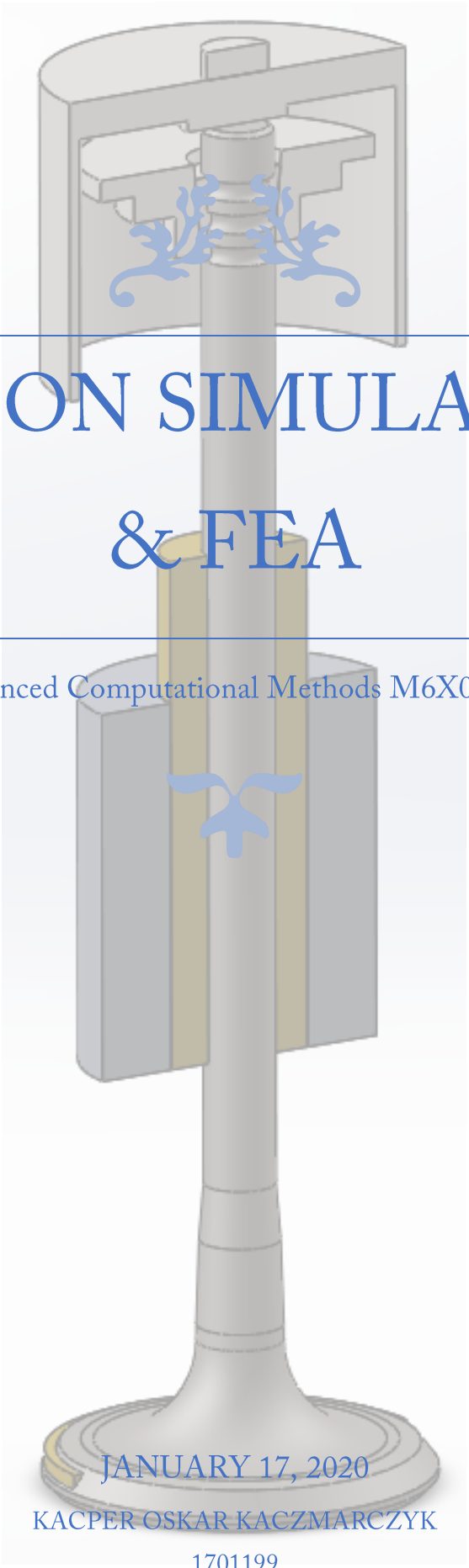




MOTION SIMULATION & FEA

Advanced Computational Methods M6X00408

JANUARY 17, 2020

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1 Introduction

This investigation focuses on the study and development of the intake valve from 1.0l GTDI Ford Fox engine. The aim was to reduce the reciprocating mass in the valve, as this would allow softer valve springs, ultimately reducing the valvetrain losses.

Author has measured multitude of components, all of which were accurately modelled in the SolidWorks Motion Simulation software. The entire assembly was run at 7,000 crank rpm to simulate the worst-case scenario and the results were then analysed and used in two static studies.

The static studies have shown how well the intake valves are engineered, making author struggle to make a reliable improvement. Initially Design Study and then Topology Study was used to improve the design.

Author has managed to lighten the intake valve by a meaningful amount. However, the geometry required to achieved this was deemed too complicated and as a result, unrealistic to implement into traditional mass production.

2 Motion Simulation

2.1 Geometry

2.1.1 Acquisition & Modelling

To ensure accurate results in later stages of this investigation, author has made sure to acquire accurate measurements of all components which may inflict stress on the intake valve, as shown in Figure 2-1 and Figure 2-2. Author has taken intake valve model from his dissertation; all other components were purposely modelled for this investigation.



Figure 2-1: Reciprocating Valvetrain Components



Figure 2-2: Valve Seat Measurement

2.1.2 Equations of important dimensions

Author has utilised the 'equations' feature as show in Figure 2-3 and Figure 2-4, to keep certain common dimensions such as stem diameter and valve guide diameter constant, as well as position the static cylinder head representation in the correct distance away from the retainer when valve is seated. This ensures the spring preload to be as intended and as a result, the spring force applied to the retainer should also be correct as the valve lifts. However, author hasn't accounted for the change of young's modulus with the raise of the spring's temperature. This could potentially exert bigger force on the retainer than in real life application.

Equations, Global Variables, and Dimensions

Name	Value / Equation	Evaluates to	Comments
Global Variables			
"Spring Free Length"	= 49	49	
"Spring Preload"	= 13	13	
"Valve Length"	= 90.06	90.06	
"Guide Offset"	= "Valve Length" - "Head Length" - ("Spr	28.06	
"Seat Width"	= 1.3	1.3	
"Head Length"	= "Head Length@Cylinder Head<1>.Par	26	
"Stem Radius"	= "Stem Radius@Intake Valve<1>.Part"	2.48	
Add global variable			
Equations - Components			
Add equation			
Dimensions - Top Level			
D1@Distance2	= "Guide Offset"	28.06mm	

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Figure 2-3: Global Variables Table

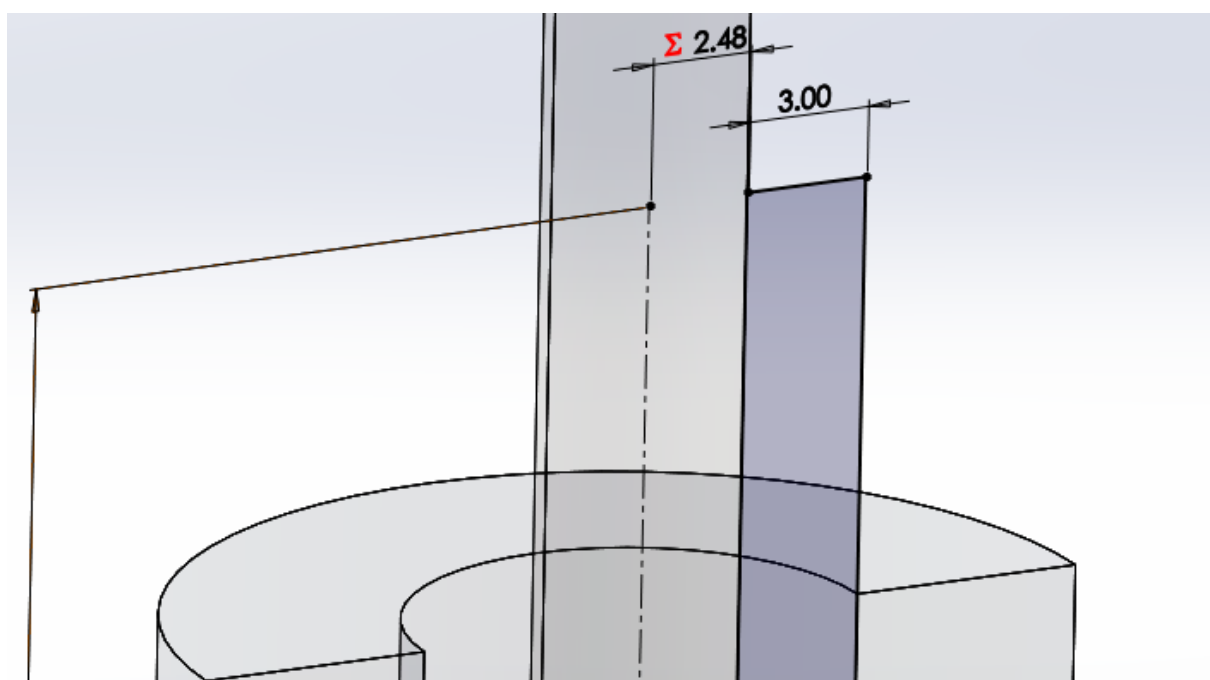


Figure 2-4: Utilised Dimension Equation

2.1.3 Component Masses

Due to the lack of information on the specific materials used in the valvetrain components as well as the limited material library provided in SolidWorks, author has measured masses of all modelled components as shown in Table 2-1. These values were then used to override individual component's mass, as shown in Figure 2-5.

Table 2-1: Reciprocating Masses

Reciprocating Masses	
Component	Mass (g)
Intake Valve	28.18
Pair of Collets	0.6
Retainer	7.97
Intake Spring	20.46
Bucket	19.98

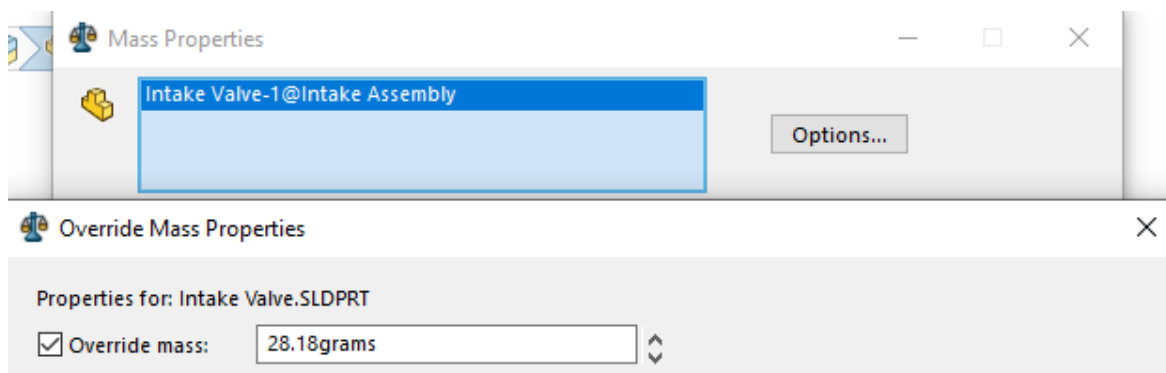


Figure 2-5: Mass Properties of Intake Valve

This was done since mass and force are directly related by acceleration, which is controlled by the camshaft. If the component mass was incorrect, the force required to accelerate it would also be incorrect. This would have resulted in very skewed and unreliable simulation results.

