117.9045	11.79045
5000	115.6754	11.567536
5050	113.4238	11.342381
5100	111.1499	11.114985
5150	108.8535	10.88535
5200	106.5347	10.653473
5250	104.1936	10.419357
5300	101.83	10.183
5350	99.44403	9.9444031
5400	97.03566	9.7035656
5450	94.60488	9.4604877

Figure 109, Ti-6AL-4V valve, spring $k=12.06$ and DLC valve bucket contact and friction forces part 3

5500	92.1517	9.2151695
5550	89.67611	8.967611
5600	87.17812	8.7178121
5650	84.65773	8.4657728
5700	82.11493	8.2114932
5750	79.54973	7.9549733
5800	76.96213	7.696213
5850	74.35212	7.4352123
5900	71.71971	7.1719714
5950	69.0649	6.90649
6000	66.38768	6.6387683
6050	63.68806	6.3688063
6100	60.96604	6.0966039
6150	58.22161	5.8221612
6200	55.45478	5.5454781
6250	52.66555	5.2665547
6300	49.85391	4.9853909
6350	47.01987	4.7019868
6400	44.16342	4.4163423
6450	41.28457	4.1284575
6500	38.38332	3.8383323
6550	35.45967	3.5459668
6600	32.51361	3.2513609
6650	29.54515	2.9545147
6700	26.55428	2.6554281
6750	23.54101	2.3541012
6800	20.50534	2.0505339
6850	17.44726	1.7447263
6900	14.36678	1.4366783
6950	11.2639	1.12639
7000	8.138613	0.8138613
7050	4.990923	0.4990923
7100	1.82083	0.182083
7150	-1.37167	-0.137167

Figure 108, Ti-6AL-4V valve, spring $k=12.06$ and DLC valve bucket contact and friction forces part 4