University of Southern Queensland

Faculty of Health, Engineering and Sciences

OPTICAL ACCESS ENGINE DEVELOPMENT

A dissertation submitted by:
Kevin Dray

In fulfilment of the requirements of
Bachelor of Mechanical Engineering

Abstract

A partial design for an optical access engine was acquired by the University of Southern Queensland from Oxford University in the UK. It is the desire of USQ or specifically certain faculty members to include an optical access engine in their engines laboratory.

The purpose of this research project is to take the existing partial design from the early 1990's of Professor Richard Stone's (Oxford University) and develop it into a useable design of which the University can invest in. for their laboratory.

Due to time constraints it was not possible for the entire engine to be designed and detail drawings produced so the engine was designed from the bottom end up to the top of the optical bore which on a normal engine is the top of the cylinder block.

Included in this dissertation is a report covering the need for engine research and the place of optical access engines in research. A literature review of various optical access engines and discussion of their types and uses. Finally a report covering the methodology of the design and the results of design calculations with a focus on balancing and component design. Detail drawings suitable for workshop manufacture and calculations are included as part of this dissertation as are recommendations for future work.

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ENG4111/ENG4112 Research Project

Limitations of Use

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Certification of Dissertation

I certify that the ideas, designs and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

K.Dray (0011120331)

(30/10/2014)

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GLOSSARY OF TERMS

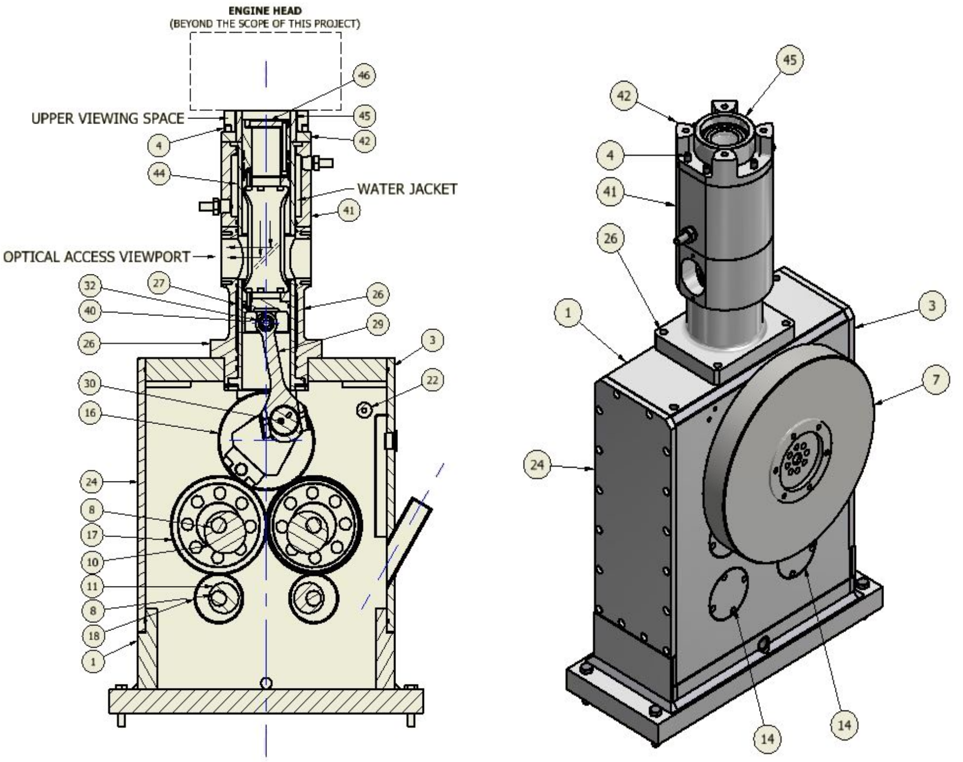


Figure 1 – Main assembly section view & isometric view.