2.2 Motion Setup

2.2.1 General Explanation

The simulation was conducted at 7,000 crank rpm since manufacturers must consider the possibility of the user accidentally over-revving their engine. Author has decided to plot the valve lift every 0.2 cam°, resulting in 105,000fps and total run time of 0.011s.

Since this investigation focuses solely on the intake valve, the only components which author considered were; intake valve, valve seat, cylinder-head (representation), valve guide, collets, retainer

and bucket. The camshaft itself wasn't modelled since it doesn't directly impact the valve, therefore it was represented as a shim on top of the bucket, as shown in Figure 2-7.

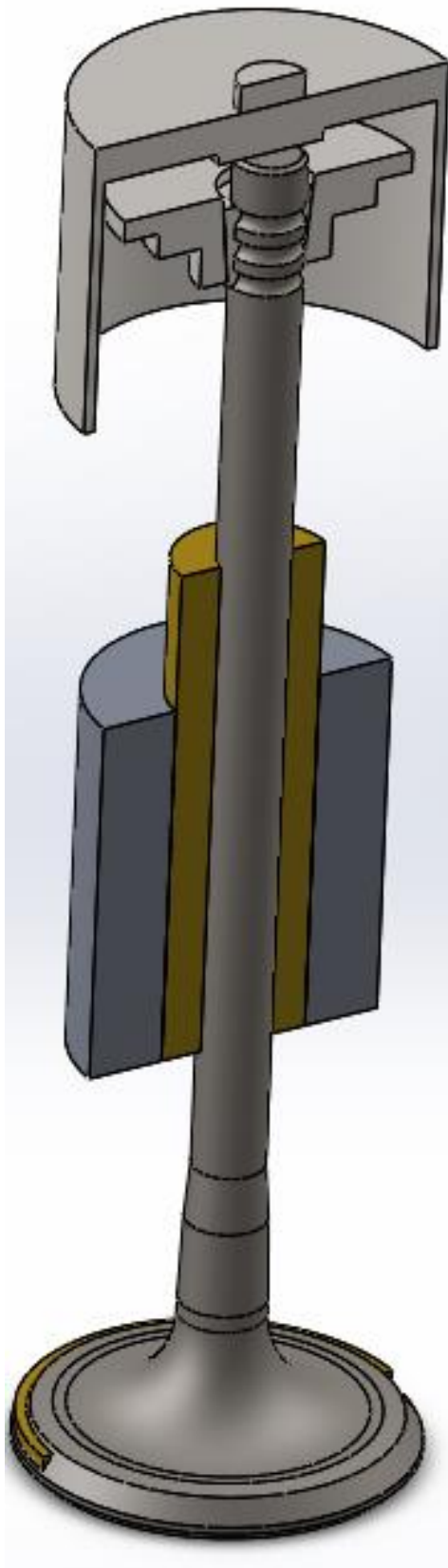


Figure 2-7: Intake Valve Assembly

When assembling, author made sure to only restrain the assembly in the Y axis, since all of the solid body contact interactions had to be modelled using 'solid body contact' feature, to ensure not only the accuracy of the forces, but also the timing as well as impact, which could occur if 'slack' was introduced into the assembly, for example due to a weak spring. Standard Steel Dry on Steel Dry friction and elastic properties were applied due to lack of reliable data.

As shown in Figure 2-6, Linear Spring feature was used to simulate the valve spring, to ensure correct distribution of the spring force, it was applied between the outer edges of cylinder-head representation and the retainer. Real free length was used to ensure correct preload, as mentioned in section 2.1.2.

Lastly, linear motor with imported data points was applied to the shim to simulate the camshaft, as shown in Figure 2-8. Linear interpolation was chosen due to its superior accuracy.

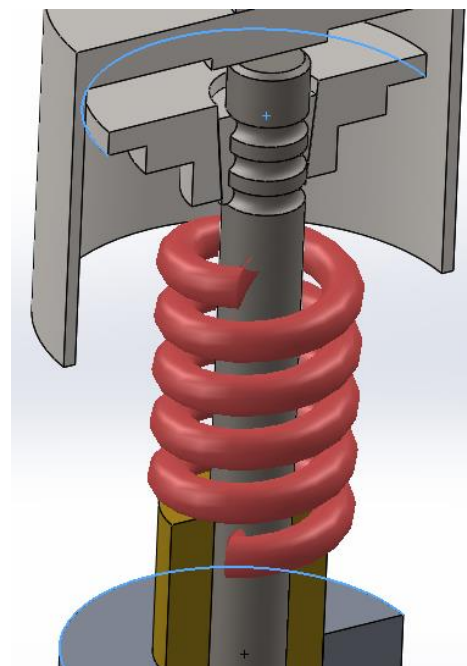


Figure 2-6: Linear Spring in the Assembly

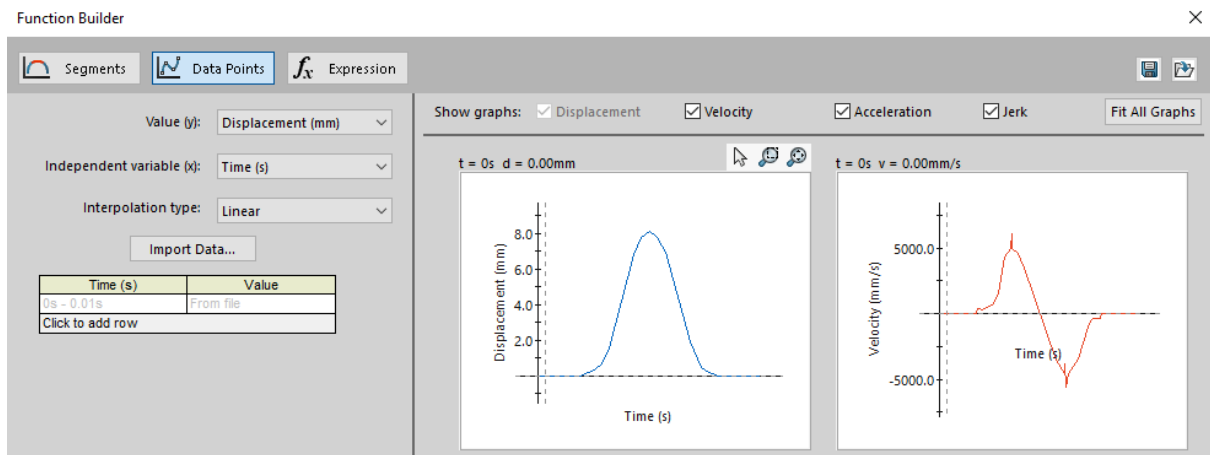


Figure 2-8: Linear Motor - Function Builder

2.2.2 Valve Lift Profile

The intake camshaft lobe was measured, as part of author's dissertation, as shown in Figure 2-9.

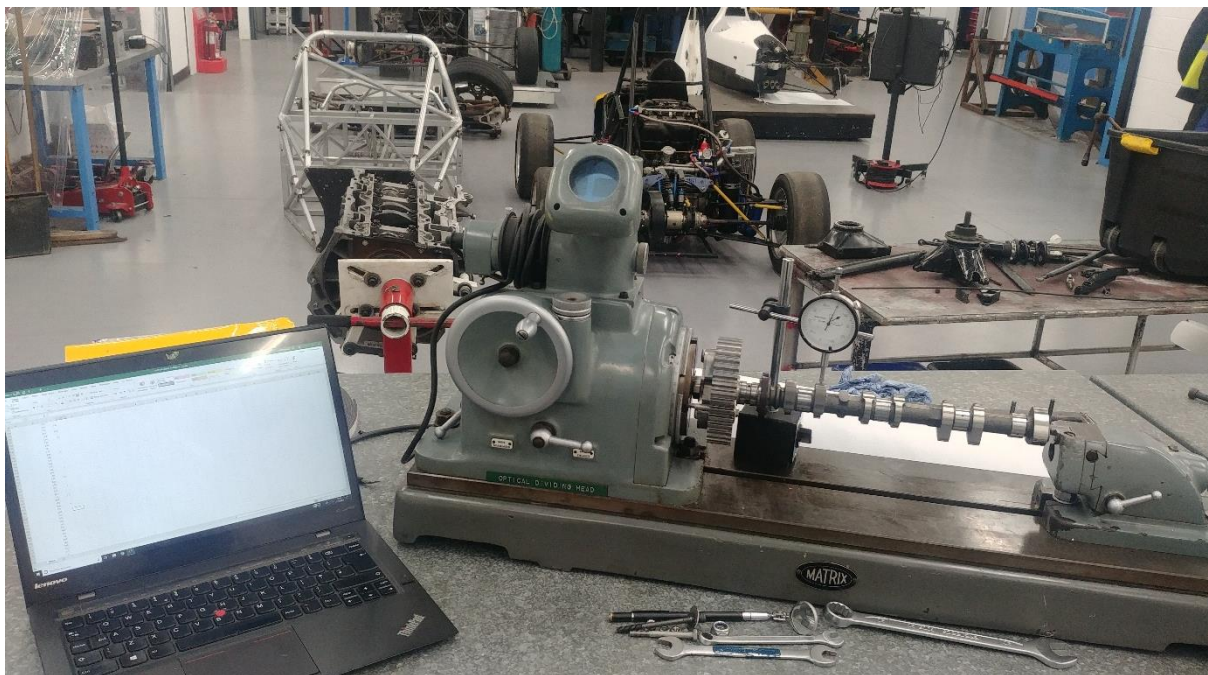


Figure 2-9: Measuring Camshaft Lift

The camshaft lift was then converted into valve lift by considering the moving contact point as shown in Figure 2-10, valve clearance of 0.27mm was also considered.

