



Prifysgol Cymru
Y Drindod Dewi Sant
University of Wales
Trinity Saint David

Student Name: Nicholas Sayers

Programme: BEng Motorsport Engineering

Module: Dissertation

Module Code: M6X00417

Lecturer: Kelvin Lake, Tim Tudor

Assignment Title: A study of the effects of Valve train friction due to valve springs

Hand out Date: 01/07/16

Submission Deadline: 25/4/2017

Contents

Introduction	3
Abstract	3
Aims and Objectives of the Project.....	4
Timing Plan.....	4
Literature Review	5
Practical Investigation	9
Creating the Model.....	10
Obtaining the Raw Data	10
Spring Rates.....	10
Model Setup.....	12
360 x 360 Matrix.....	12
360 x 360 Matrix Result.....	15
Valve Velocity and Accelerations	16
Spring Rate Calculation	18
Primary Model Results	19
Power Loss	19
Standard Spring Rates.....	19
Minimum Spring Force	21
Adaptive + 20% Force.....	24
Power Loss Comparison	26
Exhaust Graphs.....	27
Combined	27
Power Loss	27
Suggested Design.....	29
The concept	29
Advantages of an 'Air Spring'	30
Disadvantages of an 'Air Spring'	31
Possible Alternate Methods.....	32
Conclusion	33
References	34
Appendix.....	35

Introduction

Abstract

This report contains an investigation into the effects of spring forces on the friction in the Valve train of an engine. Friction in combustion engines is one of the biggest sources of wasted energy. In this modern age, the focus for all manufacturers is efficiency, with diesel engines being phased out by most companies by 2025, and petrol engine emissions being more and more closely monitored efficiency is essential. This study considers the potential gain from altering spring rates to reduce the force on the cams/valves when possible. In order to do this it is important to outline a number of key points of focus.

The key points are what will guide and therefore limit the project to a certain path or paths. For example, the time at which the spring rates can be altered are limited, at certain points there is a force that must be exerted in order for the valve train to work, but this will be further explained below.

The task

In order to be able to accurately calculate the effects of any changing in valve train design/systems it was important to be able to recreate/simulate the valve train. This was done with the creation of a 'Valve Train Friction Model' which can work out the friction caused by the spring rate on the valves. This friction model was created in excel and allowed a mathematical representation of the mechanics of a valve train. A large proportion of the knowledge needed for these calculations was gained from our lectures on engine design and powertrain over the last two years of study.

The model created in excel would allow for all of the important parts of the valve train to be predicted, these included Valve travel/lift, cam shape, spring compression, and other factors, however these needed to be validated so actual measured data was used for the cam shape/input numbers for the model.

The point of this model was to be able to calculate the effect of the spring rate on friction in an engine's valve train. The reason for doing so is to be able to theorise a better way of controlling valves in a production 4 cylinder, so that a more efficient way of doing the same job could be created, therefore potentially creating a more efficient engine, with a quantified and validated plan for improvement. Success would be defined by the ability of the model to produce results that are reasonable, and the performance/efficiency gains of the new design.

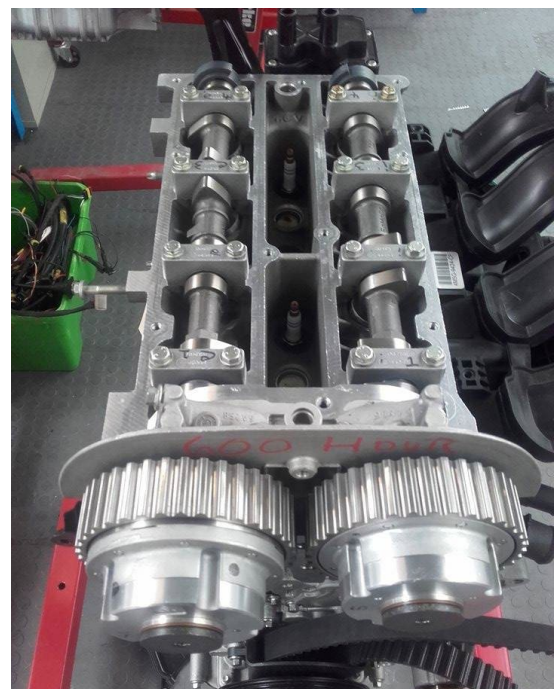


Fig.1: The valve train (from above) of a Ford Zetec 1.6 4 cylinder

Aims and Objectives of the Project

- . To look into the workings of a typical 4 cylinder Valve Train
- . Understand the best method of potentially improving the Valve train
- . Develop a method for simulating the valve train
- . Quantify any current failings
- . Quantify potential improvements
- . Obtain results of improvements for real world comparison
- . Demonstrate the potential of the suggested changes to valve train design
- . Conclude as to the relevance of the findings and discuss next steps

Timing Plan

For the project to run smoothly there were many aspects that required careful planning?

1. Most importantly was to obtain a test engine to complete the recording of data, this was to be completed by November 2016 and was thankfully made easy by the great facilities available to me at UWTSD Swansea. 01/11/16 – **Completed on Time**
2. Record raw camshaft data and begin modelling 360 x 360 matrix for valve lift. – 01/01/17
- This stage involved many hours of deliberation and patience to fully grasp the concepts at hand within the project as this was a familiar subject but being taken to a level of depth I had not previously attempted. – Data was recorded on time although 360 X 360 Matrix required more sessions of mentoring.
3. The third main stage of the project relied on good communication with mentor staff in order to ensure the course stayed on track. Regular meeting were undertaken with consistent project each week. These weeks involved gradually progressing and improving the model to become more inclusive and automated. – 01/04/17
4. This stage was the final write up of the Project. By this point the model was completed and as automated as possible within the time frame and other commitments at university. The program (Microsoft excel) was found to have some limitations in regards to automating processes and certain aspects could not be made as slick as possible. 25/04/2017 – Completed on time

Literature Review

The undertaking of a Friction model requires a vast amount of research into the theory of internal combustion engine calculations. This model focused on specifically the Valve train friction, so a good understanding of valve train dynamics and systems was essential.

The main text used was “INTERNAL COMBUSTION ENGINE MODELING” (J.I.Ramos, 1989). This book gave me plenty of information on the types of models used in internal combustion engine cycles.

When considering the use of pneumatics and air springs in a valve train, the book “Engineering applications of Pneumatics and hydraulics” by IAN C turner was of great help.

[3] http://www.hydraforce.com/svtraining/files/sdjskif_Chap8.pdf

	Power Loss (kW) (FMEP figure in kPa)	
	2000 revs/min	5000 revs/min
Bearings	0.90	5.00
Piston Skirt	0.95	5.35
Piston Rings	1.17	2.96
Valve Train	1.55	2.60
Total	4.57 (54.8)	15.91 (76.4)

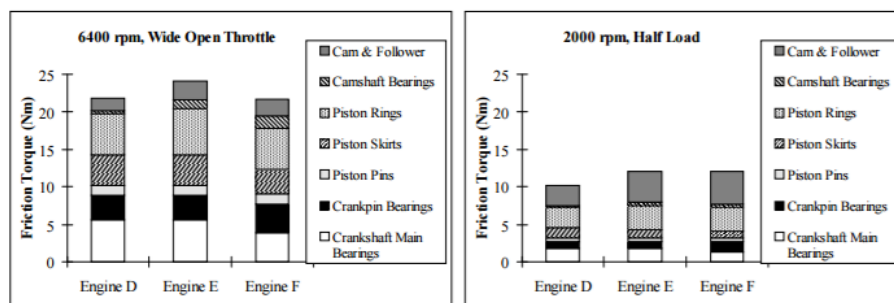
Table 1 : Results from GM FLARE software for a 5.0 litre gasoline engine for an SAE-10W/30 engine oil

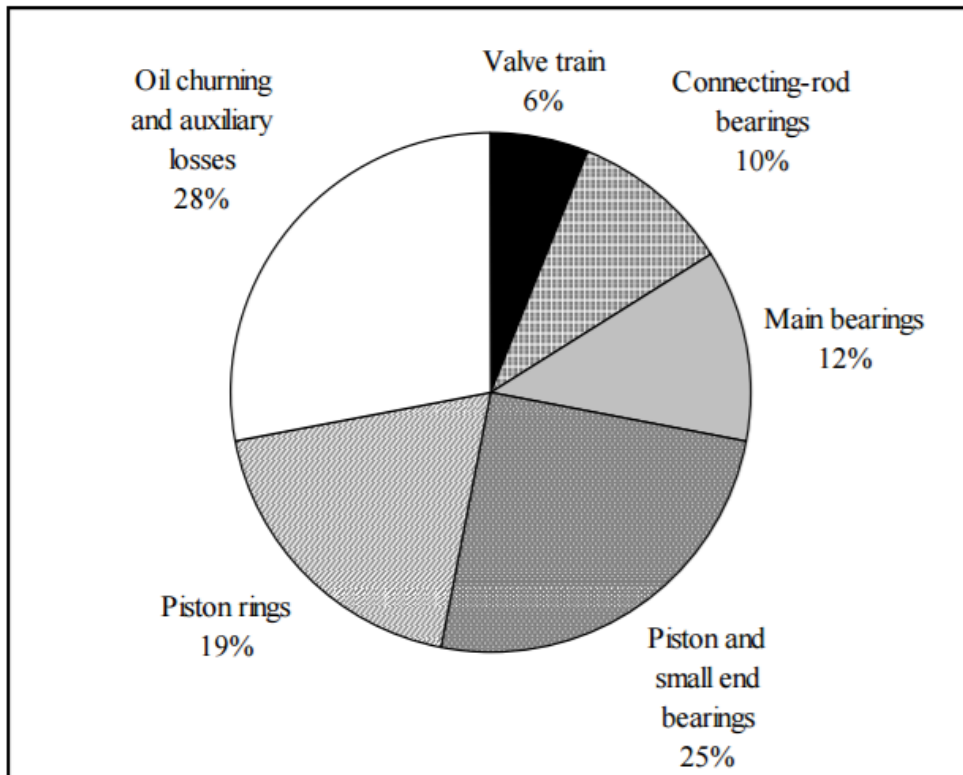
	Power Loss (kW) (FMEP in kPa)		
	SAE-10W/30	SAE-15W/40	SAE-20W/50
Bearings	0.55	0.59	0.63
Piston Assembly	0.52	0.64	0.80
Valve Train	0.38	0.29	0.14
Total	1.45 (34.8)	1.52 (36.5)	1.57 (37.7)

Table 2 : Results from Shell engine friction model for Mercedes Benz M111 2.0 litre gasoline engine at 2500 revs/min

Table 3 shows some results from the Leeds University model¹⁷, as applied to a 1.8 litre Ford Zetec engine.

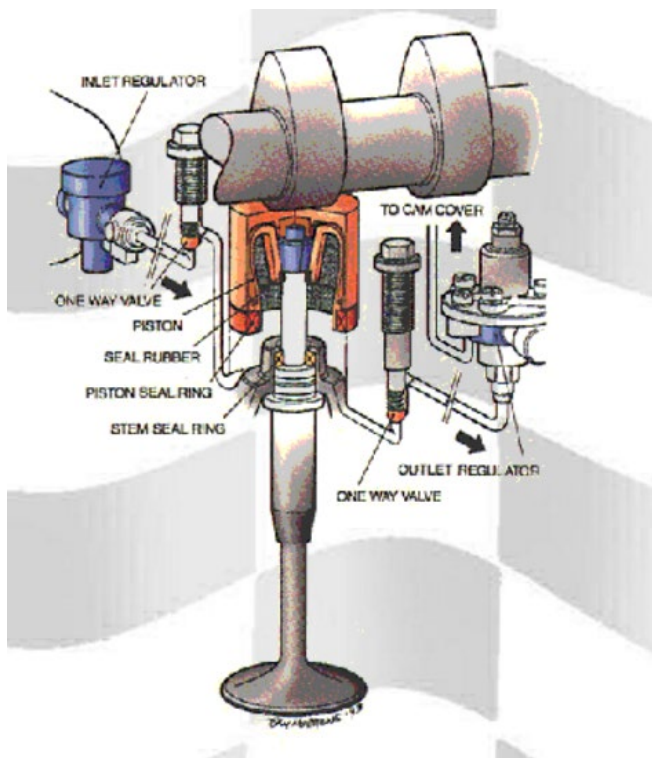
2 Research into FMEP Values





3 Typical Mechanical Losses in a Gasoline Engine

Graphs and information such as the above image show the importance of the valve train is considerably less than many other parts of the engine, however the majority of these have been tried and tested and improved to a very good degree, seemingly little effort has been put into reducing valve train friction using the method proposed in this project.



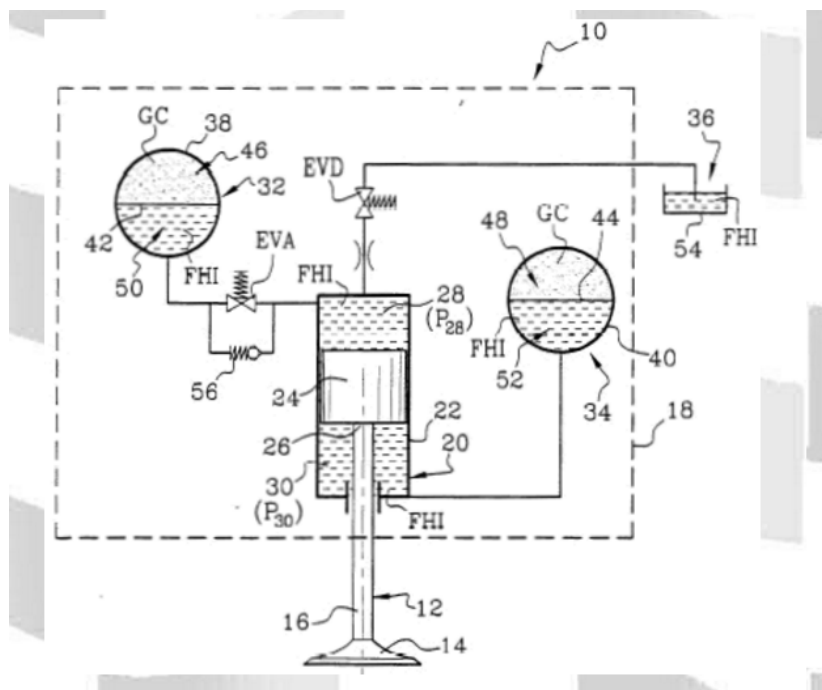
This image courtesy of the "f1 Dictionary" shows the design of the pneumatic valves used in formula one as early as the 1980's.