ITEM	DESCRIPTION	QTY
1	MAIN CRANKCASE	1
2	DRIVE ATTACHMENT	1
3	SIDE COVER (OIL FILL)	1
4	TIE ROD	6
5	FLYWHEEL SIDE MAIN BEARING HOUSING	1
6	ENCODER SIDE MAIN BEARING HOUSING	1
7	FLYWHEEL	1
8	COUNTER BALANCE SHAFT	4
9	COUNTER BALANCE SHAFT WASHER	8
10	PRIMARY BALANCE WEIGHT	2
11	SECONDARY BALANCE WEIGHT	2
12	COUNTER BALANCE SHAFT SPACER LONG	4
13	COUNTER BALANCE SHAFT SPACER SHORT	4

ITEM	DESCRIPTION	QTY
15	PRIMARY COUNTER BALANCE SHAFT GEAR	1
16	CRANKSHAFT DRIVE PINION	1
17	PRIMARY COUNTER BALANCE SHAFT GEAR	1
18	SECONDARY COUNTER BALANCE SHAFT GEAR	1
19	BEARING CAP (PRIMARY, FLYWHEEL SIDE)	2
20	6040DU - LEAD/PTFE BEARING (MODIFIED)	2
21	THRUST COLLAR	1
22	OIL SUPPLY FITTING	1
23	ENCODER ATTACHMENT	1
24	CRANKCASE SIDE COVER	1
25	SECONDARY COUNTER BALANCE SHAFT GEAR	1
26	LOWER BARREL	1
27	LOWER CYLINDER SLEEVE	1
28	BIG END LOWER HALF	1
29	CONROD	1
30	ENGINE CRANKSHAFT	1
31	LITTLE END BEARING	1
32	PISTON EXTENSION	1
33	PISTON HEAD	1
34	OPTICAL WINDOW COLLAR	1
35	LOWER PISTON	1
36	CRANK BALANCE WEIGHT	2
37	SLIDING BUSH	2
38	LOW FRICTION PISTON RINGS	2
39	WRIST PIN	1
40	UPPER BARREL	1
41	OPTICAL ACCESS COLLAR	1
42	MAIN ENGINE ASSEMBLY	4
43	MAIN ENGINE ASSEMBLY	1
44	OPTICAL ACCESS SLEEVE	1
45	OPTICAL ACCESS WINDOW	1
46	WC60DU - LEAD/PTFE THRUST BEARING (MODIFIED)	2
47	4x12 MACHINE DOWEL	4
48	5 x 20 MACHINE DOWEL	4
49	12 x 20 MACHINE DOWEL	2
50	1/4" BSPT PRESSURE PLUG	2
51	2.5 x 12 MACHINE DOWEL	2
52	4 x 20 MACHINE DOWEL	9
53	COUNTER BALANCE SHAFT BEARING	4
54	COUNTER BALANCE SHAFT BEARING	4