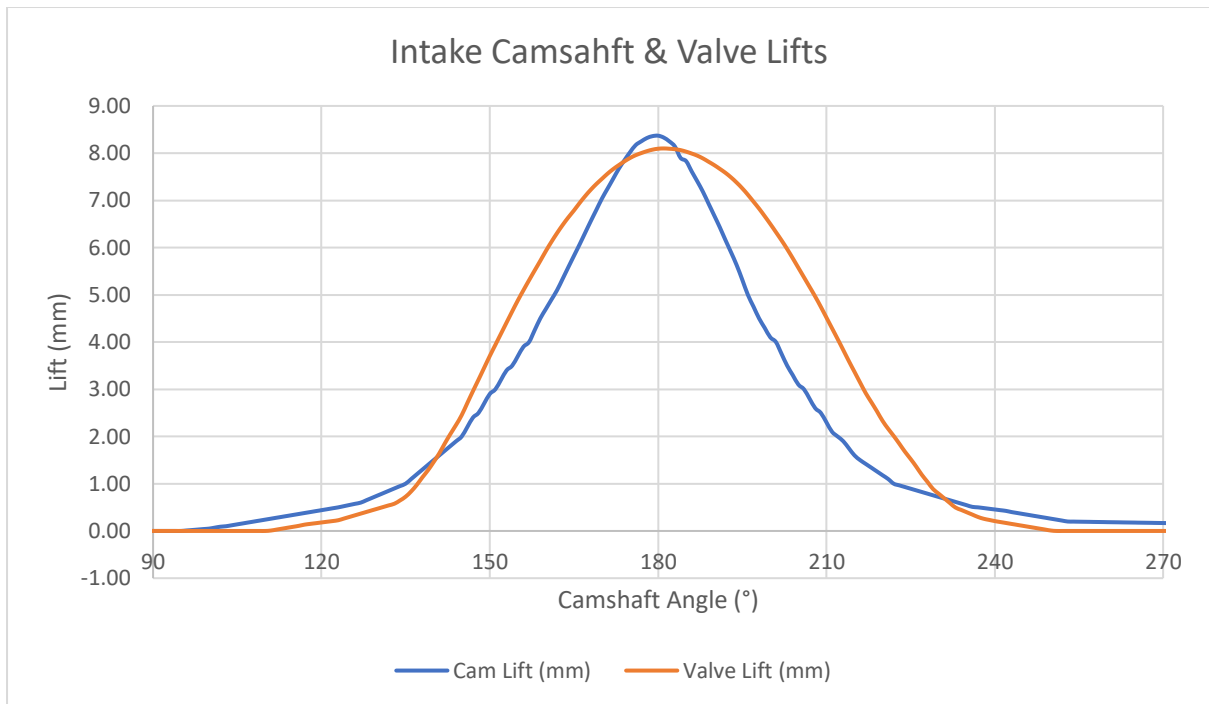


Figure 2-10: Intake Camshaft & Valve Lifts

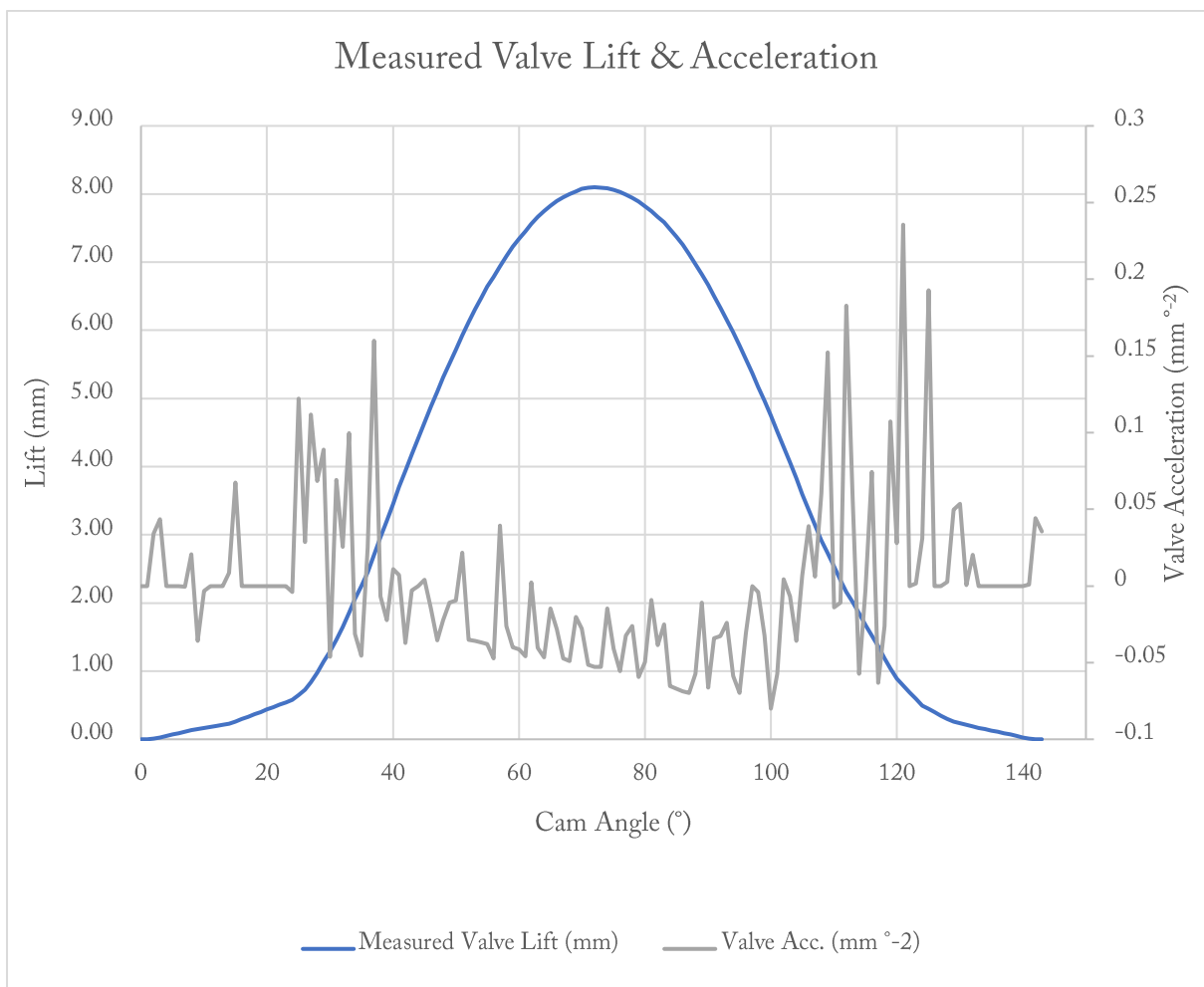


Figure 2-11: Measured Valve Lift & Acceleration

However, as Figure 2-11 shows, the acceleration gained by double differentiation of valve lift was unusable, as it would have resulted in numerous high stress impacts. This was caused by poor resolution of just 1° as well as the accuracy of the dial gauge which was only to $\pm 0.01\text{mm}$. To improve this, author would have had to use digital gauge with accuracy of at least $\pm 0.001\text{mm}$ as well as digitally controlled rotary chuck, which would allow for a reliable step of at least 0.2° .

Since such tool wasn't available, author has curve fitted 3 sets of 6th order polynomial curves in Matlab, which when joined at 40° and 100° create the measured valve lift. The lift profile had to be divided into 3 sections to achieve R^2 value of 1.0.

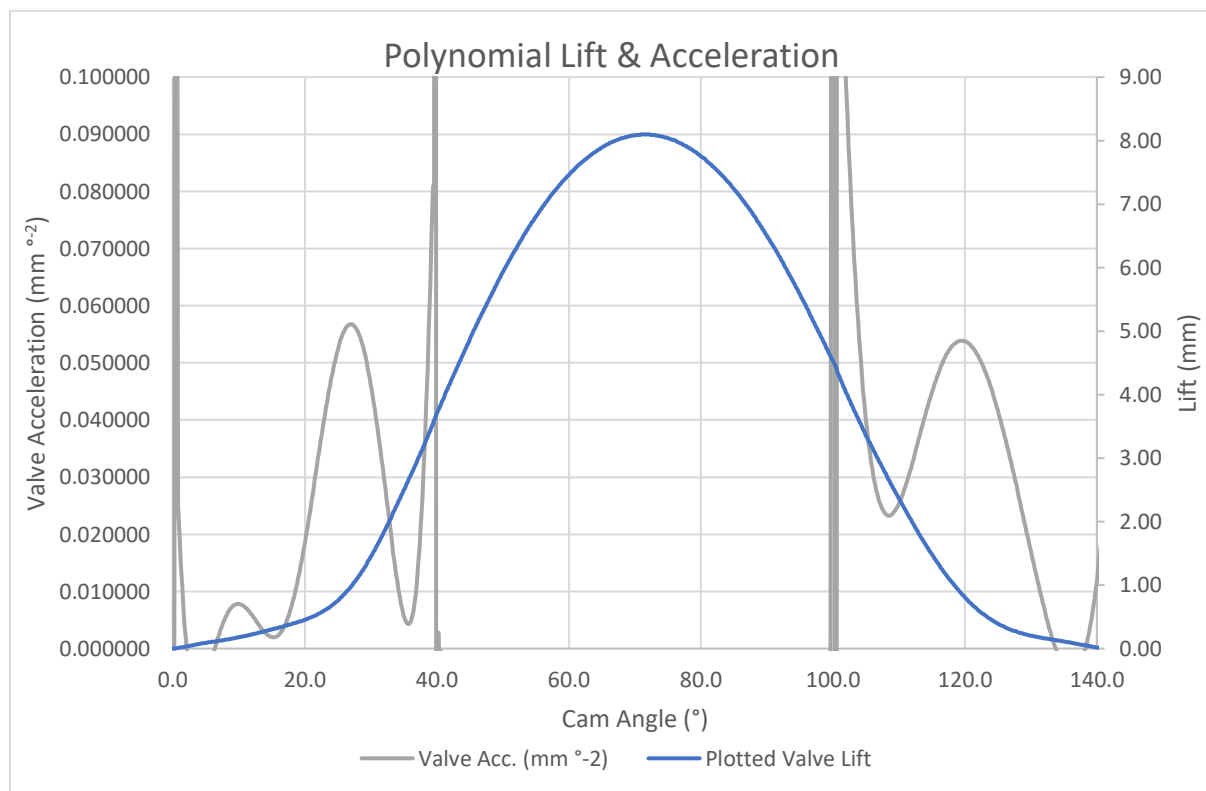


Figure 2-12: Polynomial Lift & Accelerations

Figure 2-12, shows that the stitched polynomial curve was much smoother, resulting in more realistic valve acceleration. However, the acceleration between the 'stitched' points of cam angle 40° and 100° , was unusable due to the 'stitching'. However, in this case this was acceptable since the biggest valve accelerations on a well-designed camshaft occur somewhere around the 'ramps', making the polynomial lift profile usable. However, this meant that any results between 40° and 100° would have to be discarded in later sections.