These were developed out of a necessity for higher specific outputs, with 3.0 litre engines being required to make in excess of 1200 bhp. The advantage of the air pressure systems was that there were no mechanical limitations of a bending springs, and no reverberations/vibrations from a spring that is changing direction/shape continually. These were found to increase the useable RPM range from 12,000 to up to 25,000 RPM. These also have controllability through the pressure of the air.

A 150 bar tank in a production car may give some concerns for health and safety

Explanation courtesy of Renault F1

"Renault has been researching computer controlled electro hydraulic valves using no camshaft, to reduce moving parts while improving valve control. Pressure in upper part of valve piston is controlled with electro actuated solenoid valve (EVA). In this way is possible to control exact time of valve opening. Oil for actuating valve movement is contained in pressurised tank". [9]



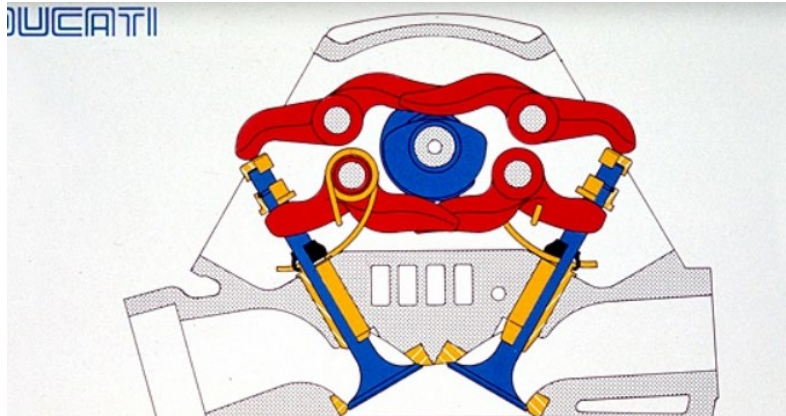


Image courtesy of 'Ducati History'

This image shows the difference in the Desmodromic system that is widely used by Ducati to increase power outputs from relatively small displacement engines. (Typically below 1300cc) these engines can produce over 200 bhp, as in the case of the Desmodeci MotoGP which is quoted at “over 250 bhp” [9].

From research however it was found that this system is not likely to suit the cause of this project.

Literary pieces on the subject were found to suggest that the Desmodromic can increase friction over a conventional Valve train system. (PIJI) The reason for the Desmodromics' use is described by Fabio Taglioni – Former Chief Director of Ducati Motorcycles Engineering as “the specific purpose of the Desmodromic system is to force the valves to comply with the timing diagram as closely as possible, in this way any energy lost is negligible, the performance curves are more uniform and dependability is way better.” This, from the man who was part of pioneering the bike shows that the purpose of this system is more aimed at performance than efficiency, more specifically at high RPM where this test will not be focusing.

Formula 1

“Formula 1 had reached the point of “one-day spring life” in the 1980s, triggering an intense search for better solutions. Several F1 engine constructors prototyped desmo systems of their own (Ferrari and Cosworth, for example) but did not pursue them.” [11]

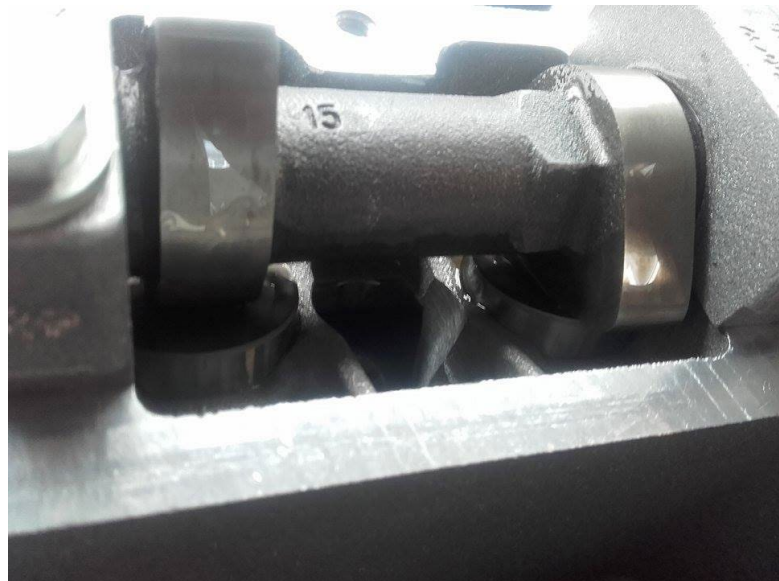
Practical Investigation

As a precursor to this study it was felt that a practical investigation into some of the forces acting upon the engine in question would help to aid in the understanding of the mathematics and theories being used in the model.

The engine that was chosen was a university engine and was therefore readily available. Having worked on these engines and conducted a number of previous tests including Gas flowing the cylinder heads it was a pleasant experience to attempt to learn more about this engine.

I began by stripping the engine down and which was essential to be able to collect the raw data for the tests, including the weights of the valve train and to measure the cam shaft.

In order to do a quick check of the forces being calculated per lobe, a reading was taken for the amount of force required to rotate the camshaft once oiled, this gave a general idea for the scale of the forces to expect the model to produce.



After a number of tests a maximum torque figure was

recorded of around 20N/M to overcome the force of the spring on a single cam lobe.

With numbers for the Friction force in the model being between 20 – 25 NM, this was at least a positive sign and gave a good feel for the engine's inner workings. Comparing the theory to actual practical results can prevent many silly mistakes and misleading results.

5 Photo of the Inlet Cam in place in the engine. The lobes were Pre oiled before any tests were carried out.



Creating the Model

Obtaining the Raw Data

In order to have a base to start from, the most important thing to measure was the cam, and specifically the shape of a cam lobe. The friction that was focused on was that on the cam lobe/Valve bucket/tappet mating faces, as there are 16 of these in a typical 4 cylinder petrol engine, there is rather a large amount of potential friction on these faces.

Measuring the cam was done with the use of an optical divider, with the cam clamped in place and reading being taken with a dial gauge at every degree of cam rotation. To allow for inaccuracies, repeat readings were taken and these were averaged out to try and remove some element of human error.

Values were recorded in mm for the shape of the cam, with the base circle of the camshaft found to be 15mm. Using the excel model it

as possible to predict the travel of the vale, when taking into account the shape of the lobe and any tolerances such as the tappet clearance. The importance of the cam lobe is it allows calculation of valve travel and therefore velocity and accelerations, which are essential for working out what the valve will be doing/forces it will be creating.

Measurements needed to then be taken for the weights of various components of the valve train, most importantly the Valves,

springs, Shims and tappets. The weight is important for the inertia forces created by the lobe/valve assembly as this is the force that directly opposes the Valve springs.



Fig.6: Measuring Camshaft Shape

Component	Weights (grams)	
	Exhaust	Intake
Spring	20.88	20.88
Collet 1	0.49	0.49
Collet 2	0.49	0.49
Bucket	24.1	24.02
Shim	7.67	7.65
Valve	33.81	37.57
Total	87.44	90.9
Total (kg)	0.08744	0.0909
Total of both in KG	0.17834	
Inertia Weight	69.33	73.01

Fig7: Component Weights

Spring Rates

The next piece of important data was the spring rates. From theoretical principles, the spring rates in a combustion engine should be a constant. Meaning no need for measuring the springs, in theory using the quoted spring rate from the manufacturer should be sufficient.

As a way of increasing the accuracy of the model, the spring rates were measured instead to A) test this theory, and B) obtain an accurate value for the spring rate.

The data in this table shows that the spring rate in N/mm is in fact not constant, as visually represented in the spring rate graph below (figure 5) the rate actually increases as more pressure is applied, however this is a linear increase for the most part.

This does complicate matters however the spring does not

operate through its' entire travel when in the engine. Between 10 and 20mm the spring rate is fairly constant, fluctuating by only 0.8N/mm and this can be neglected for the purpose of these tests.

Finally the initial spring compression was found inside the engine by comparing an uncompressed spring with one assembled in the cylinder head. The initial compression in this instance was found to be 10mm for both inlet and exhaust valves.

With these values all found it was possible to begin modelling the travel of the valve and work out a lot of the information required to predict the path it would take.

To work out the valve's travel, the most effective way was to create a matrix of 360 X 360 degrees, this was to be able to obtain the maximum downwards length created by the cam, and this would give the effective lift of the valve and is explained further in the next section.

Dince (mm)	Difference	Force in Pounds (lbs)	Force in KG	Force in Netwons	N/mm
0	2.2	0	0	0	0
1	3.4	2.2	0.997904401	9.789442171	9.789442
2	3	5.6	2.540120293	24.91858007	12.45929
3	3.2	8.6	3.900899021	38.2678194	12.75594
4	2.6	11.8	5.352396331	52.50700801	13.12675
5	3.4	14.4	6.531737896	64.07634876	12.81527
6	2.8	17.8	8.073953788	79.20548666	13.20091
7	3.3	20.6	9.344013934	91.6647767	13.09497
8	3	23.9	10.84087054	106.34894	13.29362
9	3.3	26.9	12.20164926	119.6981793	13.2998
10	3.1	30.2	13.69850586	134.3823425	13.43823
11	3.1	33.3	15.10464388	148.1765565	13.4706
12	3.2	36.4	16.5107819	161.9707705	13.49756
13	3.3	39.6	17.96227921	176.2099591	13.55461
14	3.3	42.9	19.45913581	190.8941223	13.63529
15	3.6	46.2	20.95599242	205.5782856	13.70522
16	3.6	49.8	22.58892689	221.5973728	13.84984
17	3.6	53.4	24.22186136	237.61646	13.97744
18	3.4	57	25.85479584	253.6355472	14.09086
19	3.6	60.4	27.39701173	268.7646851	14.14551
20	3.8	64	29.0299462	284.737723	14.23919
21	3.8	67.8	30.75359926	301.6928087	14.36632
22	3.6	71.6	32.47725232	318.6018452	14.4819
23	4	75.2	34.11018679	334.6209324	14.54874
24	3.8	79.2	35.92455843	352.4199182	14.68416
25	4.6	83	37.64821148	369.3289546	14.77316
26	4.2	87.6	39.73473887	389.7977883	14.99222
27	4.8	91.8	41.63982909	408.4867233	15.12914
28	11.8	96.6	43.81707505	429.8455063	15.35163
29	-108.4	108.4	49.16947138	482.3525143	16.63285

Fig.4: Recorded spring data

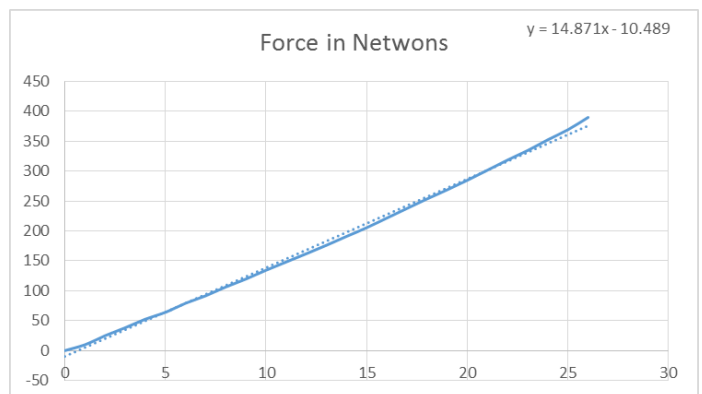


Fig.5: Spring Rate Graph In required area of motion

Model Setup

360 x 360 Matrix

In order to work out the valve lift it is important to work out the distance known as OB, this is the length from the centre of the camshaft, to the furthest point from the centre of the cam, along the vertical axis. In simpler terms this is how far down the edge of the cam is from the centre point of the camshaft (axis of rotation). (Highlighted with the RED arrow in figure 5)

To obtain this distance OB, it was essential to think about what OB actually is. This distance (referred to as the downwards length in the vertical axis), is the vertical component of the distance and angle at the contact point. For instance if the camshaft is rotated a certain angle, (in this case Phi (ϕ)) (Blue arrow) (ϕ) is chosen as the symbol for camshaft rotation), and there is another angle to consider (known as theta (θ)) which is the angle at which the contact occurs. The point at which OB is measured, is the maximum value, which will be the point furthest from the centre of camshaft rotation in the vertical direction.

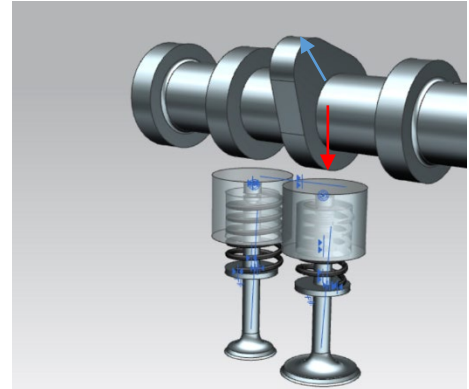


Fig.8: Figure of Cam and valve assembly

Explain more simply, Phi (ϕ) is the angle that the camshaft itself is rotated from the vertical. (0 = up, 180 = down) and the angle Theta (θ) is the angle within the cam from the cam rotation angle, to the point of contact. If this distance can be obtained at any degree of the cam being turned, at any degree within the cam, it is possible to obtain the maximum value, i.e. the one angle where the downwards distance is greatest. This downwards distance is what would be in contact with the Tappet/Valve, and is therefore what will create valve travel.

To do this the formula for the vertical distance needed to be written into a matrix of 360 (degrees of ϕ) x by 360 (degrees θ), this gives the vertical distance at any cam rotation, and any angle within the cam. The formula from the matrix is as follows
=A3*COS(RADIANS((180-(\$C3+D\$2))))

\$A3 is the Measured cam shape.

COS(Radians is telling the model to consider the vertical aspect, to work out the maximum vertical distance.

The 180 in the formula changes the direction of the vertical calculation, without this, it would work out the highest vertical distance in the upwards direction.

