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by Jon Lewis

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Investigation of a Ford 1.6 GDI Connecting Rod for High RPM Use

Powertrain Systems

Jon Clive Lewis
College of Engineering
University of Wales Trinity Saint David
Swansea UK
P146739

JLewis.146739@student.uwtsd.ac.uk

Abstract - The ford ecoboost engine series have now become common place in all ford production vehicles, therefore it would make great sense for ecoboost engines to be used in race applications. This investigation aims to see what impact raising the rpm limit has on the connecting rods as well as finding a viable engine configuration at this higher rpm. The biggest benefit to raising the RPM of an engine is greater power can be achieved however greater stresses are also incurred.

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2 Nomenclature

Pr – Power

T – Torque

W - Angular velocity (RPM'S)

P – Pressure (N/M²)

D – Diameter (M)

M – Piston Mass (KG)

R – Crank Radius (M)

Θ – Crank Angle

Lr – Rod Length (M)

X – Stretch (M)

A – Area (M²)

L – Length (M)

E – Youngs Modulus (GPa)

Rn – Number of Rotations

GDI – Gasoline Direct Injection

RPM – revolutions per minute

WLTP – World Harmonized Light Vehicle Test Procedure

CO₂ – Carbon Dioxide

UTS – Ultimate Tensile Strength

TDC – Top Dead Centre

F_{gas} – Gas Force (N)

F_{inertia} – Inertia Force (N)

F_{net} – Net Force (N)

EGR – Exhaust Gas Recirculation

3 Project Aims

The major aims of this investigation are:

1. To simulate the raise in rev limit from 6000RPM to 9000RPM for a ford GDI engine.
2. To analyse and redesign possible weak point at this higher RPM.
3. To maintain a viable safety factor.

4 Introduction

The power an engine can produce is limited by the overall size of the engine or displacement and the speed the engine can achieve. However As vehicle manufacturers are faced to comply with the demand for lower emission vehicles enforced by government regulations [1], engine downsizing has become more and more an occurrence. Due to this engineering shift manufactures such as ford have concentrated on small capacity turbocharged powerplants to power their vehicles. Therefore an alternative to increasing the capacity of an engine would be to increase the RPM limit the engine can reach.

This investigation will focus on the ford 1.6 ecoboost engine as ford engines have long been used for race application for both hobbyist use and more serious competition use. Ford engines are a popular choice due to being inexpressive and plentiful with the older generation engines such as the ford Kent, pinto and duratec models still being used in competitions. the factory 1.6 ecoboost unit has a redline of 6000RPM. This investigation will look to push the limit of the engine to 9000RPM to find the short comings of the engines connecting rods and see what changes can be made to ensure safe operation. The major focus will be to investigate the stresses on a factory connecting rod by means of hand calculations and Solidworks Simulation. The aim is to find the conditions the factory unit operates at its maximum RPM then with the aid of Solidworks simulation find a viable connecting rod design that approximately replicates these condition at 9000rpm.

5 Research

5.1 Engine Components

5.1.1 Piston

A Piston is a Hollow Cylinder that is attached to a connecting rod via a pin know as a gudgeon pin. Pistons are the initial components that the combustion forces act upon. The crown of the Piston transfers the forces resulting from the combustion of a fuel air mixture into a compressive force which then transfers the linear movement of the piston via the connecting rod into rotary movement by means of a crank shaft [2].

5.1.2 Gudgeon Pin

A gudgeon pin is a hardened steel pin generally made from a forged steel alloy. The gudgeon pin acts as a bearing allowing the connecting rod to pivot as the piston moves [3].

5.1.3 Connecting rods

Connecting rods are the straight sections that connect the piston to the crankshaft. It joins the crank pin to the piston pin with the small end is attached to the piston and the big end to the crank. The job of the con rod is to covert the linear motion of the piston moving down the cylinder bore into rotary motion to rotate the crankshaft and there after the driveline of the vehicle. Connecting rods are generally produced by forging from wrought steel or powdered

metal or alternatively they can be cast [4]. Connecting rods are generally the weakest part of a turbo charged engine when the power is increased [5] [6] [7] [8].

5.1.4 Crankshaft

The crankshaft converts the vertical motion of the pistons into horizontal rotational movement [9]. The crankshaft rotates a flywheel which via a gearbox rotates the wheels. The crankshaft keeps the pistons in sequence as they are physically linked together in a stepped orientation. These steps or Crankpins are positioned so the pistons can reach either TDC or BDC [10].

5.2 Torque and Rpm

The idea of a race engine is to produce the maximum amount of power allowable for a given race series. The power an engine produces is a function of its RPM and torque. Therefore, an increase in either torque or RPM and the power will increase.

$$Pr = (T \cdot W) / \text{radians per sec}$$

The torque an engine produces is result of the force of combustion multiplied by the distance the piston is moved. Therefore, the larger the combustion area the larger the torque. This is one reason American vehicle manufacturers such as Ford and General Motors have long held the tradition that displacement is key to high power figures with the continual release of such powerplants as the Ford Coyote V8 [11] used in the Mustang and F-150 pickup which varies in displacement from 4.6 to 5.8L. As well as the incredibly popular LS series of motors released by Chevrolet and GM with releases such as the 7L LS7 or the 6.2L LS3 engines.

5.3 Emissions

However environmental regulations have begun to crack down on emissions produced, such as the relevantly new WLTP or world harmonised light vehicle procedure which as of 1st of January 2019 has become mandatory for all new vehicles sold in Europe [12]. This procedure has come into effect after the European parliament put forth regulation 443/2009 emissions standards [1] which states that all current light vehicles produced must produce no more than 130g CO₂/km. Furthermore as of 2020 emissions for fleets will drop from 120g CO₂/km to 95g CO₂/km (see appendices). Due to this and other factors such as fuel consumption, packaging and popularity. Ford has begun selling smaller engine vehicles to its European markets.

5.4 Ford Eco-Boost

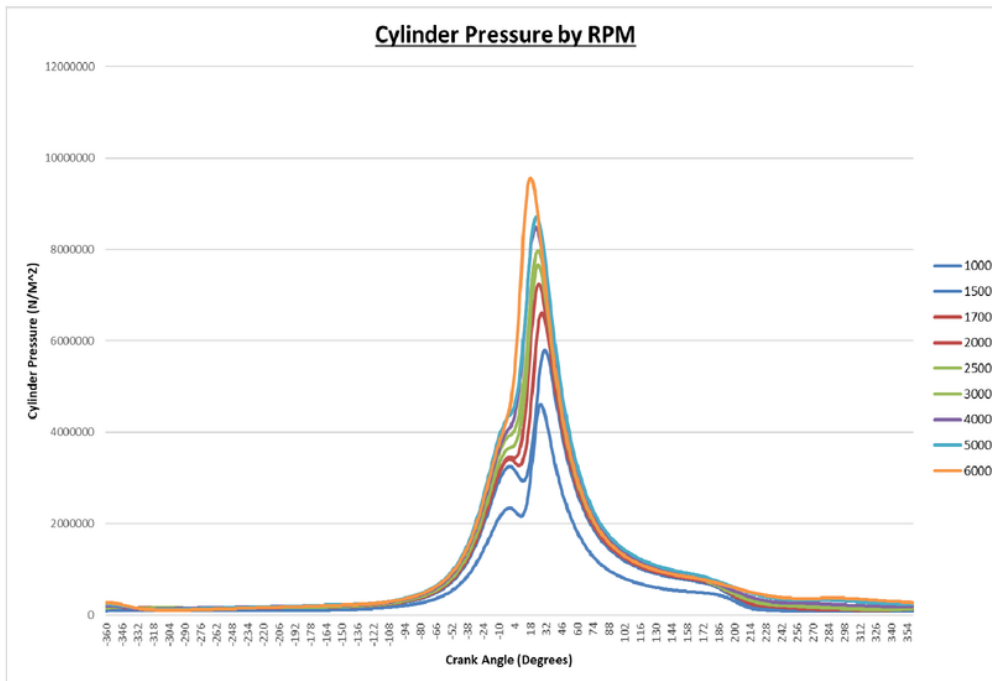
The Ford EcoBoost engine series debuted in 2009 replacing the older Duratec range. The Ford EcoBoost engine series comes in 3, 4 and 6 cylinder versions and various displacements ranging from a 1L inline 3 cylinder engine to a 3.5L V6 engine [13]. The Ford EcoBoost engines are a series of turbo charged units with direct gasoline injection or GDI. The EcoBoost series can be split into two sectors, with the 1-1.6L engines being the low emission economical versions targeted for use with the European markets and the 2-3.5L versions being targeted for sale in America and the rest of the world where emissions regulations are less stringent, and performance is more of a concern. An additional note would be the highest inline 4 cylinder units being the 2L and the 2.3L are not technically EcoBoost units as they are built on the Mazda L platform and have only mild changes to the cylinder heads and fuel injection system. The 1.6L EcoBoost engine has recently started to be used in both Formula Ford and the BTCC [14] [15].

