

Matt Powell Major Project Report

by Matthew Thomas Powell

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MASS OPTIMIZATION OF A CONNECTING ROD

Focusing on utilising composite materials.

University of Wales Trinity Saint David

School of Engineering

Motorsport Engineering and Design (Beng)

Major Project (L6)

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MAY 4, 2019

By Matthew Powell

DECLARATION

I, Matthew Powell declare that I am the sole author of this Project; that all references cited have been consulted; that I have conducted all work of which this is a record, and that the finished work lies within the prescribed word limits.

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Abstract

This report investigates in detail the connecting rods role in the power cylinder system and calculates the effect of mass reduction on friction and safety factors, further two different types of connecting rod are reverse engineered to analyse there fatigue safety factors and their displacement due to compressive and tensile forces. To calculate how the connecting rods, respond to tensile and compressive forces first calculations are performed using Excel from real cylinder vs crank angle pressure data, secondly models of both connecting rods are produced in CAD and the analysed in FEA as a comparison to the hand calculations.

Secondly carbon fibre technology is reviewed, previous carbon connecting rod designs are examined. Tensile tests are performed to ascertain material properties, a bladder mould carbon rod is designed and analysed using the measured material properties to ascertain whether it would be a suitable material to replace steel rods in a high efficiency production engine. Carbon connecting rods are a feasible way of reducing mass and therefore reducing friction, although special resins must be used to withstand the heat. A proposed design for production was produced from data from reverse engineering the GDI and Pankl connecting rods and in accordance with the safety factors.

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Notation

<u>Symbol</u>	<u>Units</u>	<u>Name</u>
D	M	Diameter
L	M	Length
A	M ²	Area
V	M/s	Velocity
R	M	Crank Throw
L	M	Connecting rod length
S	M	Piston travel
θ	Radians	Crank angle
φ	Radians	Rod angle
ω	Rads/sec	Crank velocity
P_{cr}	Nm?	Critical buckling load
σ_y	Nm ²	Material yield stress
λ_i		Eigenvalue load multiplier
K_0^{NM}		Stiffness matrix (base state)
K_{Δ}^{NM}		Stiffness matrix (from incremental loading pattern)
V_i^M		Eigenvectors
M		Degrees of freedom in model
N		Degrees of freedom in model
A	M ²	Cross sectional area
E	Nm ²	Youngs modulus
L_e	M	Effective length
r	M	Radius of gyration
E_1		Youngs modulus 0° to fibre orientation
E_2		Youngs modulus 90° to fibre orientation
V_f		Fibre volume fraction
E_f		Youngs modulus of the fibres
E_m		Youngs modulus of the matrix
ν_1		Poisson's ratio
G_1		In plane shear modulus

Definitions

PCS – Power Cylinder System

NVH –NOISE, Vibration, and harshness

RPM – Revolution per minute or RPS for seconds

TDC – When the piston is fully up (i.e. minimum volume in the cylinder or clearance volume)

BDC – When the piston is fully down (i.e. volume in the cylinder is maximum)

ATDC – After TDC when the piston is travelling towards BDC

ABDC – After BDC when the piston is travelling towards TDC

OPM – Original part manufacturer

ICE – Internal combustion engine

UD – Unidirectional

FRP – Fibre reinforced plastic

CFRP – Carbon fibre reinforced plastic

Bore – The diameter of the cylinder

CAD – Computer Aided Design

FEA – Finite element analysis

UTS – Ultimate tensile strength

FSF – Fatigue safety factor

UV – ultraviolet

Pre-preg – pre impregnated carbon fibre

1 INTRODUCTION

Powertrain development in the automotive industry is key to succeeding, due to the demand for downsized engines with greater efficiency and a higher specific power than the previous generation while meeting the WLTP emission standards. To achieve greater efficiency and power reciprocating cylinder components have to exert less force through the piston to the cylinder walls, thus more efficiently transferring the force from combustion to a rotational motion seen by the crankshaft. A critical reciprocating cylinder component which transfers the oscillating piston motion into the rotational motion of the crankshaft is the connecting rod; it must not only provide the strength and stiffness to withstand the gas forces from combustion and the inertia forces from the piston assembly but must also be of minimal mass.

This paper investigates the I-beam connecting rod used in a Ford 1.6GDI engine, a titanium H-beam rod and how the design of a structure may affect its compressive strength. The tensile strength of unidirectional carbon fibre is analysed, and a connecting rod is designed and analysed using FEA.

Aim

- To ascertain whether it is suitable to replace a steel connecting rod with one optimized for mass utilizing carbon fibre

Objectives

1. Reverse engineer a 1.6 Ford GDI connecting rod
2. Reverse engineer a titanium Pankl rod
3. Perform tensile testing on UD carbon fibre to ascertain material properties
4. Design a connecting rod suitable for manufacture from carbon fibre

The primary focus of this report is to reverse engineer two styles of connecting rod utilising real cylinder pressure data to ascertain fatigue safety factors

2 LITERATURE REVIEW

2.1 CONNECTING ROD

As detailed by (Hanke, et al., 2009) the PCS which contains the piston, piston rings, gudgeon pin, and connecting rod is one contributor to friction within the engine. Reducing friction is always one focus for emission reduction, reducing the weight of the moving components results in a lighter engine and thus, leads to an additional CO_2 reduction for the vehicle. Given that (Lapp, et al., 2010) details to more efficiently transfer the combustion force in the cylinder to the rotational motion seen by the crankshaft the PCS components must exert less force on the cylinder walls; this is completed mainly by lighter reciprocating components. A critical component in the PCS is the connecting rod which transfers the oscillating movement from the piston to rotational motion seen by the crankshaft. The rod must not only have a high enough strength and stiffness to withstand the gas and inertia forces, the hydrodynamic performance of the small and big end needs to be optimised to meet tougher NVH requirements and improve friction and wear properties (Hanke, et al., 2009).

As engines are downsized further to meet tougher emission regulations, this will lead to greater engine loading and speed. It is argued by (Lapp, et al., 2010) that although many engine components are adequately optimised for their intended use, traditionally the highly loaded reciprocating and rotational cranktrain components are over-designed. Due to the connecting rod mass affecting the mass of the components of the cranktrain, reducing the connecting rod mass would allow components such as the crankshaft, balance shaft, piston, and gudgeon pin mass to be reduced.

Connecting rods experience gas and inertia forces as well as axial and bending stresses; bending stresses occur due to the rotational inertia caused by the crankshaft rotating as well as cylinder wall deformation. Due to the connecting rod being submitted to millions of repetitive cyclic forces, typically it is designed for infinite life with the endurance limit being the primary design criterion (Afzal & Fatemi, 2004).

Mass production vehicles usually have connecting rods that are manufactured from drop forged steel alloys. In motorsport applications the standard connecting rod is replaced for one manufactured from titanium to reduce mass or, in some instances such as drag racing where the bearing forces need to be reduced to stop failure an aluminium connecting rod is used.

2.2 PLASTICS WITHIN THE ICE

(Guadette, 1985) states that plastic, when applied to materials, has a wide meaning, it covers formable unfilled resins to continuous fibre composites. Plastic engine components have their origins in the late 1970s, Matthew Holtzberg's company Polimotor Research Inc research in co-operation with Amoco Chemicals Corp created two engines that utilized both metal and plastic components. Version 1 was based on a 2.3 litre Pinto engine where the original Ford engine weighed 188kg whereas the Polimotor engine came in at 69kg, the engine produced 320 BHP the metal components included the piston crown, cylinder liners, combustion chamber, bearings, valves and seats, and utilised the standard ford crankshaft. Most of the other components were made of glass reinforced polyamide-imide thermoplastic resin, such as the block, connecting rods, and the piston skirts.

Version 1 was never fitted to a vehicle, whereas, version 2 which was based on a Cosworth BDA engine was raced in a Lola T616 over two seasons. A total of 10 different polyamide-imide components were developed and fitted to this engine. The crankcase, water jackets, and valve covers are manufactured from UD continuous graphite epoxy material whereas the connecting rod is manufactured using a woven continuous graphite fabric, the other dynamic plastic components are comprised from a polyamide-imide resin. These include the timing pulleys, intake valve stems, valve spring retainer, overhead bucket tappets, rocker arm, piston skirt, number two piston ring and oil scraper, gudgeon pin, and the roller lifter. Table 1 shows the metal component mass, the Polyamide-imide component mass, and the reduction in mass given as a percentage.

Component	Metal Component Mass (g)	Polyamide Component Mass (g)	Percent Reduction
Timing gear	750	265	65
Intake valve	140	30	79
Valve spring retainer	22	13	41
Bucket tappet	48	19	60
Rocker arm	108	60	44
Piston	500	391	22
Piston ring	16	4	75
Gudgeon pin	105	65	24
Connecting rod	743	380	49
Roller lifter	144	79	45

Table 1 – Polyimide component masses compared to original component mass

2.3 LIGHTWEIGHT DESIGN OF A HYBRID CONNECTING ROD

Like Polimotor Daimler Benz AG have also spent years developing fibre reinforced connecting rods, the main challenge faced by both (Guadette, 1985) and (Kuch & Kümmerlen, 1991) was to develop a suitable fibre reinforced plastic material for a highly stressed component in chemically aggressive hot service fluid. Whereas the Polimotor Research Inc. connecting rod only utilised Kevlar reinforced plastic, Daimler Benz thought it was necessary to differentiate material between differently stressed areas due to the complex high alternating loads and thus, the excessive bulk and weight needed for a quasi-isotropic structure that would be required by the FRP. Several iterations of the hybrid connection rod were designed and tested, the final iteration of the connecting rod was 30% lighter than that of the original metal component. The connecting rod consisted of 5 main sections as shown in figure 1. Section A acts as a compressive shank and is designed with 77% 0° UD layers and 23% $+45^\circ$ layers with a total thickness of 11mm. Section B is a loop designed to withstand the tensile forces and is 1.25mm of 0° UD fibres which are connected to section C which is a small end made from titanium, D1 is crankshaft bearing, and D2 is the connecting rod cap which are both manufactured from titanium. On figure 1 the two red circles highlight a crescent-shaped spacer that applies tension to the tensile loop and therefore compresses the shank section. This is designed so that during operation only compressive and tensile forces occur.

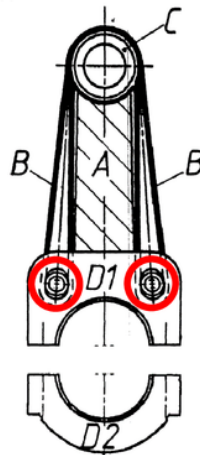


Figure 1 – Hybrid connecting rod

Table 2 gives an overview of the tests conducted on the CFRP connecting rod. Tests 1-5 demonstrate the hybrid connecting rods ability to withstand varying magnitude of compressive and tensile stresses, Test 6 increases the cycles from 5-50 million. If a steel component can withstand 50 million cycles without failure it is believed that the component has an infinite fatigue life. Test 7 increases the loads to simulate the maximum force seen in the connecting rod of a 2.3L engine.

	Type of stress	Load (KN)	Number of load cycles	Frequency (Hz)	Temperature
1	Tension/Compression	+16/-50	3	Static	Room temp
2	Tension	+2/+14	5×10^6	70	Room temp
3	Compression	-2/-40	5×10^6	70	Room temp
4	Tension	+2/+14	5×10^6	70	120°C
5	Compression	-2/-40	5×10^6	70	120°C
6	Tension/Compression	+14/-40	5×10^7	70	120°C
7	Tension/Compression	+17/-45	5×10^7	70	120°C

Table 2 – Tensile and compressive tests

Although the connecting rod was 30% lighter than the original and was tested up to 5×10^7 cycles, it was concluded that due to high fabrication and material costs a split hybrid connecting rod was not practical, whereas unsplit (press fit) versions could be utilised in high-efficiency engines successfully.

2.4 COMPOSITE TECHNOLOGY OVERVIEW

Composites are comprised of two or more materials that work together, each constituent material retains its unique identity and contributes its own structural properties; given this, upon combination, the resulting material has superior properties than those of its constituents (Dorworth, et al., 2009).

Fibre	+	Matrix	=	Composite
Tensile strength		Compressive strength		Increased strength
Flexural stiffness		Interlaminar shear		Increased stiffness
Somewhat Brittle		Low density		lightweight

Table 3 – Composite technology overview

2.4.1 Fibre reinforcements

Carbon fibre is either made from polyacrylonitrile (PAN) which is an acrylic fibre material or is oil/ coal based (Pitch). PAN is often used to make standard and intermediate modulus fibres which have a notable tensile strength. Pitch fibres are often used to make high and ultra-high modulus carbon fibres that are very stiff but somewhat brittle, they also tend to have a higher density than PAN fibres. Composites can be categorized by the form of reinforcement used; structural composites used continuous fibres whereas non-structural components may use chopped fibres.

2.4.2 Matrix technology

The matrix acts to bond and/or encapsulate the fibres; this enables the transfer of loads from fibre to fibre and provides resistance to fibre buckling under compressive loads. The matrix also moderately protects the fibres from degradation due to environmental effect which include moisture, chemical, UV radiation, abrasion, and impacts. Selecting the right matrix is important depending on the loading the component is placed under and the temperature it operates in. The matrix has a major influence on the shear properties of the composite, inter laminar shear strength is important for components that experience bending loads, whereas in-plane shear strength important under tensile loads.

To cure the matrix heat and pressure is applied to the component, for flat sheets this is normally done using a heated press while more complex components laid in moulds are cured using an autoclave. During the curing process, the molecules in the resin chemically react together and are joined by cross-linking; this forms a rigid 3-dimensional structure. Once the links are formed the molecules are locked in and cannot be melted or reshaped by the application of heat and pressure, however, when a resin has a low number of cross-links it may still be possible to soften at elevated temperatures. Table 4 shows typical properties of different thermoset resins available.

Matrix system	Tensile strength (Mpa)	Youngs modulus (Gpa)	% Elongation to failure	Cost	Initial cure temp °C
Polyester	20.7 - 75.8	2.8-3.4	1-5	low	121
Vinyl Ester	68.9-82.7	3.4-3.9	3-12	low-med	93
Epoxy	48.3-89.6	2.7-3.7	2-9	medium	177
Phenolic	13.8-62.1	4.1	1-2	medium	60-232
Bismaleimide (BMI)	48.3-89.6	3.3-4.3	1-3	high	190-288
Cyanate Ester (CE)	48.3-89.6	2.8-3.4	2-4	very high	121-177
Polyimide (PI)	34.5-117.2	1.4-4.8	1-4	very high	316-399

Table 4 – Matrix system properties

2.4.3 Youngs Modulus

Unlike most metals which are isotropic materials where their mechanical properties are the same in all orientations, carbon fibre is an orthotropic material, and the lamina has anisotropy properties and behaves differently depending on the direction of the force.

Carbon fibre has a higher Young's modulus when the applied force is parallel to the fibres (Christine Demerchant, 2019).

E_1 is the Young's modulus of the composite parallel to the fibre direction, and a simple approximation can be calculated by a 'rule of mixtures' expression otherwise known as the Voigt expression which ignores the difference in Poisson's ratio between the fibre and matrix direction.

Equation 1
$$E_1 = E_f V_f + 1 - V_f E_m$$

The Young's modulus perpendicular to the fibres E_2 is calculated by

Equation 2
$$E_2 = 1/V_f E_f + 1 - V_f E_m$$

Composite laminates are usually made from fibres at different orientations, to calculate Young's modulus E_x of a ply at angle θ using equation 3.

Equation 3
$$E_x = E_1 \cos^4 \theta + G_1 - 2V_1 E_1 \sin 2\theta \cos 2\theta + E_2 \cos^4 \theta$$

(Anand, 2016)

2.4.4 Ply Orientation

Laminate performance is almost totally dependent on the orientation of the fibres and is closely controlled during engineering and manufacturing; there is an internationally recognised system for controlling the fibre orientation on engineering drawings and manufacturing processes. The recognised standard uses a system whereby the fibre orientation will be in the counter clockwise direction to the 0° axis which is normally parallel to the primary load direction.

2.4.4.1 Layup Terms

A composite layup generally comes under two terms, Symmetry or balance. A symmetrical laminate is a mirror image when looking from the mid plane. It is often difficult to achieve a symmetrical laminate when multi-axial or stitched-biaxial forms are used. A balanced laminate is when it is made up from equal numbers of plus and minus-angled plies, this arrangement is used to reduce twisting under applied loads.

2.4.4.2 Quasi-isotropic

A quasi-isotropic layup is both symmetrical and balanced, and the industry standard for this layup consists of an equal number of plies at each of the 0° , 90° , $+45^\circ$, and -45° angles. This layup behaves similar to steel; it is also possible to achieve these properties with fewer fibre angles such as 0° , $+60^\circ$, and -60° ; or with additional fibre angles such as 0° , $+22.5^\circ$, $+45^\circ$, $+67.5^\circ$, 90° , -22.5° , -45° , -67.5° .