\$C3 + D\$2 Considers the Cam rotation and the Angle within the cam, for all angles from 0 – 360 degrees.

The difficulty of this was programming the model to alter to the relative values, the use of the \$ command in the correct place allows the model to change to the next degree value of one column, and ensures that this change is only in one direction. In this way, the model is automated once the original formula is written in. This means that with this Model for OB, if another cam is chosen, or if measurements are redone, all that requires changing is the A column. These Formulae and columns are shown below in figure 6.

Fig.9: The OB Maximum Matrix for Valve Travel

	A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T
1		CamLift	theta	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
2	s	23.812	8.812	01=\$A3*CO	-23.81	-23.6	-23.78	-23.75	-23.72	-23.68	-23.63	-23.58	-23.52	-23.45	-23.37	-23.29	-23.2	-23.1	-23	-22.89
3		23.766	8.766	1	-23.76	-23.75	-23.73	-23.71	-23.68	-23.64	-23.59	-23.53	-23.47	-23.4	-23.33	-23.25	-23.16	-23.06	-22.96	-22.85
4		23.68	8.68	2	-23.67	-23.65	-23.62	-23.59	-23.55	-23.5	-23.45	-23.39	-23.32	-23.24	-23.16	-23.07	-22.98	-22.87	-22.76	-22.65
5		23.548	8.548	3	-23.52	-23.49	-23.46	-23.42	-23.37	-23.32	-23.26	-23.19	-23.12	-23.03	-22.94	-22.85	-22.75	-22.64	-22.52	-22.4
6		23.393	8.393	4	-23.34	-23.3	-23.26	-23.22	-23.17	-23.1	-23.04	-22.96	-22.88	-22.79	-22.7	-22.6	-22.49	-22.37	-22.25	-22.12
7		23.195	8.195	5	-23.11	-23.07	-23.02	-22.97	-22.91	-22.84	-22.77	-22.69	-22.6	-22.51	-22.4	-22.3	-22.18	-22.06	-21.93	-21.8
8		22.9125	7.9125	6	-22.73	-22.74	-22.69	-22.63	-22.56	-22.49	-22.41	-22.33	-22.23	-22.13	-22.02	-21.91	-21.79	-21.66	-21.53	-21.39
9		22.722	7.722	7	-22.55	-22.5	-22.44	-22.38	-22.3	-22.23	-22.14	-22.05	-21.95	-21.84	-21.73	-21.61	-21.48	-21.35	-21.21	-21.07
10		22.468	7.468	8	-22.25	-22.19	-22.13	-22.06	-21.98	-21.89	-21.8	-21.7	-21.6	-21.49	-21.37	-21.24	-21.11	-20.98	-20.83	-20.68
11		22.193	7.193	9	-21.92	-21.86	-21.79	-21.71	-21.62	-21.53	-21.44	-21.33	-21.22	-21.11	-20.98	-20.85	-20.72	-20.58	-20.43	-20.27
12		21.911	6.911	10	-21.58	-21.51	-21.43	-21.35	-21.26	-21.16	-21.06	-20.95	-20.84	-20.72	-20.59	-20.46	-20.32	-20.17	-20.02	-19.86
13		21.622	6.622	11	-21.22	-21.15	-21.07	-20.98	-20.89	-20.78	-20.68	-20.56	-20.44	-20.32	-20.19	-20.05	-19.9	-19.75	-19.6	-19.43
14		21.349	6.349	12	-20.88	-20.8	-20.71	-20.62	-20.52	-20.42	-20.3	-20.19	-20.06	-19.93	-19.79	-19.65	-19.5	-19.35	-19.19	-19.02
15		21.051	6.051	13	-20.51	-20.43	-20.33	-20.24	-20.13	-20.02	-19.9	-19.78	-19.65	-19.52	-19.38	-19.23	-19.08	-18.92	-18.76	-18.59
16		20.779	5.779	14	-20.16	-20.07	-19.97	-19.87	-19.76	-19.65	-19.53	-19.4	-19.27	-19.13	-18.98	-18.83	-18.68	-18.51	-18.35	-18.17
17		20.529	5.529	15	-19.83	-19.73	-19.63	-19.52	-19.41	-19.29	-19.17	-19.03	-18.9	-18.75	-18.61	-18.45	-18.29	-18.13	-17.96	-17.78
18		20.259	5.259	16	-19.47	-19.37	-19.27	-19.16	-19.04	-18.91	-18.78	-18.65	-18.51	-18.36	-18.21	-18.05	-17.89	-17.72	-17.54	-17.37
19		19.989	4.989	17	-19.12	-19.01	-18.9	-18.78	-18.66	-18.53	-18.4	-18.26	-18.12	-17.97	-17.81	-17.65	-17.48	-17.31	-17.13	-16.95
20		19.742	4.742	18	-18.78	-18.67	-18.55	-18.43	-18.3	-18.17	-18.04	-17.89	-17.74	-17.59	-17.43	-17.27	-17.1	-16.92	-16.74	-16.56
21		19.509	4.509	19	-18.45	-18.33	-18.21	-18.09	-17.96	-17.82	-17.68	-17.53	-17.38	-17.23	-17.06	-16.9	-16.72	-16.54	-16.36	-16.17
22		19.254	4.254	20	-18.09	-17.98	-17.85	-17.72	-17.57	-17.45	-17.31	-17.16	-17	-16.84	-16.67	-16.5	-16.33	-16.15	-15.96	-15.77
23		19.031	4.031	21	-17.77	-17.65	-17.52	-17.39	-17.25	-17.1	-16.96	-16.8	-16.64	-16.48	-16.31	-16.14	-15.96	-15.78	-15.59	-15.4
24		18.8135	3.8135	22	-17.44	-17.32	-17.19	-17.05	-16.91	-16.76	-16.61	-16.45	-16.29	-16.13	-15.95	-15.78	-15.6	-15.41	-15.22	-15.03
25		18.609	3.609	23	-17.13	-17	-16.87	-16.73	-16.58	-16.43	-16.28	-16.12	-15.95	-15.78	-15.61	-15.43	-15.24	-15.05	-14.86	-14.66
26		18.407	3.407	24	-16.82	-16.68	-16.54	-16.4	-16.25	-16.1	-15.94	-15.78	-15.61	-15.44	-15.26	-15.08	-14.89	-14.7	-14.5	-14.3
27		18.3201	3.3201	25	-16.6	-16.47	-16.32	-16.18	-16.02	-15.87	-15.7	-15.54	-15.36	-15.19	-15.01	-14.82	-14.63	-14.44	-14.24	-14.03
28		18.013	3.013	26	-16.19	-16.05	-15.9	-15.75	-15.6	-15.44	-15.28	-15.12	-14.95	-14.78	-14.61	-14.43	-14.24	-14.05	-13.86	-13.65
29		17.831	2.831	27	-15.89	-15.74	-15.6	-15.44	-15.28	-15.12	-14.95	-14.78	-14.61	-14.43	-14.24	-14.05	-13.86	-13.66	-13.46	-13.25
30		17.671	2.671	28	-15.6	-15.46	-15.3	-15.15	-14.99	-14.82	-14.65	-14.48	-14.3	-14.11	-13.92	-13.73	-13.54	-13.34	-13.13	-12.92
31		17.498	2.498	29	-15.3	-15.15	-15	-14.84	-14.68	-14.51	-14.33	-14.16	-13.97	-13.79	-13.6	-13.4	-13.21	-13	-12.8	-12.59
32		17.332	2.332	30	-15.01	-14.86	-14.7	-14.54	-14.37	-14.2	-14.02	-13.84	-13.66	-13.47	-13.28	-13.08	-12.88	-12.68	-12.47	-12.26
33		17.1695	2.1695	31	-14.72	-14.56	-14.4	-14.23	-14.06	-13.89	-13.71	-13.53	-13.34	-13.15	-12.96	-12.76	-12.56	-12.35	-12.14	-11.93
34		17.021	2.021	32	-14.43	-14.28	-14.11	-13.94	-13.77	-13.59	-13.41	-13.23	-13.04	-12.85	-12.65	-12.45	-12.24	-12.04	-11.82	-11.61
35		16.8745	1.8745	33	-14.15	-13.99	-13.82	-13.65	-13.48	-13.3	-13.11	-12.93	-12.74	-12.54	-12.34	-12.14	-11.93	-11.72	-11.51	-11.29
36		16.7595	1.7595	34	-13.89	-13.73	-13.56	-13.38	-13.21	-13.02	-12.84	-12.65	-12.45	-12.26	-12.06	-11.85	-11.64	-11.43	-11.21	-11
37		16.6165	1.6165	35	-13.61	-13.44	-13.27	-13.09	-12.91	-12.73	-12.54	-12.35	-12.15	-11.95	-11.75	-11.54	-11.33	-11.12	-10.9	-10.68
38		16.4985	1.4985	36	-13.35	-13.18	-13	-12.82	-12.64	-12.45	-12.26	-12.07	-11.87	-11.67	-11.46	-11.25	-11.04	-10.82	-10.61	-10.38
39		16.381	1.381	37	-13.08	-12.91	-12.73	-12.55	-12.36	-12.17	-11.98	-11.78	-11.58	-11.38	-11.17	-10.96	-10.75	-10.53	-10.31	-10.09
40		16.27	1.27	38	-12.82	-12.64	-12.46	-12.28	-12.09	-11.9	-11.7	-11.5	-11.3	-11.1	-10.89	-10.67	-10.46	-10.24	-10.02	-9.792
41		16.161	1.161	39	-12.56	-12.38	-12.2	-12.01	-11.82	-11.63	-11.43	-11.23	-11.02	-10.81	-10.6	-10.39	-10.17	-9.95	-9.726	-9.499
42		16.1115	1.1115	40	-12.34	-12.16	-11.97	-11.78	-11.59	-11.39	-11.19	-10.99	-10.78	-10.57	-10.36	-10.14	-9.919	-9.696	-9.47	-9.241
43		15.9635	0.9635	41	-12.05	-11.86	-11.67	-11.48	-11.29	-11.09	-10.89	-10.68	-10.47	-10.26	-10.05	-9.828	-9.607	-9.383	-9.156	-8.927
44		15.8795	0.8795	42	-11.8	-11.61	-11.42	-11.23	-11.03	-10.83	-10.63	-10.42	-10.21	-9.993	-9.776	-9.558	-9.334	-9.108	-8.88	-8.649
45		15.792	0.792	43	-11.55	-11.36	-11.17	-10.97	-10.77	-10.57	-10.36	-10.15	-9.938	-9.723	-9.504	-9.282	-9.058	-8.831	-8.601	-8.368
46		15.7192	0.7192	44	-11.31	-11.12	-10.92	-10.72	-10.52	-10.31	-10.1	-9.892	-9.678	-9.46	-9.24	-9.016	-8.79	-8.561	-8.33	-8.096
47		15.643	0.643	45	-11.06	-10.87	-10.67	-10.47	-10.26	-10.06	-9.844	-9.631	-9.414	-9.195	-8.972	-8.747	-8.52	-8.29	-8.057	-7.822
48		15.577	0.577	46	-10.82	-10.62	-10.42	-10.22	-10.01	-9.803	-9.593	-9.374	-9.156	-8.935	-8.711	-8.484	-8.255	-8.023	-7.789	-7.552
49		15.515	0.515	47	-10.58	-10.38	-10.18	-9.973	-9.764	-9.552	-9.337	-9.119	-8.899	-8.676	-8.45	-8.222	-7.991	-7.758	-7.522	-7.284
50		15.468	0.468	48	-10.35	-10.15	-9.943	-9.734	-9.523	-9.309	-9.092	-8.872	-8.65	-8.424	-8.197	-7.967	-7.734	-7.499	-7.262	-7.024
51		15.4105	0.4105	49	-10.11	-9.906	-9.698	-9.488	-9.274	-9.058	-8.839	-8.618	-8.393	-8.166	-7.937	-7.705	-7.471	-7.235	-6.996	-6.755
52		15.361	0.361	50	-9.874	-9.667	-9.457	-9.244	-9.023	-8.811	-8.593	-8.367	-8.14	-7.911	-7.681	-7.447	-7.212	-6.974	-6.734	-6.492
53		15.292	0.292	51	-9.628	-9.419	-9.208	-8.993	-8.778	-8.562	-8.337	-8.111	-7.884	-7.656	-7.428	-7.199	-6.969	-6.728	-6.486	-6.243

As you can see, the column A is the measured cam data, this can be easily altered and the

The values produced by the matrix at this stage give the OB distance at all points, these are not yet the Maximum value. For this project all that is required is the maximum distance, as this is the furthest from the centre of the cam, i.e. the outermost downward point on the