6 Data

Initially pressure data from the GDI engine was provided by Malcolm McDonald (UWTSD Lecturer). This data was at various intervals between 1000 to 6000rpm. This data was modelled in excel to produce a peak pressure graph and then to estimate the pressure at 9000RPM. This estimation was done by calculating the average pressure increase per 1000RPM and then multiplying the current pressure values to suit.

RPM	Max Pressure N/M^2	Difference (2-1)	Multiplier (2/1)
1000	7248850		
2000	7649800	400950	1.0553
3000	7979125	329325	1.0431
4000	8481425	502300	1.0630
5000	8723500	242075	1.0285
6000	9560600	837100	1.0960

Table 1 Initial cylinder pressure values

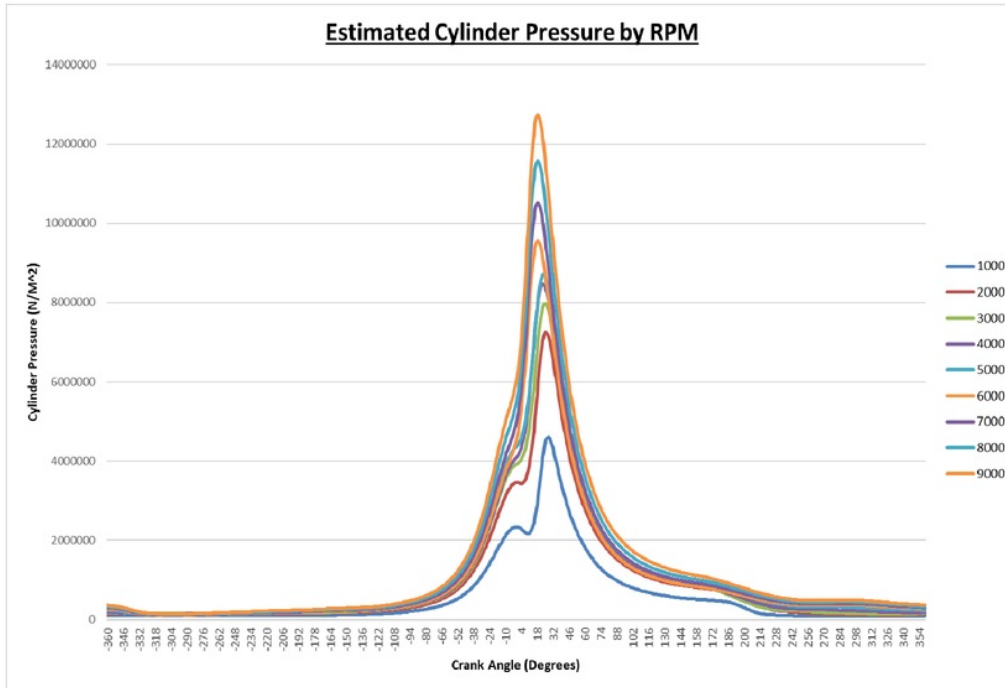


Graph 1 Cylinder pressures by RPM

As a maximum difference of 1.096 was found, to be safe it was rounded to 1.1 and then values for 7000, 8000 and 9000RPM were calculated.

RPM	Max Pressure N/M^2	Multiplier
7000	10516660	1.1
8000	11568326	1.1
9000	12725158.6	1.1

Table 2 Estimated cylinder pressures



Graph 2 Estimated and recorded cylinder pressures

From the Cylinder pressure the force of the gas on the piston or F_{gas} can be found by multiplying the Cylinder pressure by the bore area, the formula is shown below.

$$F_{gas} = P * \left(\frac{\pi * D^2}{4} \right)$$

RPM	Max Pressure N/M^2	Fgas N
1000	7248850	36436.69
2000	7649800	38452.09
3000	7979125	40107.46
4000	8481425	42632.29
5000	8723500	43849.09
6000	9560600	48056.82
7000	10516660	52862.5
8000	11568326	58148.75
9000	12725158.6	63963.62

Table 3 Gas Force at each RPM

6.1 Inertia Forces

The total force applied to the piston or the Net force the piston receives equates to the Gas Force minus the Inertia Force. The inertia force varies in accordance to the pistons position.

The formula to find the Inertia force ($F_{inertia}$) is shown below.

$$F_{inertia} = MRW^2 \cos \theta + \frac{R}{Lr} MRW^2 \cos 2\theta$$

RPM	Crank Angle at Max Fgas Occurs (Degrees)	Rod Angle at Max Fgas Occurs (Radians)	Angular Velocity	Inertia Force (N)		
1000	28	0.1366	104.72	246.99		
2000	26	0.1275	209.44	989.17		
3000	25	0.1229	314.16	2226.91		
4000	23	0.1136	418.88	3963.34		
5000	24	0.1182	523.60	6189.35		
6000	18	0.0897	628.32	8939.21		
7000	18	0.0897	733.04	12167.25	Piston Mass (Kg)	0.3475
8000	18	0.0897	837.76	15891.92	Rod Length (M)	0.134
9000	18	0.0897	942.48	20113.21	Crank Radius (M)	0.0407

Table 4 Inertia forces at each RPM

6.2 Net Forces

From the Gas forces and the inertia forces the net forces on the piston can now be found.

The net forces are split into two sections, maximum tensile and maximum compressive forces. This is explained by whether the piston is in a compression stroke or power stroke.

Engine Speed RPM	Gas Force N	Inertia Force N	Net Force	
			Max compressive force N	Max tensile force N
1000	23100.51	249.31	22910.88	201.65971
2000	36436.69	997.26	35647.07	-340.304726
3000	40107.46	2243.83	38297.31	-1449.63278
4000	42632.29	3989.03	39300.82	-3054.12014
5000	43849.09	6232.86	38730.79	-5057.6529
6000	48056.82	8975.32	40101.45	-7617.41938
7000	52862.50	12216.41	42035.22	-10720.933
8000	58148.75	15956.12	44007.73	-14310.2835
9000	63963.62	20194.47	46067.06	-18383.1856

Table 5 Net forces at each RPM

6.3 Rod Forces

To calculate the forces that the connecting rod faces, a cross sectional area known as a shank is used. This shank uses the minimum cross sectional area i.e. where the rod is thinnest and therefore weakest. The shank cross sectional area was found to be 142mm². The forces on the rod are again split into compressive and tensile forces however the compressive force on the rod is found by dividing the gas force by the cross sectional area and the tensile force on the rod is found by dividing the inertia force by the cross sectional area.

RPM	Gas Force N	Inertia Force N	Rod max compressive force	Rod max tensile force	
1000	23100.51	249.31	162.68	1.76	
2000	36436.69	997.26	256.60	7.02	
3000	40107.46	2243.83	282.45	15.80	
4000	42632.29	3989.03	300.23	28.09	
5000	43849.09	6232.86	308.80	43.89	
6000	48056.82	8975.32	338.43	63.21	
7000	52862.50	12216.41	372.27	86.03	Rod area mm ²
8000	58148.75	15956.12	409.50	112.37	
9000	63963.62	20194.47	450.45	142.21	142

Table 6 Rod Shank Forces at each RPM

6.4 Safety Factor

The safety factor of an engine part is a critical design point as it determines the overall component strength. The safety factors of the rod is found by dividing the tensile or compressive force by the ultimate tensile strength of the material. In this case the UTS of a steel con rod is approximately 750 MPa.