Bowditch Piston Assembly

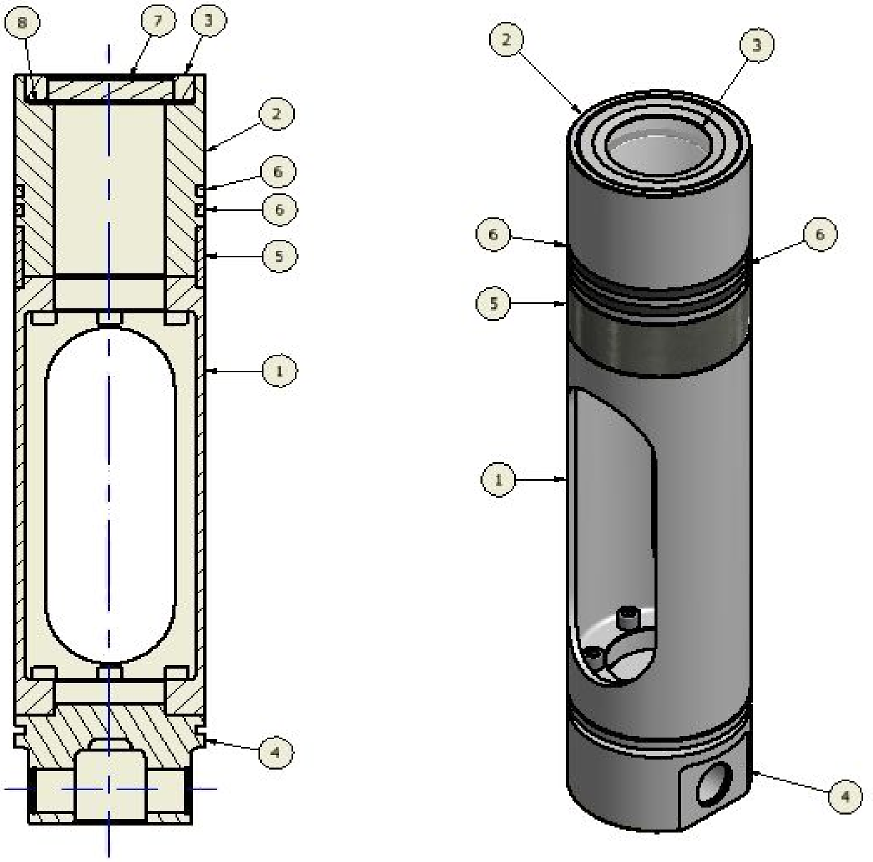


Figure 2 –Bowditch piston assembly section view & isometric view (Design revision A) Final design may differ slightly.

ITEM	DESCRIPTION	QTY
1	PISTON EXTENSION	1
2	PISTON HEAD	1
3	OPTICAL WINDOW COLLAR	1
4	LOWER PISTON	1
5	SLIDING BUSH	2
6	LOW FRICTION PISTON RINGS	2
7	OPTICAL ACCESS WINDOW	1
8	4 x 20 MACHINE DOWEL	4

Crankshaft Assembly

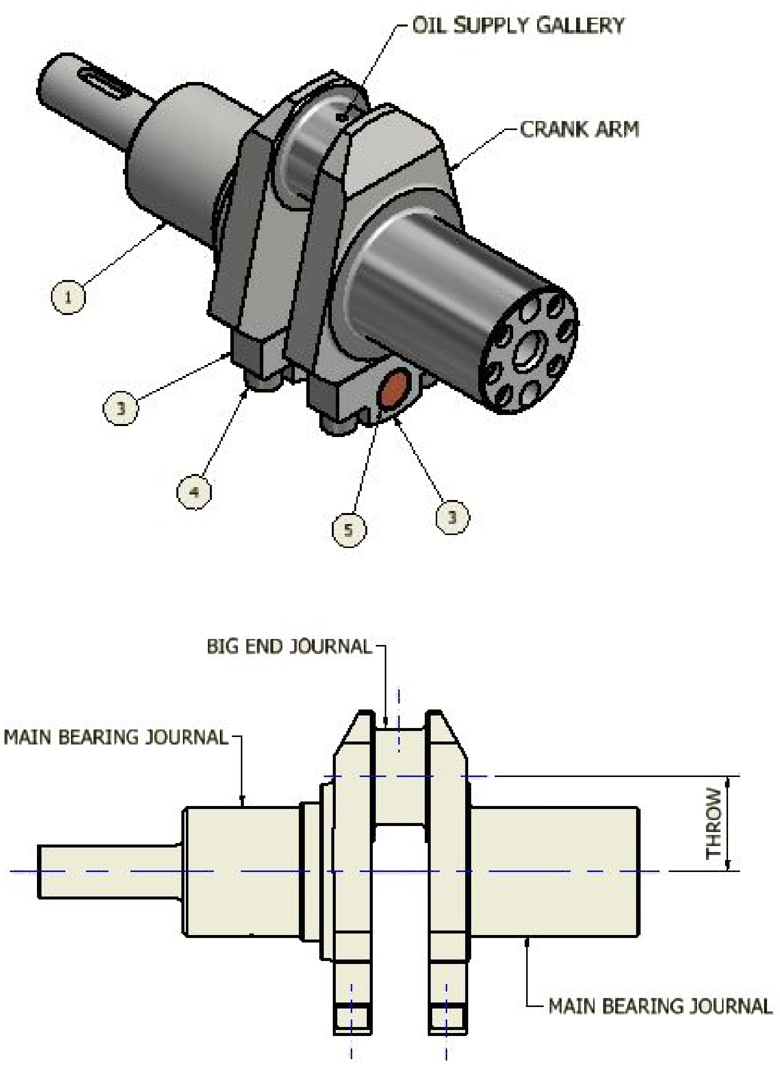


Figure 3 – Crankshaft assembly side view & isometric view. (Design revision A) Final design may differ slightly.