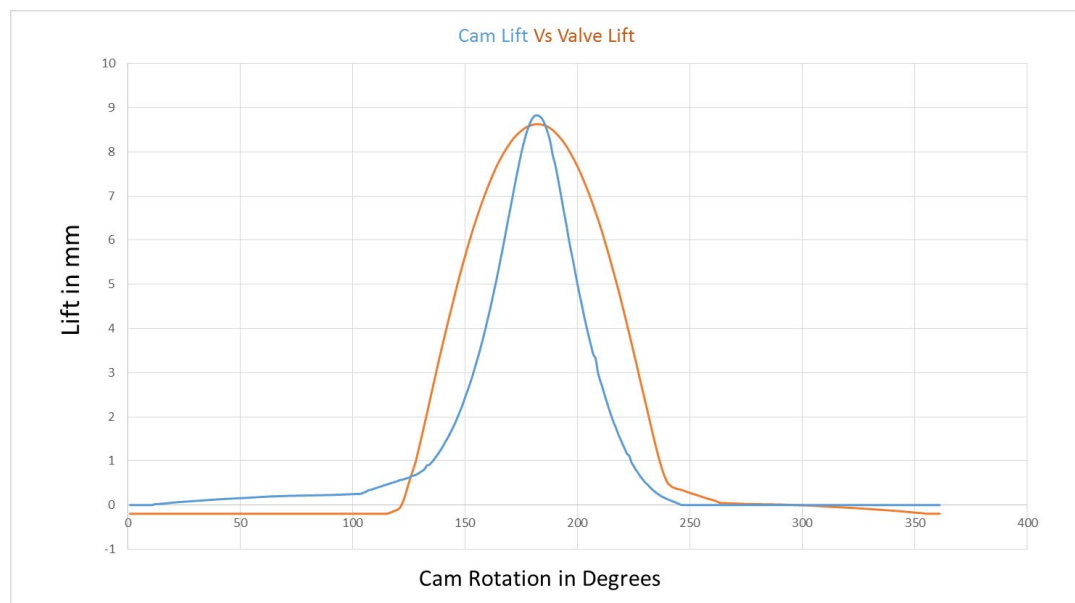
349	21.439	6.439	346	-20.8	-20.89	-20.97	-21.05	-21.11	-21.18	-21.23	-21.28	-21.32	-21.36	-21.39	-21.41	-21.43	-21.44	-21.44	-21.43	-21.41	-21.39
350	21.632	6.632	347	-21.14	-21.22	-21.29	-21.36	-21.42	-21.48	-21.53	-21.57	-21.61	-21.64	-21.66	-21.68	-21.69	-21.69	-21.68	-21.66	-21.64	-21.61
351	21.961	6.961	348	-21.48	-21.56	-21.63	-21.69	-21.75	-21.8	-21.84	-21.88	-21.91	-21.93	-21.95	-21.96	-21.96	-21.95	-21.93	-21.91	-21.88	-21.84
352	22.231	7.231	349	-21.82	-21.89	-21.96	-22.01	-22.07	-22.11	-22.15	-22.18	-22.2	-22.22	-22.23	-22.23	-22.23	-22.22	-22.19	-22.16	-22.11	-22.07
353	22.491	7.491	350	-22.15	-22.21	-22.27	-22.32	-22.37	-22.41	-22.44	-22.46	-22.48	-22.49	-22.49	-22.49	-22.48	-22.46	-22.44	-22.41	-22.37	-22.32
354	22.74	7.74	351	-22.46	-22.52	-22.57	-22.62	-22.65	-22.68	-22.71	-22.73	-22.74	-22.74	-22.74	-22.73	-22.71	-22.68	-22.65	-22.62	-22.57	-22.52
355	22.981	7.981	352	-22.76	-22.81	-22.86	-22.89	-22.93	-22.95	-22.97	-22.98	-22.98	-22.98	-22.97	-22.95	-22.93	-22.89	-22.86	-22.81	-22.76	-22.71
356	23.192	8.192	353	-23.02	-23.06	-23.1	-23.14	-23.16	-23.18	-23.19	-23.19	-23.18	-23.16	-23.14	-23.1	-23.06	-23.02	-22.97	-22.91	-22.84	-22.77
357	23.389	8.389	354	-23.26	-23.3	-23.33	-23.36	-23.37	-23.39	-23.39	-23.39	-23.37	-23.36	-23.33	-23.3	-23.26	-23.21	-23.16	-23.1	-23.03	-22.96
358	23.542	8.542	355	-23.45	-23.48	-23.51	-23.53	-23.54	-23.54	-23.54	-23.53	-23.51	-23.48	-23.45	-23.41	-23.37	-23.31	-23.25	-23.18	-23.11	-23.03
359	23.672	8.672	356	-23.61	-23.64	-23.66	-23.67	-23.67	-23.67	-23.66	-23.64	-23.61	-23.58	-23.54	-23.5	-23.44	-23.38	-23.31	-23.24	-23.15	-23.07
360	23.768	8.768	357	-23.74	-23.75	-23.76	-23.77	-23.76	-23.75	-23.74	-23.71	-23.68	-23.64	-23.59	-23.54	-23.48	-23.41	-23.33	-23.25	-23.16	-23.06
361	23.822	8.822	358	-23.81	-23.82	-23.82	-23.82	-23.81	-23.79	-23.76	-23.73	-23.69	-23.64	-23.59	-23.53	-23.46	-23.38	-23.3	-23.11	-23.01	-22.9
362	23.83	8.83	359	-23.83	-23.83	-23.83	-23.82	-23.8	-23.77	-23.74	-23.7	-23.65	-23.6	-23.54	-23.48	-23.39	-23.31	-23.22	-23.12	-23.02	-22.91
363																					
364	Max Travel (OB MAX)			15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15
365	Travel - Valve Clearance			14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8	14.8
366																					
367	Travel - Base (cam Lift)			0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
368	VALVE LIFT			-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2	-0.2
369																					
370	Cam Lobe			0	0	0	0	0	0	0	0	0	0	0	0.02	0.02	0.025	0.03	0.035	0.04	0.045
371																					0.05

the matrix were correct. The aim of this matrix was to produce a path for the valve travel, research of these types of graph gave an expectation of the shape that was required. After researching the typical behaviour of valve lift vs. Cam lift there were a few key areas that were focused on.

The shape of the graph produced would be very indicative of whether the model was working correctly. There are a few factors that influence the movement of the valve in relation to the camshaft shape. Firstly the valve movement should be of a similar shape to the cam, however in this particular engine a valve/tappet clearance of 0.2mm is chosen. In a rotating assembly, a clearance of this type leads to the object that is being affected, being out of phase. This gap means that any movement of the valve is delayed as this gap is taken up.

This clearance also means that there is a delay in the starting of valve movement, for a brief period there is no movement of the valve whilst cam rotation occurs. The model must take into account this clearance, for this reason, where the base circle of the camshaft is 15mm, the valve travel is shown as 14.8mm. In the real case this does not happen as the valve will be fully closed, but the benefit of this gap is that the valve spring is able to push the valve closed unopposed. If there were no valve gap, the valve would be constantly in contact with the cam lobe. Researching has highlighted that there are many issues that arise when a valve is not able to close with sufficient force so it is important that this model recreate this.

In order to create a graph of Valve Lift vs. Cam lift, these two sets of data must be isolated from the shape of the cam, i.e. by subtracting the base circle of the cam (15mm) you are left with the travel of the cam lobe, and the travel of the valve. This produces a graph as follows.



360 x 360 Matrix Result

The above graph shows the pattern that is produced for valve travel. This pattern follows the expected path for valve travel, and has the characteristics expected from research undertaken.

0 – 100 Degrees rotation: Very little change is happening, on this section of the cam shape, the distance measured is simply the base circle, give or take very small differences, the valve lift in the model at this point is negative due to the Valve clearance, in the real world situation it is not actually negative this just symbolises that there is no contact between the cam lobe and the valve bucket/tappet.

100 – 125 degrees: This is where the shape of the cam begins to change, and the valve will begin to lift, as stated earlier there is a delay and a point where the cam distance increases, but the valve does not travel. Due to the valve clearance, this means that the travel of the valve will not be in time with the shape of the cam lobe.

125 – 150 Degrees: Due to the eccentric shape of the cam shaft, the measured cam shape, and predicted valve lift begin to separate from this point. Whereas the Cam lift was measured from a single fixed point, the valve lift is being predicted at the actual point of contact, this point of contact changes angle (θ) and therefore does not maintain alignment with the cam. It is for this reason that the cams are shaped in this more elliptical/oval shape. This shape means that rather than just opening and closing, like the shape of the cam, the valve has a duration of lift.

150 – 180 Degrees: During this phase the cam is reaching maximum lift, but the valve does not achieve this same maximum value due to the valve clearance. However, the width of the curve of the Valve lift is far greater than that of the Cam lift.

For the second half of the cam lobe the pattern is much the same. The important part of this comparison is the so called 'area under the graph'. From 125 to 240 degrees the valve is opening. The shape of the cam lobe means that although there is only a maximum opening or around 8.8mm, for only 3 degrees of rotation, the valve spends around 40 Degrees of rotation above 8mm of opening. This means the Valve can stay open for longer and allow more air into the engine.

These patterns for the graph show an expected representation of valve travel, this gave confidence that the matrix was working, and giving back useful data for the travel of the valve. The aim of this project is to focus on the time where the cam and valve bucket/tappet are in contact with each other. This matrix has provided a model of when this occurs and for how long. The next stage was to look further into what is occurring within a rotation of the cam lobe and the accelerations and forces occurring.

Valve Velocity and Accelerations

Calculating valve velocity and accelerations are essential to work out the force being created by the cam shaft. The way in which valve springs work is to provide enough force to close the valve when needed. This force needs to be sufficient to close the valve at the highest engine speeds. In order to calculate the required spring rate for this maximum speed (and all other speeds) it is essential to be able to calculate the rate at which the valve is accelerating, as force in its simplest term is Mass * acceleration, knowing the acceleration is vital.

Acceleration is defined in the oxford dictionary as “The rate of change of velocity per unit of time.” In this case, the unit of time is degrees of rotation per second, but firstly the velocity is required.

The velocity is the speed at which the valve is travelling, for this model, this was calculated per degree of cam rotation. Therefore the velocity was simply the change in valve lift from one degree to the next.

These Acceleration and Velocity calculations produced a number of graphs giving a visual representation for the rate at which the valve moves. These were as follows.

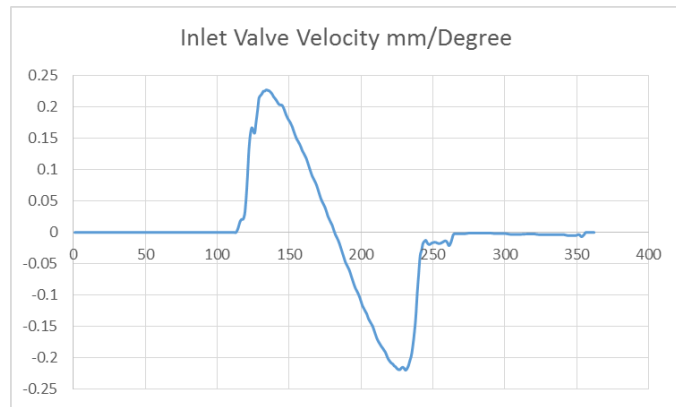
	A	B	C	D	E	F
1	Crank Angle	Cam Rotation	Cam Lift	Inlet Valve Lift	Inlet Valve Lift AVG	Inlet Valve Velocity mm/Degree
2	0	0	0	-0.2	-0.2	=E3-E2
3	2	1	0	-0.2	-0.2	0
4	4	2	0	-0.2	-0.2	0
5	6	3	0	-0.2	-0.2	0
6	8	4	0	-0.2	-0.2	0
7	10	5	0	-0.2	-0.2	0
8	12	6	0	-0.2	-0.2	0
9	14	7	0	-0.2	-0.2	0
10	16	8	0	-0.2	-0.2	0
11	18	9	0.02	-0.2	-0.2	0
12	20	10	0.02	-0.2	-0.2	0
13	22	11	0.025	-0.2	-0.2	0
14	24	12	0.03	-0.2	-0.2	0
15	26	13	0.035	-0.2	-0.2	0

Fig. 9: Velocity Calculations

As shown in the figure above, some averaging was done for the Valve lift results, the raw data was recorded numerous times and averaged out, and then that data was averaged further. Whilst this may have dampened some of the Maximum or minimum values, this choice was made as the data was drastically smoothed with such a small averaging. This smoother data allowed for some human/equipment error in the measuring and gave for extremely promising and convincing graphs.

This valve velocity is important as this is then multiplied by the rotational velocity and Mass of the Valve assembly to give a value for the force that the spring must overcome.

This figure shows the rate of change of inlet valve velocity. The horizontal axis being the Degrees of Camshaft rotation, and the vertical axis being the Velocity in millimetres per degree. This data was found to be acceptably accurate for this study. Further improvements would be possible with repeat readings.



The pattern of the graph follows very closely to that which would be expected, considering the shape of the camshaft.

The low flat points of the graph donate the Base circle of the cam shaft.

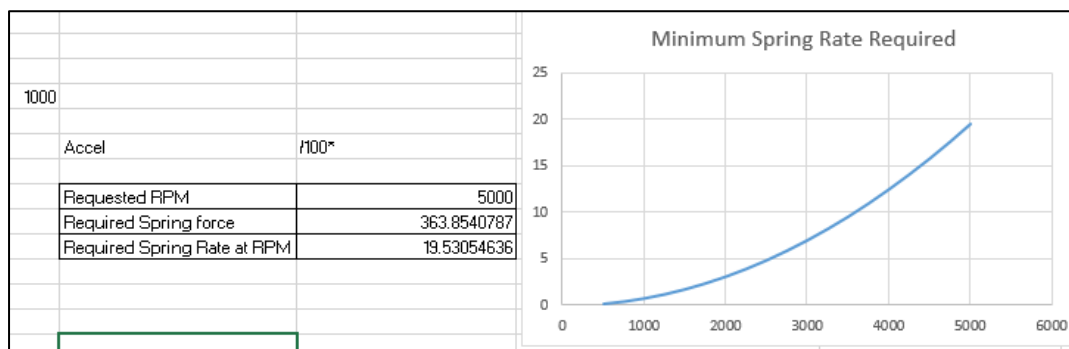
In the case of the exhaust valves, the same process was repeated for calculating velocity and acceleration, however the dimensions were altered accordingly. The smaller exhaust Valve had a lower mass than Inlet and more importantly a different cam profile, with a maximum lift value of 9.14mm as opposed to the 8.83 mm lift of the inlet camshaft.

These differences meant that another model was required for the Exhaust cam calculations, simply calculating the figures for one inlet cam lobe and multiplying by 16 to get the overall values would have given a ball-park figure, however this would not have been sufficiently accurate for a model considering such small differences in efficiency and friction.

Spring Rate Calculation

One of the key benefits of the model created is the automation of spring rate calculations. For this project to be a success there would be far more benefit from being able to vary spring rates according to engine speed. Adaptive/Variable spring rates would provide the greatest reduction in frictional losses throughout the engines RPM range.

In order to calculate the minimum spring force required to close the valves and overcome the inertia of the valve itself, a calculator was created.



This aspect of the model allows for easy calculation of the spring force required at the RPM ranges of the engine, from 1000 to 10,000 RPM engine speed. The float speed of the standard engine with standard spring rate of 14.8 NM was found to be at 4363 RPM Cam Speed for the intake, this is half the engine speed which comes to 8726RPM, and for the exhaust is 4314Cam RPM and 8629 Crank RPM respectively. These values are the theoretical maximum rpm of the engine, where it is not possible for it to overcome the inertia of the Valves. Meaning that the exhaust valves cannot be opened/closed in time, dramatically limiting efficiency and preventing further RPM.