Engine Speed RPM	Rod max compressive force N	Rod max tensile force N	Safety factor compressive	Safety factor tensile	
1000	162.68	1.76	4.61	427.17	
2000	256.60	7.02	2.92	106.79	
3000	282.45	15.80	2.66	47.46	
4000	300.23	28.09	2.50	26.70	
5000	308.80	43.89	2.43	17.09	
6000	338.43	63.21	2.22	11.87	
7000	372.27	86.03	2.01	8.72	Rod UTS (MPa)
8000	409.50	112.37	1.83	6.67	
9000	450.45	142.21	1.67	5.27	750

Table 7 Safety Factors at each RPM

6.5 Rod Stretch

To calculate the possible stretch the conrod might receive the net forces are multiplied by the rod length (L) and the shank area (A) is multiplied by the young's modulus (E) and then the two results are divided. The formula is shown below.

$$X = \left(\frac{F_{net} * L}{A * E} \right)$$

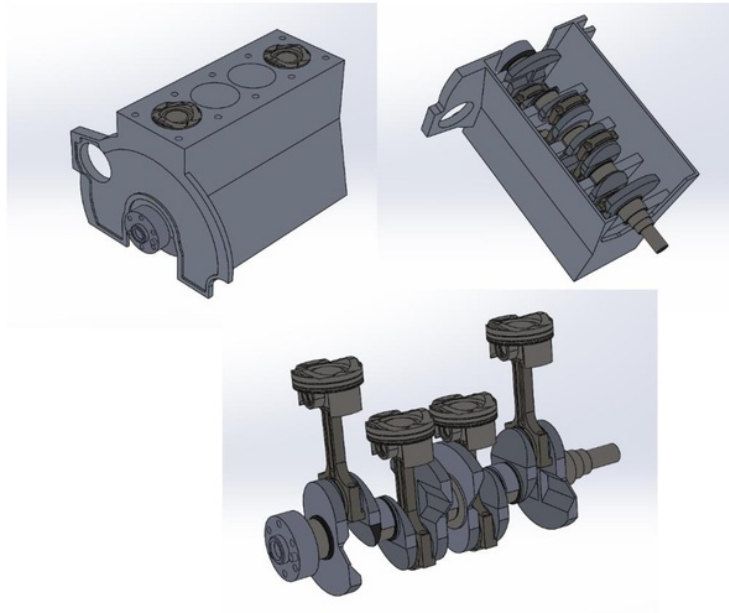
Using this formula the stretch the rod receives at each RPM can be calculated, with negative values equating to rod compression. Please note the greyed cell at 1000rpm is to be negated as the minimum stretch value does not go into compression.

Engine Speed RPM	Net Force		Max Stretch mm	Min Stretch (Compression) mm		
	Max compressive force N	Max tensile force N				
1000	22910.88	201.66	0.10810	0.0009515		
2000	35647.07	-340.30	0.16819	-0.0016057		
3000	38297.31	-1449.63	0.18070	-0.0068398		
4000	39300.82	-3054.12	0.18543	-0.0144103		
5000	38730.79	-5057.65	0.18274	-0.0238636		
6000	40101.45	-7617.42	0.18921	-0.0359413		
7000	42035.22	-10720.93	0.19834	-0.0505847	Youngs Modulus (GPa)	2.00E+11
8000	44007.73	-14310.28	0.20764	-0.0675204	Rod Length (M)	0.134
9000	46067.06	-18383.19	0.21736	-0.0867376	Rod Area (M^2)	0.000142

Table 8 Rod Stretch at each RPM

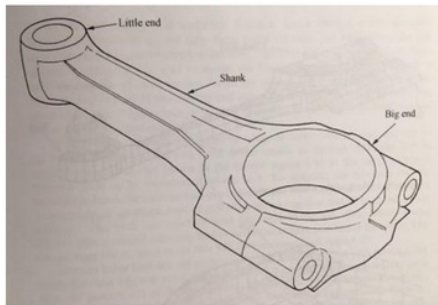
7 Solid works Model

To test the stresses and find the short comings of the piston assembly, a simplified version of the 1.6 ecoboost was modelled in Solidworks. This model was created using measurements taken from a disassembled unit and reverse engineered to get accurate version of the rotating assembly i.e. Crankshaft, piston, gudgeon pin, connecting rod etc. Dimensions for the key components used can be found in the appendices. As the focus of this report was on the internals of the engine, a simplified engine block was used. This block maintained key measurements such as crank position in correspondence to the cylinders and cykinder head, cylinder sizing, cylinder spacing etc.



Picture 1 Images of the engine construction used [16]

The ecoboost connecting rods utilise a one piece design which is then Fractured mid-way at both the big end and small end to create a matting face with a unique break pattern to ensure that the end caps have error-proof assembly and are fitted the correct way [17]. Also the break pattern gives the matting surface a third dimension and a larger surface area which when compared to conventional connecting rods gives a better stronger connection. For simulation purposes the connecting rods were modelled as a single piece including the caps.



Picture 2 Connecting Rod Layout [18]



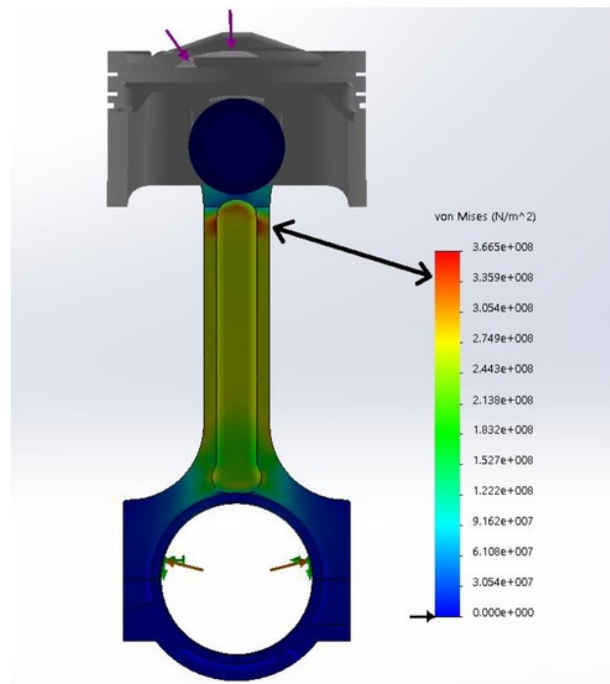
Picture 3 Fracture Type Connection [19]

To validate the design, the rod was weighed (see appendices) and validated using Solidworks evaluation tools. The real rod was found to be 367g and the modelled rod was 375g, a slight discrepancy of 8g was found. This could be due to a slight inaccuracy of the scales used as household scales were used in place of a smaller scale. As these scales are used for larger masses this could be the cause of the difference. Additional considerations would be imperfections due to component wear or the forging processes i.e. rough surfaces, excessive material etc. Other points would be slight differences in the material density as the model uses an ideal figure where as in real world situations this figure can differ. The final point would be the Solidworks model lacks the threads for the bolts which would add a small

weight. Due to the weight difference being merely a few grams, the difference will be omitted for the purposes of the simulation.