2.5 FINITE ELEMENT ANALYSIS

As previously mentioned by (Lapp, et al., 2010) components are often overengineered; this is to stop premature failure of a component and increase customer safety. Both (Anderson & Yukioka, 2012) and (Yadav, et al., 2016) state that one of the main design concerns for a connecting rod is its buckling strength which is heavily affected by the shanks cross-section. Due to the demand for lightweight connecting rods to reduce fuel consumption and NVH, new connecting rods generally have lesser material with varying CSA. Theoretical methods like the Merchant-Rankine formulae and Johnson's buckling equation can be helpful during the initial design phase to get an approximate FOS, utilising the minimum cross section using Johnson's equation 4 yet (Anderson & Yukioka, 2012) concluded that for a complex 3-dimensional shape it wasn't useful to calculate critical buckling force.

Equation 4

$$P_{cr} = \sigma_y A \left[1 - \left(\frac{\sigma_y}{4\pi^2 E} \right) \times \left(\frac{l_e}{r} \right)^2 \right]$$

With modern technologies such as CAD and FEA programs, it is possible to analyse how a component will perform in its intended scenario and ensure that adequate safety factors and displacements are achieved, without manufacturing prototypes (Simscale, 2019). Although hand calculations can be useful in the initial design phase, uncertainty from Johnson's equation can be reduced using the Eigenvalue buckling method utilised by FEA software which is well suited to compute complex 3-dimensional geometry. The Eigenvalue buckling method estimates the loads whereby the model stiffness matrix becomes singular the problem shown in equation 5 has nontrivial solutions.

Equation 5

$$K^{MN} v^M = 0$$

The Eigenvalue problem is solved utilising an incremental loading pattern Q^N . The loading is scaled by λ_i , therefore, the magnitude of load isn't important. Therefore, from equation 4.

Equation 6

$$(K_0^{NM} + \lambda_i K_\Delta^{NM}) v_i^M = 0,$$

2.6 ANALYSIS OF PROBLEMS (FORCES TENSION, COMP ECT.)

The connecting rod is subjected to complex loading conditions and has to withstand a high number of cyclic loads to the order of 10^8 cycles. During each cycle the loads alter from compression from combustion, tensile forces due to the inertia, and due to the complex motion, the inertia and compressive forces induce bending moment. At any point on the connecting rod the stress it undergoes consists of two components, a bending stress and axial stress components (Shenoy, 2004).

Figure 2 is a simple diagram that shows how the tensile force transfers around the connecting rod around the small end and is translated through the cap.

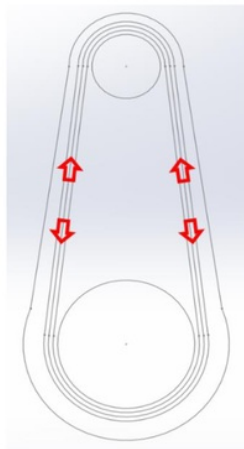


Figure 2 – Tensile stress

Figure 3 shows how the compressive force is distributed in the shank section of the connecting rod.

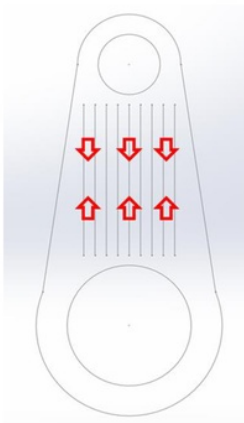


Figure 3 – Compressive stress

Figure 4 highlights the complexity of stress from the bending moment, one side of the connecting rod experiences compressive stress while at the same time the opposite side is submitted to a tensile stress and these stress alternate during each cycle.

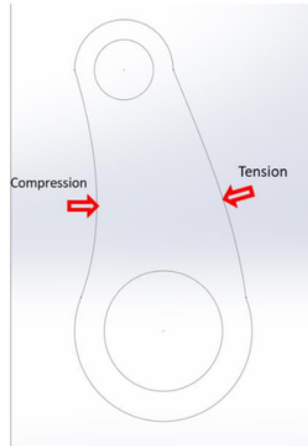


Figure 4 – bending stress

(Lapp, et al., 2010) split the connecting rod into eight critical sections, and describes the failures that occur in each area the causes of the failures. Figure 5 highlights the areas of concern. Section 1 is the apex of the small end, failures seen here are caused by the inertia force and usually consist of cracks due to fatigue and bore deformation. The bottom of the small end (2) can wear excessively which can lead to the gudgeon pin seizing as is caused by the compressive force. The shank of the connecting rod (3) fails due to the compressive force, failures include bending or buckling. Sections 4, and 5 are the transition areas between the shank and the big end and suffer from cracks due to fatigue and bored distortion due to high compressive force. 6 is the joint faces between the big end and cap; failures include bulging, gap opening, and bore deformation which as caused by both inertia and compressive forces. The connecting rod bolts (7) fail from the alternating compressive and inertia forces; the bolts can become loose which causes a reduction in the clamping forces which can lead to bore distortion. The cap (8) suffers from cracks due to fatigue and bore deformation due to the inertia forces.



Figure 5 – Critical connecting rod sections

2.7 DESTRUCTIVE COUPON TESTING

The properties of a composite material must be validated through destructive coupon testing; it must be performed to an accepted industry standard and supported by data analysis.

Tensile and compressive tests designed for plastics and metals cannot typically be used for composites, ASTM and ISO have created standard methods for the testing of composites (Dorworth, et al., 2009). Before the testing can be conducted, the specimen must be prepared, the quality checked, and a test piece manufactured.

2.7.1 Laminate preparation

Lengths are cut from the reel of UD carbon fibre, which comes in either 0.125mm or 0.250mm thickness; each layer is then stacked on top of one another with the fibres orientated relative to one another. It is vital that the layers are aligned with reference to a datum as even a couple of degrees misalignment can affect the mechanical properties dramatically. Once the stack of prepreg is prepared it needs to be cured to set the matrix, this can be done either in an autoclave or in a heated press. Both processes last several hours and in the case of autoclaving the specimen must be placed into a vacuum bag before being placed into the autoclave whereas when using the heated press, the component is placed between two metal plates. Both methods heat and pressure are applied, and in the case of the autoclave a vacuum is also applied, temperatures range from 120-175°C for epoxy resins.

2.7.2 Quality checking

During the manufacturing process, it is possible to introduce defects into the laminate; typical defects include voids and delamination's, longitudinal cracks may also occur but are less common. Voids are caused if the prepreg is not allowed to warm to room temperature before completing the layup process, as this introduces moisture into the stack. Delaminations are caused either by entrapped air or the inclusion of the backing sheet. Longitudinal splitting and delamination may also occur in multidirectional laminates due to the thermal stress induced during the cooling process.

2.7.3 Specimen manufacture

The test specimens as defined by the chosen standard and test, are cut from the laminate using a diamond-tipped saw or diamond tipped cutting tool for a CNC machine. Autoclaved laminates will have a feathered edge which needs to be removed. High temperatures are produced while dry cutting which may lead to degradation and damage at the machined edge, this can be prevented with the use of a coolant but subsequent drying steps are required. Tensile and compression tests require tabs to be bonded to the ends of the sample; this diffuses the stresses produced by gripping by the jaws of the test machine, tabs may be of aluminium, GFRP, or CFRP depending on the testing standards.

2.7.4 Coupon tests

The overall goal of a test procedure is to gather data that will represent the actual structure actually being built. It is important to consider the components operational life, taking all the environmental conditions into consideration. The ability to obtain accurate and repeatable data can be affected by specimen preparation, environmental conditioning, test setup and instrumentation. Coupon testing can be split into the following categories tensile, compressive, shear, and fatigue. The data generated by the tests will give material property values for stress, strain, extension, UTS, and Young's modulus.

2.7.4.1 Tensile

For a tensile test, a rectangular coupon is held at each end by a wedge or hydraulic grips; the test specimen is then loaded in uniaxial tension. The specimens gauge length is the section of the sample between the two jaws where the deformation or strain is measured; the materials response is measured using extensometer or strain gauges.

2.7.4.2 Compressive

Compressive testing utilises a rectangular coupon which is held by tapered wedge grips, which may be conical, pyramid-shaped, or rectangular. Load is introduced into the specimen through end loading, shear loading, or a combination of both. Results include compressive modulus and strength.

2.7.4.3 Shear

Although possible to conduct shear testing of a composite, it has been a very difficult area to define a consistently accurate test. Shear modulus is less problematic than shear strength where issues such as the non-linear behaviour of the fibre matrix interface, uneven stress distribution, and nominal stress causes the results to be questionable.

2.7.4.4 Fatigue

A cyclic load is applied to a specimen until failure is achieved; the number of cycles applied to the component until failure is known as fatigue life and is recorded. Stress or the log of stress is plotted against the log of fatigue life to develop an S-N curve, which defines the fatigue resistance of the material. The recorded data provides a large amount of scatter; thus, a large number of tests will need to run to define the fatigue resistance.

3 STRUCTURAL TESTING 3D PRINTED COMPONENTS

As carbon fibre comes in UD form, as well as woven, and non woven forms. Utilising UD carbon, it is theoretically possible to change the orientation of the fibre depending on the section of the connecting rod to achieve the desired strength and stiffness. 50% scale rods were created using CAD and were 3D printed, 4 rods were produced and consisted of a rod with a section of the shank completely cut out, the 3 other rods were produced with the cut-out section braced in 90° , 45° , and 20° respectively. Compressive and tensile tests were planned to be carried out using a Hounsfield H20k-W test machine, due to unforeseen circumstances this was not possible due as it was broken and unfixable.

3.1 COMPRESSIVE TESTING

To analyse the effect of the bracing on the compressive strength a new test was designed utilising a valve spring compressor that included a load cell and extensometer. To achieve this, additional components were designed and manufactured and include a gudgeon pin, main journal pin, and two sections that transferred the load through the pins but allowed the connected rod to float to represent real conditions accurately. The assembly is shown in Figure 6.

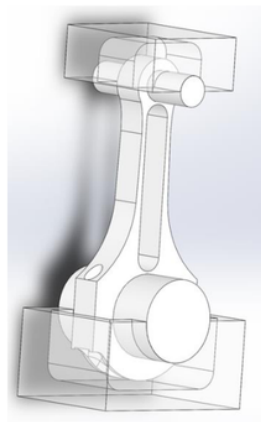


Figure 6 – Plastic connecting rod compression test assembly

Figure 7 shows the assembly during a test in the valve spring compressor.



Figure 7 – Plastic connecting rod test

Figure 8 shows the results from the compression test conducted, the load cell measures in Lb and values for displacement were taken at 10lb intervals. The plain rod failed at force of 827N, the 90° and 20° failed at 1735 N and 1765 N respectively which is 1.79% difference but the 90° displaced 0.36mm more than the 20° which suggests that 90° is less stiff than the 20° although it was not possible to calculate Young's modulus due to the CSA changing throughout the whole length of the shank. The 45° failed at 1935 which is 11.53% higher than what was achieved by the 90° rod and 9% higher than the 20°.

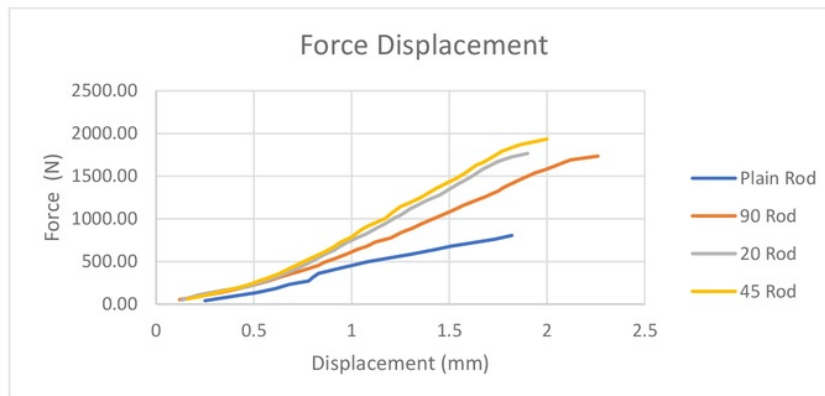


Figure 8 – Force displacement chart from plastic connecting rod compression test

A bending experiment was also conducted shown in Figure 9, and 10. The results were determined to be unreliable as each weight was weighed before being added to improve accuracy, during this process, it was noted that the reading on the DTI gauge increased due to the elastic properties of the ABS plastic used.

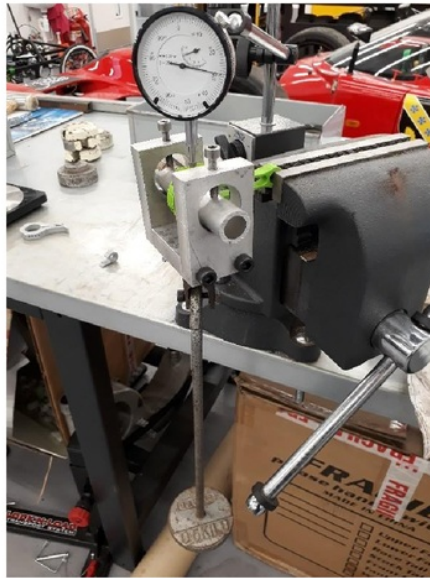


Figure 9 – Bending test rig



Figure 10 – Bending test

4 REVERSE ENGINEERING GDI AND PANKL CONNECTING RODS

During the connecting rod design, an important factor to calculate is the compressive and tensile forces the rod will see from normal engine operation; these alternating loads will cause the rod to displace under tension effecting piston to valve clearance and compression that the rod must be stiff enough not to buckle.

4.1 CYLINDER PRESSURE DATA

Cylinder pressure against crank angle from 1000-6000 RPM was obtained from the GDI engine investigated, from the cylinder pressure data and measurements for bore, rod length, and the masses for the individual components in the piston connecting rod assembly; it is possible to calculate the gas force, inertia force, side force, and friction force. Utilising material data, the CSA of the shank, and these forces it is possible to calculate the fatigue safety factors of several sections of the connecting rod and the tensile and compressive displacement. Cylinder pressure against crank angle from the provided data is shown below for three RPM sites with the peak pressure ATDC shown for 1000 and 6000 RPM below in Figure 11, from this graph it can be seen that the peak pressure at 6000 RPM occurs earlier than at 1000 RPM. The ignition timing is retarded at low RPM's due to lower turbulence which causes a slower flame speed resulting in increased time for the unburned zone to heat up and pre-detonate, this is caused by the lower piston velocity. At higher RPM's the ignition can be advanced towards MBT timing which is the point where maximum torque is output by the engine this is due to the increased piston velocity, thus resulting in less time for the unburned zone to heat up and cause pre-detonation.

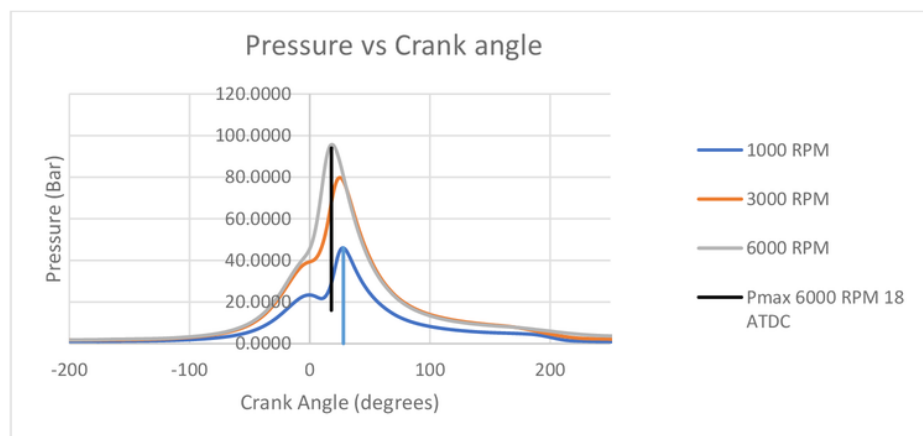


Figure 11 – Cylinder pressure vs crank angle chart

4.1.1 Calculations

In order to calculate the maximum inertia, net force, and friction forces and average cylinder pressure across the four-cylinder is taken piston acceleration must first be calculated. To calculate piston acceleration in Excel first the crank angle must be converted to radians, from the crank angle it is possible to calculate the rod angle utilising the following equation.

Equation 7
$$\varphi = \sin^{-1} \left(\frac{R}{L} \right) * \sin \theta$$

From rod angle

Equation 8
$$\text{Piston travel (mm)} = R + L - (R \cdot \cos \theta) - (L \cdot \cos \varphi)$$

From piston travel the velocity can be calculated using equation 9.

Equation 9
$$\text{Piston velocity (mm/s)} = \text{piston travel}_2 - \text{piston travel}_1$$

Piston velocity in mm/s is converted to mm/sec using equation 10.