With a variable rate spring, the total amount of spring rate acting upon the engine from 1000, to 10,000 rpm changes from 282.5N/mm to 140.7 N/mm. This gives an indication of the level of extra force put on the valve train by the constant high spring rate.

Exhaust float speed at 4314	Engine Float Speed	8629.751901
Float RPM^2	18618154.47	4314.875951
	Spring Force Check	281.65674

Primary Model Results

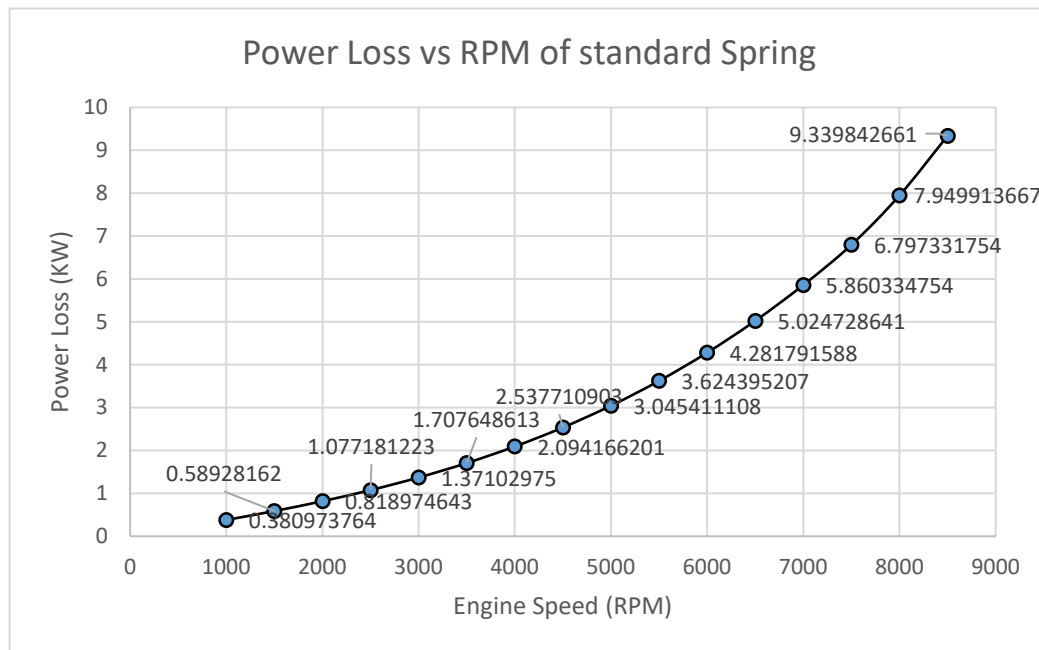
Power Loss

Standard Spring Rates

Constant Spring Rate				
Crank RPM	Cam RPM	Average Max contact Force	Inlet Cam FMEP total * 8 Cam Lobes	Power Loss in KW
1000	500	273.1729691	0.057146065	0.380973764
1500	750	268.6402515	0.058928162	0.58928162
2000	1000	262.2944469	0.061423098	0.818974643
2500	1250	254.1355553	0.064630873	1.077181223
3000	1500	244.1635766	0.068551488	1.37102975
3500	1750	232.3785109	0.073184941	1.707648613
4000	2000	218.7803582	0.078531233	2.094166201
4500	2250	203.3691184	0.084590363	2.537710903
5000	2500	186.1447916	0.091362333	3.045411108
5500	2750	167.1073778	0.098847142	3.624395207
6000	3000	146.2568769	0.10704479	4.281791588
6500	3250	123.593289	0.115955276	5.024728641
7000	3500	99.11661407	0.125578602	5.860334754
7500	3750	72.82685211	0.135946635	6.797331754
8000	4000	44.72400312	0.149060881	7.949913667
8500	4250	14.8080671	0.164820753	9.339842661
			Total Across RPM Range	56.5007161
			Total From 1k to 3k RPM	4.237441

In order to quantify any benefit of lowering friction a base line was required. The above table of data shows that recorded values for power loss at each 500RPM engine speed amount to a total of 56.6KW. This value has no real quantity in the real world application as power loss is a constant, however for the purpose of comparison it is very useful. The work done by the engine to turn the camshafts is effectively the power used simply to rotate the camshafts, and equates to the power loss. The total of the recorded Values at the RPM points calculated comes to a total of 56 KW, from the sum of various points.

The important Values are the RPM points themselves, as the RPM increases the Friction increases as expected, however, the points of interest for the model and the following graphs are particularly the lower rpm ranges, especially between 1000 and 3000/4000 RPM/. This is the point where the engine will be operating for most of its lifespan. This RPM range is also the range that the car will be run in an emissions test typically. These tests consist of low loading of the engine and low RPM's. This key area is of enormous focus to manufacturers and calibration engineers, this vital section has a large effect on fuel economy and emissions, reducing unnecessary friction here is expected to have the best result.



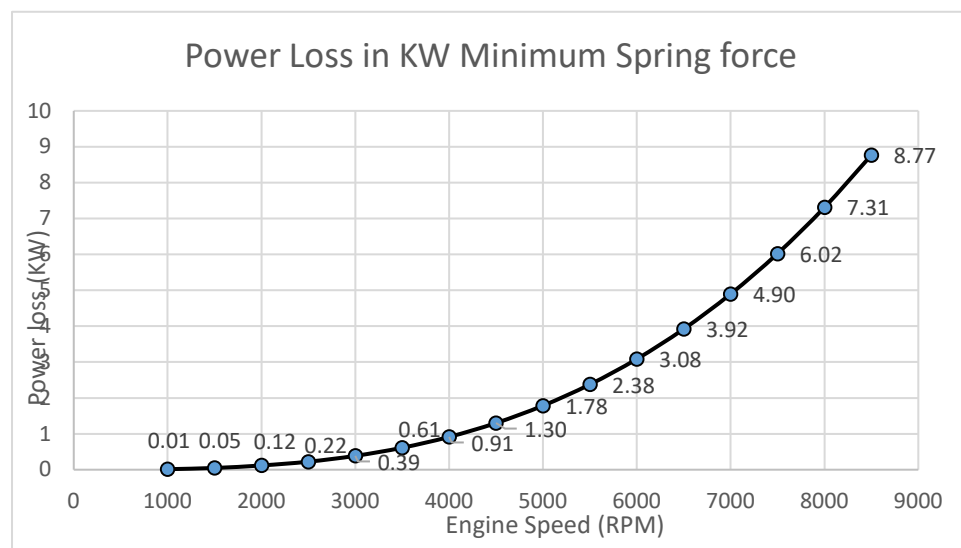
The above graph shows the Power loss due to friction as the engine RPM increases, the graph shape is correct and as expected. The most interesting part of the graph to focus on is the initial section, below 6,000 RPM (3000 Cam RPM) where the friction is certainly present. This friction starts at a value of 0.56KW from 1,000RPM and climbs steadily. At this point the Valve spring is stiff enough to return the valve when the engine is at 8200+ RPM, due to this there is a lot of excess force on the cam lobe. As is shown in the comparison of the “ideal” valve spring rate below, this value for power loss is dramatically higher than is theoretically possible.

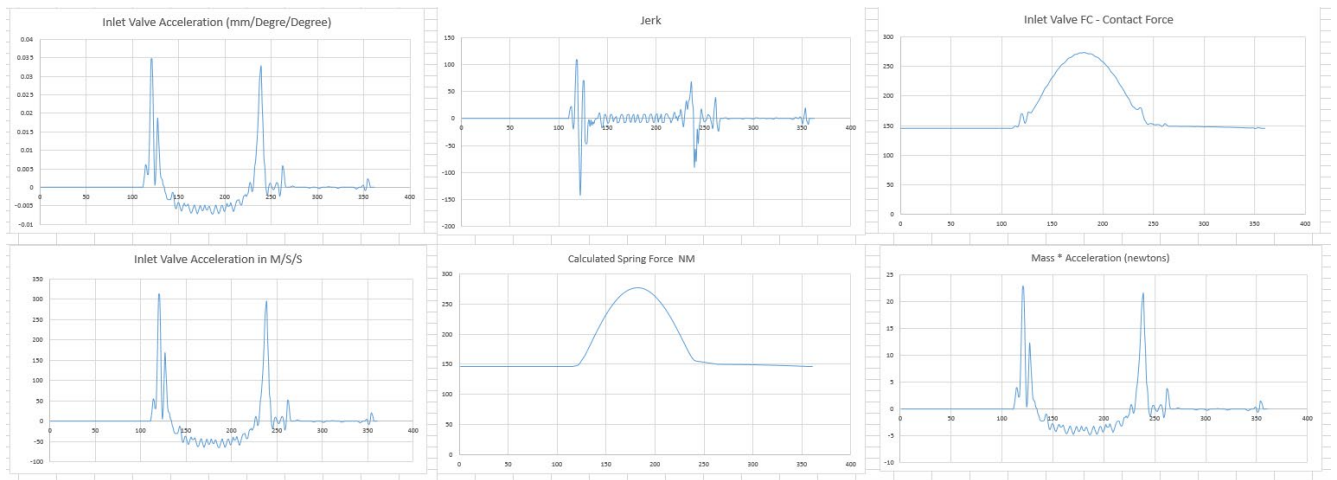
Minimum Spring Force

Adaptive Spring Rate				
Crank RPM	Cam RPM	Average Max contact Force	Inlet Cam FMEP total * 8 Cam Lobes	Power Loss in KW Minimum Spring force
1000	500	-0.043378725	0.002141363	0.014275753
1500	750	-0.399760821	0.004818322	0.04818322
2000	1000	2.752503882	0.008863399	0.118178653
2500	1250	-0.247850342	0.013384803	0.223080044
3000	1500	-0.353181822	0.019274323	0.385486455
3500	1750	-0.470895989	0.026235041	0.612150956
4000	2000	-0.619606194	0.034265923	0.913757935
4500	2250	-0.780699086	0.043368002	1.301040068
5000	2500	-0.954174664	0.05354128	1.784709337
5500	2750	-1.158646281	0.064784721	2.37543978
6000	3000	-1.375500585	0.077099361	3.083974423
6500	3250	-1.604737575	0.090485198	3.921025246
7000	3500	-1.864970604	0.104941199	4.897255941
7500	3750	-2.137586319	0.120468398	6.023419881
8000	4000	-2.422584722	0.137066795	7.310229047
8500	4250	-2.738579162	0.154735355	8.768336782
Total Power Lost Across RPM Range (KW)				41.78054352
Power Loss Between 1k and 3k RPM				0.789204125

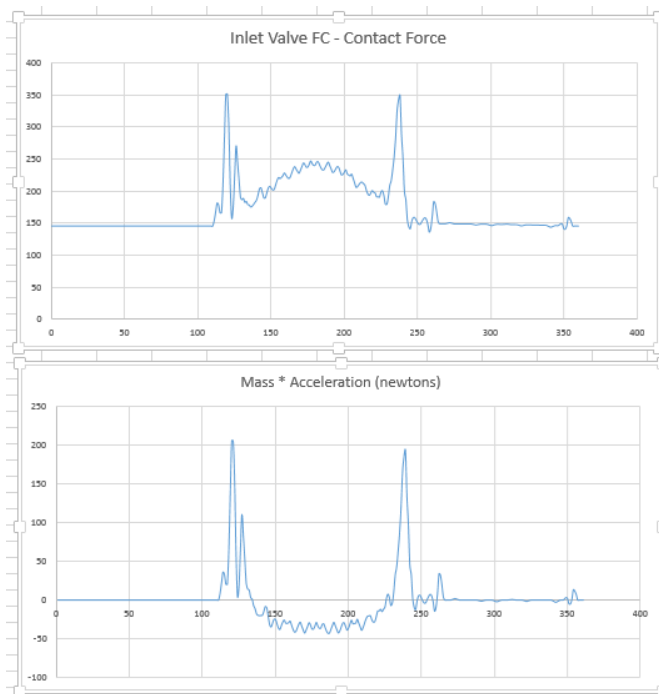
Calculating the minimum spring force was done considering the float speed of the engine. The model created a number of graphical outputs to highlight which forces were acting on the cam lobe, most importantly for calculating the spring rate is the contact force. Contact force is the amount of force that the spring is applying to the tappet and thus the camshaft and is how capable the spring is of returning the valve to its closed position and resisting the valve/cam inertia. In this way, calculating the minimum spring force for valve closing is done by finding the point where the Valve inertia force (Mass * acceleration) is equal to the reactionary force of the spring. I.e. when the force opening the valve = the force closing the vale. It is at this point where in theory, the minimum friction will occur.

In theory this would mean that the value for friction would be zero at all rpm's if the spring rate was matched exactly to the inertia force of the valve. However in practise the data recorded does not give such straight line results. The friction increases dramatically with RPM even if the contact force does not, this is due to the friction being a squared factor of the RPM, meaning that the effect of friction as RPM increases will increase dramatically, from a friction efficiency standing, high RPM is a very detrimental to the performance of Internal Combustion.