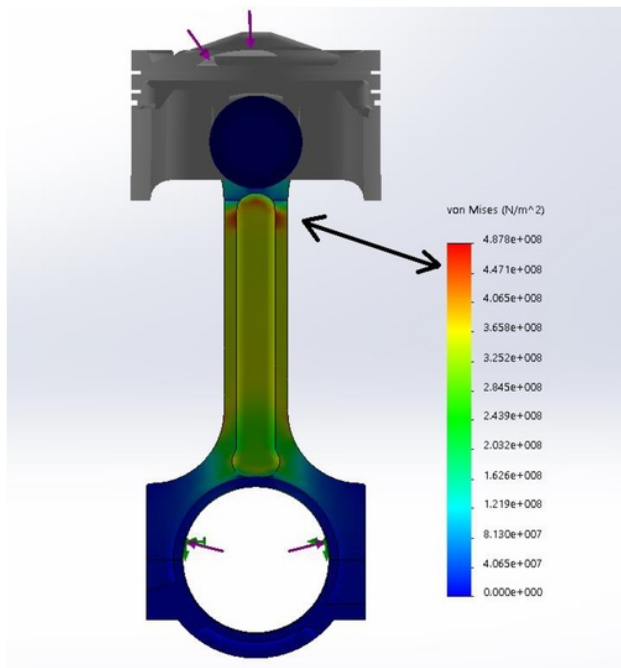
7.1 Static Simulation

For the initial testing of the factory connecting rod the piston assembly was modelled, the connecting rod was fixed at the big end where the rod connects to the crankshaft. Next a pin connection was made between the gudgeon pin and the small end of the connecting rod. Another two pin connections were made with the gudgeon pin and the two areas of the piston that contacts the gudgeon pin. Next the Gas force was applied to the top of the piston, the figure used for the gas force was 48056.82N which was the maximum gas force at 6000rpm. An additional force of 8975.32N was applied to the big end of the rod in the opposite direction of the gas force. Finally the assembly was meshed, and the simulation was run.



Picture 4 Factory rod simulation results @ 6000RPM

The results of the simulation shown in picture 4 show that the maximum compressive stress on the factory rod to be approximately $3.36 \times 10^8 \text{ N/M}^2$ or 336000000 N/M^2 which equates to 336 N/mm^2 , only 2 N/mm^2 difference from the calculated stress shown in table 6. This is approximately a 0.59% difference. Next the simulation was repeated but with the gas and inertia force values changed in accordance with table 6 to simulate the stresses at 9000RPM.



Picture 5 Factory rod simulation results @ 9000RPM

The results of the simulation shown in picture 5 show that the maximum compressive stress on the factory rod to be approximately $4.47 \times 10^8 \text{ N/M}^2$ or 447000000 N/M^2 which equates to 447 N/mm^2 , only 3 N/mm^2 difference from the calculated stress shown in table 6. This is approximately a 0.66% difference.

8 Design

Once the factory connecting rod design was validated with the calculations in section 7 a new rod could be designed to match the performance of the factory unit at this new higher operating speed. The new designs would cover three areas; material change, thickness change and design change.

8.1 Material choice

The first method of improving the rod would be to maintain the overall design and dimensions but to change the material properties. The current rod is made from a powder forged aluminium alloy. This material and method is used due to the inexpensive nature of the Aluminium and the large production volume capability. A disadvantage to this method would be the large initial costs for the dies/moulds used to produce the rods which can cost thousands and as these dies can wear out the cost can be repeated when replacements are required. Therefore this method therefore is only a viable production method for large scale production [20]. Two other commonly used materials for stronger connecting rods are billet 4340 steel and a titanium alloy Ti-6Al-4V.

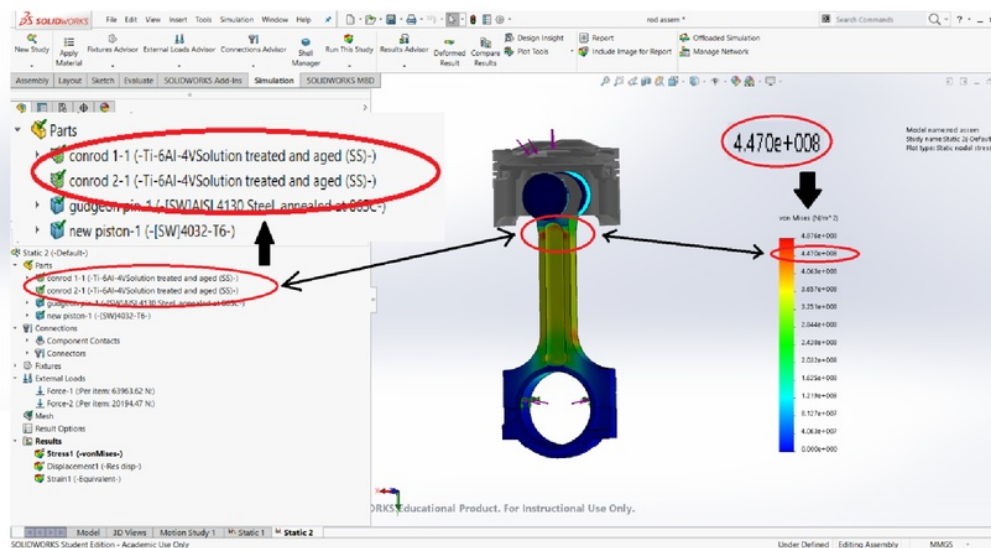
4340 steel rods are the most common type used for high performance applications, they are milled from a single piece of forged steel using CNC machines in a CAD CAM process. The disadvantage to this process is the long production time to low production volume when compared to press forged factory rods.

Ti-6Al-4V grade titanium is used by many high end automobile manufacturers since the early nineties such as Ferrari, Bugatti and Koenigsegg for its low density and high strength applications [21]. The major disadvantage with titanium is the cost and therefore can only really be used in small batch production.

	4032 Aluminium	4340 Steel	Ti-6Al-4V Titanium
Tensile strength - N/m ²	380000000	745000000	1050000000
Yield Strength - N/m ²	315000000	470000000	827370880
Mass Density - Kg/m ³	2680	7850	4428.78
Poisson's ratio	0.34	0.285	0.31
Elastic Modulus - N/m ²	7.90E+10	2.05E+11	1.05E+11

Table 9 The Material characteristics used in Solidworks model.

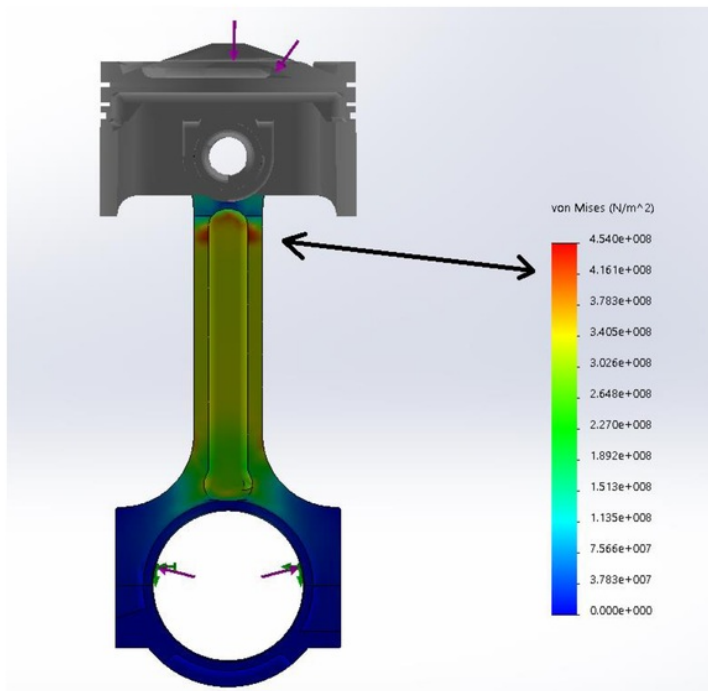
The results at 9000rpm for 4340 steel and the titanium alloy were from testing the three different material the results were 4.469×10^8 N/m² and 4.470×10^8 N/m² respectively. Comparing this to the factory rod at 4.471×10^8 N/m², changing the material by its self doesn't increase the strength enough.