ITEM	DESCRIPTION	QTY
1	ENGINE CRANKSHAFT	1
2	4 x 20 MACHINE DOWEL	4
3	CRANK BOB-WEIGHT	2
4	BROACHED SOCKET HEAD CAP SCREW - METRIC -	4
5	TUNGSTEN INSERT	2

General Terms

No.	Term	Description
1	ATDC	Crank position After Top Dead Centre
2	BDC	Bottom Dead Centre - Lowest position of piston
3	BTDC	Crank position Before Top Dead Centre
4	Expansion Phase	Portion of the power stroke after combustion
5	FEA	Finite Element Analysis - Computer Stress Analysis
6	ICE	Internal Combustion Engine
7	OAE	Optical Access Engine
8	OHV	Overhead Valves - Engine valve configuration
9	TDC	Top Dead Centre - Highest position of piston
10	Compression Phase	Portion of the compression stroke before ignition
11	Counter Balance Shaft	Separate rotating shaft to counter reciprocating inertia forces
12	HCCI	Homogeneous Charge Compression Ignition
13	Optical Bore	Upper cylinder sleeve made from transparent quartz
14	Journal	Shaft component of a bearing arrangement
15	Oil Bore	Hole in the crankpin to which oil feeds the big end bearing
16	DOHC	Double Overhead Cam

Mathematical Nomenclature

The nomenclature listed below covers that used in this dissertation body only and does not extend to the detailed calculations contained in the appendix. Whilst all efforts have been made to maintain a consistency it was not possible to use the exact same nomenclature in the design software.

Term	Description	Units
m_{rm}	Reciprocating mass	kg
r	Crank throw	m
x	Piston position	m
$\dot{x}$	Piston velocity	m/s
$\ddot{x}$	Piston accelerations	m/s^2
ω	Angular Acceleration	radians/s
θ	Crank Angle	radians
Fy_{rm}	Vertical reciprocating force	N
m_1	Upper conrod equivalent mass	kg
$m_{(conrod)}$	Conrod mass	kg

Term	Description	Units
m_2	Lower conrod equivalent mass	kg
m_p	Piston mass	kg
r_1	Upper conrod mass radius	m
r_2	Lower conrod mass radius	m
Fy_{rp}	First order reciprocating forces	N
Fy_{rs}	Second order reciprocating forces	N
Fx_{bm}	Horizontal forces due to bob-weight	N
m_b	Effective bob-weight mass	kg
r_b	Bob-weight mass radius	kg
m_{be}	Crankshaft & lower conrod mass (excl. bob-weight)	kg
Ts	Shaking torque	N.m.
m_{pcb}	Primary balance weight	kg
m_{scb}	Secondary Balance weight	kg
r_{pcb}	Primary balance weight radius	m
r_{scb}	Secondary balance weight radius	m
ϕs	Stoichiometric ratio	
ϕ_e	Equivalence ratio	
γ	Poly=tropic constant	
a_x	Weibe function constant	
m_x	Weibe function constant	
C_{fuel}	Burnt fuel constant	
Xb	Mass burn fraction	
T_f	Final gas temperature (combustion)	K
T_o	Initial charge temperature	K
M_{fuel}	Mass of fuel	kg
Q_{fuel}	Heating value of fuel	kj/kg
Cv_{air}	Specific heat value of air (constant volume)	kj/kg.K
P_f	Final cylinder pressure	Mpa
P_o	Initial cylinder pressure	MPa
V_o	Initial cylinder volume	M^3
T_o	Initial gas temperature	M^3
V_f	Final cylinder volume	M^3
$F_{g\theta}$	Piston gas force with respect to crank angle	MPa
P_θ	Cylinder pressure with respect to crank angle	MPa
D_p	Piston diameter	M
$\emptyset$	Conrod angle	Radians
$F_{c\theta}$	Conrod force with respect to crank angle	N
$Q_{rf\theta}$	Axial force in the web	N
φ	Angle between conrod and crankarm	Radians
$M_{bo\theta}$	Moment in the crankpin at the oil bore	N.m.
ψ	Angle of the oil bore to the crankpin tangent	Radians