Author couldn't implement the method of moving average due to the lack of resolution, if more cam angle points were recorded author believes the moving average would have resulted in much more usable profile.

2.2.3 Valve Springs

To find out the two different valve spring rates author used a spring rate tester.

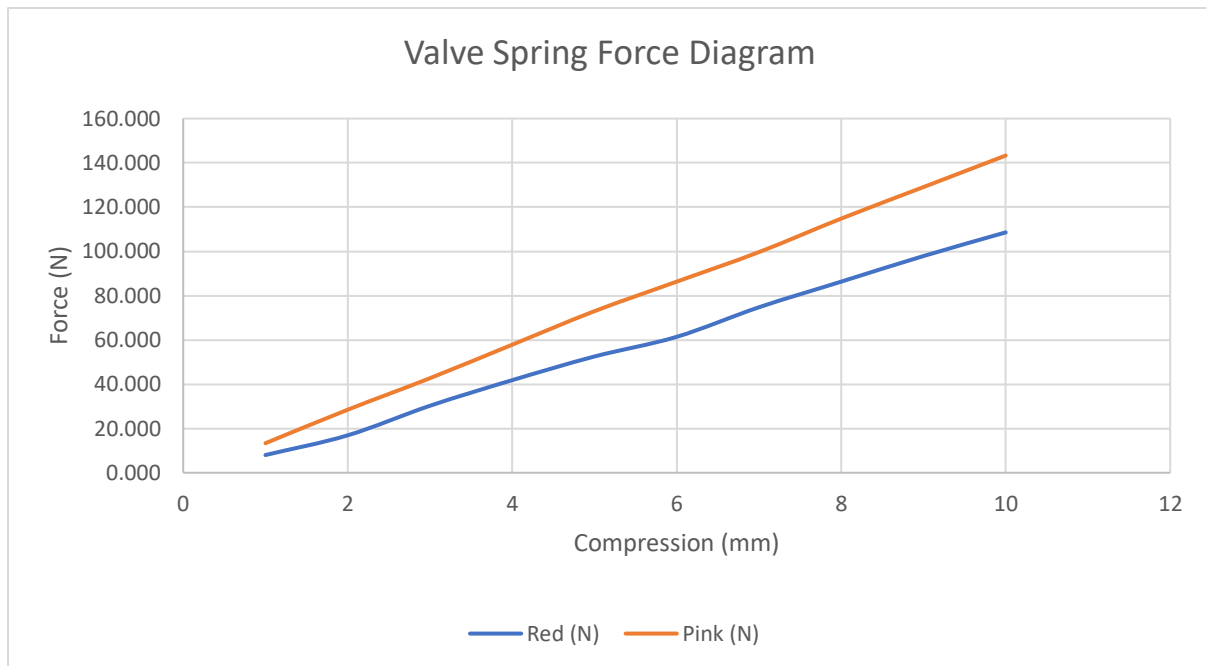


Figure 2-13: Valve Spring Force Diagram

Figure 2-13 shows the recorded spring forces of the red and pink valve springs (paint markings). Force was then differentiated in respect to compression, to plot the spring rate every 1mm of compression, as shown in Figure 2-14.

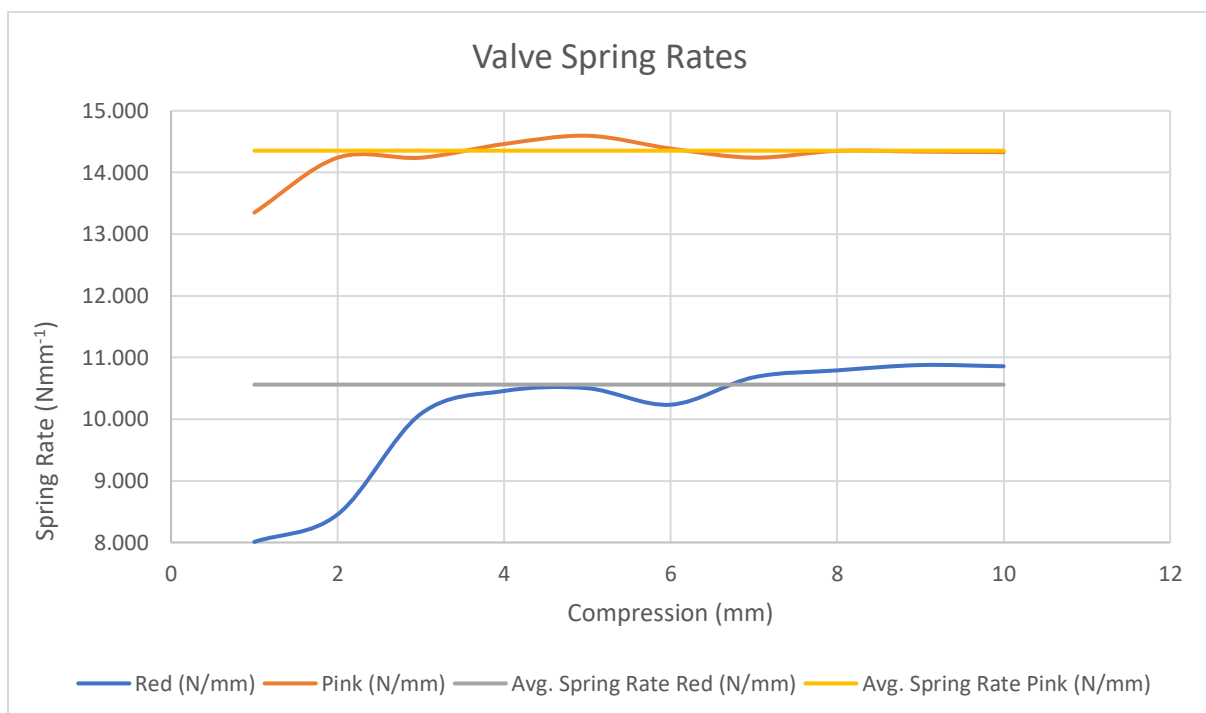


Figure 2-14: Valve Spring Rates

The instantaneous spring rates showed to be considerably lower at low compression, this may have been caused by extensive endurance testing, which the spring's engine was subjected to. Since the history of the supplied prototype engine was lacking, author has decided to disregard spring rates at compression of 1mm and 1mm + 2mm, for pink and red springs respectfully. It is possible that the spring rates have deteriorated due to extensive fatigue. However, author had no way of checking if this was the case as author wasn't able to find more information.

Since the pink spring was considerably stiffer, author has assumed it to be intended for the intake side, as intake section of valvetrain almost always has heavier total reciprocating masses due to bigger valves.

Author could have improved this recording by improving the resolution of compression.