Equation 10
$$\text{Piston velocity} \left(\frac{\text{mm}}{\text{second}} \right) = \text{Piston velocity (mm/s)} \times 6 \times \text{RPM}$$

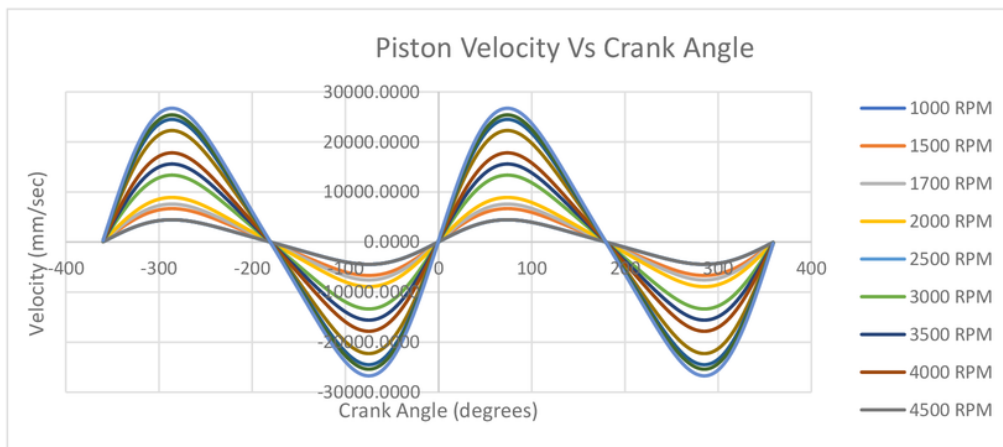


Figure 12 – Piston velocity Vs crank angle chart

From piston velocity acceleration can be calculated but first crank angle velocity (ω) must be calculated in radians/second using equation 11, acceleration in M/s^2 is defined as the rate of change in velocity per unit of time.

Equation 11
$$\omega = \left(2\pi \frac{\text{RPM}}{60} \right)^2$$

Equation 12
$$\text{Piston acceleration} = (R \cdot \omega^2 \cdot \cos \theta) + \left(\frac{R}{L} R \cdot \omega^2 \cdot \cos 2\theta \right)$$

Gas force is calculated using equation 13.

Equation 13

$$\text{Gas Force} = F_{\text{gas}} = \text{Pressure} \cdot \frac{\pi \cdot D^2}{4}$$

Max Gas Force	
RPM	N
1000	23100.51
1500	29085.62
1700	33223.10
2000	36436.69
2500	38532.76
3000	40107.46
3500	41499.94
4000	42632.29
4500	42376.94
5000	43849.09
5500	46740.99
5700	47880.39
6000	48056.82

Table 5 – Max gas force at all RPM

To calculate the inertia force, the reciprocating mass must first be calculated. A general assumption is the 1/3rd of the rod + gudgeon pin + piston mass is assumed to be reciprocating but from accurate CAD the connecting rod is split at its COG, and it was found the 41% of rod mass plus the piston and gudgeon pin can be assumed at reciprocating mass.

Equation 14

$$\therefore \text{Reciprocating mass} = 41\% \text{ rod} + \text{piston} + \text{gudgeon pin}$$

From reciprocating mass and piston acceleration.

Equation 15

$$\text{Inertia force} = \text{Reciprocating mass} \times \text{piston acceleration (mm/sec)}$$

Equation 16

$$= (M \cdot R \cdot \omega^2 \cdot \cos\theta) + \left(\frac{R}{L} \cdot M \cdot R \cdot \omega^2 \cdot \cos 2\theta\right)$$

1st order2nd order

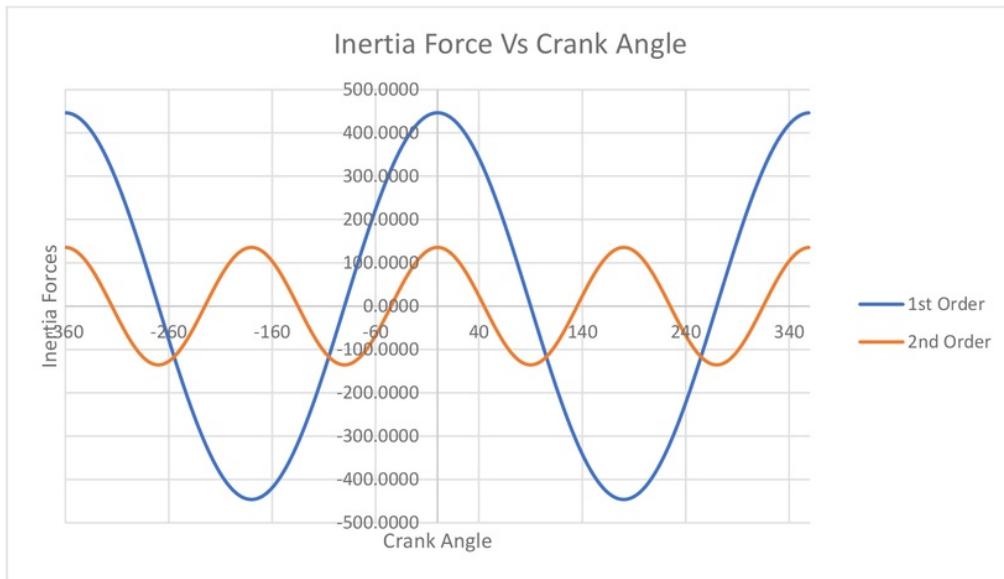


Figure 13 – 1st and 2nd order inertia forces Vs crank angle

TDC is at -360, 0, and 360° at this point the first order inertia forces combine to an increased positive value, thus increasing the force seen on the connecting rod; this results in the 1st order forces to cancel at BDC -180, 180° this results in a downwards force.

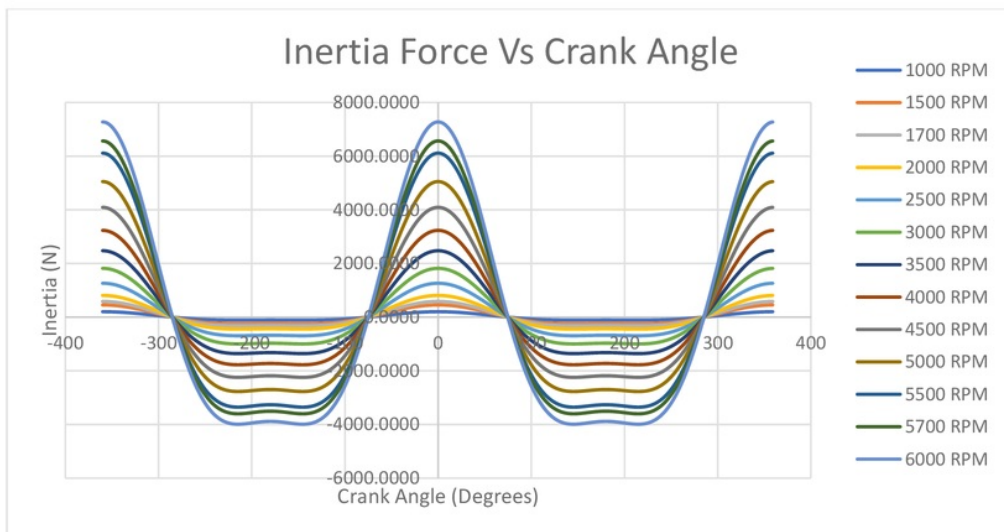


Figure 14 – Inertia force at each RPM Vs crank angle

From gas force and inertia force

$$\text{Equation 17} \quad \text{Net Force} = \text{Gas Force} - \text{Inertia Force}$$

Maximum inertia force occurs at max RPM on the exhaust stroke at TDC, at this point, the connecting rod experiences its largest tensile load; this is due to their being little or no gas pressure which cushion the inertia forces. A large inertia force causes the rod to stretch if too high it can cause the piston to valve clearance issues.

From net force

$$\text{Equation 18} \quad \text{Piston Side Force} = F_{\text{side}} [N] = F_{\text{net}} \cdot \tan \phi$$

And rod force

$$\text{Equation 19} \quad \text{Rod force } F_{\text{rod}} = \frac{F_{\text{net}}}{\cos \phi}$$

From side force, where μ = Co efficient of friction

$$\text{Equation 20} \quad \text{Friction force} = \mu \cdot F_{\text{side}}$$

The maximum compressive force happens ATDC and is calculated from F_{net} , but as the rod angle is not vertical when the maximum force occurs the rod angle must be taken into consideration. Therefore the maximum compressive force is the maximum rod force (F_{rod}); as the maximum tensile force is seen at TDC, therefore, there is 0° rod angle the minimum value for F_{net} show the highest tensile forces as negative values show compression and positive compression. The maximum forces for each RPM are found using the min, and maximum function in Excel and are shown in table 6.

$$\text{Equation 21} \quad \text{Max Compressive Force} = \text{Max } F_{\text{rod}}$$

$$\text{Equation 22} \quad \text{Max Tensile Force} = \text{Min } F_{\text{net}}$$

RPM	Max Gas Force N	Max Net Fcomp N	Max Net Ftensile N
1000	23100.51	23053.53	172.30
1500	29085.62	28836.91	347.41
1700	33223.10	32777.16	347.41
2000	36436.69	35587.90	738.85
2500	38532.76	36991.02	1496.31
3000	40107.46	37759.05	2348.01
3500	41499.94	38120.23	3444.75
4000	42632.29	38042.25	4662.23
4500	42376.94	36630.27	6048.06
5000	43849.09	36636.44	7572.47
5500	46740.99	37565.77	9284.72
5700	47880.39	37862.27	10054.34
6000	48056.82	36785.70	11237.48

Table 6 – Maximum Gas, Net, and Tensile forces at each RPM

From the maximum compressive and tensile forces and using SolidWorks to calculate the CSA, it is possible to calculate maximum stress in tension and compression using equations 23 + 24.

Equation 23
$$\text{Max Compressive Stress} = \frac{\text{Max compressive force}}{\text{CSA}}$$

Equation 24
$$\text{Max Tensile Stress} = \frac{\text{Max tensile force}}{\text{CSA}}$$

RPM	Max Stress Comp N/mm ²	Max Stress Tensile N/mm ²
1000	153.69	0.69
1500	192.25	1.15
1700	218.51	2.32
2000	237.25	4.93
2500	246.61	9.98
3000	251.73	15.65
3500	254.13	22.97
4000	253.62	31.08
4500	244.20	40.32
5000	244.24	50.48
5500	250.44	61.90
5700	252.42	67.03
6000	245.24	74.92

Table 7 – Maximum compressive and tensile stress at each RPM

Each connecting rod was split into four separate parts in CAD shown below in Figures 15, and 16, utilizing the mass properties in SolidWorks it was possible to calculate the inertia force acting on each section of the connecting rod due to the weight above it in Excel, this was done to ascertain the FSF at different points of connecting rod shank.

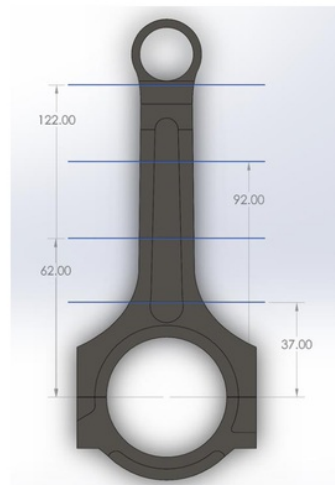


Figure 15 – GDI connecting rod sections

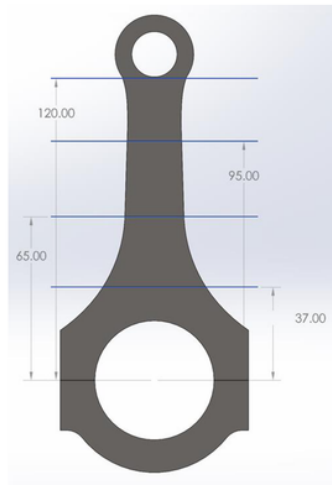


Figure 16 – Pankl connecting rod sections

Tables 8 + 9 show the CSA and the mass acting above the component section, this includes the gudgeon pin and piston.

GDI	CSA mm ²	Mass Kg	Mass Above (Kg)
point A	440	0.0332	0.461
point b	185	0.0664	0.527
point c	180	0.0429	0.57
point d	265	0.038	0.608

Table 8 - Mass above sections GDI connecting rod

Pankl	CSA mm ²	Mass Kg	Mass Above (Kg)
point A	434.41	0.03292	0.46032
point b	104.18	0.02255	0.48287
point c	107.71	0.01421	0.49708
point d	448.24	0.02788	0.52496

Table 9 - Mass above sections Pankl connecting rod

Using the masses given above in tables 8+ 9 and equation 16, the inertia force seen at each section of the rod was calculated at all RPM and the values are shown below in Figures 10 and 11.

Finertia Max GDI									
RPM	Overall Inertia	Point A	Fnet	Point B	Fnet	Point C	Fnet	Point D	Fnet
1000	349.91	268.00	22884.09	306.66	22852.88	331.61	22832.73	353.70	22814.89
1500	787.29	603.00	28623.21	689.97	28556.52	746.11	28513.46	795.83	28475.34
1700	1011.23	774.52	32607.88	886.23	32519.15	958.34	32461.87	1022.20	32411.15
2000	1399.63	1072.00	35543.90	1226.62	35698.61	1326.43	35638.56	1414.80	35258.40
2500	2186.92	1675.00	37117.53	1916.60	36913.40	2072.54	36781.64	2210.63	36664.97
3000	3149.16	2412.00	38069.52	2759.90	37775.58	2984.46	37585.84	3183.31	37417.83
3500	4286.36	3283.00	38687.70	3756.53	38282.08	4062.18	38020.26	4332.84	37788.41
4000	5598.51	4288.01	38910.78	4906.48	38374.01	5305.71	38027.53	5659.21	37720.72
4500	7085.62	5427.01	37728.62	6209.77	37067.26	6715.03	36640.35	7162.44	36262.33
5000	8747.67	6700.01	38109.84	7666.38	37282.04	8290.17	36747.71	8842.52	36274.55
5500	10584.69	8107.01	39532.12	9276.32	38492.35	10031.10	37821.19	10699.45	37226.88
5700	11368.48	8707.33	40050.42	9963.23	38921.08	10773.90	38192.09	11491.74	37546.58
6000	12596.65	8707.33	40140.12	9963.23	38998.73	10773.90	38261.97	11491.74	37609.58

Table 10 – Maximum inertia forces acting at each section of the GDI connecting rod

Finertia Max Pankl										
RPM	Overall Inertia	Point A	Fnet	Point B	Fnet	Point C	Fnet	Point D	Fnet	
1000	302.2154146	267.85	22884.21	280.98	22873.61	289.25	22866.94	305.47	22853.84	
1500	679.9846829	603.00	28623.46	689.97	28600.82	650.80	28586.55	795.83	28558.56	
1700	873.4025483	774.10	32608.22	812.02	32578.10	835.92	32559.11	882.80	32521.87	
2000	1208.861659	1071.42	35544.38	1123.91	35500.67	1156.98	35473.12	1221.87	35419.08	
2500	1888.846342	1674.09	37118.30	1756.10	37049.01	1807.78	37118.30	1909.18	36919.67	
3000	2719.938732	2410.69	38070.62	2528.79	37970.84	2603.21	37907.97	2749.21	37784.60	
3500	3702.138829	3281.22	38689.23	3441.96	38551.54	3543.25	38464.77	3741.99	38294.54	
4000	4835.446634	4285.68	38912.80	4495.62	38730.59	4627.92	38615.77	4887.49	38390.49	
4500	6119.862147	5424.06	37731.11	5689.77	37506.61	5857.21	37365.14	6185.73	37087.57	
5000	7555.385366	6696.37	38112.95	7024.41	37831.95	7231.13	37654.88	7636.70	37307.46	
5500	9142.016293	8102.61	39536.03	8499.54	39183.08	8749.66	38960.66	9240.41	38524.28	
5700	9818.978822	8702.61	40054.67	9128.93	39671.31	9397.57	39429.73	9924.66	38955.76	
6000	10879.75493	8702.61	40144.41	9128.93	39756.96	9397.57	39512.81	9924.66	39033.78	

Table 11- Maximum inertia forces acting at each section of the Pankl connecting rod

From tensile and compressive stress, along with values for UTS, it is possible to calculate stress, strain, Young's modulus, and the maximum displacement in tensions and compression.

Equation 25
$$\text{Stress} = \frac{F}{A}$$

Equation 26
$$\text{Strain} = \frac{x}{L}$$

Equation 27
$$\text{Youngs Modulus} = \frac{\text{stress}}{\text{strain}} = \frac{\left(\frac{F}{A}\right)}{\left(\frac{x}{L}\right)} = \frac{FL}{xA}$$

Equation 28
$$\text{Max Compression displacment} = \frac{MF_{\text{comp}}.L}{E.CSA}$$

Equation 29
$$\text{Max Tensile displacment (stretch)} = \frac{-MF_{\text{tens}}.L}{E.CSA}$$

The static safety factors of the shank are calculated using the following equation.

Equation 30
$$\text{Static Safety Factor} = \frac{UTS}{\text{Max Compressive Stress}}$$

Tables 12 + 13 show the safety factors, the displacement due to compression and tension for both the GDI and Pankl connecting rods. The connecting rods minimum CSA was used to calculate the static safety factor as this is the critical section of the connecting rod. The calculations were conducted for both steel and titanium this was done as the GDI rod is manufactured from steel and the Pankl is a titanium connecting rod. As can be seen from the tables there is no difference in the static safety factor between the GDI and Pankl rods this is due to the CSA not being taken into account. If both connecting rods were to be remanufactured from steel the Pankl rod would displace 32.8% more in compression and 34.6% in tension, although this is a lot less than comparing the displacement from the materials the connecting rods are manufactured from whereby the Pankl connecting rod displaces 141% more than the GDI connecting rod in compression and 146% more in tension. If the Pankl connecting rod were to be remanufactured from either steel or carbon fibre, which has a much higher Young's modulus, it would be possible to increase the compression ratio through a thinner head gasket as there would be a lower chance of piston to valve contact through displacement due to tension.