Term	Description	Units
σ_{bfn}	Alternating bending stress in the web	MPa
$M_{brf\theta}$	Bending moment in the web	N.m.
σ_{qfn}	Alternating compressive stress in the web	MPa
σ_{bon}	Alternating bending stress at the oil bore	MPa
$M_{t\theta}$	Axial torque in the main journal or crankpin	N.m.
τ_n	Alternating torsional stress	MPa
W_p	Torsion modulus of the main journal	mm^3
α_b	Crankpin fillet bending stress factor	
α_t	Crankpin fillet torsion stress factor	
β_b	Journal fillet bending stress factor	
β_t	Journal fillet torsion stress factor	
β_q	Journal fillet radial compression stress factor	
γ_b	Crankpin oil bore bending stress factor	
γ_t	Crankpin oil torsion stress factor	
σ_{bh}	Factored alternating bending stress in crankpin fillets	MPa
σ_{bg}	Factored alternating bending stress in crankpin oil bore	MPa
σ_{bo}	Factored alternating bending stress in main bearing fillets	MPa
τ_h	Alternating torsional stress in crankpin fillets	MPa
τ_g	Alternating torsional stress in the main bearing fillets	MPa
σ_{to}	Alternating torsional stress in the crankpin oil bore	MPa
σ_{vcp}	Equivalent alternating stress in the crankpin fillet	MPa
σ_{vj}	Equivalent alternating stress in the main bearing fillet	MPa
σ_{vbo}	Equivalent alternating stress in crankpin oil bore	MPa
σ_{dwcp}	Fatigue strength for the crank pin	MPa
σ_{dwbg}	Fatigue strength for the main bearing	MPa
σ_b	Minimum material tensile strength	MPa
Q	Acceptability factor	
σ_m	Mean Stress (Fatigue Analysis)	MPa
σ_a	Alternating Stress (Fatigue Analysis)	Mpa
Se	Endurance stress	Mpa
nf	Fatigue factor of safety	
K_{yy}	Conditions constant for conrod buckling (Roark's)	
S	Sommerfeld number	
ω_{co}	Conrod angular velocity	Radian/s
ω_{cr}	Crankshaft angular velocity	Radian/s
ω'	Relative angular velocity of the conrod to the crankpin	Radian/s
ω_l	Angular velocity of the load	Radian/s
F_{bs}	Balance mass radial force	N
L_{90}	Bearing life (90th percentile)	hr
M_c	Per unit bending stress in optical window	N

Chapter 1

Introduction

As the report title suggests this project involves the research of Optical Access Engines and the design of (at least in part) of an Optical Access Engine for the University of Southern Queensland's engine laboratory.

What is an Optical Access Engine? Put simply an Optical Access Engine is an engine (in this case a reciprocating internal combustion engine) where the combustion process can be viewed from outside the engine body. The means of doing this has changed over the years, however in modern engine designs this is currently achieved by viewing the process through a transparent piston crown and mirror positioned at 45° to the path of the piston. See figure 1.1 for a better understanding.

It is important to note that it is assumed in this report that the reader possesses a basic understanding of the operation of an internal combustion engine. Specifically a spark ignition, petrol engine as found in most cars these days. To learn more about the internal combustion engine the reader is referred to the following excellent texts: Introduction to Internal Combustion Engines by Richard Stone, Internal Combustion Engines Fundamentals by J.B. Heywood, Internal Combustion Engines by Colin R Ferguson.

1.1 Background: The need for on-going research.

Why perform engine research? In short to further engine development. The internal combustion engine since its inception into general use around the Mid-19th century (Stone,1999) has undergone a myriad of design changes.