2.3 Results & Peak Stresses

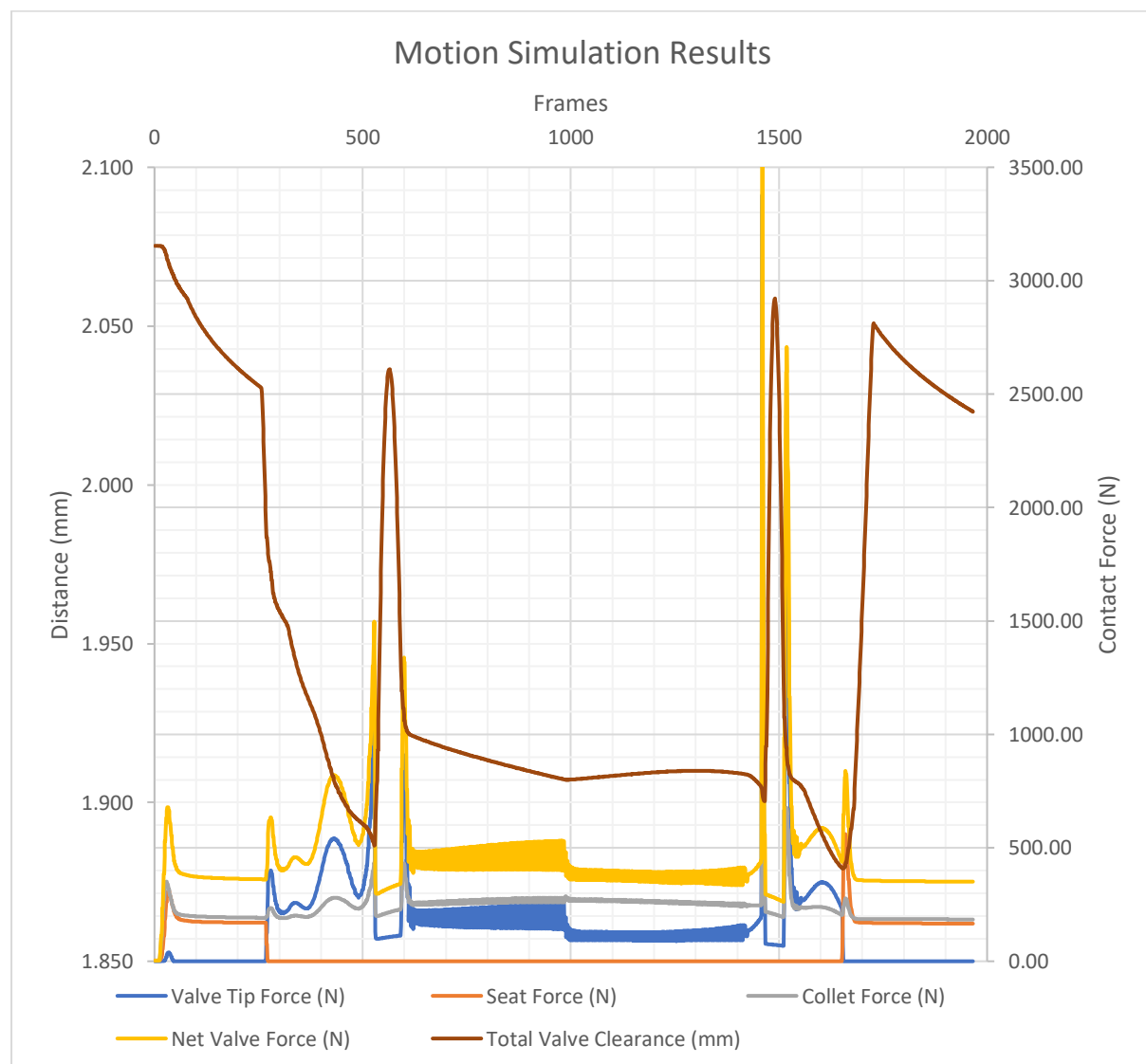


Figure 2-15: Motion Simulation Results

Figure 2-15 shows the results of the motion simulation. Firstly, it shows that the model has ran correctly as the seat force and the tip force increase and decrease respectfully as the valve is seated at frame 1670, vice versa occurs at frame 270. The figure also shows that the forces at the collet meeting faces experience residual forces from the tip and the seat faces on top of the spring force, since the collet face line follows both of the mentioned profiles. Lastly, as author expected, Figure 2-15 shows the peak tip force to occur at the maximum valve acceleration at frame 432 and maximum seat force at frame 1663, since this is when the valve contacts the seat.

Author has acknowledged that the shown results are likely to be overestimated, since the simulation assumes all components to be infinitely stiff, resulting in no strain and as a result, forces are directly transferred through the components instead of being dissipated, e.g. the retainer doesn't flex so all of the spring force is transferred to the collets and as a result into the tip of the valve. Additionally, this also means that the location of stress concentrations in later work will be inaccurate, as the components can't flex, resulting in inaccurate applications of the force's direction. Furthermore, the motion simulation assumes no moments and no shear forces. As a result, the valve isn't rotating and the valve guide doesn't exert shear force on the valve stem. This has been done due to the lack of information about the working clearances (stem to guide clearance is likely to decrease as components heat up) and working friction coefficients.

This has caused author to create specific methodology which is discussed in chapter 3.

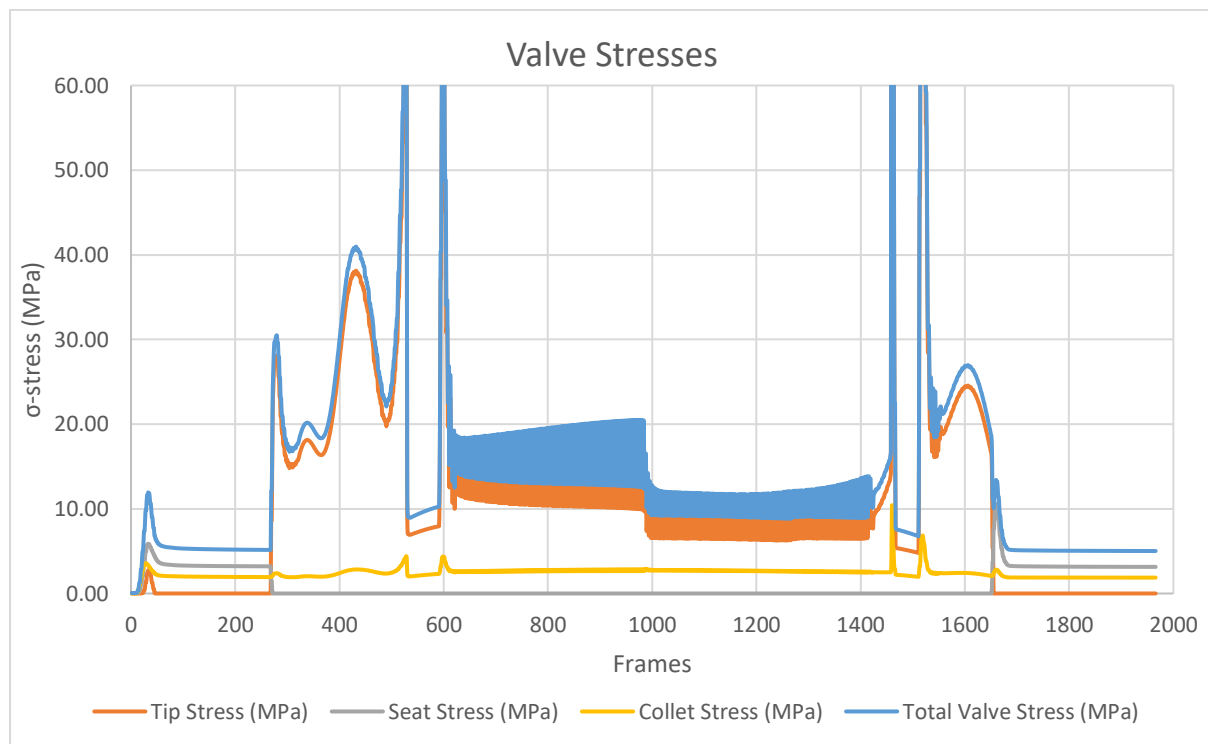


Figure 2-16: Valve Stresses

Figure 2-16 shows the valve stresses which were calculated by dividing the valve forces from Figure 2-15 by corresponding areas, which were measured directly from the model as shown in Table 2-2.

Table 2-2: Surface Areas

Surface Areas	
Face Type	Area (m ²)
Seat	0.00005289
Tip	0.00001421
Collet Faces	0.00009857

Author has calculated the applied stresses as they account for the area not just the force. The importance of this is well shown in Figure 2-16 since the magnitude at the peak tip stress is 3.58 times bigger than the peak seat stress, despite having almost the same force.

Since it wasn't possible to correctly import the motion loads into FEA study, author has decided to divide the valve into two stress groups, top and bottom. This was done due to the big difference of topology. Figure 2-16 clearly shows that the peak total stress applied into the top stress group occurs at frame 432. Since the bottom stress group only has single source of stress, the seat stress has been directly used to determine the peak group stress, which occurred at frame 1663.

3 Finite Element Analysis and Topology Optimisation

3.1 FEA

3.1.1 Setup

Author has decided to use ALSI 4340 normalized steel, since it represented the Silchrome 1 material the best (Kaufman, 2006).

Due to the fixtureless nature of the poppet valve, author had to use inertial relief option so that the forces balance, making the results realistic in this particular scenario.

Loads were applied according to the results from chapter 2.3, the load locations can be seen in Figure 3-2 and Figure 3-1. The magnitude of applied loads can be seen in Table 3-1.

Table 3-1: Load Magnitudes

Loads	Force (N)	
	Frame 432	Frame 1663
Seat	N/A	560.88
Tip	541.94	N/A
Collet	280.26	277.37

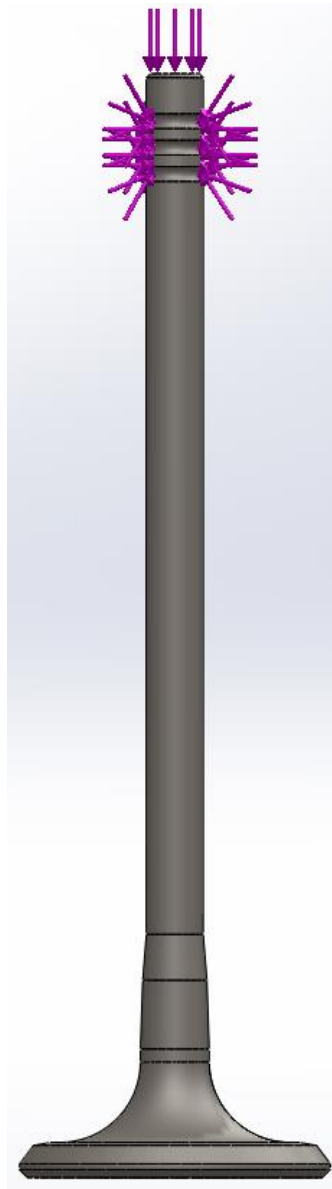


Figure 3-1: Loads location at frame 432

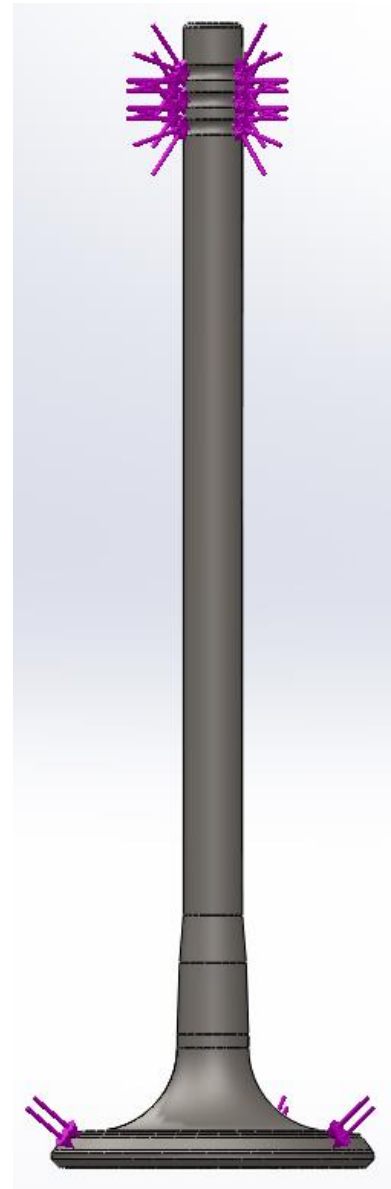


Figure 3-2: Loads location at frame 1663

After testing various different meshes, author has decided to use mesh parameters shown in Figure 3-3. The chosen mesh is a compromise between accuracy and computational time. This was important since author knew that both studies will be used in iterative development. Author knows that the mesh quality is high since 99.9% of elements have aspect ratio below 3.