GDI						
CSA	Material	UTS	youngs modulus	Safety Factor	Comp dist	Stretch
mm ²		MPa	MPa		mm	mm
150	4340 steel	745	200000	2.65	0.189	0.026
	Ti6Al4V Grade 5	1000	110000	3.55	0.343	0.048

Table 12 – Material properties and GDI safety factors and displacements

Pankl						
CSA	Material	UTS	youngs modulus	Safety Factor	Comp dist	Stretch
mm ²		MPa	MPa		mm	mm
109	4340 steel	745	200000	2.65	0.251	0.035
	Ti6Al4V Grade 5	1000	110000	3.55	0.457	0.064

Table 13 - Material properties and Pankl safety factors and displacements

4.1.2 Analysis

The calculations outlined in the previous section were completed by reducing the con-rod mass from 100% to 50% in 10% increments to analyse the effects of optimising the mass of the rod, specifically the effect on skirt friction seen by the piston and the effect of the fatigue safety factors this analysis was conducted for both connecting rod examined.

4.1.2.1 Friction

As suggested in (Lapp, et al., 2010) paper, table 14 and Figure 17 show that as the mass of the connecting rod reduced there is also a reduction in friction at maximum RPM, for every 10% mass removed from the connecting rod there is a 0.52% reduction in friction.

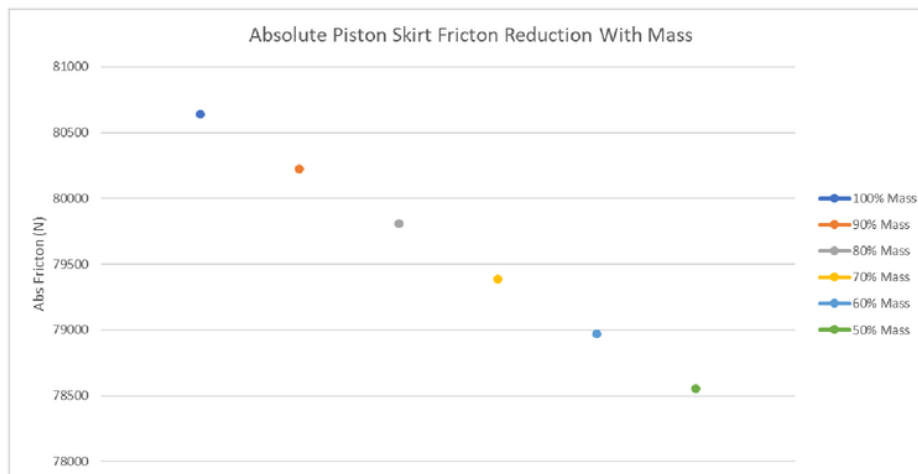


Figure 17 – Piston Skirt friction with mass reduction Chart

	Total friction	Friction / °	Reduction %
100% Mass	80640.35	112.00	
90% Mass	80223.05	111.42	0.52
80% Mass	79805.76	110.84	1.04
70% Mass	79388.47	110.26	1.56
60 % Mass	78971.17	109.68	2.09
50% Mass	78553.88	109.10	2.61

Table 14 – Total friction and friction per ° with reduced mass

Tables 15, and 16 show the sum of the friction forces on each stroke for connecting rod with 100% and 50% mass.

Friction 100% Connecting Rod Mass														
	Friction Forces	1000 RPM	1500 RPM	1700 RPM	2000 RPM	2500 RPM	3000 RPM	3500 RPM	4000 RPM	4500 RPM	5000 RPM	5500 RPM	5700 RPM	6000 RPM
intake	-360 / - 180	1963.16	2777.50	3035.03	3187.35	3423.17	3788.08	4131.43	4486.92	4777.25	5230.77	5624.38	5773.38	6034.31
compression	-180 / 0	-6917.87	-9595.08	-10149.43	-10364.42	-11055.81	-12021.60	-12426.43	-13119.07	-14031.61	-14826.83	-15350.74	-15443.53	-15558.40
expansion	0 / 180	25564.86	36392.25	38996.59	40393.05	42759.24	45055.45	45652.88	46778.92	48507.42	49730.16	49856.24	49350.81	48444.94
exhaust	180 / 360	-2080.01	-2814.23	-3162.90	-3747.48	-4212.76	-4657.99	-5432.90	-6481.11	-7537.36	-8792.35	-9803.23	-10142.62	-10602.70
Total friction (Abs)		36525.90	51579.06	55343.95	57692.29	61450.98	65523.12	67643.63	70866.01	74853.63	78580.11	80634.59	80710.33	80640.35
Total friction		18530.13	26760.44	28719.27	29468.50	30913.85	32163.95	31924.97	31665.66	31715.70	31341.75	30326.64	29538.03	28318.14
Average F Abs		50.73	71.64	76.87	80.13	85.35	91.00	93.95	98.43	103.96	109.14	111.99	112.10	112.00

Table 15 – Friction forces at 100% rod mass

Friction 50% Connecting Rod Mass														
	Friction Forces	1000 RPM	1500 RPM	1700 RPM	2000 RPM	2500 RPM	3000 RPM	3500 RPM	4000 RPM	4500 RPM	5000 RPM	5500 RPM	5700 RPM	6000 RPM
intake	-360 / - 180	1948.67	2744.89	2993.15	3129.39	3332.61	3657.68	3953.93	4255.09	4483.84	4868.53	5186.08	5302.62	5512.69
compression	-180 / 0	-6903.38	-9562.47	-10107.56	-10306.46	-10965.25	-11891.19	-12248.93	-12887.24	-13738.20	-14464.59	-14912.44	-14972.77	-15036.79
expansion	0 / 180	25550.37	36359.65	38954.71	40335.09	42668.69	44925.05	45475.38	46547.09	48214.01	49367.93	49417.93	48880.05	47923.32
exhaust	180 / 360	-2065.52	-2781.63	-3121.03	-3689.52	-4122.20	-4527.58	-5255.41	-6249.28	-7243.95	-8430.12	-9364.93	-9671.86	-10081.08
Total friction (Abs)		36467.95	51448.65	55176.45	57460.46	61088.75	65001.50	66933.65	69938.69	73679.99	77131.18	78881.38	78827.30	78553.88
Total friction		18530.13	26760.44	28719.27	29468.50	30913.85	32163.95	31924.97	31665.66	31715.70	31341.75	30326.64	29538.03	28318.14
Average F Abs		50.65	71.46	76.63	79.81	84.85	90.28	92.96	97.14	102.33	107.13	109.56	109.48	109.10

Table 16 – Friction forces at 50% rod mass

In Figure 18 it can be seen that apart from at 1000 RPM the reduction in connecting rod mass reduces the friction over the intake stroke. From tables 15 + 16 it can be seen that during the compression stroke friction is reduced from 6,033 N to 5512 N which is a 9.46% reduction at 5500 RPM.

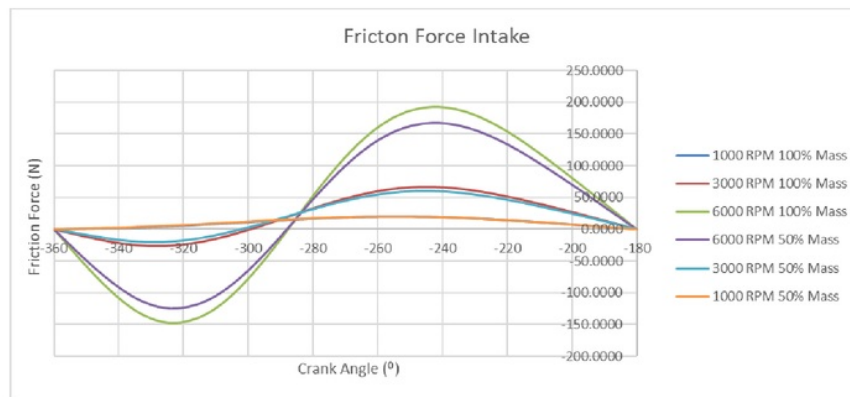


Figure 18 – Intake stroke friction force chart

Figure 19 shows that at 1,000 RPM there is a minute reduction in friction, at 3000 RPM less friction is produced from the connecting rod between -180° to -80° , between -80° and 0° the lower weight connecting rod causes a small increase in friction. Whereas, at maximum RPM the lighter connecting rod produces less friction throughout the whole stroke. From tables 15 + 16 at 6000 RPM friction is reduced from 15558 N to 15036 N which is a 3.47% reduction during the compression stroke.

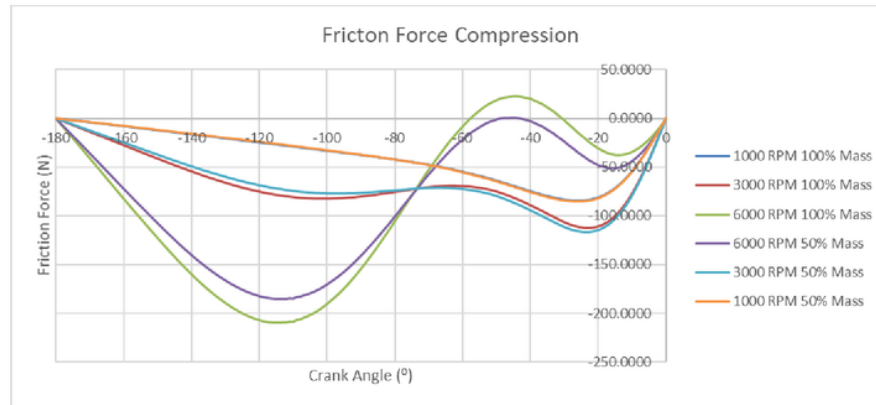


Figure 19 – Compression stroke friction forces chart

Figure 20 shows friction over the expansion stroke; there is little or no benefit of a reduced mass connecting rod at 1000 and 3000 RPM. From tables 15 + 15 at 6000 RPM total friction is reduced from 49,856 N to 49,418 N which is a 0.88% reduction of friction, from Figure 15 it can be seen that although there is an overall reduction in friction between 20° - 80° ATDC friction is higher with the reduced mass connecting rod.

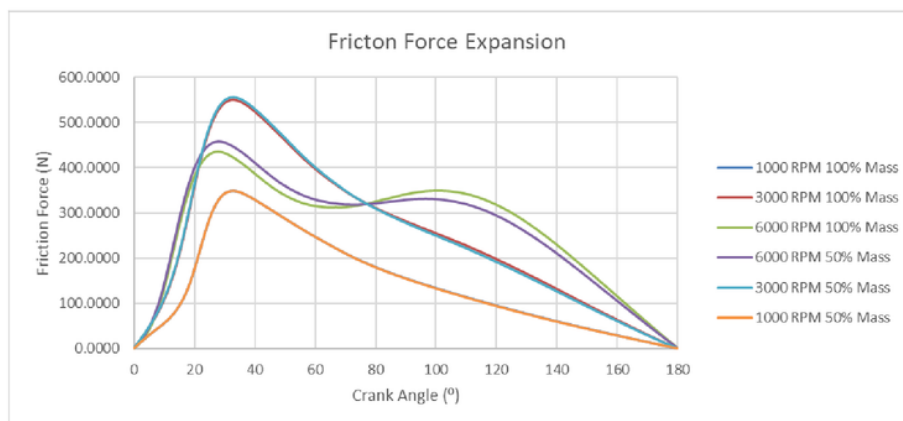


Figure 20 – Expansion stroke friction forces chart

The friction forces over the exhaust stroke are shown in Figure 21; friction force is reduced at 3000 RPM and 6000 RPM. From tables 15 + 16 it can be seen that total friction during this stroke is reduced from 10,602 N to 10,081 N which is a 5.17% reduction.

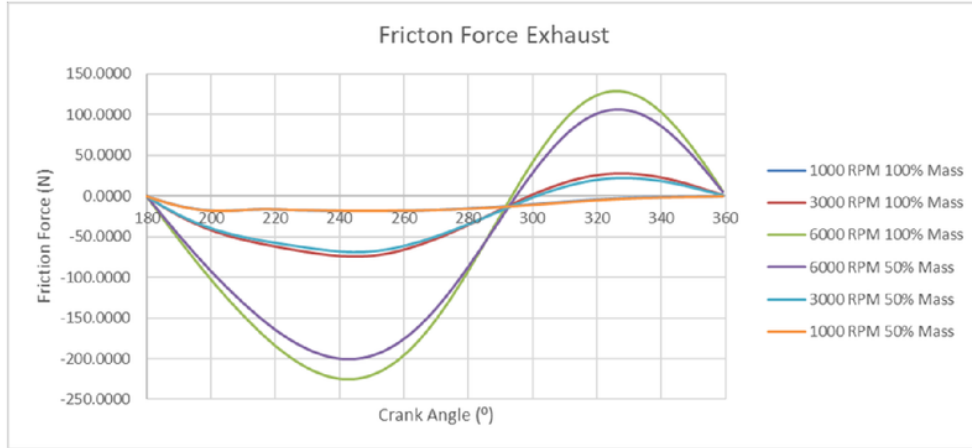


Figure 21 – Exhaust stroke friction force chart

4.1.2.2 Fatigue Safety Factors

Due to the connecting rod having to withstand up to 10^8 cycles it is important when designing and analysing the connecting rod that the FSF is taken into account, this analysis is generally conducted in the critical CSA utilising maximum stress values. To calculate the FSF first a mean, an alternating, endurance limit, and allowable range must be calculated (Grolier, 2000).

Equation 31

$$\sigma_{min} = \frac{F_{net_{comp}}}{CSA}$$

Equation 32

$$\sigma_{max} = \frac{F_{net_{tension}}}{CSA}$$

Equation 33

$$Actual\ range\ of\ stress = \sigma_{max} - \sigma_{min}$$

Equation 34

$$Mean\ stress = \frac{(\sigma_{max} + \sigma_{min})}{2}$$

Equation 35

$$Endurance\ limit = 0.45 \times UTS$$

Equation 36

$$Allowable\ range = (2 \times endurance\ limit) \times \left(\frac{UTS - mean\ stress}{UTS} \right)$$

Equation 37

$$FSF = \frac{Allowable\ Range}{Actual\ Range}$$

Tables 17 + 18 display the calculations and safety factors for the GDI connecting rod at 100% and 50% mass, see appendix 9.5 for screen shots from Excel calculations. Due to the mass above each section being used to calculate the inertia force for each section of the rod, the net forces differ marginally throughout the connecting rod. Although this appears to have little or no effect of the FSF of the GDI rod; this is due to there being a maximum difference in mass of 0.09 Kg difference at point D and therefore has little effect on the inertia force.

GDI Calculations 100% Rod Mass										
Section	Sigma min	singma max	Stress Range	Mean Stress	allowable range	Fatiguge saftey factor	GDI	mm ²	Kg	Mass Above
A	-83.51	25.51	110.89	55.45	620.60	5.60	point A	440.49	0.03317	0.46057
B	-199.32	60.89	261.30	130.65	552.92	2.12	point b	184.56	0.06643	0.527
C	-204.84	62.58	277.16	138.58	545.78	1.97	point c	179.58	0.04288	0.56988
D	-139.08	42.49	185.64	92.82	586.96	3.16	point d	264.5	0.03797	0.60785

Table 17 – GDI fatigue safety factors for each section at 100% rod mass

GDI Calculations 50% Rod Mass										
Section	Sigma min	singma max	Stress Range	Mean Stress	allowable range	Fatiguge saftey factor	GDI	A mm ²	Kg	Mass Above
A	-87.29	21.38	110.83	55.41	620.63	5.60	point A	440.49	0.016585	0.443985
B	-208.33	51.03	262.83	131.41	552.23	2.10	point b	184.56	0.033215	0.4772
C	-214.11	52.44	274.43	137.21	547.01	1.99	point c	179.58	0.02144	0.49864
D	-145.37	35.61	185.05	92.53	587.23	3.17	point d	264.5	0.018985	0.517625

Table 18 – GDI fatigue safety factors for each section at 50% rod mass

Figure 22 shows FSF graphically.

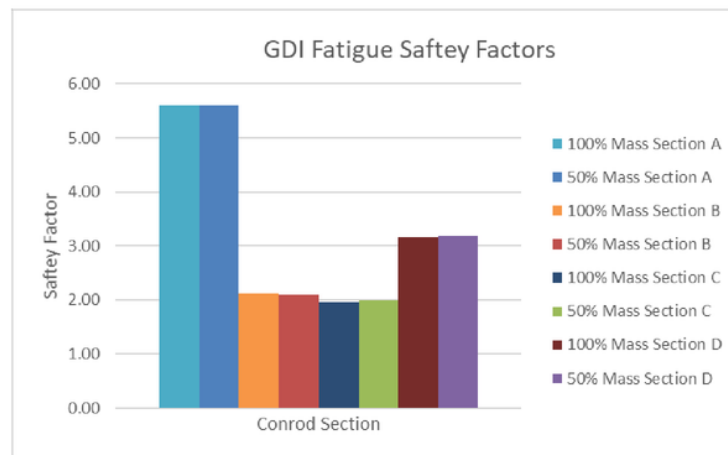


Figure 22 – GDI fatigue safety factors chart

Tables 19 + 20 display the calculations and safety factors for the Pankl connecting rod at 100% and 50% mass, see appendix 9.5 for screen shots of Excel calculations. As for the GDI connecting rod, there is little or no effect on the FSF due to a marginal difference in inertia force as there is a 0.048 Kg at point D.