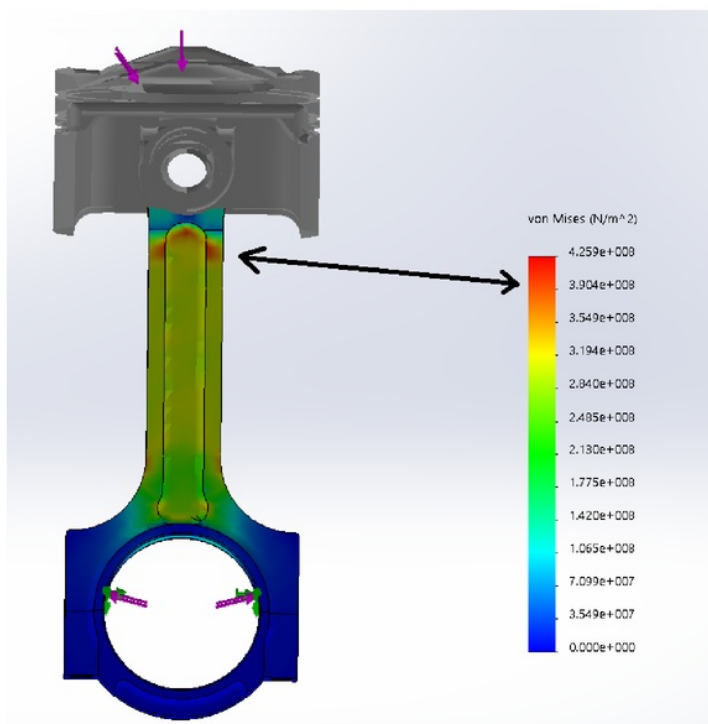
Picture 6 Titanium Rod @ 9000RPM

8.2 Material Thickness

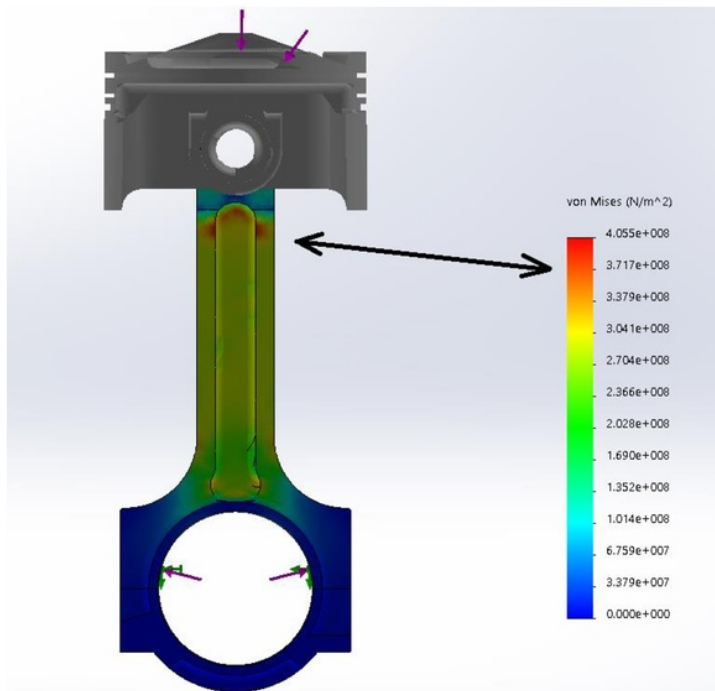
To see the effects of the material thickness compared to the strength of the rod the factory unit was used. The model thickness was increased from 22mm by 2mm three times whilst the material choice stayed the same for all three runs.



Picture 7 Factory conrod with 21mm shank @9000RPM



Picture 8 Factory conrod with 22mm shank @9000RPM



Picture 9 Factory conrod with 23mm shank @9000RPM

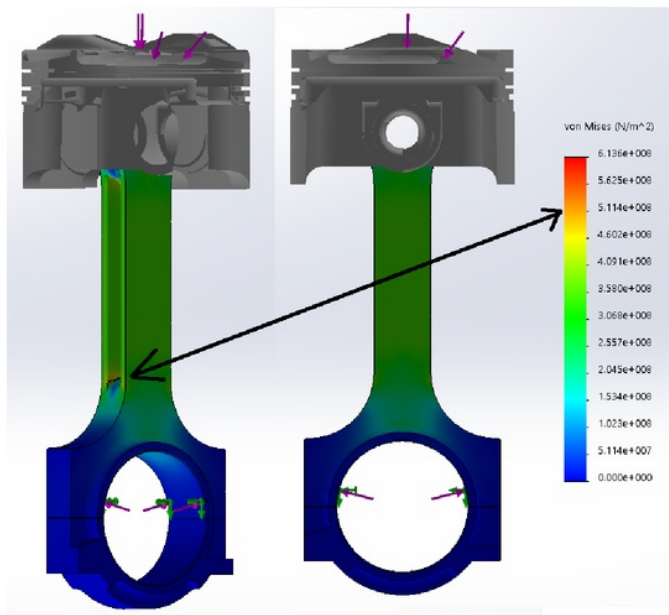
Shank size mm	Tensile strength N/mm ²	Difference
20	447	
21	416	31
22	390	26
23	372	18

Picture 10 Shank size differences

From the results of these simulations it is clear that increasing the thickness of the material can dramatically increase the strength of the connecting rod.

8.3 Design Change

The factory unit uses an I beam design; other popular designs are H beam rods and Solid rods. The advantage of the I beam design is that it is generally the lightest design which reduces reciprocating mass which should aid in high revving applications [22]. The advantage to running a solid rod would be that the additional material strengthens the rod making it much stronger but at the cost of making it much heavier. As weight can act as a major penalty in a high revving engine the solid rod design was disregarded, and H and I beam designs were concentrated on. Using examples from existing rods a new H beam design was modelled and simulated. To maintain a scientific result the shank area was kept approximately the same and so too is the overall weight of the rod. The new design has a shank area of 0.6 mm² more than the factory rod and approximately 0.4g heavier.



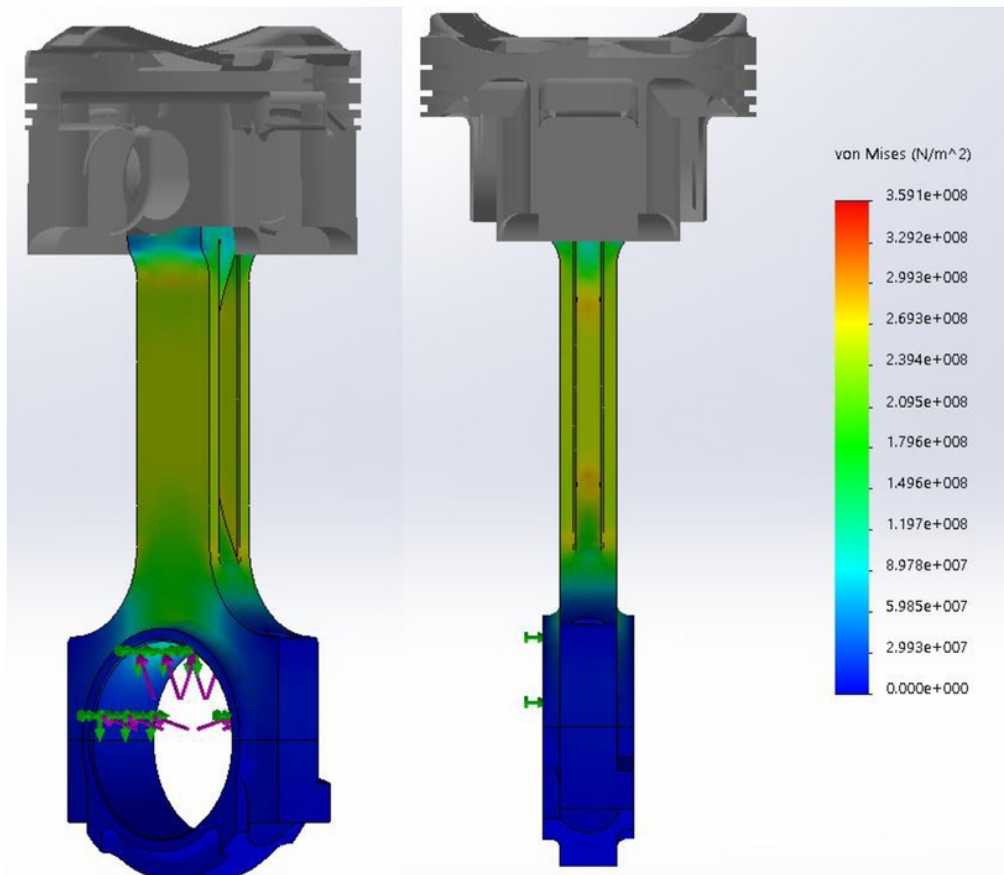
Picture 11 H beam rod design @9000RPM

From the results of the H beam model simulation it can be seen that the peak stresses are higher but only on a small fillet section, which could be easily smoothed out to even out the stress concentration. Therefore, looking at the remainder of the forces the peak appears to be closer to 358 N/mm^2 which is equal to 89 N/mm^2 less than the factory unit, which equates to approximately 19% reduction in stress.