Mesh Details	
Study name	Frame 432 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Blended curvature-based mesh
Jacobian points	At Nodes
Max Element Size	0.5 mm
Min Element Size	0.14 mm
Mesh quality	High
Total nodes	331867
Total elements	229691
Maximum Aspect Ratio	4.4603
Percentage of elements with Aspect Ratio < 3	99.9
Percentage of elements with Aspect Ratio > 10	0
% of distorted elements (Jacobian)	0
Time to complete mesh(hh:mm:ss)	00:00:10
Computer name	MAINRIG

Figure 3-3: Mesh Details

3.1.2 Results

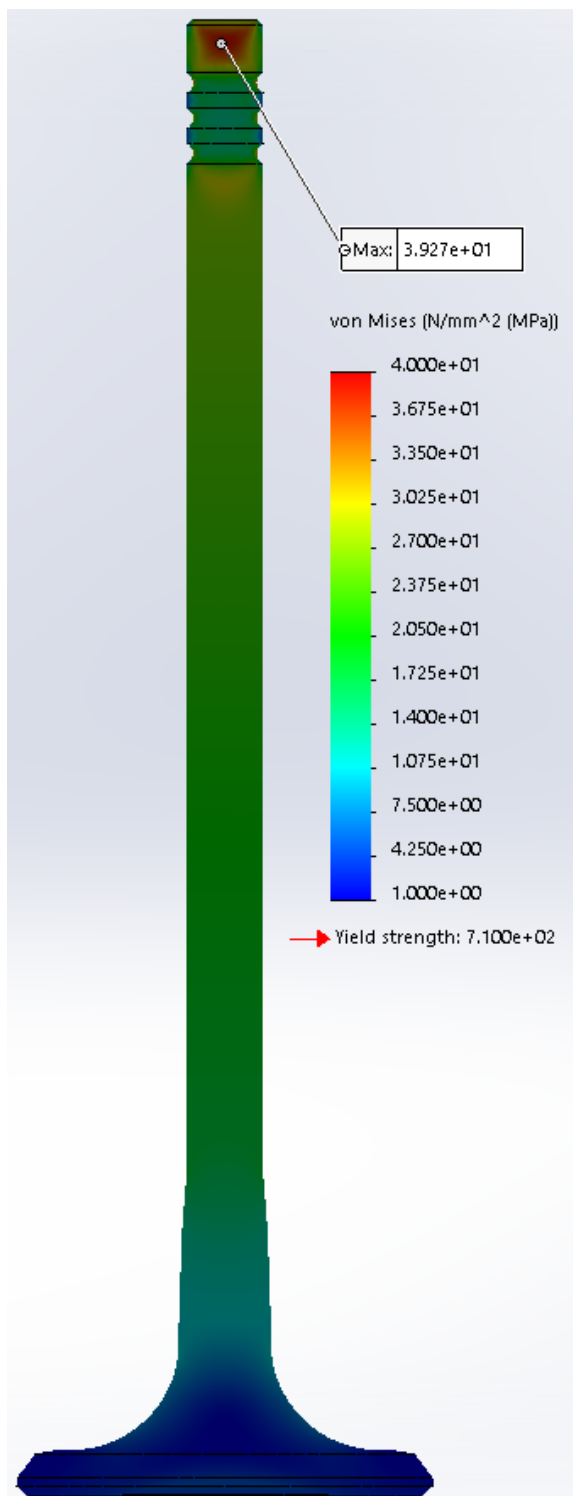


Figure 3-5: Section Stress Plot frame 432

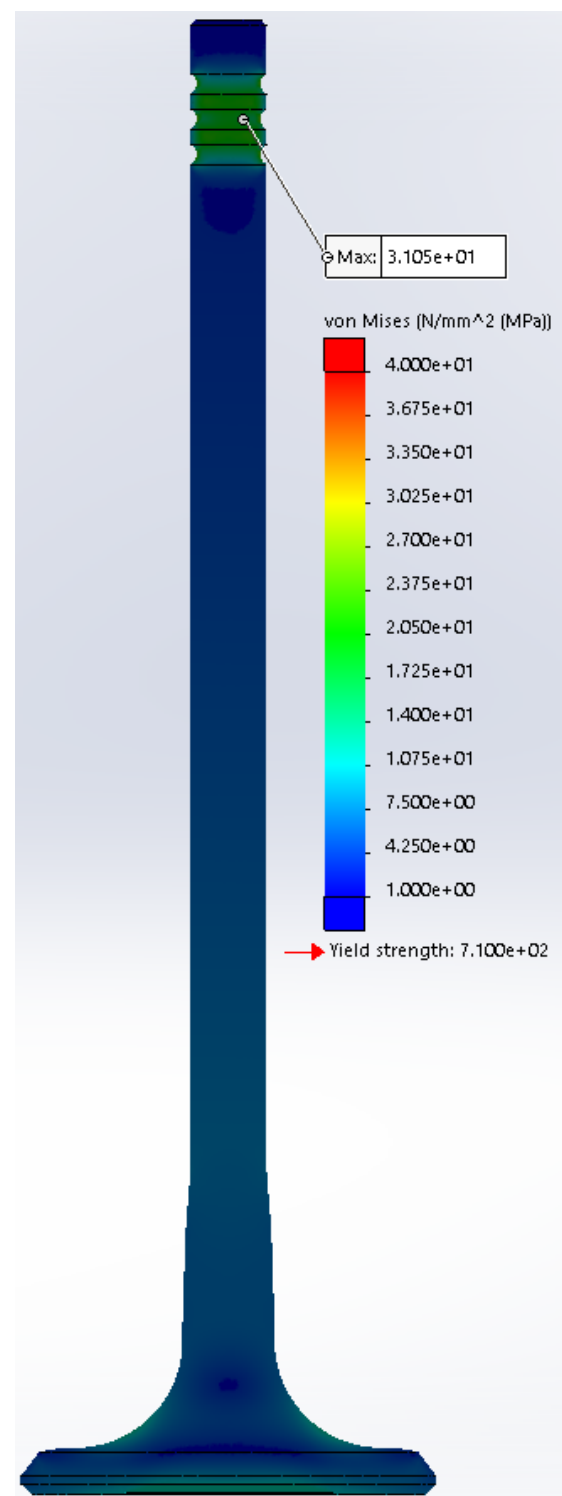


Figure 3-4: Section Stress Plot frame 1663

Figure 3-4 and Figure 3-5 show the results of the simulation. They prove what author expected: higher stress in the head of the valve at frame 432 due to 3.58 times higher total stress input as well as much closer distance between the load faces. However, these results also contradicted author

expectation of higher stress in the stem of the valve at frame 1663. Nonetheless, both of those plots show that it will be very difficult to improve the design, due to the dynamic nature of a valve.

As mentioned in chapter 2.3, the development methodology created for this investigation is to set minimum target factor of safety (FOS) values based on the minimum values calculated by the simulation. This ensures that all of the unaccounted-for factors (some of which have already been mentioned) such as fatigue and heat cycling, effectively won't change and as a result the reliability of the valve should not deteriorate.