Pankl Calculations 100% Rod Mass										
Section	Sigma min	singma max	Stress Range	Mean Stress	allowable range	Fatiguge saftey factor	GDI	mm^2	Kg	Mass Above
A	-84.68	25.87	112.44	56.22	849.40	7.55	point A	434.41	0.03292	0.46032
B	-341.53	104.33	454.09	227.05	695.66	1.53	point b	104.18	0.02255	0.48287
C	-353.10	107.87	469.24	234.62	688.84	1.47	point c	107.71	0.01421	0.49708
D	-82.07	25.07	109.22	54.61	850.85	7.79	point d	448.24	0.02788	0.52496

Table 19 – Pankl fatigue safety factors for each section at 100% rod mass

Pankl Calculations 50% Rod Mass										
Section	Sigma min	singma max	Stress Range	Mean Stress	allowable range	Fatiguge saftey factor	GDI	mm^2	Kg	Mass Above
A	-88.51	21.68	112.38	56.19	849.43	7.56	point A	434.41	0.01646	0.44386
B	-356.97	87.43	453.54	226.77	695.91	1.53	point b	104.18	0.01128	0.455135
C	-369.07	90.40	468.79	234.39	689.05	1.47	point c	107.71	0.00711	0.46224
D	-85.78	21.01	109.04	54.52	850.93	7.80	point d	448.24	0.01394	0.47618

Table 20 - Pankl fatigue safety factors for each section at 50% rod mass

Figure 23 shows FSF graphically.

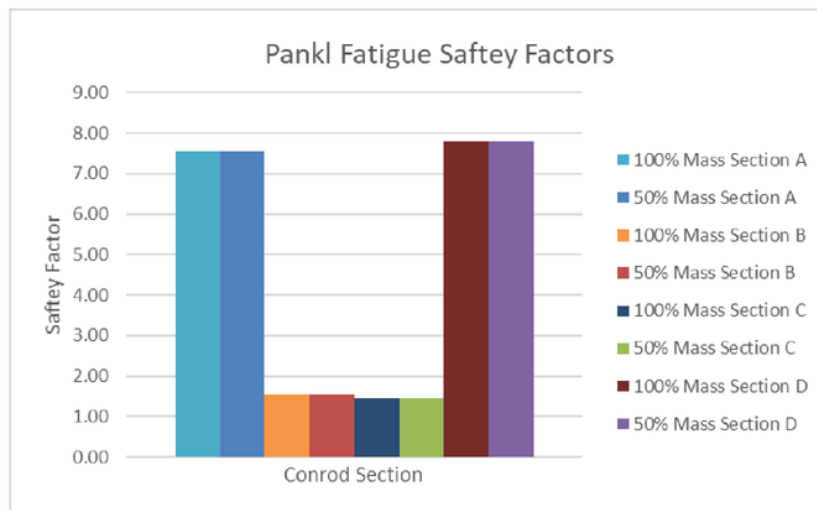


Figure 23 – Pankl fatigue safety factors chart

4.2 FEA

FEA analysis was conducted on both connecting rods; this was done using both Solidworks CAD package and using FEmap FEA package. With both models set up and constrained the same way, the only difference is there is more control over the mesh sizing in FEmap. When analysing compression, the main journal was fixed, and the compressive force of 42,000 N was applied to the lower surface of the small end, for tension the main journal is fixed in the same way with 11,000 N of force applied to the upper inner surface of the small end journal.

4.2.1 GDI

Figures 24, and 25 show the connecting rod's displacement being analysed in compression. Maximum displacement seen in Solidworks is 0.257mm and 0.122mm in FEmap and this compares to a displacement of 0.169mm from the calculations conducted in Excel. Solidworks displacement prediction is 52% higher and Femap is 38% lower than the hand calculations preformed.

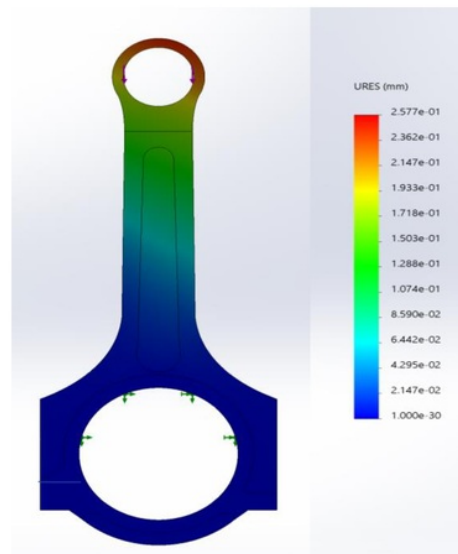


Figure 24 – GDI compressive displacement Solidworks

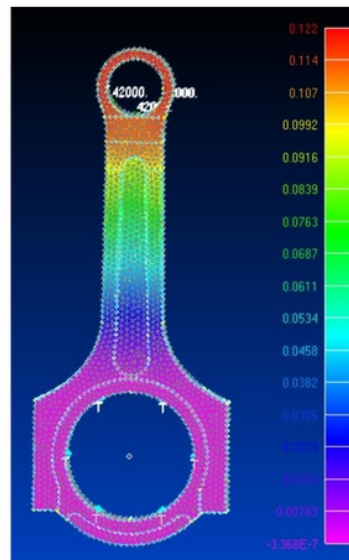


Figure 25 – GDI compressive displacement FEmap

Figures 26, and 27 show displacement from the tensile forces. A maximum displacement of 0.117mm is seen in Solidworks and 0.102mm in FEmap, this compares to 0.05mm calculated in Excel. Solidworks predicted extension is 134% higher and FEmap's is 104% higher than the hand calculations.

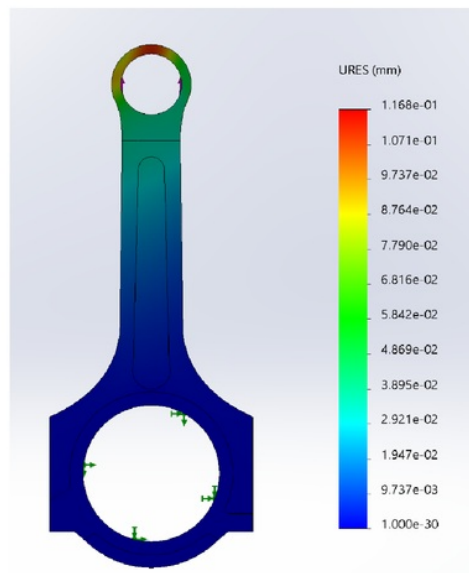


Figure 26 – GDI tensile displacement soldworks

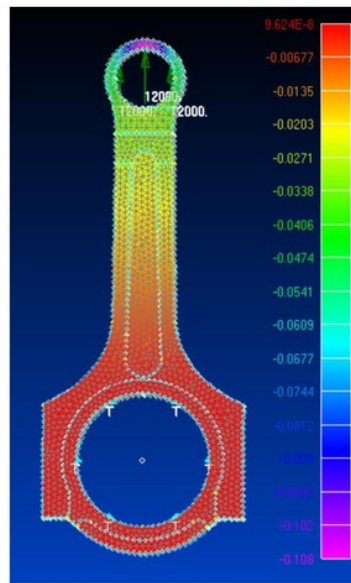


Figure 27 – GDI tensile displacement FEmap

Figure 28 shows the FOS produced in Solidworks, this shows a minimum safety factor of 2.35 which is 25% lower than the 2.95 from the hand calculations.

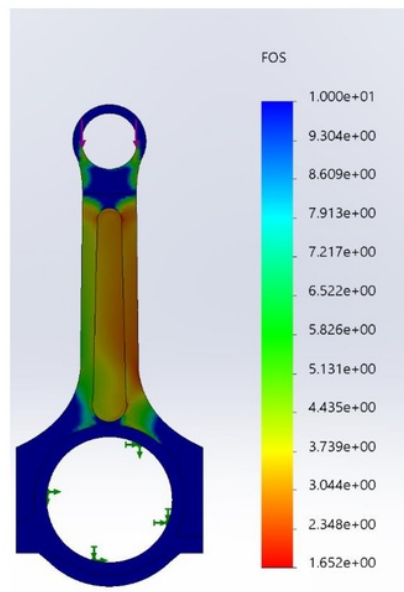


Figure 28 – GDI safety factor Solidworks

4.2.2 Pankl

The connecting rod's displacement was analysed in compression and is shown in Figures 29, and 30. The maximum displacement is 0.201mm in Solidworks and 0.237mm in FeMap, which compares to 0.41mm calculated in Excel with Solidworks prediction being 104% lower than the hand calculations while FEmaps prediction is 73% lower.

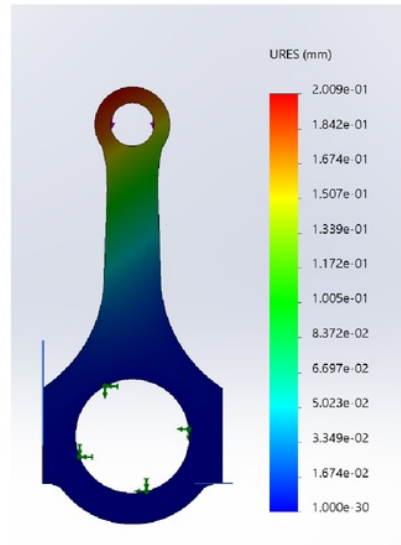


Figure 29 – Pankl compressive displacement Solidworks

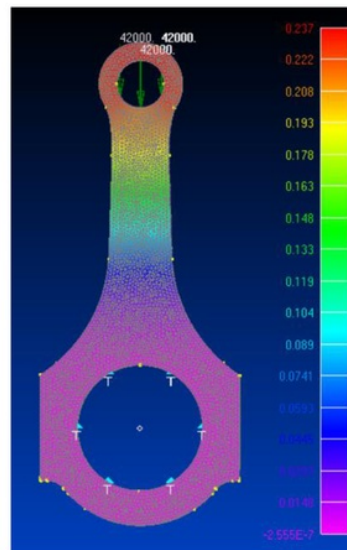


Figure 30 – Pankl compressive displacement FEmap

Figures 31, and 32 show displacement from the tensile forces. A maximum displacement of 0.057mm is seen in Solidworks and 0.103mm in FEmap; these values are 53% and 15% lower respectively than then 0.122mm displacement calculated in Excel.

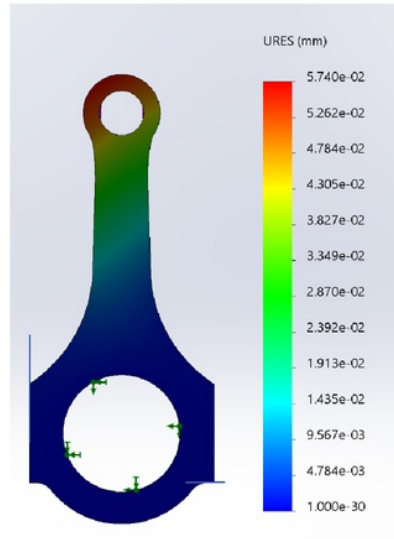


Figure 31 – Pankl tensile displacement Solidworks

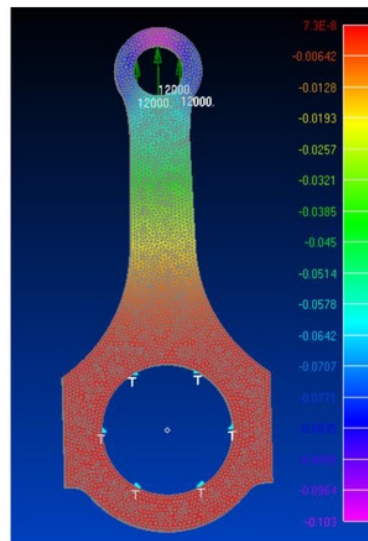


Figure 32 – Pankl tensile displacement FEmap

The FOS is shown in Figure 33, this shows a minimum safety factor of 5 which is 26% higher than the 3.96 from the hand calculations.

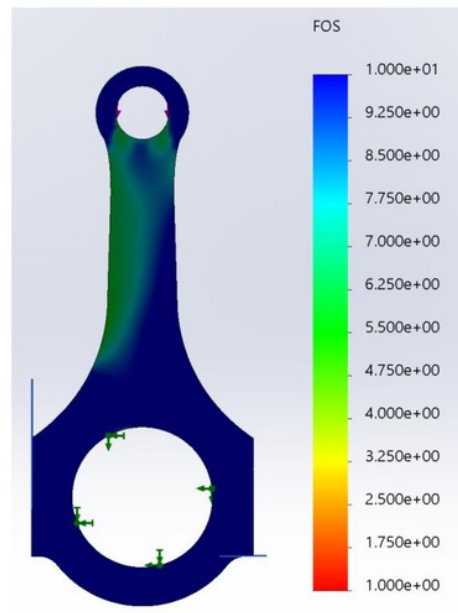


Figure 33 – Pankl safety factor Solidworks

5 CARBON FIBRE TESTING

This section details the destructive tensile carbon tested completed by the author with sections on testing standards, initial results, issues with test results, and the steps conducted in order to calibrate the results from the Denison test machine used. This Denison test machine is manufactured to ASTM E4. It is a computer-controlled precision testing machine and is suitable for a wide range of materials for compression, tension, bending, and shearing tests for further information including full machine specifications see appendix 9.2.

Equations 38 + 39 display another way Young's Modulus can be calculated

$$\text{Equation 38} \quad K = \frac{F}{\Delta L}$$

$$\text{Equation 39} \quad E (K) = \frac{KL}{A}$$

5.1 TESTING STANDARDS

ASTM tensile testing standards were used and were obtained from (Hodgkinson, 2000) book on mechanical testing of advanced fibre composites. Current testing standards for UD carbon fibre are ASTM D 3039³ (0° + 90°), the dimensions of the sample sizes are shown in Figure 34 with A referring to the 0° fibres and B the 90°. As explained in section 2.7.3 a sheet was made up that measured 300mm x 300mm; a CNC milling tool was then used to cut the individual test coupons. Once the individual coupons were machine both the surface at the end of the test specimen and the composite tabs which were manufactured 2 plys thin (0.25), were keyed using sandpaper, cleaned with acetone, two plys of 45° were used in between the coupon and tab, and finally taped up and returned to the heated press to bond the two components together.

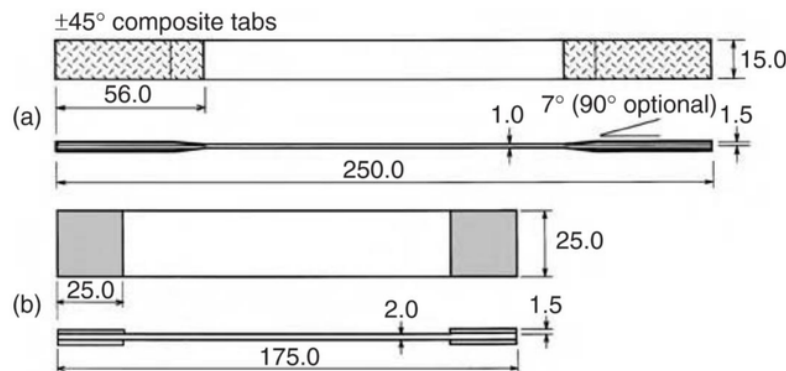


Figure 34 – ASTM 3039 tensile testing dimensions

5.2 INITIAL RESULTS

Both the 0° and 90° coupons loaded into the Denison with the tabs all the way in the jaws so only the gauge length was protruding from the jaws, each time the length was checked to improve accuracy. Once the specimens were fitted between the jaws the Denison was set to displace by 20mm/min for all test conducted.

5.2.1 0° UD Tests

Figure 35 shows the force-extension graph from the recorded data, Table 21 shows the recorded results for load, displacement, stress, strain, Young's modulus, it also includes the calculated Young's modulus calculated from equation 38, and 39.