8.4 Combined Design

From a culmination of these three design methods (9.1, 9.2, 9.3) a new rod was designed. This final design used 4340 steel, had a 24mm shank and a H beam design. The choice to use 4340 forged steel over titanium was due to effectiveness of the forged material and also the large price margin between the two materials with titanium rods costing anywhere between two to three times as expensive forged steel rods [23].

Initially the design suffered stress concentrations similar to picture 11 (See appendix) this appeared to be due to sharp contours in the rods design, so the rod was revised multiple times adding various fillets and chamfers to reduce possible stress concentrations.



Picture 12 Final rod design tested @9000RPM

From the results the peak stress in the rod was approximately 270 N/mm², roughly 60% of that of the factory rod. For comparison this newly deigned rod reaches what the factory unit achieves at 3000RPM at 9000RPM.

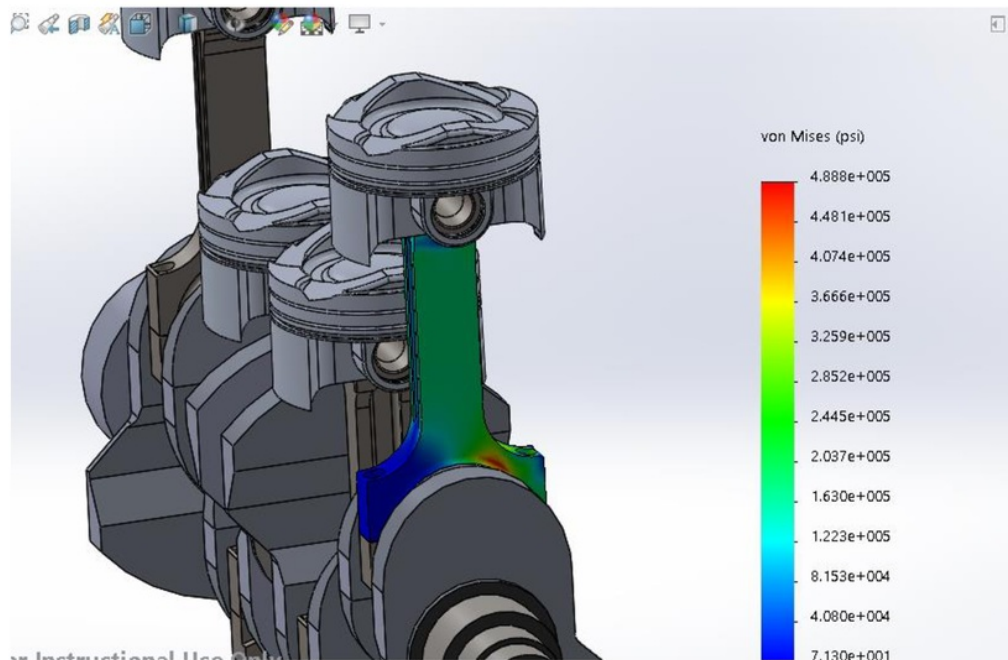
8.5 Motion study

Using the Solidworks simulation suite the engine speed was simulated at 6000RPM by adding a rotary motor to the end of the crankshaft. Contact constraints were added to necessary components such as between the pistons and the cylinders etc. the assembly constraints positioning piston 1 and 4 at TDC were relaxed to allow the rotating assembly free movement. The simulation would run for two revolutions which equates to one full combustion cycle. The time the simulation would run was calculated by converting the revolutions per minute into revolutions per second and then the total simulation cycles i.e. 2 divided by the revolutions per second. From this the simulation time for two rotations at 6000RPM equals 0.02 seconds, for 9000RPM the simulation time will be 0.0134 seconds. The formula is shown below.

$$\text{Simulation Time} = \frac{Rn}{\text{RPM}/60}$$

To ensure true results, the measured maximum pressure data was imported from excel in the form of a .csv file and applied to the top of the pistons to simulate Realtime cylinder

pressures. The gas force recorded for 2 revolutions was split into 4 forces, one for each cylinder. These forces were created by offsetting the recorded gas force by 180 degrees from -360 to 359, this simulated the firing order of the engine (1,3,4,2).



Picture 13 Motion Simulation result with final rod design @9000rpm

From doing this motion simulation it appears that the stress values are quite as severe as the static simulation however, this test does show up a new concentration just above the crank journal, but the rod still appears to be in a safe operational range.

9 Conclusion

This report has shown that the strength of a measured connecting rod can be confirmed by means of simple calculations. When designing a conrod the three main design factors are the size of the rod, the shape of the rod and the material of the rod. There are various material available for use of constructing connecting rods but of the three selected 4340 forged fillet steel is the best. This is due to the construction technique being suited to small production runs, perfect for race use. The other reason is the affordability for a mid to high range performance use such as for competition use in events such as British touring car racing, Formula Ford or various armature events.

The final working connecting rod design had the largest rod shank area helping to achieve the strongest design. As shown in section 9.2 the larger the shank area the greater the rod strength. However, increasing the rod area also increases the rod's mass which when trying to achieve high RPM's can be very counterproductive. Therefore, the shank size of a rod creates a trade off between strength and lightness.

The H Beam design proved a welcome improvement to the strength of the rod. However, further testing would be needed to determine if a H beam design is in fact superior than an I beam design or if in fact the designs used were just more effective due to material spacing and smoother surface finishes i.e. better use of fillets and chamfers. Hopefully this report has

shown that with a few simple design changes the strength of a connecting rod can be vastly improved.

The derived safety factor for the final rod design equates to the UTS of 4340 steel which is approximately 1000mpa divided by the max compressive force of 270 N/mm². Therefore a compressive safety factor of 3.7 is found, keeping to the goal of matching the factory safety factor of 2.2 at a low 6000RPM and then some.

10 Further work

Methods such as shot peening, work hardening and heat treating should be investigated to see the effects they have on a connecting rods strength. Generally connecting rods and crank shafts are indeed shot peened however simulating the effects in Solidworks would lead to much more accurate models but would be a very time consuming endeavour.

As this report focuses on what is deemed to be the weakest component addition studies should be made to simulate stresses on the surrounding components such as the engine block. It is possible that running the engine at this new operating speed can lead to cracks developing in the block or warping of the head to block mating surfaces which can lead to head gasket failure or even total engine failure. Cracks in the block develop due to too high of a cylinder pressure or cylinder temperature, it is also possible for cracks to develop overtime and therefore fatigue testing would be required to determine the lifespan at this new engine speed.

An additional consideration would be difference in cylinder temperature due to the increased engine speed and friction. Increased temperatures effect the tolerances on the components due to the expansion of metals at high temps. High engine block temperatures can also cause head bolt stretch leading to head gasket failure. Another issue would be the possible reduced lifespan of overlooked components such as oil pumps, water pumps, various fluids, belts and gaskets.

There are methods to strengthen the connection between the head and block such as changing out the cylinder head bolts for a stud and nut design such as the ones provided by ARP bolts, changing out these bolts for studs increases the clamping force [24]. There are also methods to reduce cylinder temperatures such as water injection, over fuelling, staged fuelling, retarding ignition timing and EGR valves. The reduction of friction is also possible via PTFE based lubricants, low friction coating, offsetting components such as the crankshaft or piston.