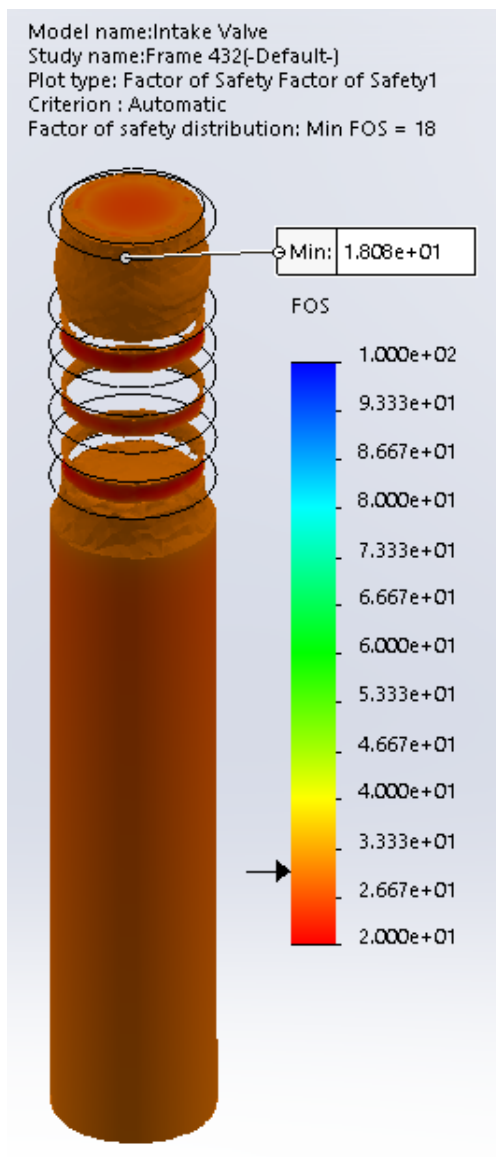


Figure 3-7: Iso-Clipping Stress Plot frame 432

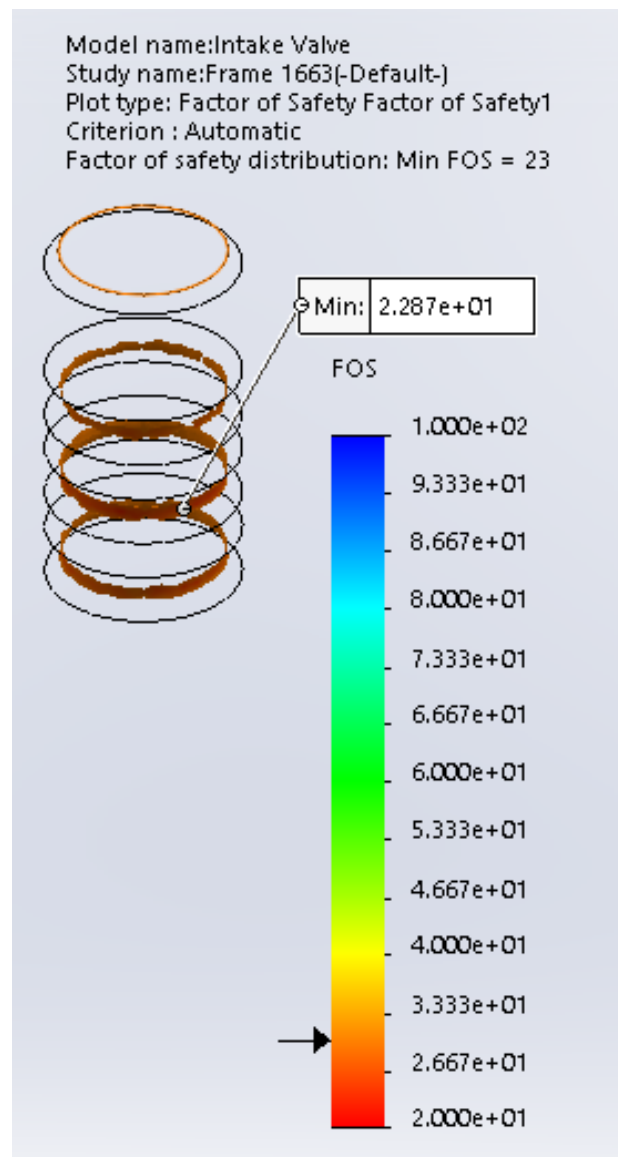


Figure 3-6: Iso-Clipping Stress Plot frame 1663

As shown by Figure 3-6 and Figure 3-7, due to much higher stress as well as stress consecration at frame 432, the FOS is much lower throughout the head and the stem, in comparison to frame 1663. Also, the minimum FOS values for frame 432 and 1663 are 18.1 and 22.9 respectfully, making them the minimum targets/ criteria. Bottom of the valve wasn't shown as it is above FOS 30 in both scenarios.

3.2 Valve Development

The aim of this investigation is to reduce the valve mass, since this would allow softer springs. It is important to remember that in modern engines such as the 1.0l GTDI Ford Fox which this valve came from, there are at least 2 valves per cylinder. Also, it is common for the cylinder count to be anywhere between 3 and 6, making spring rate reduction worth essentially anywhere between 6 to 12 times. Additionally, the friction rises to the speed squared. All of these factors effectively cause a big difference in valvetrain friction from a small spring rate decrease.

Before developing the valve, author believed it to be of great importance to make sure that any changes could have a possibility of being introduced by Ford Motor Company in mass production. Therefore, organic shapes which require additive manufacturing will not be considered, as well as anything else which cannot be done with traditional machining and grinding, which the current factories are already setup to do.

3.2.1 Wasted Core

The current traditional way of manufacturing engine poppet valves is to friction weld the stem onto a blank which is then forged and ground into shape, naturally the weld joint as well as different blank material isn't simulated in this investigation. Author believes it to be feasible to predrill the stem, effectively wasting the core and reducing the valve mass. Therefore, author has setup a design study which aims at optimising the depth and side of the cylinder which could be removed as shown in Figure 3-8.

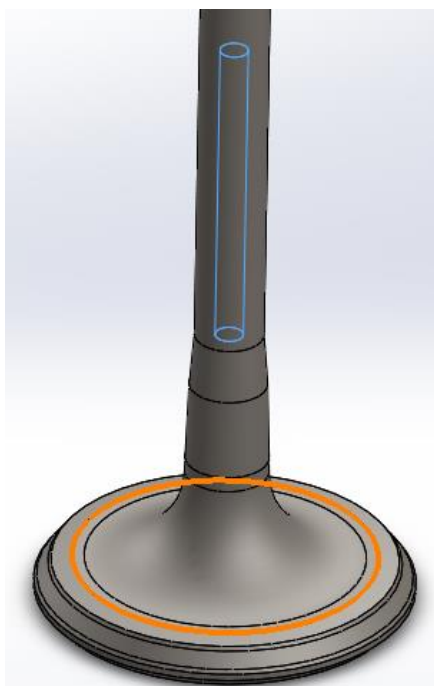


Figure 3-8: Wasted Core

However, as Figure 3-9 shows, no combination of diameter and depth would result in valve with minimum FOS of 18.0, when simulated at frame 432.

		Current	Initial	Optimal	Iteration 1	Iteration 2	Iteration 3	Iteration 4	Iteration 5	Iteration 6	Iteration 7	Iteration 8	Iteration 9
Diameter		0.1mm	0.1mm		4mm	0.1mm	4mm	0.1mm	4mm	0.1mm	2.05mm	2.05mm	2.05mm
Depth		0.1mm	0.1mm		50mm	50mm	0.1mm	0.1mm	25.05mm	25.05mm	50mm	0.1mm	25.05mm
Minimum Factor of Safety ¹	> 18.000000	17.878599	17.878599		7.830619	4.531437	5.627211	17.878599	9.167692	6.158180	16.591099	11.264915	17.918911
Mass3	Minimize	29.48689 g	29.48689 g		24.55459 g	29.48381 g	29.47703 g	29.48689 g	27.01581 g	29.48535 g	28.19139 g	29.4843 g	28.83785 g

Figure 3-9: Wasted Core Design Study Results

3.2.2 Topology Study

Due to poor results of traditional design studies, author has decided to run a topology study which would show where the material can be taken away. Once this is done, author can decide how to implement it so it is feasible to manufacture.

3.2.2.1 Setup

The topology study was setup exactly the same as the static study at frame 432. However, there were three additional topology study specific parameters:

- Goals: set to best stiffness to weight ratio, as the aim was to minimize the mass while retaining the structural integrity. The aim for the reduction of mass was set to 10% since the previous chapters showed that the investigated component has already been well designed. Naturally, the mass wasn't overridden for this study.
- Manufacturing Controls:
 - User decided to preserve the regions of the valve tip, collet faces, all valve cuts as well as everything past the valve seat. This was to make sure that the flow, compression ratio or other valvetrain components wouldn't change. 1mm aerial depth was also set to protect the preserved regions.
 - Quarter symmetry was applied since this significantly cuts down the computational time. In this case quarter symmetry was used since the entire component is revolved around single axis.

3.2.2.2 Results

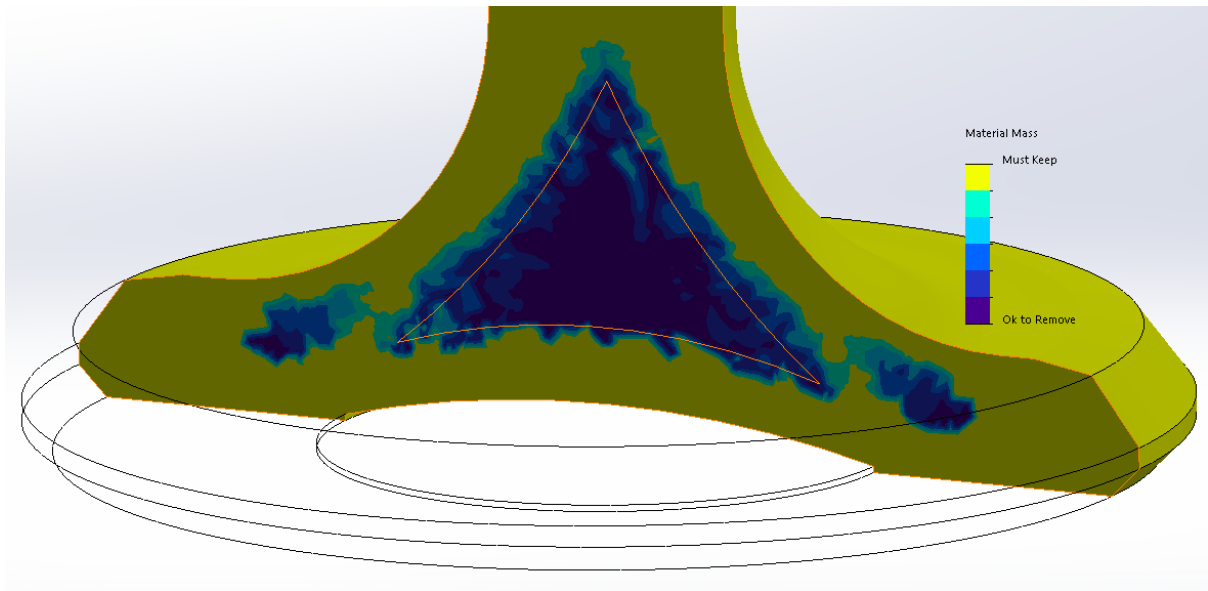


Figure 3-10: Topology Result

Figure 3-10 shows that the only place where material can be removed without negatively impacting the stiffness of the valve is in the core of the valve which is originally forged. This proves that the valve design done by Ford is very good and there is no design alteration which would keep the reliability, improve performance and be feasible in mass production, all at the same time, while keeping the original material.