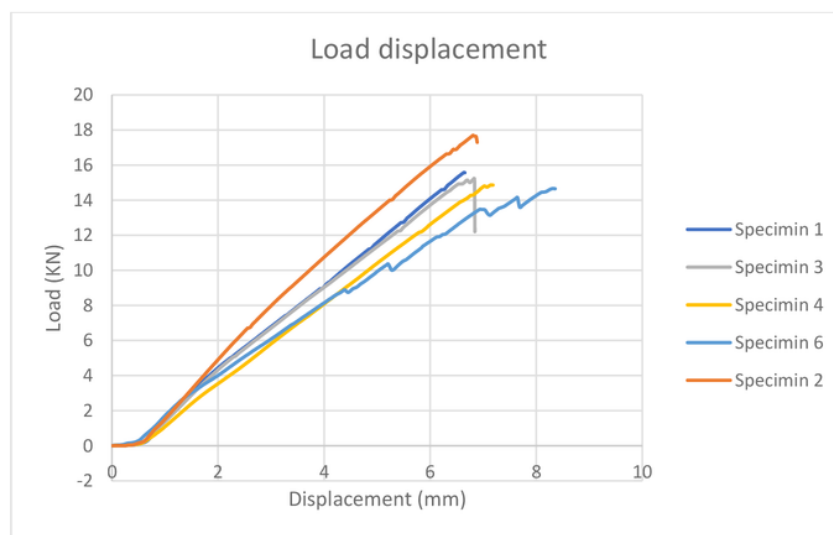


Figure 35 - 0° UD load displacement chart

Specimin	Load (N)	Disp (mm)	Stress (N/mm ²)	Strain	E (Gpa) Measured	E (Gpa) Calculated
1	15575	6.66	1038.33	0.033	7.67	31.20
2	17695	6.89	1179.67	0.034	9.80	34.26
3	15250	6.84	1016.67	0.034	7.83	29.71
4	14865	7.19	991.00	0.036	ERROR	27.58
6	14660	8.36	977.33	0.042	ERROR	23.39
Average	15609	7.19	1040.60	0.036	8.43	28.96
% difference	20.70259209	25.58978212	20.70	25.590	27.77053455	46.48

Table 21 - 0° material properties measured and calculated

5.2.2 90° Tests

Figure 36 shows the force-extension graph from the recorded data, Table 22 shows the recorded results from the Denison as in section 5.2.1

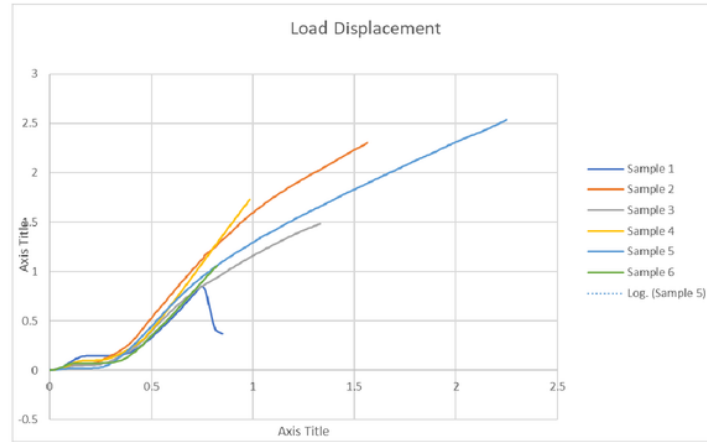


Figure 36 - 90° UD load displacement chart

Specimin	Load (N)	Disp (mm)	A	Stress (N/mm ²)	Strain	E (Gpa) Measured	E (Gpa) Calculated
1	855	0.85	48.30	17.70	0.01	127.43	2.60
2	2305	1.57	59.64	38.65	0.01	93.75	3.08
3	1485	1.33	60.90	24.38	0.01	ERROR	2.29
4	1730	0.99	60.90	28.41	0.01	102.13	3.60
5	2535	2.25	58.80	43.11	0.02	ERROR	2.39
6	1060	0.82	60.27	17.59	0.01	12.54	2.67
Average	1661.67	1.30	58.14	28.31	0.01	83.96	2.77
% difference	196.49	173.18	26.09	145.13	90.05	916.19	34.98

Table 22 - 90° material properties measured and calculated

5.3 ANALYSIS OF ORIGINAL TESTS

When originally analysing the data recorded by the Denison the values for extension particularly in the 0° fibre direction look unrealistic high. The results from the carbon testing were compared to a specification sheet provided by McLaren (appendix 9.3) as it wasn't possible to obtain one from the original material manufacture. Although all the material dimensions were input before running the sample, there were 4 occasions over the 9 test's where the program was unable to calculate a Young's modulus and when it did it was incredibly low for the 0° and high for the 90°. It also didn't line up with the calculated values which in this section were calculated from $E = \text{stress/strain}$ which were both values recorded by the Denison and not calculated. It was observed that the test specimen and tabs appeared to be slipping, this was especially notable in the 0° samples.

The Young's modulus for the 0° UD according to McLaren's data should have been between 130-170 Gpa, the Denison results range from 7.67-9.8 Gpa which is a 27.77% difference over the 6 tests, and the calculated results range from 23.39-34.26 Gpa which is a 46.48% difference if the results from the runs with errors this reduces to 18.28 %.

When analysing the 90° samples it should be noted that the samples originally produced for the author to test were accidentally used by someone else. A new sheet was produced, but the CNC machine was out of order, and the samples were cut using a band saw by the machine shop operator. Unfortunately, the samples were not cut straight and as can be seen in table 22 the CSA areas range by 26.09%, as the CSA area is taken into account to calculate E the author decided to test the samples anyway while more were being manufactured. Due to the uneven geometry, the load and displacement values have a 196% and 173.18% between the highest and lowest values. Again, the Young's modulus is unrealistic whereas the datasheet provided by McLaren shows values for E of 9 Gpa the Denison measured values range from 12.54-127.43 Gpa which is a 916% difference, and the calculated values range from 2.29 - 3.6 Gpa which is a 34.98% difference but is a third of what Young's modulus should be.

5.4 CARBON RETESTS

Due to the inconstant results and slipping of the composite tabs, it was decided that all the tests need to be repeated, and the end tabs were investigated to reduce slippage. When no tabs were attached, the coupon would consistently fail prematurely in the section between the jaw. It was found the best way to negate slipping was to key the ends of the coupon and super glue emery tape to the end's and put shims in the bottom jaws and in between tests, the jaws were removed and cleaned with a wire brush.

Note: as the specification sheet was unavailable to the carbon tested and it was unclear whether the material was within its shelf life 0° , a carbon fibre strip was purchased from easy composites to be tested as a comparison and an investigation was conducted on whether 45° UD would behave in the same way as 90° UD.

5.4.1 0°

Figure 37 shows the force-extension graph from the recorded data, Table 23 shows the recorded results from the Denison, along with calculated values for stiffness, and Young's modulus which were calculated using equations 38 + 39.

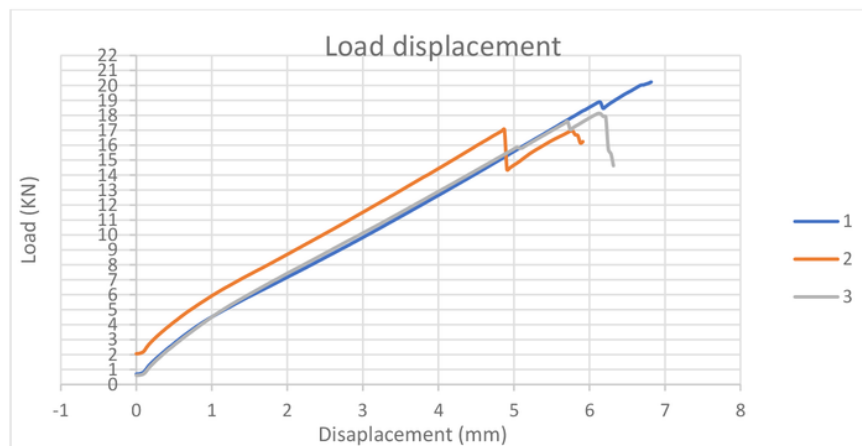


Figure 37 - 0° UD retest force displacement chart

0°										
Specimen	Load (kN)	Load (N)	K	Displacement (mm)	Stress (N/mm ²)	Strain	A (M ²)	E Measured (Gpa)	E (K)	E (Stress/strain)
1	20.23	20230.00	2750000	6.83	112388888.89	0.0198	0.000018	7.47	2.29E+10	5.68E+10
2	17.06	17060.00	2833333.333	5.91	94777777.78	0.016866667	0.000018	7.47	2.36E+10	5.62E+10
3	18.12	18120.00	2250000	6.32	100666666.67	0.018066667	0.000018	7.47	1.88E+10	5.57E+10
Average	18.47	18470.00	2611111.111	6.35	102611111.11	0.018244444	0.000018	7.47	2.18E+10	5.62E+10
% difference	18.58	18.58	25.93	14.51	17.49	17.39			25.93	1.87

Table 23 - 0° retest material properties calculated and measured

5.4.2 45°

Figure 38 shows the force-extension graph from the recorded data. Table 24 shows the recorded results from the Denison

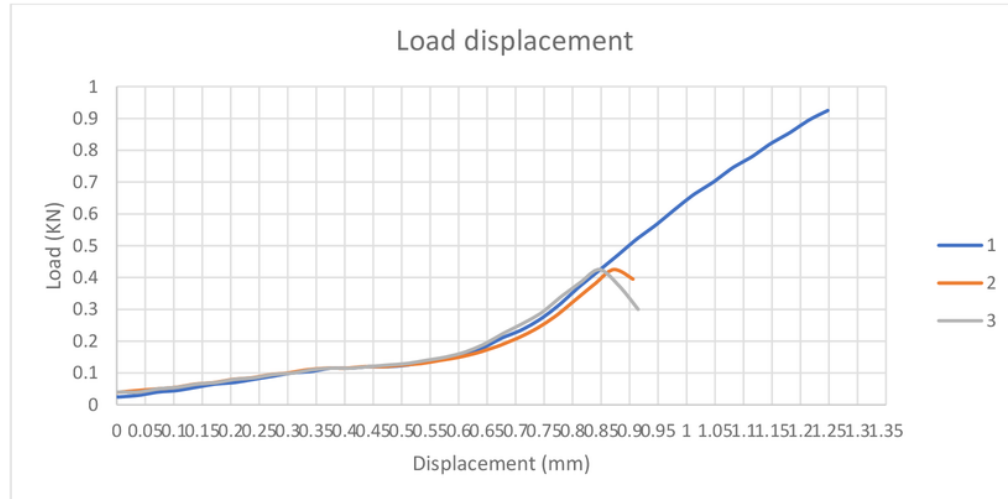


Figure 38 - 45° UD load displacement chart

45										
Specimen	Load (Kn)	Load (N)	K	L (M)	Stress (N/mm ²)	Strain	A (M ²)	E Measured	E (K)	E (Stress/strain)
1	0.925	925	1366666.667	0.15	51388888.89	0.003533333	0.000018	4.28	1.139E+10	14544025157
2	0.425	425	1319444.444	0.15	23611111.11	0.002533333	0.000018	1.83	1.1E+10	9320175439
3	0.425	425	1184210.526	0.15	23611111.11	0.0026	0.000018	2.06	9.868E+09	9081196581
Average	0.591666667	591.66667	1290107.212	0.15	32870370.37	0.002888889	0.000018	2.723333333	1.075E+10	10981799059
% difference	117.6470588	117.64706	15.40740741		117.6470588	39.47368421		133.8797814	15.407407	60.15538291

Table 24 - 45° UD material properties measured and calculated

5.4.3 90°

Table 25 shows the recorded results from the Denison. Please note, due to a manufacturing error only one sample was produced.

90										
Specimen	Load (Kn)	Load (N)	K	Displacement (mm)	Stress (N/mm ²)	Strain	A (M ²)	E Measured	E (K)	E (Stress/strain)
1	0.395	395	1192307.692	0.921	21944444.44	0.373763441	0.000018	1.67	0.00814995	5.87E+07

Table 25 -90° retest material properties measured and calculated

5.4.4 Easy composites 0°

Figure 39 shows the force-extension graph from the recorded data. Table 26 shows the recorded results from the Denison

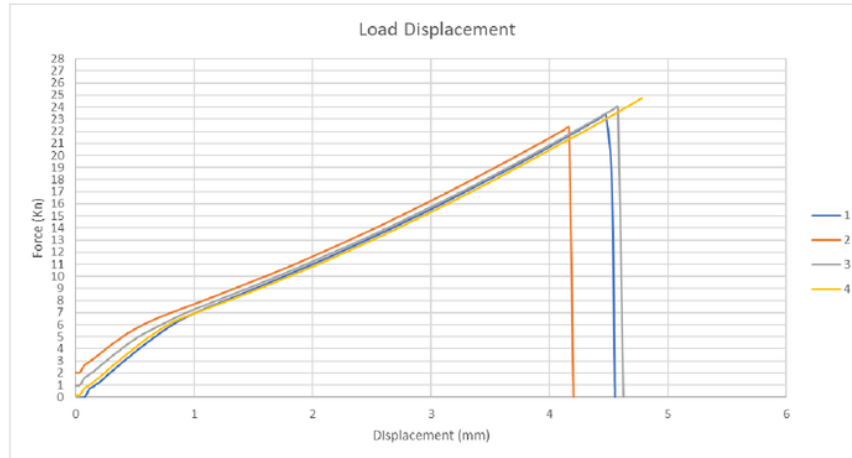


Figure 39 – Easy composites 0° UD load displacement chart

Specimen	Easy Composite 0									
	Load (Kn)	Load (N)	K	Displacement (mm)	Stress (N/mm ²)	Strain	A (M ²)	E Measured (Gpa)	E (K)	E (Stress/strain)
1	23.36	23360.00	4400000	4.55	9733333.33	0.0153125	0.000024	8.71	3.30E+10	6.36E+10
2	22.35	22350.00	4600000	4.21	93125000.00	0.0155	0.000024	8.41	3.45E+10	6.01E+10
3	24.05	24050.00	4666666.667	4.63	100208333.33	0.0159375	0.000024	8.67	3.50E+10	6.29E+10
4	24.75	24750.00	4333333.333	4.78	103125000.00	0.015625	0.000024	8.77	3.25E+10	6.60E+10
Average	23.63	23627.50	4500000	4.54	984479166.67	0.01559375	0.000024	8.64	3.38E+10	6.31E+10
% difference	10.74	10.74	7.69	12.01	0.11	4.08		4.280618312	7.692307692	9.852348993

Table 26 – Easy composites 0° UD material properties measured and calculated

5.5 DENISON CALIBRATION

From both sets of tensile tests, it is clear that the Denison was giving unrealistic values for displacement; as shown in Figure 40 the test rig was last calibrated in 2015, although the author was assured that it had been done at the start of the year.

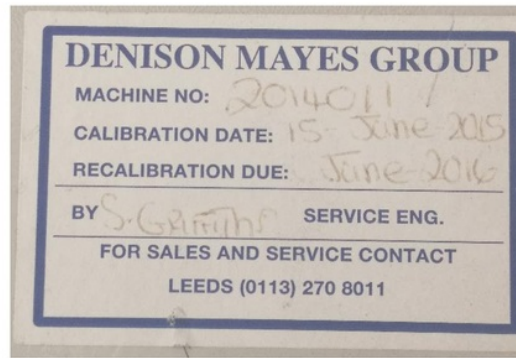


Figure 40 – Denison calibration sticker

Adaptors were manufactured so Hounsfield dog bone specimens could be used, a total of 16 tests were completed using 4 different material. The extension from the Denison test rig was reduced by the values shown in table 27, Young's modulus is then calculated as by the equations given in section 5 the values were adjusted to be as close as possible to material data from AZO Materials. As can be seen in the table over all of the tests there is only a 1% variance in the modification factor, but unfortunately when the extension is reduced by the same amount for carbon fibre test previously completed it returns values for Young's modulus between 300-400 Gpa.

Test	Material	Extension Reduction %	E Gpa (Calculated)	E (AZOM)
1	Brass	90	120	102-125
2	Aluminium	91	69.8	69
3	Aluminium	90.2	70	69
4	Aluminium	90.5	71	69
5	Aluminium	90.4	68	69
6	0.54% Carbon steel	90.4	197	200
7	0.54% Carbon steel	90.33	202	200
8	0.54% Carbon steel	90.3	204	200
9	0.54% Carbon steel	90.25	204	200
10	0.54% Carbon steel	90.31	202	200
11	0.54% Carbon steel	90.35	205	200
12	0.4% Carbon steel	90.42	199	200
13	0.4% Carbon steel	90.3	204	200
14	0.4% Carbon steel	90.2	194	200
15	0.4% Carbon steel	90.3	203	200
16	0.4% Carbon steel	90.34	203	200

Table 27 – Denison calibration with Hounsfield test pieces properties and adjustment value

5.6 GOM CORRELATE

GOM Correlate is a free digital image correlation and evaluation software for materials research and component testing (GOM, 2018). The carbon samples were sprayed with a yellow base coat, a mist of black paint is applied to create data points that can be picked up by a video camera, the tensile test is carried out as normal, but a web camera was set up to record. The video was uploaded to the GOM software which breaks the video down into frames, reference dimensions are input which allows the software to analyse the change in distance between the dots. Figure 41 shows a screenshot from the software, the data labels show the actual displacement of the test sample.