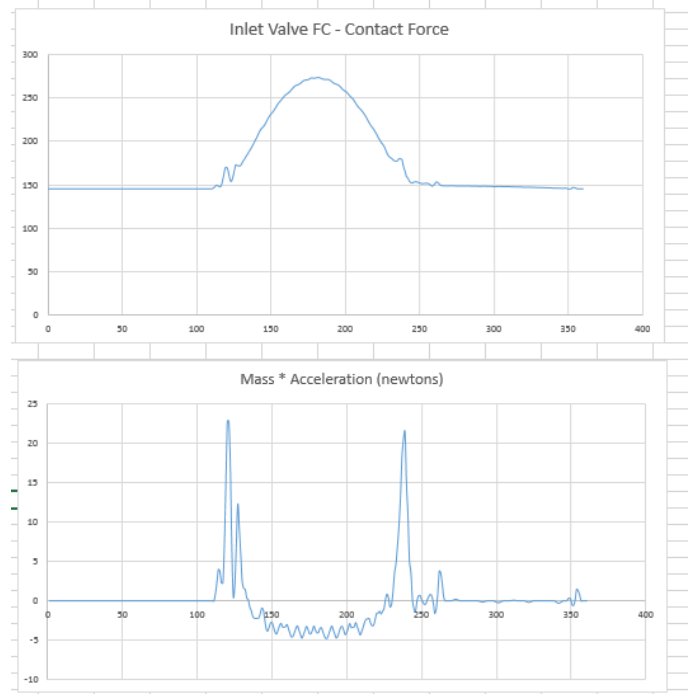




An example of the many graphs produced to analyse forces on the valve train

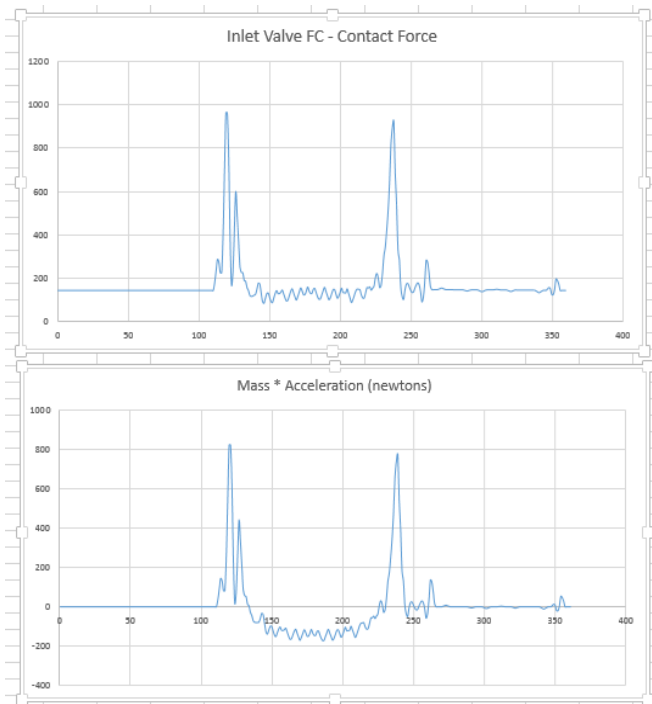


*Contact force and Mass * acceleration Graphs of the intake cam at 1500 RPM Decreasing Contact Force curve and Increasing M*A curve.*

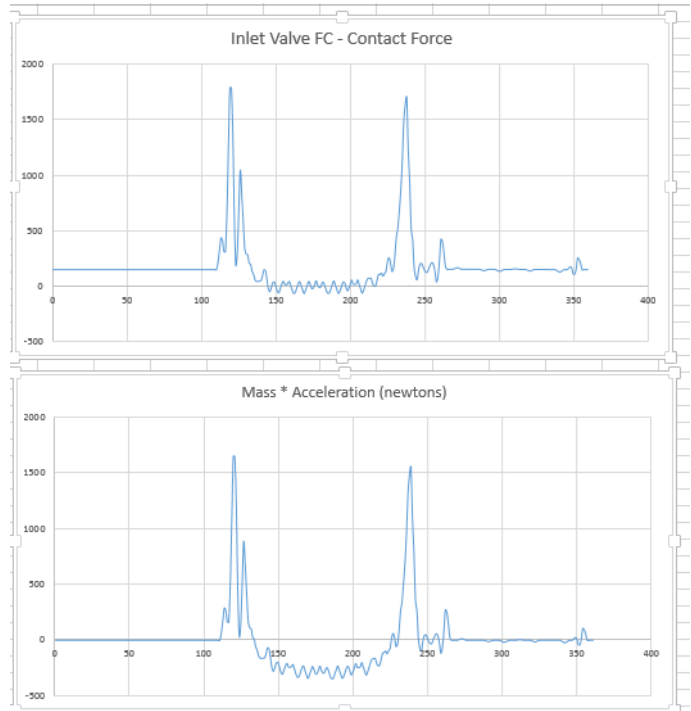


*Contact force and Mass*acceleration Graphs of the Intake Cam at 500 rpm, visibly large contact force curve and low M*A curve.*

Contact Forces Continued



Contact force at 3000 RPM

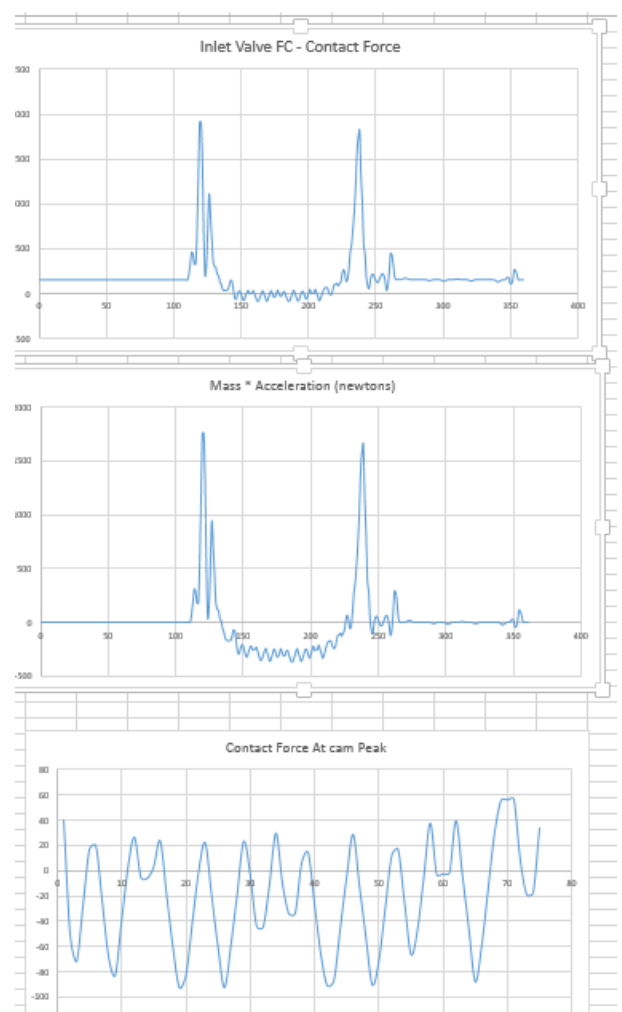


Contact force at 4250RPM

These graphs show the altering trend of the contact force curves as the RPM increases up to the float speed RPM. Initially at low rpm there is a strong contact force produced by the spring at the surface of the cam lobe. This equates to the larger amount of friction at lower Engine/Cam Speeds.

The three graphs on the right show the overall forces on a cam lobe at 4400 rpm, slightly above the calculated float speed of around 4362 RPM. At this point the graph of "Contact force at cam peak" shows that there is more area under the horizontal axis on the graph. This indicates a negative force on the cam lobe. As this is not possible in the real world application, the true meaning of this is that the spring can no longer overcome the valve's force. At this point the maximum mechanical rev limit of the engine is reached.

The problematic character of the variable valve spring will be to maintain sufficient clamping force and spring force to return the valves in a controlled fashion allowing a good seal, whilst reducing friction as much as possible.

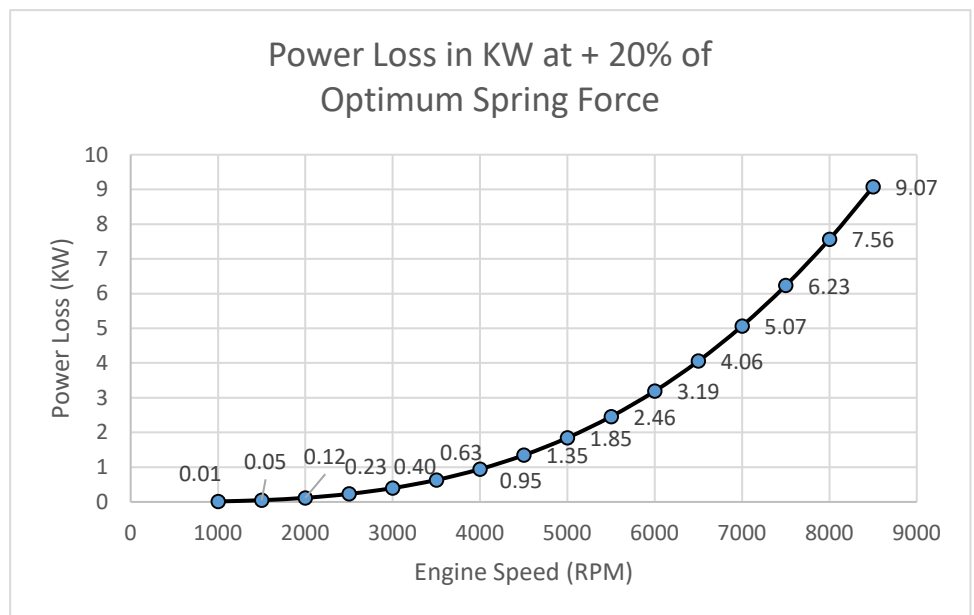


4400RPM Contact and Mass * Acceleration Graphs

Adaptive + 20% Force

1	Adaptive Spring Rate				
2	Crank RPM	Cam RPM	Average Max contact Force	Inlet Cam FMEP total * 8 Cam Lobes	Power Loss in KW +20% Force
3	1000	500	0.682541965	0.002215054	0.014767024
4	1500	750	1.112348902	0.004984667	0.049846667
5	2000	1000	2.752503882	0.008863399	0.118178653
6	2500	1250	4.29380731	0.013848066	0.230801095
7	3000	1500	6.18754973	0.019941852	0.39883703
8	3500	1750	8.433731144	0.027144756	0.633377649
9	4000	2000	11.01001553	0.035453596	0.945429222
10	4500	2250	13.93873891	0.044871554	1.346146629
11	5000	2500	17.21990128	0.055398632	1.846621063
12	5500	2750	20.83116663	0.067031644	2.457826945
13	6000	3000	24.79487096	0.079773775	3.190951006
14	6500	3250	29.11101429	0.093625025	4.057084433
15	7000	3500	33.7572606	0.10858221	5.067169807
16	7500	3750	38.75594589	0.124648514	6.2324257
17	8000	4000	44.10707018	0.141823937	7.563943303
18	8500	4250	49.78829744	0.160105294	9.072633348
19				Total Power Lost Across RPM Range (KW)	43.22603957
20				Total from 1k to 3k RPM	0.812430469

As the “optimum” spring force would achieve the results shown below at valve float speed, this would not be a reliable condition to maintain with zero margins for error. Allowing for a percentage overlap of spring force vs rpm would be the most sensible option. In typical valve train technology a safety factor of 10% is undertaken, however with such an experimental method of valve control being proposed, 20% extra force was taken to be a sensible margin.



This 20% extra however was not directly reflected in the overall power loss. Whereas ultimate peak power loss has risen from 8.46 to 9.07, (107% increase) of the theoretical minimum amount, the overall frictional losses are substantially lower.

Total RPM	82.12243	Total RPM	85.71298	Total RPM	91.99614	Total RPM	110.5562
1 - 3k	1.54751	1 - 3k	1.611057	1 - 3k	1.729251	1 - 3k	8.648234
% Original Total	74.28119		77.5289		83.21213		100
%Original 1 - 3k	17.89395		18.62874		19.99542		100

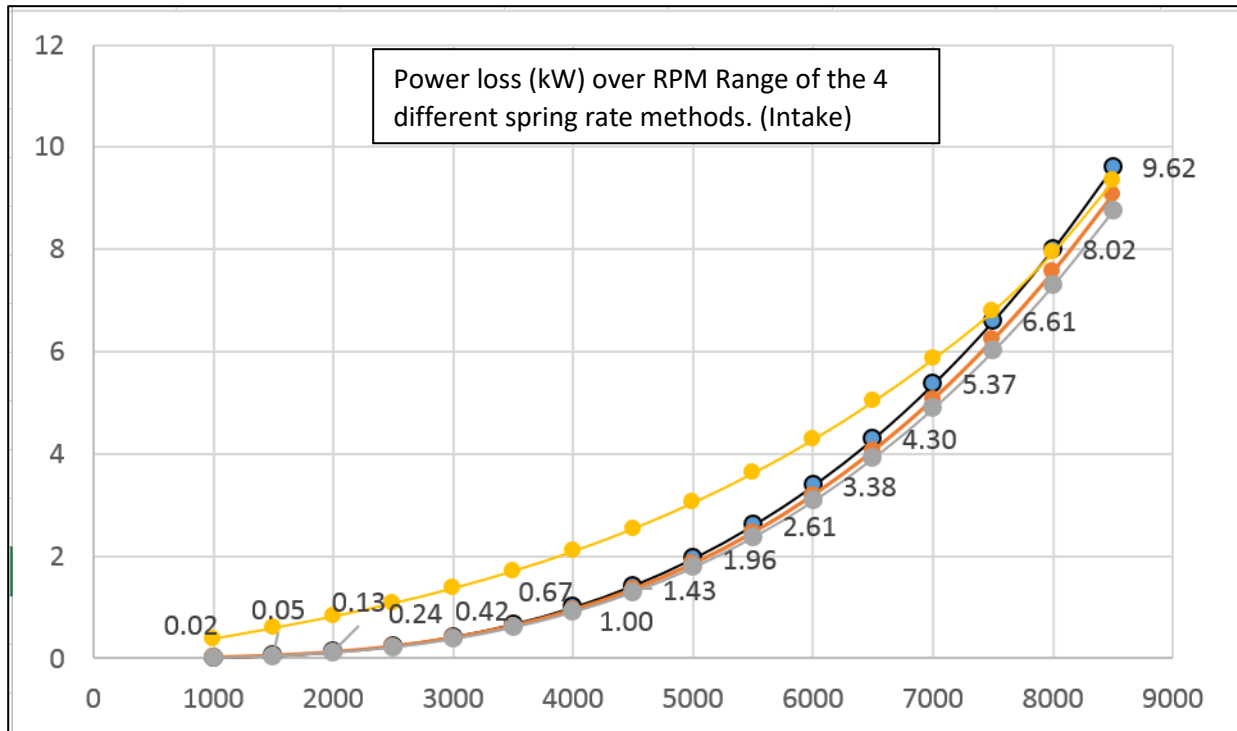
As mentioned, the theoretical minimum vale gives a much lower power loss across the entire rev range and the frictional losses from 1000 – 3000 rpm are only 17% of the original valve spring. The values set at 120% of the minimum stiffness would provide the valve with sufficient clamping force to operate safely without alarming damage and the drop in performance/efficiency is not linear. The 20% increase in spring rates at each individual rpm has only decreased efficiency by >0.8% in the 1 to 3k rev range, or >0.5kw. Over the entire rev range the increase is only around 3.6kw and this is still dramatically improved from the 110.5562KW power loss recorded at each rpm sport with std. spring setups. As a direct comparison in the area in which the Emissions tests would be carried out, the project seems to have merit to it.