3.2.3 Final Design

Author has decided to disregard the manufacturing feasibility to find out what the mass of an optimal design would be. Therefore, author has implemented the removal sections from Figure 3-10 and re-ran static studies at frames 432 and 1663. The results are shown in Figure 3-11 and Figure 3-12.

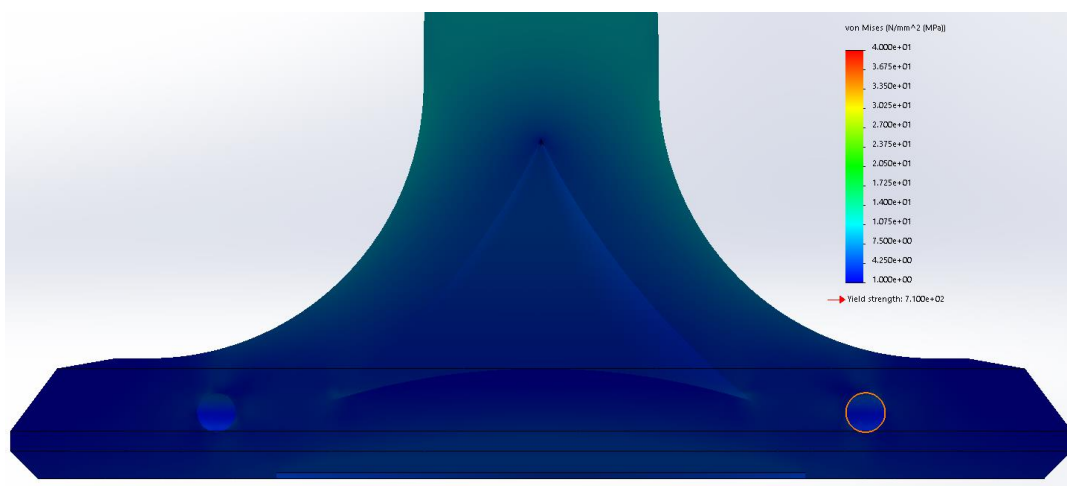


Figure 3-11: Section Stress Plot of Wasted Valve at frame 432

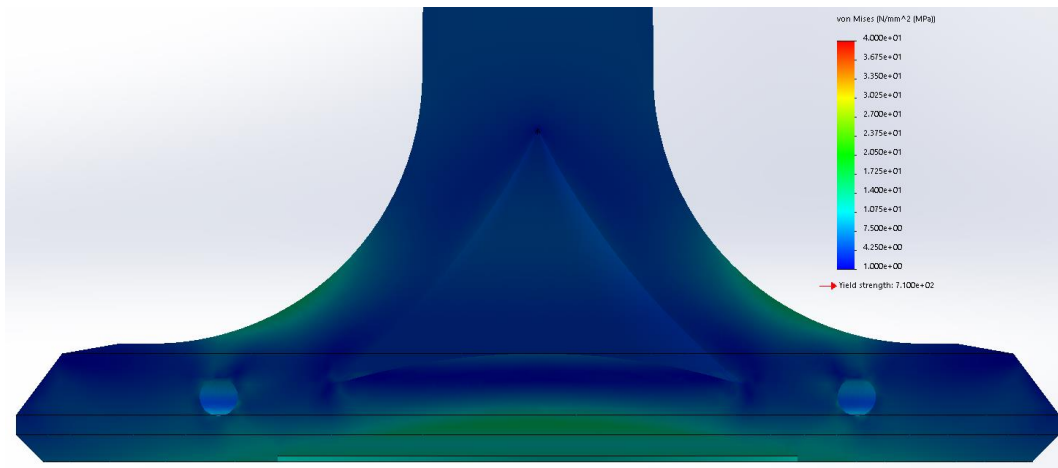


Figure 3-12: Section Stress Plot of Wasted Valve at frame 1663

The core wasting has clearly caused the stress to increase at the bottom section of the valve, especially at frame 1662. However, the stresses didn't increase anywhere near the peak stresses, therefore FOS around the wasted core stayed over double the minimum required, as shown by Figure 3-13 and Figure 3-14, making the new design theoretically reliable.

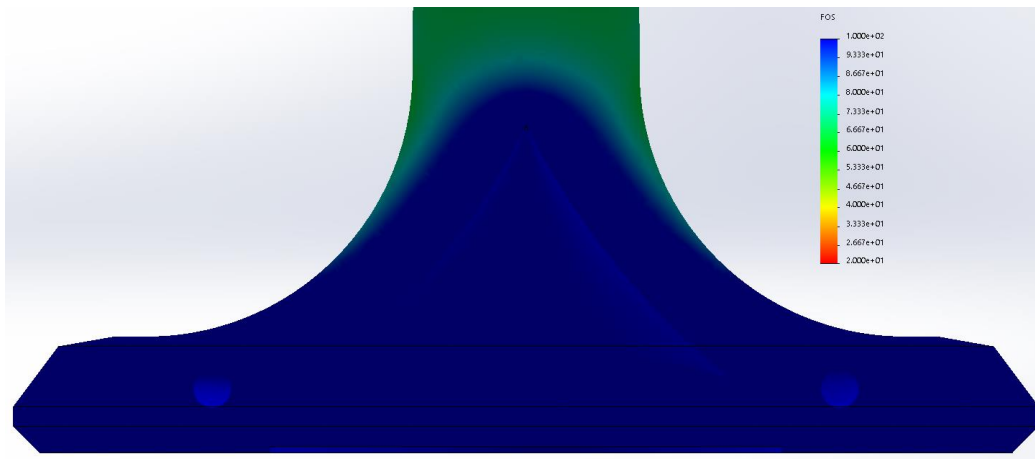


Figure 3-13: Section FOS Plot of Wasted Valve at frame 432

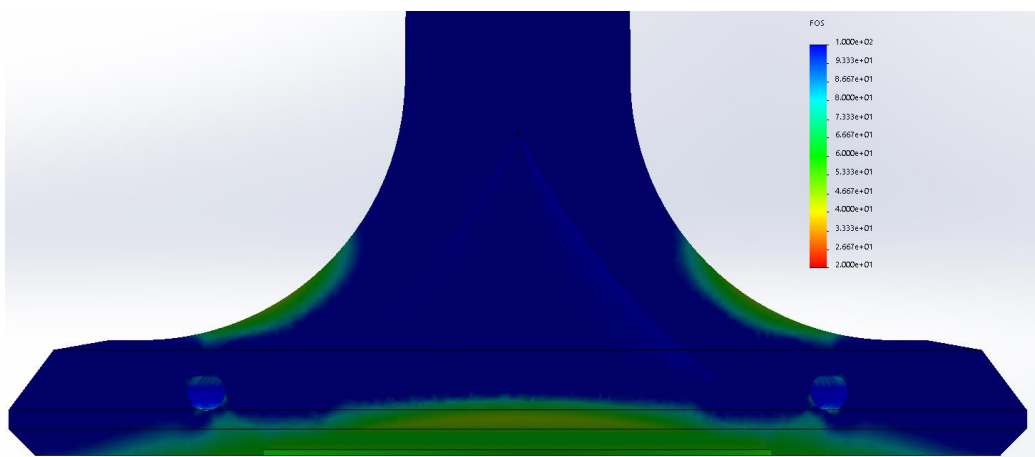


Figure 3-14: Section FOS Plot of Wasted Valve at frame 1663

By dividing the original valve volume to the new 'wasted core' volume, a mass decrease of 4.34% or 1.22g was calculated, making the mas of the new valve 26.96g. Author believes this change to be way too small for real life implementation to be even considered due to the challenging geometry.

4 Conclusion

This investigation has shown many weaknesses in author's original methodology. Many were caused by lack of experience with mechanical development. However, author was able to suggest many improvements, most of which would be relatively easy to implement, possibly increasing the accuracy of the end results by a big margin.

Due to lack of data, author was forced to create a plethora of assumptions, such as no elevated temperature and incorrect material. Most if not all are most likely detrimental to the accuracy.

However, by studying original design, author was able to create FOS targets which stopped author from implementing design changes which would definitely have negative impact on the reliability of the component.

Author used Topology study to find out the locations of which material can be removed and after implementing this into the design, the results showed minimal increase in stress. This has resulted in component which was 4.34% lighter and theoretically as reliable as before. However, the geometry was too organic with too little access to even consider implementing in real life.

Naturally, more time and research would have to be done before any design implementations should be seriously considered. Nevertheless, author believes this investigation to be a good starting point, from which both the model and understanding can be built up on.

5 References

Kaufman, D., 2006. Understanding Valve Design and Alloys.

To access the CAD files please go to:

<https://uowtsd->

[my.sharepoint.com/:f:/g/personal/1701199_student_uwtsd_ac_uk/EpBPnG3tOgBBiac1I79C5TkBn_bI_Gw8nZ-W5ard4G9QaQ?e=7aV4Af](https://uowtsd-my.sharepoint.com/:f:/g/personal/1701199_student_uwtsd_ac_uk/EpBPnG3tOgBBiac1I79C5TkBn_bI_Gw8nZ-W5ard4G9QaQ?e=7aV4Af)