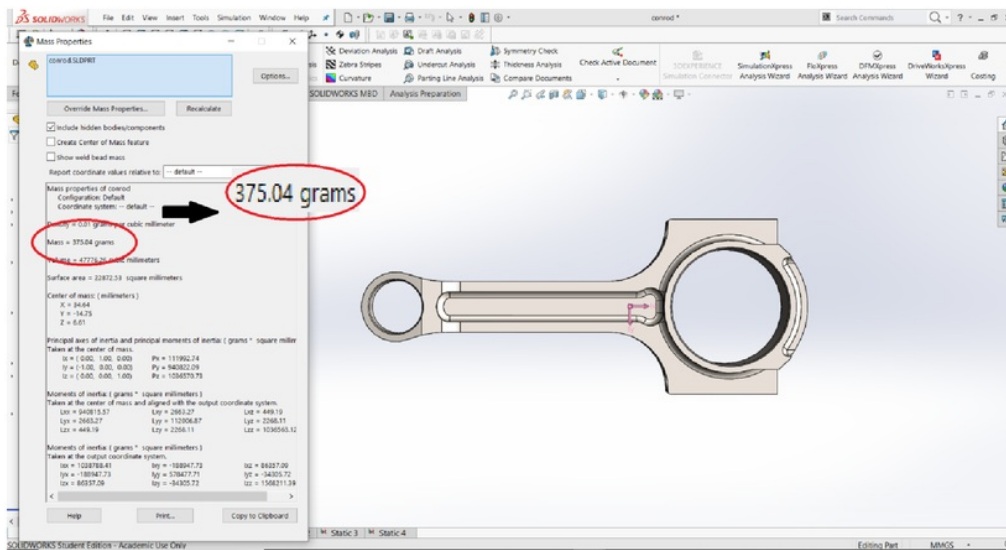
11 Appendices

5.6.2009	EN	Official Journal of the European Union	L 140/5
the overall objective of the European Community of 120 g CO₂/km as average emissions for the new car fleet. This Regulation sets the average CO₂ emissions for new passenger cars at 130 g CO₂/km, by means of improvement in vehicle motor technology, as measured in accordance with Regulation (EC) No 715/2007 and its implementing measures and innovative technologies. From 2020 onwards, this Regulation sets a target of 95 g CO₂/km as average emissions for the new car fleet, in accordance with Article 13(5). This Regulation will be complemented by additional measures corresponding to a reduction of 10 g CO₂/km as part of the Community's integrated approach. Article 2 Scope 1. This Regulation shall apply to motor vehicles of category M₁ as defined in Annex II to Directive 2007/46/EC ('passenger cars') which are registered in the Community for the first time and which have not previously been registered outside the Community (new passenger cars). (f) 'specific emissions of CO₂' means the CO₂ emissions of a passenger car measured in accordance with Regulation (EC) No 715/2007 and specified as the CO₂ mass emissions (combined) in the certificate of conformity. For passenger cars which are not type-approved in accordance with Regulation (EC) No 715/2007, 'specific emissions of CO₂' means the CO₂ emissions measured in accordance with the same measurement procedure as specified for passenger cars in Regulation (EC) No 692/2008, or in accordance with procedures adopted by the Commission to establish the CO₂ emissions for such passenger cars; (g) 'specific emissions target' means, in relation to a manufacturer, the average of the specific emissions of CO₂ permitted in accordance with Annex I in respect of each new passenger car of which it is the manufacturer or, where the manufacturer is granted a derogation under Article 11, the specific emissions target determined in accordance with that derogation. 2. For the purposes of this Regulation 'a group of connected manufacturers' means a manufacturer and its connected undertakings. In relation to a manufacturer, 'connected undertakings' means: (a) undertakings in which the manufacturer has, directly or			

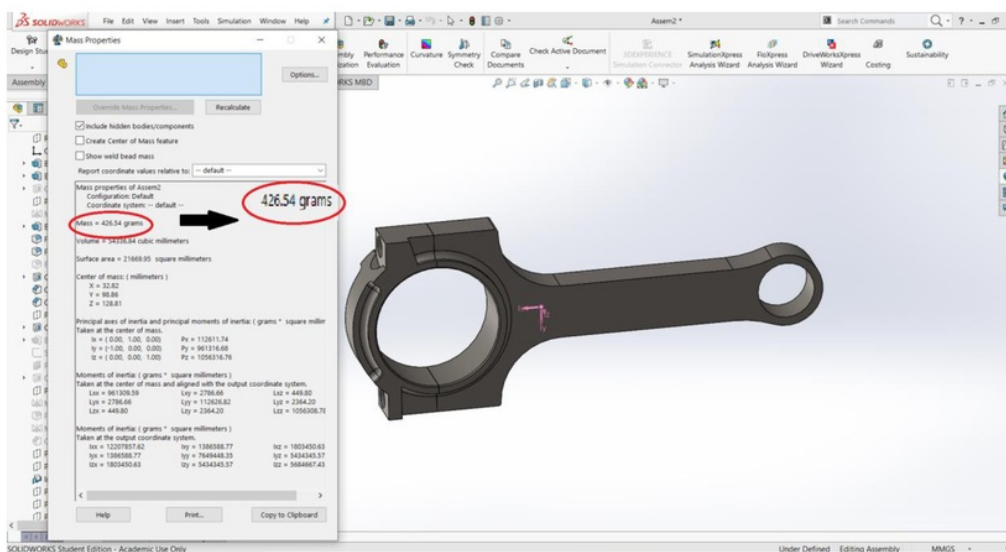
Picture 14, Article 1 of the European emission standards regulation 443/2009



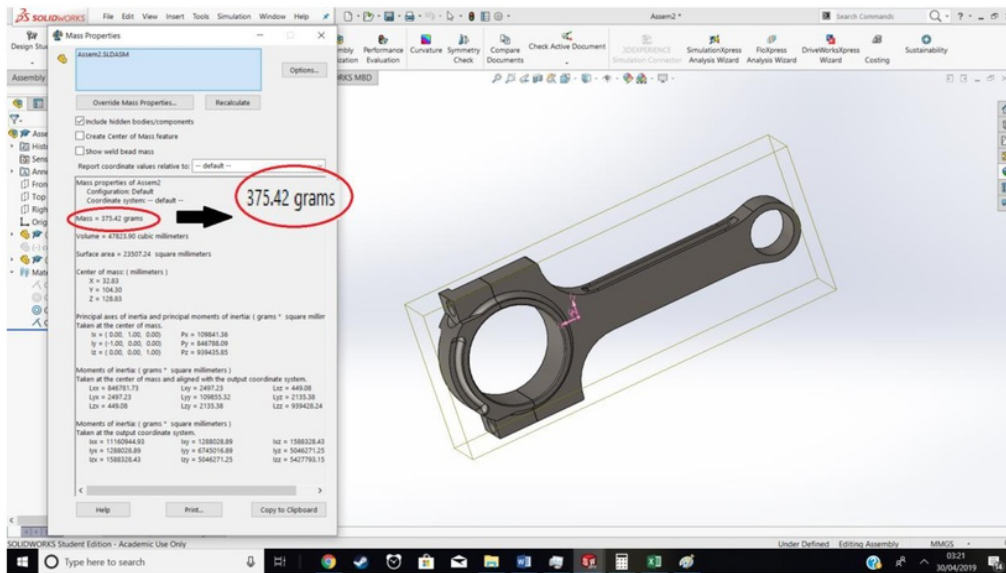
Picture 15 Recorded Weight and area of the 1.6 rod [25]