With a potential to realistically produce 1/5th of the original frictional losses in the valve train, this could reduce the valve train losses from a typical 5 – 10% (typically around 6 – 7%) down to as little as 1 – 2% in a production engine.

The figures below display the data for the 140% stiffness of the valve spring. This in practise will more than likely be an excessive safety margin, however the benefits in the lower region of the RPM range show that even with the 140% margin, the gains are still significant and the 19.99542% of the original friction between 1- 3k will be effective and noticeable in real world tests.

Adaptive Spring Rate				
Crank RPM	Cam RPM	Average Max contact Force	Inlet Cam FMEP total * 8 Cam Lobes	Power Loss in KW +40% Force
1000	500	1.416422637	0.002349308	0.015662054
1500	750	2.635002239	0.005285943	0.052859431
2000	1000	5.665690549	0.009397232	0.12529643
2500	1250	8.852641483	0.014683175	0.24471959
3000	1500	12.74780374	0.021143773	0.422875452
3500	1750	17.35117731	0.028779024	0.671510555
4000	2000	22.6627622	0.037588929	1.002371441
4500	2250	28.68255841	0.047573488	1.42720465
5000	2500	35.41056593	0.058732702	1.957756721
5500	2750	42.84678478	0.071066569	2.605774196
6000	3000	50.99121494	0.08457509	3.383003614
6500	3250	59.84385643	0.099258266	4.301191516
7000	3500	69.40470923	0.115116095	5.372084443
7500	3750	79.67377335	0.132148579	6.607428934
8000	4000	90.65104879	0.150355716	8.01897153
8500	4250	102.3365355	0.169737508	9.618458771
			Total Power Lost Across RPM Range (KW)	45.82716933
			Total from 1k to 3k RPM	0.861412957

Power Loss Comparison



The above figure shows the result of the Power Loss across the RPM range of the engine, calculated using the model after working out the frictional loss (FMEP) caused by the excess spring rates.

The orange line represents the standard spring system, at the original spring rate. This line clearly indicates the region in which engine friction is being increased by the traditional method of valve spring control. The remaining curves on the graph represent the power loss across all RPM ranges below Valve float, this is the effective RPM range of the engine and where the car will be driven.

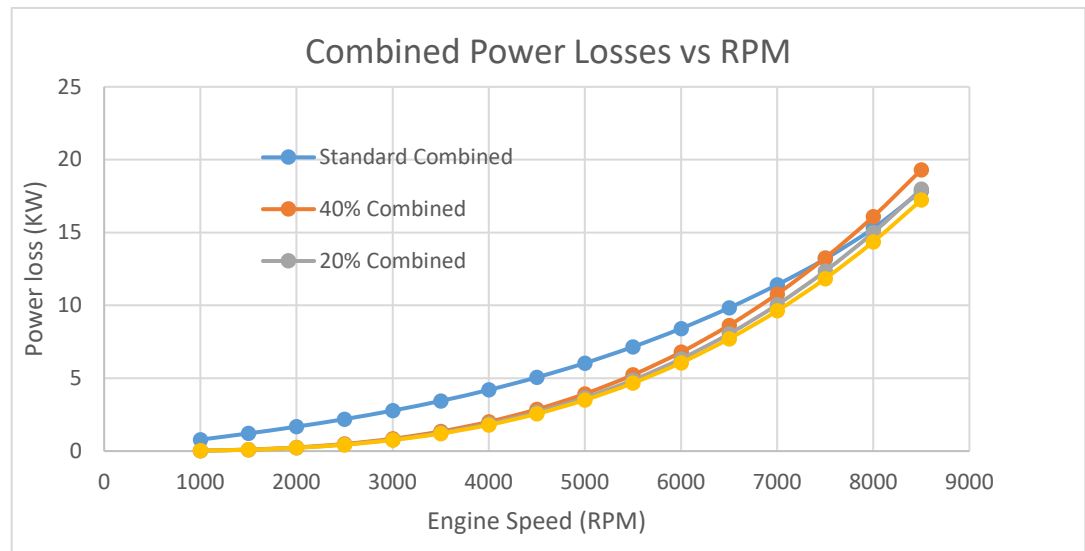
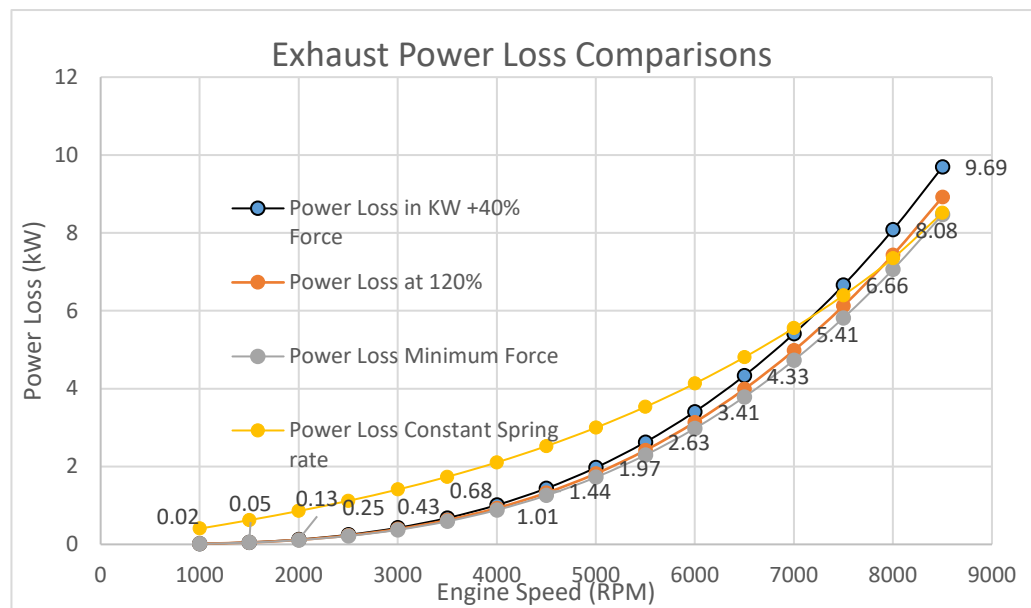
In this operating range, the standard valve spring is optimized only for the maximum RPM needed, in order to prevent valve float at maximum RPM of around 8000 rpm the force of the spring must be higher than that of the inertia of the valve during acceleration, this higher force will create a clamping effect on the valve, sealing it properly against the cylinder head. Generally a safety factor of 10% is observed [4] (P.W.Kett), this means the actual spring force will be around 10% higher than is required for the valve to close at maximum rpm. Interestingly, this means that if Ford were observing this safety factor, the intended float speed of the engine is anywhere from 8,000 – 8,800 RPM.

The importance of this factor becomes prevalent when looking at the adaptive spring rates. The entire purpose of an adjustable spring rate is to be able to apply the absolute minimum amount of spring force on the cam lobe, this creating less friction. The valve, however still needs to maintain full functionality. Many problems can arise with poor combustion and material lifespan if there is Insufficient clamping force on the valve.

Exhaust Graphs

Combined

Power Loss



The most evident difference in the curves for the power loss graphs is the increase from 1000- 7000 rpm, the shape of this curve is important however as it highlights that a standard spring is in no way calibrated to a lower engine speed, the Adaptive spring rate follows the needs of the engine, this large area under the graph is purely wasted power. This wasted power unfortunately is in the most used part of the engine. If this power were to be regained, this could transform the driveability of a vehicle from a customer point of view. Combustion engines typically lose up to 30% of their power to friction. In a modern 100 KW Vehicle, which is 30% efficient the original potential of the vehicle is 300kw, from the calorific value of the fuel. 4% of this is 12kw or 16 horsepower. A gain of this type without increasing any pressures on the engine itself would be astronomical in terms of car development.

Optimum IN KW Loss	Optimum EXH KW Loss	Combined Optimum KW Loss	20% Intake	20% Exhaust	20% Combined
0.014275753	0.013787386	0.028063139	0.014767024	0.01452048	0.029287508
0.04818322	0.046532428	0.094715648	0.049846667	0.04900663	0.098853302
0.118178653	0.110299088	0.22847774	0.118178653	0.11616387	0.234342527
0.223080044	0.215427906	0.43850795	0.230801095	0.22688257	0.457683661
0.385486455	0.372259421	0.757745876	0.39883703	0.39205307	0.790890104
0.612150956	0.591134173	1.203285129	0.633377649	0.62256576	1.255943411
0.913757935	0.882392701	1.796150637	0.945429222	0.92931099	1.874740212
1.301040068	1.256375546	2.557415614	1.346146629	1.32317913	2.669325755
1.784709337	1.723423245	3.508132582	1.846621063	1.81506053	3.661681591
2.37543978	2.293876339	4.669316119	2.457826945	2.41584556	4.873672509
3.083974423	2.978075367	6.06204979	3.190951006	3.13642459	6.327375599
3.921025246	3.786360869	7.707386115	4.057084433	3.98768798	8.044772415
4.897255941	4.729073384	9.626329325	5.067169807	4.98052609	10.0476959
6.023419881	5.816553451	11.83997333	6.2324257	6.12582928	12.35825498
7.310229047	7.059141611	14.36937066	7.563943303	7.43448793	14.99843123
8.768336782	8.467178402	17.23551518	9.072633348	8.91739238	17.99002573

Data table of Power loss figures in KW across the engine RPM range for both the "ideal calculated" spring rate, and values set at 20% higher.

40% intake	40% exhaust	40% Combined	Standard Intake	Standard Exhaust	Standard Combined
0.0156621	0.01577887	0.031440922	0.380973764	0.404570904	0.785544668
0.0528594	0.05325368	0.106113112	0.58928162	0.622068176	1.211349796
0.1252964	0.12623095	0.251527375	0.818974643	0.857819634	1.676794277
0.2447196	0.24654482	0.491264405	1.077181223	1.117910005	2.195091228
0.4228755	0.42602944	0.848904892	1.37102975	1.408424017	2.779453767
0.6715106	0.67651897	1.348029528	1.707648613	1.7354464	3.443095012
1.0023714	1.00984756	2.012219004	2.094166201	2.10506188	4.199228081
1.4272046	1.43784936	2.865054011	2.537710903	2.523355188	5.06106609
1.9577567	1.97235852	3.930115242	3.045411108	2.99641105	6.041822158
2.6057742	2.62520919	5.230983387	3.624395207	3.530314196	7.154709403
3.3830036	3.40823552	6.791239138	4.281791588	4.131149353	8.412940941
4.3011915	4.33327167	8.634463186	5.024728641	4.80500125	9.82972989
5.3720844	5.41215178	10.78423622	5.860334754	5.557954615	11.41828937
6.6074289	6.65671001	13.26413894	6.797331754	6.396094177	13.19342593
8.0189715	8.0787805	16.09775203	7.949913667	7.349363328	15.29927699
9.6184588	9.69019741	19.30865618	9.339842661	8.51450843	17.85435109

Table of data for Power loss at 140% of the calculated "Ideal spring rate" for each RPM compared with the standard spring rate Power Loss data.

Suggested Design

The power loss figures found by the use of this model show a very real advantage to being able to reduce friction in a valve train. Not only could reducing friction increase power and efficiency, many other factors could also be improved. Less mechanical friction will lend itself toward a less noise polluting engine, less friction would create less heat requiring less cooling. Other benefits such as lightening the camshafts may be possible due to less stress being acted upon the material throughout its lifecycle. Considering all of the points made for the Advantages of a lower friction design are clear and a new design for the valve train must be considered.

The concept

There are a number of different ways that have been used over the years to control valves in internal combustion engines. The methods focused on in this application are those that would potentially reduce friction, and be applicable to a production vehicle, each have their merits and downsides, the final chosen method would depend on the intended purpose of the vehicle and valve system.

Pneumatic Valve Control

For a number of years peak levels of motorsport have used pneumatically controlled valves in their racing engines. Competitions such as the MotoGP™ use a compressed air that is stored in a tank to control the movement of valves. The success of this relies on a number of factors that are unique to this particular type of motorsport. Most importantly is the length of time that the Valves must be controlled for.

A typical MotoGP race will be around 30 laps of a circuit, “races vary in length between 95-130km and normally last between 40-45 minutes” (MotoGP official , 2017) with the addition of a formation lap and any normal stoppages, a MotoGP motorcycle is typically not required to run for more then 60 – 90 minutes. For this reason, the advantages of an air controlled valve outweigh any disadvantages. The compressed air acts in the same way as a typical mechanical spring, and for this application, the air is used primarily to save weight in the cylinder head, and room in the motorcycle. With the bikes only running for up to 90 minutes, it is sensible to use a compressed air tank to store the air at a high enough pressure that it can replenish the air in a sealed unit in the cylinder head of the motorcycle.

The advantages of this system for a production car, would be that the air pressure in a unit such as is used in MotoGP and Formula 1 can be altered accordingly to achieve a desired “spring rate” with the compression of the air. Air of a certain pressure will compress a certain amount and it is possible to calculate and predict this. For a production car however, there are a significant number of other variables to consider that may affect the successfulness and relevance of this type of design.

Advantages of an 'Air Spring'

Adjustable spring rate – this is the key point of this investigation, the glaring advantage of the air based spring is the simplicity of the concept of controlling the spring rate. Adjusting the pressure of the air the valve stem/actuator/plunger would be travelling in will directly alter the effective spring rate. This can allow for an infinitely adjustable spring rate and matching of spring rate to the required clamping forces for proper valve sealing.

Adjusting the valve this way could mean that all of the potential savings in friction would be achieved.