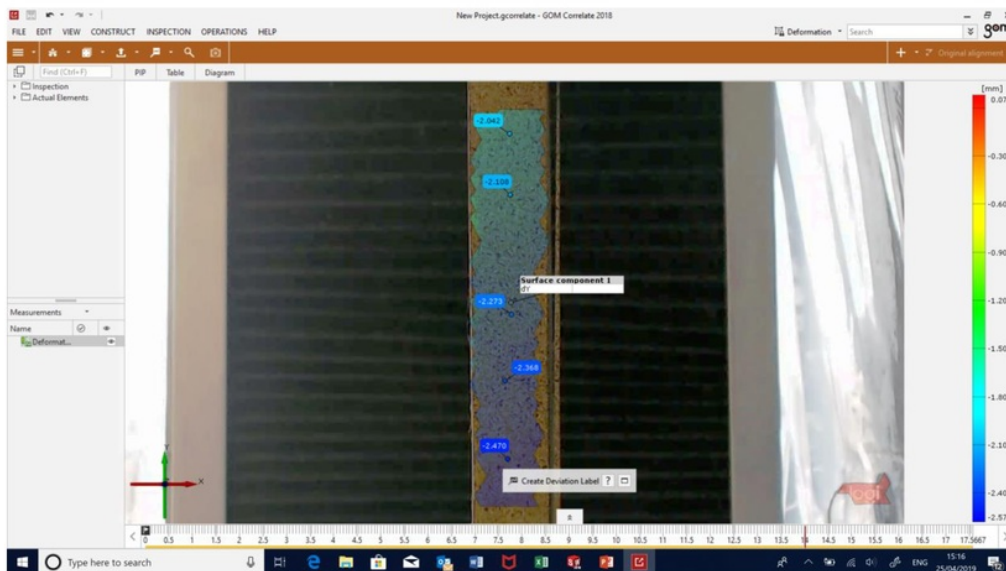


Figure 41 – GOM correlate screen grab showing actual displacements

Test	Displacement (mm)
1	2.47
2	2.62
3	2.547
4	2.518
Average	2.539

Table 28 – GOM correlate test results

The average displacement for the 0° fibres was 6.35mm for the same material tested during the GOM calibration; the average displacement of the carbon tests was divided by average displacement from the GOM testing $\frac{6.35}{2.539} = 2.5$. This value was used to adjust the maximum displacement measured by the Denison and the Youngs modulus was recalculated shown in tables 29-32.

0									
Specimen	Load (Kn)	Load (N)	F/x	L (M)	Stress (N/mm ²)	Strain	A (M ²)	E Measured	E Recalculated
1	20.23	20230	2750000	0.15	1123888889	0.0198	0.000018	7.47	1.44176E+11
2	17.06	17060	2833333.333	0.15	947777777.8	0.016866667	0.000018	7.47	1.42729E+11
3	18.12	18120	2250000	0.15	1006666667	0.018066667	0.000018	7.47	1.41528E+11
Average	18.47	18470	2611111.111	0.15	1026111111	0.018244444	0.000018	7.47	1.42811E+11
% difference	18.58	18.58	25.93		17.49	17.39			1.87

Table 29 - 0° carbon properties with adjustment factor

90								
Specimen	Load (Kn)	Load (N)	L (M)	Stress (N/mm	Strain	A	E Measured	E Recalculated
1	0.855	855	0.175	17100000	0.002866667	0.00005	127.43	1.49E+10
2	2.305	2305	0.175	46100000	0.002333333	0.00005	93.75	4.94E+10
3	1.485	1485	0.175	29700000	0.0026	0.00005	ERROR	2.86E+10
4	1.73	1730	0.175	34600000	0.002733333	0.00005	102.13	3.16E+10
5	2.535	2535	0.175	50700000	0.002933333	0.00005	ERROR	4.32E+10
6	1.06	1060	0.175	21200000	0.002606667	0.00005	12.54	2.03E+10
Average	1.661666667	1661.667	0.175	33233333.33	0.002678889	0.00005	55.975	3.13E+10
% Difference	169.59	169.59		196.49	17.14		916.19	231.21

Table 30- 90° carbon properties with adjustment factor

Easy Composite 0									
Specimen	Load (Kn)	Load (N)	F/x	L (M)	Stress (N/mm ²)	Strain	A (M ²)	E Measured	E Recalculated
1	23.36	23360	4400000	0.18	973333333.3	0.0153125	0.000024	8.71	1.61454E+11
2	22.35	22350	4600000	0.18	931250000	0.0155	0.000024	8.41	1.52605E+11
3	24.05	24050	4666666.667	0.18	1002083333	0.0159375	0.000024	8.67	1.59705E+11
4	24.75	24750	4333333.333	0.18	1031250000	0.015625	0.000024	8.77	1.6764E+11
Average	23.6275	23627.5	4500000	0.18	984479166.7	0.01559375	0.000024	8.64	1.60351E+11
% difference	10.74	10.74	7.69		0.11	4.08		4.28	5.80

Table 31- Easy composites 0° carbon properties with adjustment factor

45									
Specimen	Load (Kn)	Load (N)	K	L (M)	Stress (N/mm ²)	Strain	A (M ²)	E Measured	E Recalculated
1	0.925	925	1366666.667	0.15	51388888.89	0.003533333	0.000018	4.28	3.69E+10
2	0.425	425	1319444.444	0.15	23611111.11	0.002533333	0.000018	1.83	2.80E+10
3	0.425	425	1184210.526	0.15	23611111.11	0.0026	0.000018	2.06	2.72E+10
Average	0.591666667	591.6666667	1290107.212	0.15	32870370.37	0.002888889	0.000018	2.723333333	3.07E+10
% difference	117.65	117.65	15.41		117.65	39.47	0.00	133.88	35.60

Table 32- 45° carbon properties with adjustment factor

With the adjustment factor applied, the calculated Young's modulus for both the 0° fibres tested are within the range from the data sheet supplied by McLaren of between 130-170 GPA. The Young's modulus is slightly lower in the coupons manufactured in house, but the results are very consistent with a variance of 1.87%. Unfortunately, due to the manufacturing error previously only 1 90° was retested therefore the original data was manipulated using the same error factor but as can be seen in table 30 there is a variance of 231% between the highest and lowest results and the highest Young's modulus with the highest being 5 times larger than the value on the data sheet.

6 CARBON CONNECTING ROD

The two plastic reinforce fibre connecting rods manufacturing was considered when designing the connecting rod; the connecting rod in (Guadette, 1985) was manufactured by making a plate of the desired thickness, and then the connecting rod was machined utilising a CNC. The hybrid rod, described in (Kuch & Kümmerlen, 1991) was manufactured in three sections. First, the tension loop was manufactured using a specially designed tape winding device and was cured in a special jig. The compression shank was manufactured as a plate and machined to the required dimensions. The titanium small and big end bearings were eroded tempered and finished; finally, all three 3 sections of the connecting rod are bonded together. With the aim of designing a carbon connecting rod, it was decided that the hybrid connecting rod would require more specialised tools and jigs that would increase the cost of manufacture and although Guenette's manufacturing process is simpler the author decided to go down another root to reduce costs and minimise waste.

6.1 DESIGN

Two main methods were considered for the design of the connecting rod, the first is known as forged carbon which was developed by Lamborghini and Callaway. Unlike traditional carbon components that are manufactured by laying up sheets of carbon, forged carbon utilises a mixture of short strand fibres (up to 200,000 per square cm) which is impregnated with resin, heat and pressure is then applied to form the part into the required geometry (Lamborghini, 2019) (Carbon Fiber Gear, 2019). The second method considered is bladder moulding, which is generally used for components with complex geometry. The manufacturing process starts with pre-preg carbon sheets, which are placed in one half of a mould. The inflatable bladder is placed in the middle then the second half of the mould is placed on top, once the mould is closed pressured is applied to the bladder causing it to expand pushing the pre-preg against the inside of the mould cavity; finally heat is applied to cure the composite (Rockwestcomposites, 2019).

Utilising the original dimensions of the Ford GDI connecting rod, i.e. length, small end, and big end, a connecting rod was designed to be manufactured utilising bladder moulding. Bladder moulding was chosen over forged carbon due to there being no orientation to the fibres, and the appropriate FEA software was unavailable; therefore, no analysis of the design could be completed. The design requires metal inserts to be bonding in for the small and big end bearings for better wear characteristics, meatal inserts are also required with a tapped inner face so the cap can be fastened using bolts to the main rod section.

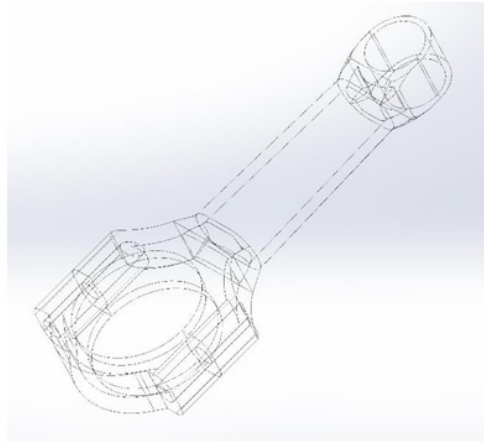


Figure 42 – Carbon rod design

6.2 ANALYSIS

The initial analysis was completed on SolidWorks analysing the connecting rod using 4340 steel as a comparison to the Excel hand calculations previously performed for the standard GDI connecting rod. The original GDI connecting rod mass was 431g, using Solidworks mass properties the proposed carbon rod when analysed in 4340 steel weighed 481g which is an 11% increase. Figure 43 shows Von Mises stress, yield point is $7.1 \times 10^8 \text{ NM}^2$ and maximum stress $4.396 \times 10^8 \text{ NM}^2$ which is 61% of the yield stress.

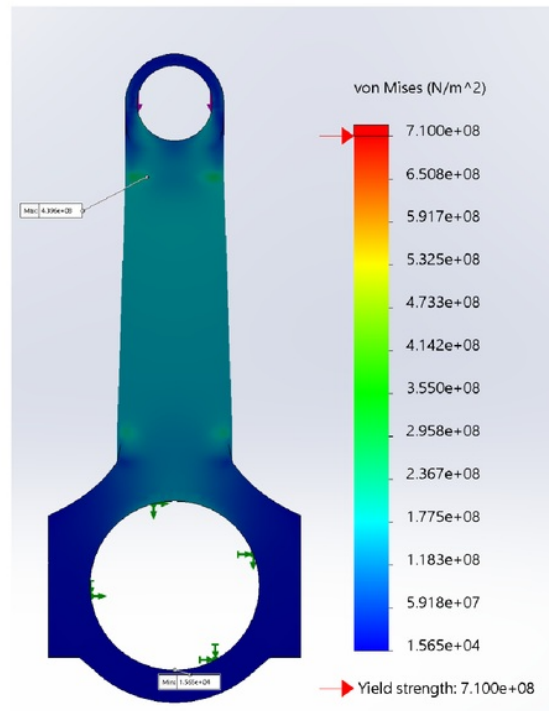


Figure 43 – Carbon rod Von Misses stress Solidworks

Figure 44 shows the displacement send from the gas force; it can be seen there is a maximum displacement of 0.107mm which is 36.7% lower than the original GDI rod displacement from the Excel calculations.

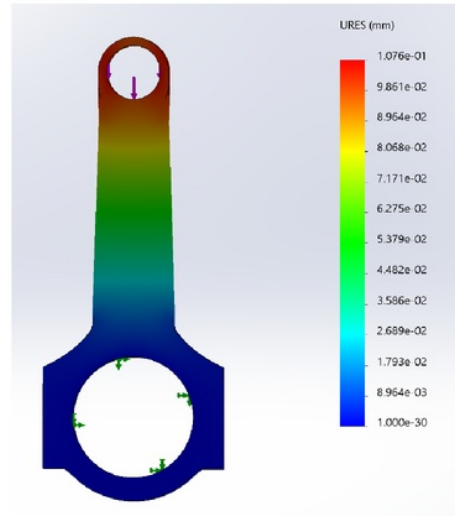


Figure 44 – Carbon rod compressive displacement Solidworks

Figure 45 shows the static FOS, the lowest value being 2.4 which is 18% lower than than the hand calculated value for the standard rod.

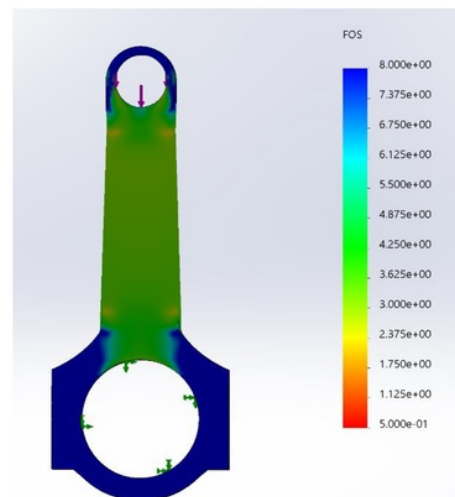


Figure 45 – Carbon rod safety factor Solidworks

The final FEA was conducted using Design Studio as it is not possible to input composite material properties into Solidworks. An analysis was performed using the material properties provided by McLaren. A quasi-isotropic layup was used and consisted of 12 0.125mm UD laminates orientated as follows; 0°, 90°, +45°, -45°, 0°, 0°, 0°, 0°, 0°, -45°, +45°, 90°, 0°.

Figure 46 shows displacement due to the gas force; it can be seen that there is a maximum displacement of 0.29mm seen at the top of the small end and 0.14mm seen in the shank itself; this compares to the 0.169mm displacement calculated in Excel and 0.122mm from FEmap for the standard connecting rod.

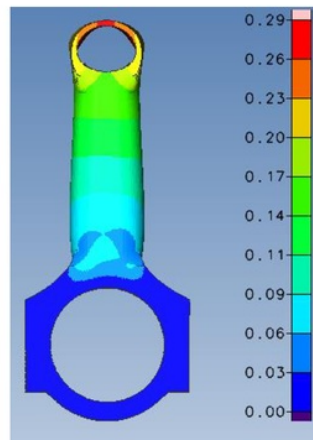


Figure 46 – Carbon rod compressive displacement Design studio

Figure 47 shows the Von Mises stress, with a maximum stress of $6 \times 10^3 \text{ NM}^2$

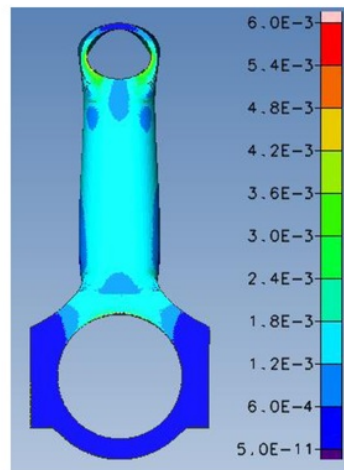


Figure 47 – Carbon rod Von Mises stress Design studio

7 CONCLUSION

Due to tougher emission regulations engines have been downsized engines with increased load. Friction is an area of focus for emission reduction, one contributor to friction in the PCS is the connecting rod (Lapp, et al., 2010) details that a reduction in reciprocating components mass allows the combustion forces to be more efficiently transferred due to a reduction in force transmitted to the cylinder walls. Traditionally the highly loaded cranktrain components are overdesigned, the connecting rod is submitted to millions of cycles over its life cycle to the order of 10^8 , and therefore the endurance limit is the primary design criterion as the connecting rod is designed for infinite life. In each cycle, the connecting rod experiences Gas and inertia forces, axial and bending stresses.

Polimotor successfully produced two versions of an engine which utilise PRF components including timing gear, intakes valves, valve spring retainers, bucket tappet, rocker arm, piston, piston rings, gudgeon pin, connecting rods, and roller lifter. In the first version of the engine based on a 2.3L Pinto engine reducing mass by 119 kg, the 2nd version was a 2.0L based on a Cosworth engine, and the mass was reduced from 143Kg to 76Kg saving 67Kg. The 2nd version was successfully raced, for two seasons, in a Lola T616.

Daimler Benz AG successfully designed, manufactured, and tested a hybrid connecting manufactured in three sections. A carbon fibre tension hoop and compressive shank; with the crankshaft bearing and cap sections manufactured from titanium which reduced the mass by 30% from the original. The connecting rod was put through a series of different tensile and compressive loads during testing, the rod successfully withstood 5×10^7 cycles at 120°C with alternating tensile and compressive loads of 14Kn and 40Kn respectively but was deemed too costly to put into production at that time; although an unsplit version could be utilised in high-efficiency engines successfully.

The compressive strength of structures was investigated by manufacturing 4 3D printed connected rods with the centre section of the shank braced in different ways. The first rod had no bracing the other rods were braced at 20°, 45°, and 90° to the vertical. The plain rod failed at 872N, while the 90° and 20° failed at 1735 N and 1765 N respectively while the 45° rod failed at 1935 N which is 11.53% higher than the 90° and 9% higher than the 20° braced rod.

From cylinder pressure data calculations were conducted to reverse engineer two different connecting rods. A standard steel I beam GDI connecting rod and a Pankl titanium H beam connecting rod. The fatigue safety factors were calculated at four different sections of both connecting rods, the static safety factors, compressive, and tensile displacements were calculated for each connecting rod. The steel rod having a minimum safety factor of 2.65 and the titanium 3.55, the GDI displaced 0.189mm due to compression and 0.026mm from tension, whereas the Pankl displaced 0.457mm due to the compressive force and 0.064mm from tension. FEA was conducted and compared to these values.

As LAPP (YEAR) stated that friction could be reduced by reducing the connecting rod mass, calculations were conducted for friction and the effect on FSF reducing the mass in 10% increments from 100% to 50%. For every 10% reduction in mass, there is a 0.52% reduction in friction. Reducing the mass on the connecting rod would also allow Ymass to be reduced in the other PCS components it is attached to ,i.e. piston, crankshaft etc. which would reduce friction further. As the connecting rod itself is a small part in the reciprocating mass the reduction in mass had little effect on the inertia force; therefore, there was little or no effect on the FSF.