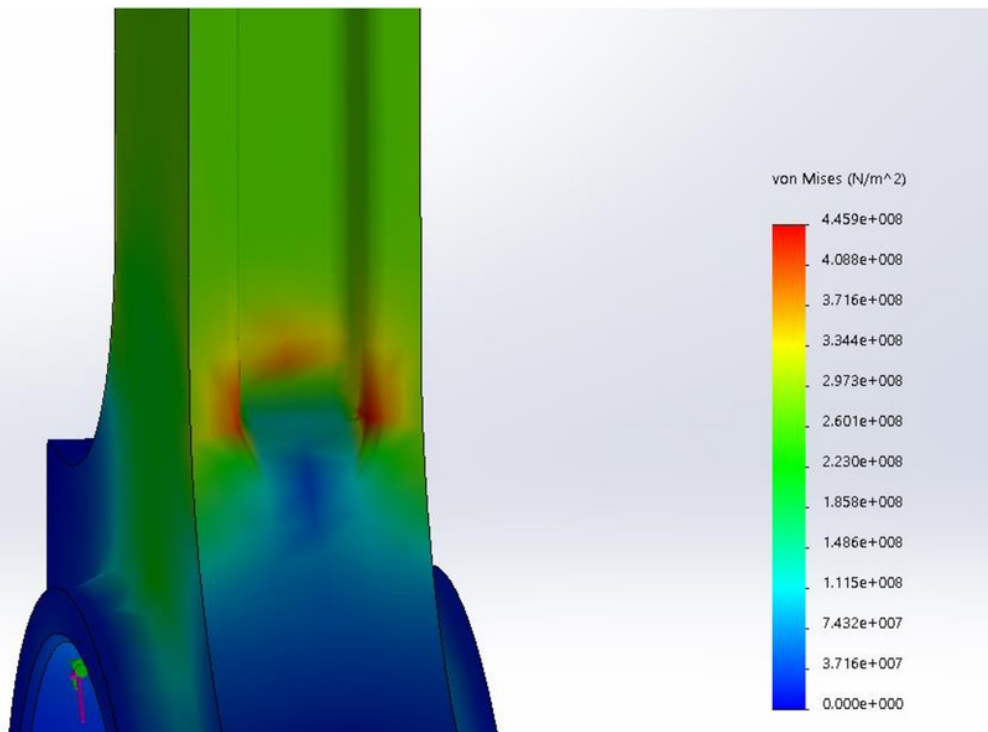
Picture 16 Weight of the Solidworks factory Conrod



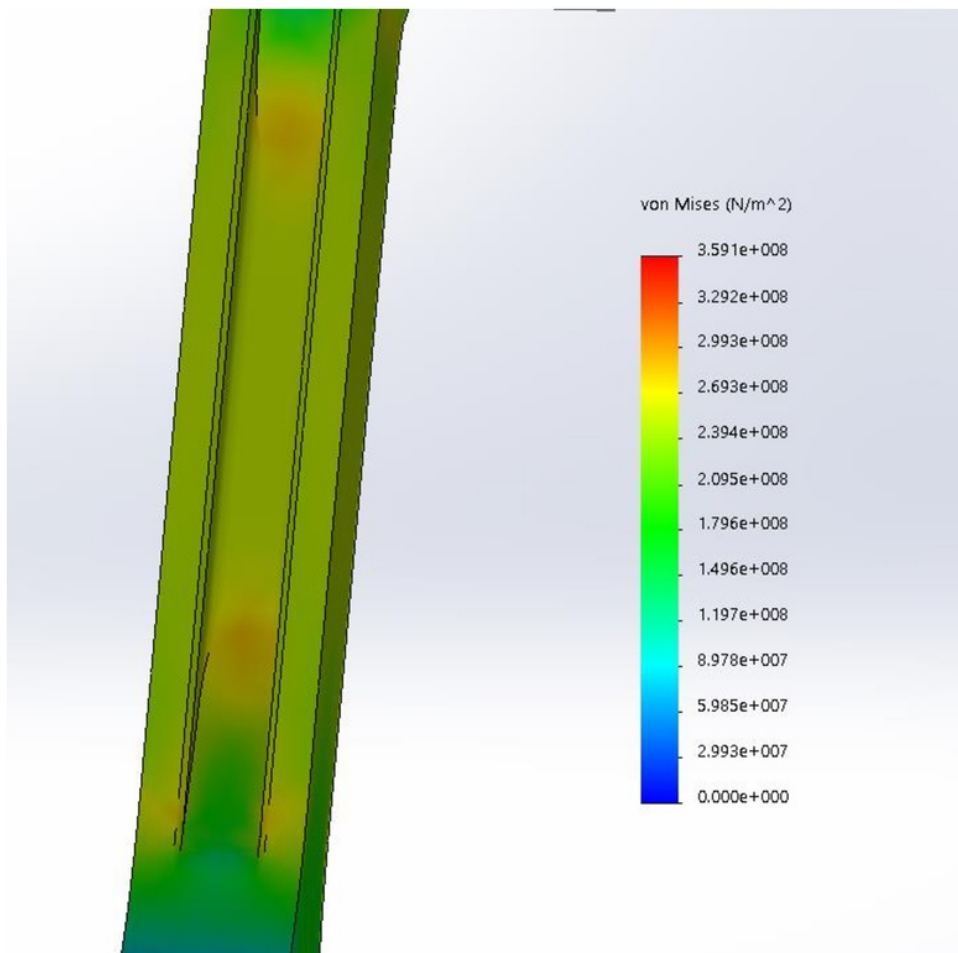
Picture 17 Example Weight of a Solid rod



Picture 18 Weight of the H beam design Conrod

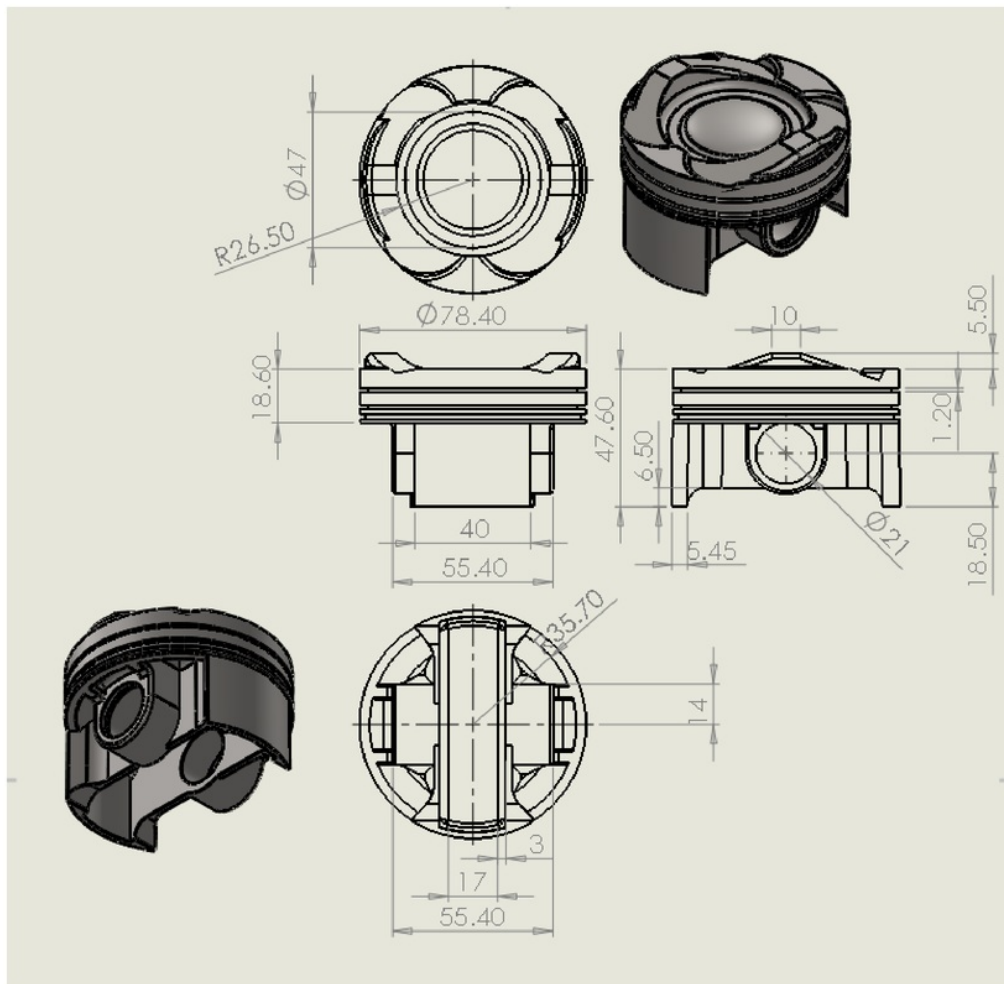


Picture 19 Final rod design issues



Picture 20 Final rod issues resolved





Picture 22 Dimensions Used in Simulation for Piston [26]

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- Further work/recommendations after conclusion