Sizing – Redesigning the valve train of an engine offers new design solutions for manufacturers. Being able to alter the design of the cylinder head could have dramatic benefits for packaging, sound deadening, heat management and many other aspects that must be considered for a competitive car package today. Changes to the cylinder head could allow for smaller and more importantly shallower engines, however the gains may be limited due to the necessity to use the existing 'poppet' valve type.

The gains in efficiency if as prominent as the study suggests will make this method of valve control a very viable option. This technology if proven effective and cost-effective could be used in Gasoline and diesel powered modes of transport, with tighter emissions being an ever increasingly important problem for car manufacturers a 4% gain in engine friction could make a large difference in a company's competitiveness and therefore future.

Also, the future of the combustion engine in recent years in becoming more and more in doubt, with electric and hybrid vehicles improving dramatically, unless every effort is made to improve efficiency and reduce emissions an increasing number of vehicles will not meet up to the standard of the emissions test and be at worst case rendered obsolete, potentially crippling some major car manufacturers.

Disadvantages of an 'Air Spring'

Whilst this method provides many answers to existing friction and various other kinds of problems for a manufacturer; this does not come without its drawbacks.

Reliability – The currently mass produced and widely used mechanical method of returning a valve in a combustion engine (the valve spring) is a tried and tested method, some theoretical losses are accepted by manufacturers due to the reliability of experience with these Valve train mechanicals.

Questions of longevity of the air controlled valves would certainly be a concern for a production manufacturer. In most cases, Air controlled valves are only used in high end motorsport, for relatively short periods of time, my research did not show of any cases where a pneumatically controlled valve was in use for over 100,000 miles, which is typically the minimum expected lifetime of a standard production engine.

Problems for reliability could come from many different sources, in order to hold pressure the Air valves must have seals. In the real world seals generally degrade and leak over time, Percentages of pressure drop would have to be calibrated and tolerances agreed upon and this would take much deliberation and experimentation. Another area for potential reliability issues would be the source of the air pressure. Unlike a motorsport application, a production car would be more likely to have an on board method of replenishing the pressure in the valves, rather than a single storage tank pre-charged for the entire vehicles cycle. This could have potentially catastrophic results if any part of these systems failed, resulting in an immobilized vehicle.

In a production car a more likely solution that one tank would be to have a tank storing the air and a compressor to recharge the air pressure. This would mean yet more potential for unreliability as currently available automotive compressors such as Air conditioning compressors and tyre inflation compressors are not known to work without fault for the entire life of a vehicle, having to replace one of these is quite commonplace in the automotive world. These issues would require vigorous testing.

Safety – Internal combustion engines rely on using a highly combustible liquid, put under pressure, and ignited with a spark, to create what is essentially a controlled explosion. Added to this most modern vehicles have upwards of 6 air bags, and 4 seat pre tensioners, all filled with some form of explosive, the modern production car is effectively, in many ways, a moving bomb. The use of a pneumatic valve system would require a storage tank of some kind, with high pressure air lines to feed the compressed air into the tank. This tank would need to be stored either near the front of the vehicle, near a warm engine that creates sparks, near the rear of the car and next to a fuel tank, or somewhere near the middle of the car, typically next to a hot exhaust. Whilst compressed air is not a particularly lethal substance, applying high pressure air to an explosion typically does not have a positive effect.

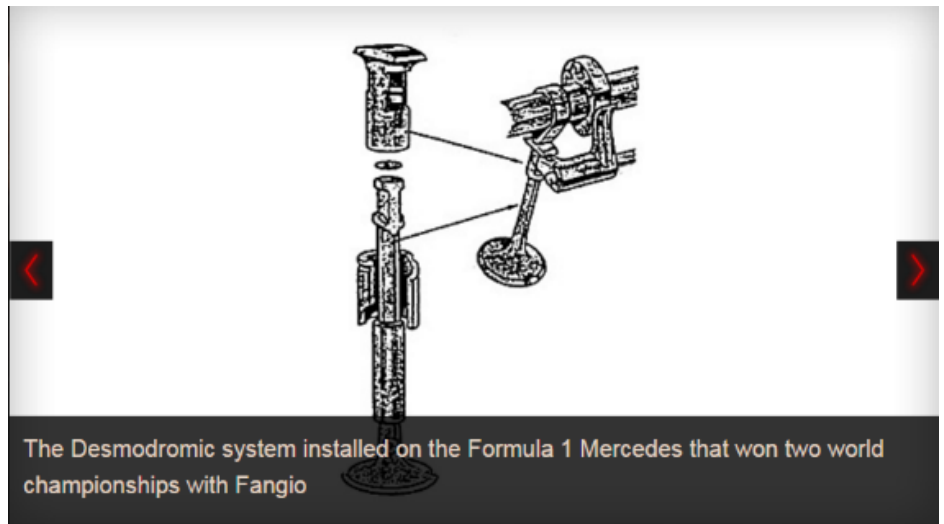
Cost – All of these considerations mean that many parts of the vehicle may need to be redesigned, this can require alterations to the production of the cars. Dealerships selling the vehicles would require extra training on the vehicles, Technicians would require the knowledge of these systems and the creation of the air controlling system itself would incur costs.

Possible Alternate Methods

Desmodromic

Air valves are not the sole method of controlling valves more accurately, other methods such as the Desmodromic system which has been used for many years in formula one and in Motorbike Racing, championed mostly by Ducati. [9]

This system uses a combination of two cam lobes to both push and pull the valve open and closed. This system effectively eliminates the issues of valve springs inability to keep pace with the engine rpm, however, as there is more mechanical mating surface, and another lobe working in the opposite direction Friction is actually increased in comparison to the conventional spring method.



The Desmodromic would allow for higher RPM but this does not suit the planned project.

Computer Controlled Electric/Hydraulic

Systems such as the Keonigsegg/Free Valve Developed “Free-Valve” technology have been in experimental development for a number of years, using high pressure oil and electronic actuators to control the exact timing, position and speed of inlet valves.

For this application it was felt that this option would be excessively expensive and not a viable option as a development step for a production ford petrol engine. The complicated nature of this method of valve control would make the engine costly and potentially unreliable.

This technology is not in use in main stream automotive applications and has not been pioneered in Motorsport in any great detail.

The conclusion is that the Pressurised air method of valve control will offer the best all round performance and efficiency gain, reduction of friction and an acceptable level of complication to the production.

Conclusion

This project has taken a look into the friction in the valve train of a relatively modern Gasoline production car engine. This engine is widely used across the world and its development is ongoing today for many purposes. Improvement of the engines efficiency can offer gains of many kinds, this study has highlighted the proportionally large gains that are potentially available from this engine, whether the focus is performance of improved efficiency, and the study has shown that these are a byproduct of each other.

The method of simulation used in this study have been based on sound mathematical and engineering based logic, through research of previous engine design improvements and with the aid of experienced engine designers and established literary experts a reliably accurate study of the effects of Valve spring rates on Valve-train friction has been undertaken. The results from this investigation have been surprising to myself as an engineer with a healthy amount of practical experience in the world of automotive engines.

The most attractive part of the results of this study are the 80% potential reductions in power loss due to this aspect of the valve train. If this could be harnessed effectively and within the near future, this solution could offer a substantial competitive edge to a manufacturer of engines, this study is of a Ford engine that is very similar to millions of cars on the road. It is reasonable to assume that any 1.6 four 4cylinder gasoline engine would see similar improvements from the reduction in friction that a variable spring rate could offer.

Recommended further Work

In order to create a more reliable model, creation of the valve train in another form of simulation would be advantageous, as well as a real world practical test using a more thorough method, perhaps a dynamometer suited to the task would be effective in reading real world friction losses due to the valve spring.

Of course, designing and creating a prototype pneumatic valve for use in a real Zetec engine would produce the real world answers and validate whether the model was an accurate representation of the real world problem/solution.

In regards to improving the current model, repeating of the initial measuring would improve accuracy and taking readings at more intervals could create more reliable data.

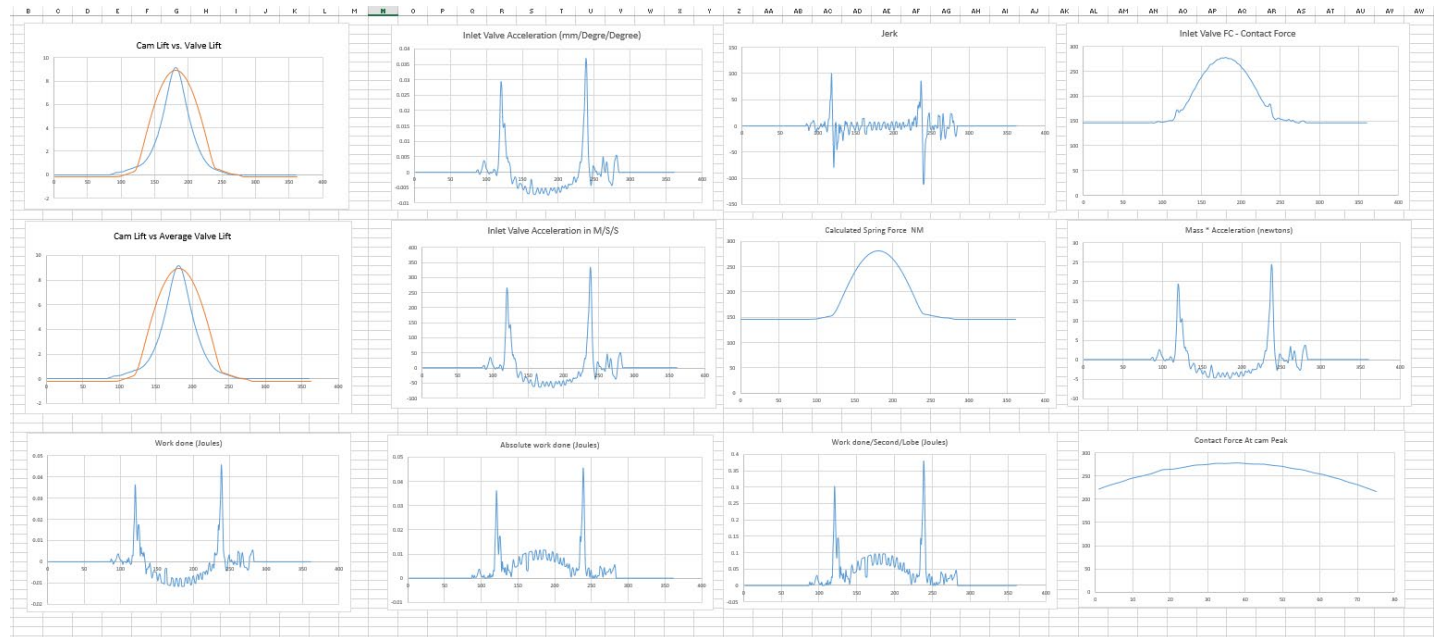
On the whole the project was very successful and improved my knowledge of engine mechanics dramatically, this is a topic that I now feel I am very engaged in and look forward to pursuing it further.

References

- [1] J.I.Ramos. (1989). *INTERNAL COMBUSTION ENGINE MODELING* . pittsburg, pennsylvania : HEMISPHERE PUBLISHING CORPORATION .
- [2] <https://en.oxforddictionaries.com/definition/us/acceleration>
- [3] http://www.hydraforce.com/svtraining/files/sdiskjf_Chap8.pdf
- [4] Motor Vehicle Science, Part 2 edited by P.W. Kett
- [5] https://energy.gov/sites/prod/files/2014/03/f8/deer12_wong.pdf
- [6]file:///C:/Users/Nick/Desktop/Mapping/BMW%20330D%20Downloaded%20map/Fuel%20efficient%20motor%20oil%20technical%20article.pdf
- [7] http://www.formula1-dictionary.net/pneumatic_valve_actuation.html
- [8] “***Jalopnik's Book Of Car Facts And History Even Gearheads Don't Know***”
By Matt Hardigree, Jason Torchinsky, Travis Okulski, M t Petr ny, Raphael Orlove, Sam Smith
- [9] <http://www.ducati.com/racing/motogp/bike/index.do>
- [10] A.B.PIJL. (2007). Analysis and Modeling of Desmodromic Valve Train. Universiti Teknikal Malaysia
- [11] <http://www.cycleworld.com/2014/04/25/cw-tech-valve-control-history-and-why-ducati-is-committed-to-desmodromics>

For extra material and background work see appendix Overleaf

Appendix



Graphs Produced From Main Model

One of the many Spreadsheets produced for automated calculations

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
Adaptive Spring Rate															
Crank RPM	Cam RPM	Average Max contact Force	Inlet Cam FMEP total * 8 Cam Lobes	Power Loss in KW Minimum Spring force											
1000	500	-0.012470857	0.002068108	0.013787386											
1500	750	0.293137416	0.004653243	0.046532428											
2000	1000	-0.049883428	0.008272432	0.110299088											
2500	1250	-0.077942856	0.012925674	0.215427906											
3000	1500	-0.112237713	0.018612971	0.372259421											
3500	1750	-0.152767999	0.025334322	0.591134173											
4000	2000	-0.199533712	0.033089726	0.882392701											
4500	2250	-0.252534855	0.041879185	1.256375546											
5000	2500	-0.311771426	0.051702697	1.723423245											
5500	2750	-0.377243425	0.062560264	2.293876339											
6000	3000	-0.448950853	0.074451884	2.978075367											
6500	3250	-0.52689371	0.087377559	3.786360869											
7000	3500	-0.611071995	0.101337287	4.729073384											
7500	3750	-0.701485708	0.116331069	5.816553451											
8000	4000	-0.79813485	0.132358905	7.059141611											
8500	4250	-0.90101942	0.149420795	8.467178402											
Total Power Lost Across RPM Range (KW)				40.34189132											
Power Loss Between 1k and 3k RPM				0.758306228											

Power Loss in KW Minimum Spring force

RPM	Power Loss (KW)
1000	0.01
1500	0.05
2000	0.11
2500	0.22
3000	0.37
3500	0.59
4000	0.88
4500	1.26
5000	1.72
5500	2.29
6000	2.98
6500	3.79
7000	4.73
7500	5.82
8000	7.06
8500	8.47

4000 RPM

4250 RPM

4500 RPM

4750 RPM

5000RPM

Adaptive Spring Calcs

Power Loss Constant Rate

Power loss Adaptive

Power loss at 120%

Power loss at 140%

Sheet6