The tensile strength of UD carbon fibre was investigated utilising a Denison material test rig; unfortunately, the rig was out of calibration causing unrealistic values for Young's modulus. Testing was conducted with metal test sections fitted with strain gauges see appendix 9.4 in an attempt to calibrate the Denison but this was inconclusive, adapters were made so that the Hounsfield dog bone test specimens could be analysed. 16 tests were carried out utilising a range of materials and the extension given by the Denison was reduced by a factor and Youngs modulus calculated this factor varied by 1% for all tests conducted but when applied to carbon fibre gave values 3 times higher than data provided by McLaren. GOM correlate is an analysis software which was used to measure the real extension values using data points and a video camera, from this the extension was reduced by dividing it by 2.5 and used to calculate strain. This resulted in values for the Youngs modulus that fell into the range supplied by McLaren.

A bladder mould connecting rod was designed utilising Solidworks, an analysis was conducted first in Solidworks as a direct comparison to the standard GDI rod utilising 4340 steel material properties. Although 11% heavier than the standard rod the FOS was increased by 2% in the critical section but for the majority of the shank was increased from 2.4 to 4. Displacement reduced by 58% from 0.257mm to 0.107mm.

To conclude from the literature review, it was ascertained that the connecting rod mass can be optimised utilising carbon fibre, although special resin is required to withstand the temperatures inside the ICE. The GDI and Pankl connecting rods were reverse engineered using cylinder pressure data from first principles and static, and fatigue safety factors were obtained, these were then compared to values ascertained in FEA. The mechanical properties of UD carbon fibre were obtained through tensile testing on a Denison rig. Although the Denison was out of calibration after several different methods of calibration, it was possible to obtain a displacement calibration factor utilising data points, a video camera, and image analysis software which when compared to data supplied by McLaren and returned similar values for Young's modulus in the fibre and matrix directions. A connecting rod suitable for manufacture utilising carbon fibre was produced in Solidworks and analysed in 4340 steel as a direct comparison to the standard GDI rod, basic carbon analysis was conducted in Design Studio but due to unforeseen delays in acquiring material properties and delays in manufacturing testing coupons the analysis was only basic.

Further work includes further FEA in Design studio to ascertain the minimum possible thickness of the carbon fibre while keeping appropriate safety factors. Manufacture of the moulds and bladder, manufacture of a carbon rod to perform mechanical testing. As (Guadette, 1985) stated that in the mid 80's it was too expensive manufacture of a production engine, the costs of the equipment, moulds, materials, and labour costs will be investigated to ascertain whether a carbon connecting rod is viable for production engines or whether it is more suitable for high-performance racing engines.

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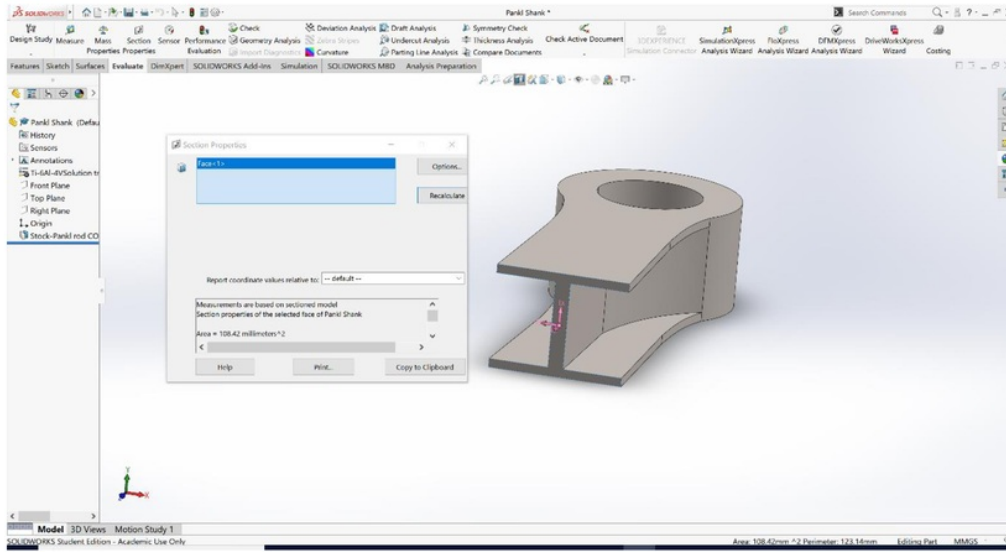
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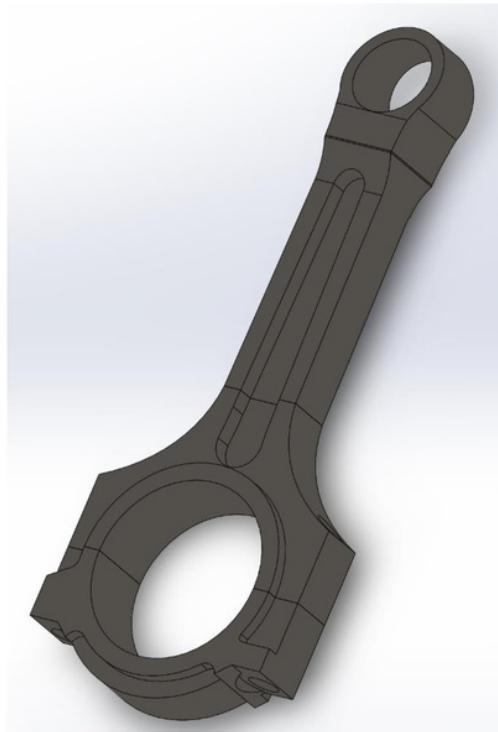
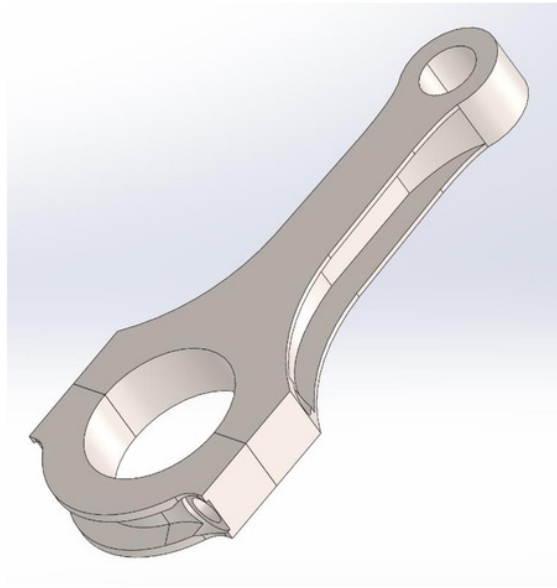
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9 APPENDIX

9.1 CAD SCREENSHOTS





9.2 DENISON MACHINE SPECIFICATIONS

Denison Machine specifications – taken from the Denison equipment directory

This model of Electromechanical Universal Testing Machine is designed and manufactured according to National Standard & ASTM E4 standard. It is computer-controlled precision Testing Machine, suitable for wide range of material for tension, compression, Bending and shearing test. It has high stability as well as high precision, equipped with PC system & printer for graph, test result display, printing & data processing. Complete with modulus for metal, spring, textile, rubber, plastic and other material testing & creep test. It is widely used in many fields such as industry factories, mineral enterprise and high schools.

Machine specifications:

Max. Load: 100kN

Load accuracy: $\pm 0.5\%$

Deformation resolution: $\pm 0.5\%$

Crosshead displacement accuracy: 0.01mm

Crosshead speed: 0.005-500mm/min.

Twin ball screw driven with close-loop servo and motor control.

Stiffness: 164kN/mm

Height: 2080mm, Width: 1020mm, Depth: 560mm

Crosshead travel: 1260mm

Weight: 1000kG

Power: 220V AC, 50Hz/60Hz

Emergency stop control

Automatic overload protection

General uses:

Sample sizes / required sample preparations:

Tensile grips for round specimen

Φ4~Φ9mm 1 set

Φ9~Φ14mm 1 set

Φ14~Φ20mm 1 set

Tensile grips for flat specimen

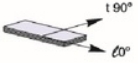
0-7mm 1set

7-14mm 1set 14-21mm 1 set

9.3 McLAREN PREPREG TECHNOLOGY DATA SHEET

PREPREG TECHNOLOGY

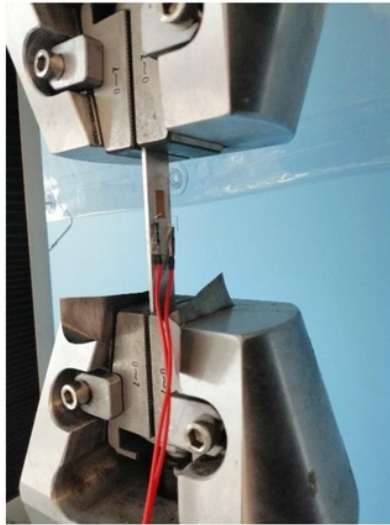
TYPICAL MECHANICAL VALUES ON EPOXY PREPREG LAMINATES

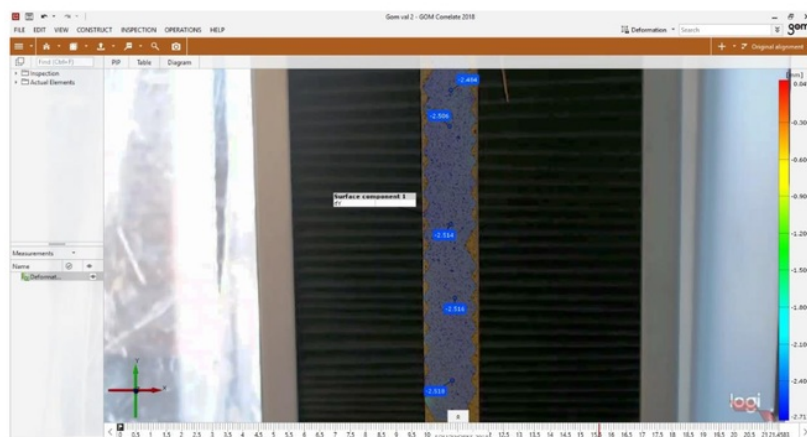
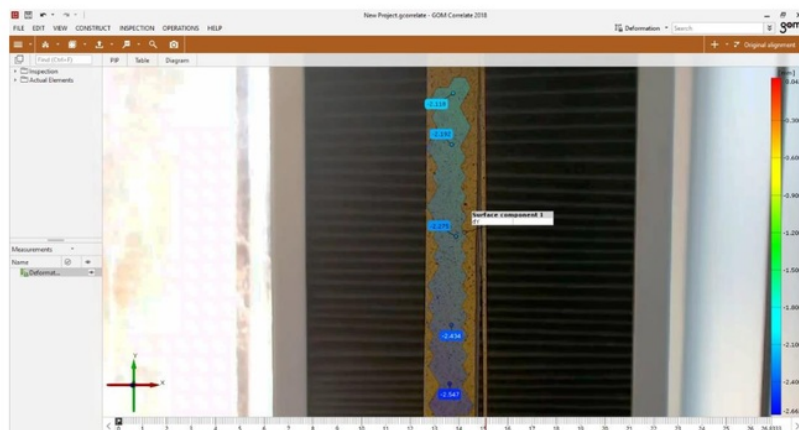
			UNITS	FIBRES							
				E-GLASS		ARAMID		HIGH STRENGTH CARBON		INTERMEDIATE MODULUS CARBON	
				UD	Fabric	UD	Fabric	UD	Fabric	UD	Fabric
 Volume content of fibres : ~ 60 % (Carbon) ~ 50 % (E-glass - Aramid)		$\sigma \ell$	MPa	1100	600	1100	500	2000	800	2400	900
		σt	MPa	35	550	35	450	80	750	80	850
		$E \ell$	GPa	43	20	60	30	130	70	170	90
		$E t$	GPa	8	19	8	30	9	65	9	90
		Poisson's ratio $\nu \ell t$		0.28	0.13	0.34	0.2	0.25	0.05	0.27	0.05
	Compression	$\sigma \ell$	MPa	900	550	250	150	1300	700	1600	800
		σt	MPa	150	500	150	150	250	650	250	750
		$E \ell$	GPa	42	17	75	31	115	60	150	80
		$E t$	GPa	10	16	5.5	30	10	55	11	75
	Flexural	$\sigma \ell$	MPa	1200	700	550	400	1800	1000	1400	1200
		$E \ell$	GPa	42	20	40	25	120	65	140	75
	In-plane shear	$\sigma \ell t$	MPa	60	55	45	40	95	80	95	80
		$G \ell t$	GPa	4	4.2	2.1	4	4.4	5.5	4.4	5
	Interlaminar shear	σ	MPa	75	50	60	50	80	70	80	70

TYPICAL THERMAL PROPERTIES OF PREPREG LAMINATES

	Units	Glass		Aramid		High strength carbon	
		UD	Fabric	UD	Fabric	UD	Fabric
Coefficient of expansion	$10^{-6} K^{-1}$	11	14	- 0.4	- 5.2	0.3 - 0.7	2 - 3
Thermal conductivity	$Wm^{-1} K^{-1}$	0.4	0.16 - 0.33	0.4	0.21	1	0.86 - 1.44

9.4 CALIBRATION PHOTOS





9.5 EXCELL CALCULATIONS

All calculations were conducted at each RPM and connecting rod mass from 100% to 50% in 10% increments.

The screenshot displays an Excel spreadsheet titled "14000 rpm calculations 100% weight - Real". The spreadsheet is organized into several columns, likely representing different engine parameters and configurations. The data is presented in a tabular format, with rows corresponding to specific engine settings or calculations. The bottom of the image shows a Windows taskbar with the date 03/06/2019 and various application icons.

1.6GdI rod calculations 100% weight - Excel

File Home Insert Page Layout Formulas Data Review View Help Tell me what you want to do

Paste Cut Copy Format Painter Clipboard Font Alignment Wrap Text Merge & Center Number Conditional Formatting Styles

B4 =("Data sheet"!'K22"/"Data sheet"!'\$U\$37)

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1														
2	RPM	Sigma min	sigma max	Stress Range	Mean Stress	allowable range	Fatigue safety factor	RPM	Sigma min	sigma max	Stress Range	Mean Stress	allowable range	Fatigue safety factor
3						GDI Section A							GDI Section C	
4	1500	-51.95	0.61	52.56	26.2799291	644.8480639	12.31	1000	-127.257361	1.84055675	129.103928	64.5519642	612.4032322	4.74
5	1000	-64.98	1.37	66.35	33.174685	640.6428069	9.66	1500	-159.018349	4.154777	163.173126	81.588665	597.0720933	3.66
6	1700	-74.03	1.76	75.78	37.8925317	636.3988835	8.40	1700	-181.084464	5.33658023	186.421044	93.2105221	586.6105301	3.15
7	2000	-80.69	2.43	83.13	41.5626894	633.0935795	7.62	2000	-198.789454	7.38627021	206.175724	103.087862	577.720924	2.80
8	2500	-84.26	3.80	88.07	44.0333867	630.8699529	7.16	2500	-205.554095	11.5410472	217.095142	108.547571	572.6071861	2.64
9	3000	-86.43	5.48	91.90	45.9050569	629.1444968	6.85	3000	-210.355149	16.619108	226.974257	113.488128	564.3615945	2.50
10	3500	-87.83	7.85	95.28	47.6040997	627.6231633	6.59	3500	-213.175637	22.6204525	235.786908	117.889045	561.971590	2.39
11	4000	-89.34	9.73	98.07	49.03492	626.308572	6.38	4000	-215.087536	29.5450089	243.230217	121.610309	561.0452323	2.31
12	4500	-89.65	12.32	97.97	48.9859381	626.4126357	6.38	4500	-208.418535	37.320903	243.803828	121.9014	560.788276	2.30
13	5000	-86.52	15.21	101.73	50.663689	624.722725	6.14	5000	-207.606871	46.1641888	253.777108	126.85855	556.3030232	2.19
14	5500	-89.75	18.40	108.15	54.0751571	621.8323587	5.75	5500	-214.364552	55.8586685	270.205221	133.10281	548.9076506	2.03
15	5700	-90.92	19.77	110.89	55.3449073	620.6895634	5.61	5700	-216.733908	59.9949798	276.22888	138.364444	545.9720004	1.97
16	6000	-91.13	19.77	110.89	55.4467195	620.5979525	5.60	6000	-217.166335	59.9949798	277.163135	138.360055	545.7774083	1.97
17														
18														
19	RPM	Sigma min	sigma max	Stress Range	Mean Stress	allowable range	Fatigue safety factor	RPM	Sigma min	sigma max	Stress Range	Mean Stress	allowable range	Fatigue safety factor
20						GDI Section B							GDI Section D	
21	1000	-123.714391	1.66154759	125.375939	62.6879095	614.0808274	4.90	1000	-86.2566595	1.3372435	87.593903	43.7969515	631.0827436	7.20
22	1500	-154.494276	3.73848208	158.23276	79.11638	599.295258	3.79	1500	-107.057249	3.0087978	110.066047	55.3302325	620.7002789	5.61
23	1700	-175.887898	4.80187254	180.68977	90.3448852	589.1890033	3.28	1700	-122.537434	3.88463372	126.42008	63.2010341	613.6190693	4.85
24	2000	-193.100103	6.64619037	199.746294	99.8381468	581.611679	2.91	2000	-133.302079	5.34897401	138.651053	69.3256266	608.1070621	4.39
25	2500	-199.265868	10.3546725	209.676361	104.67318	576.1447377	2							