By developing engine technology, gains can be made on many fronts (efficiency, performance, reliability, new designs/prototypes and possibly new or modified fuels).

There is a belief that fossil fuels cannot be consumed at their current rates, this leads to the need for more efficient engines to be created and engine/combustion research is a means to achieve this.

Further to the previous statement, emission standards in most countries become stricter by the year (Johnson, 2006), this forces manufactures to design & build cleaner emission engines and the only means to do this involves engine / combustion research.

1.2 The Optical Access Engines Role in Combustion Research.

Possibly the first optical access engine could have been created by Nicolaus Otto when developing one of his prototype internal combustion engines. The optical access component consisted of a glass cylinder sleeve simply to view the mechanical process, however the concept is there.

In the 1930's General Motors engineers Rassweiler & Withrow were utilizing quartz windows in the cylinder heads and high speed cameras of the time to study combustion in an attempt to solve the engine knock problem (Richter,2008).

In 1960 Fred Bowditch & Lloyd Withrow at General Motors patented an engine design including an elongated piston with an opening in the trunk to place a 45° mirror bolted to the engine block. Included in the head of the piston was a transparent quartz window allowing the combustion process to be viewed from beneath via the 45° mirror (see below).

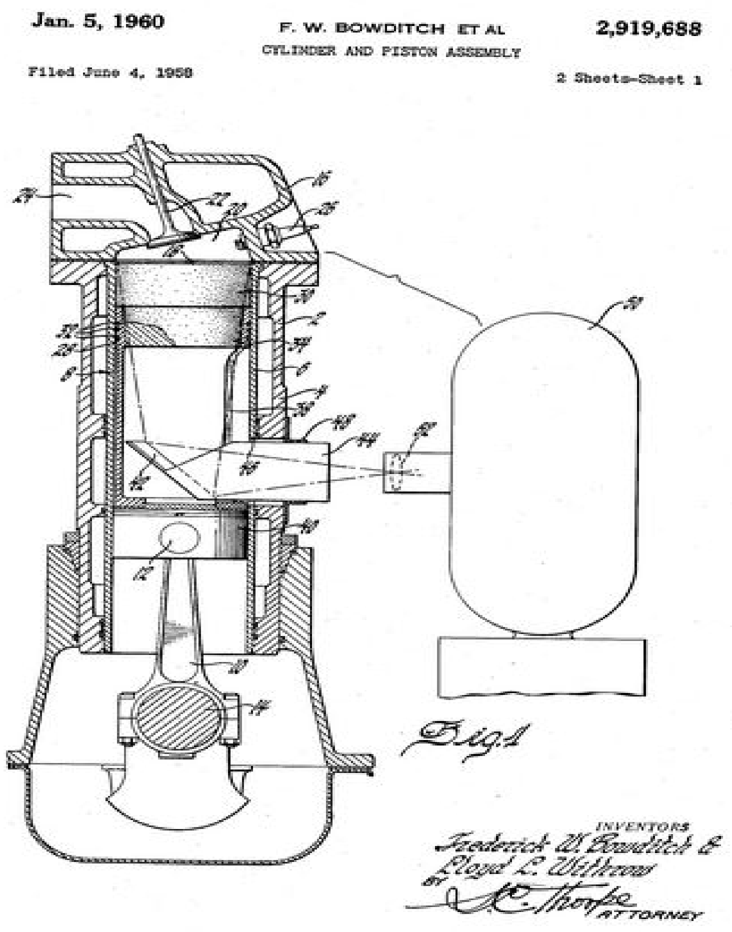


Figure 1.1: Bowditch Piston Patent Application Drawing (Google Patents, 2012)

It is this basic design applied to a modern engine that is the concept behind our optical access engine design.

As said optical access engines are by no means a new concept, however their use in engine research has increased in recent years. With improvements in high speed camera technology, pressure transducers, IR cameras and the diagnostic functions of lasers the research possibilities of these engines is greater than ever before.

1.3 The USQ Optical Access Engine Project (Background)

An existing optical access engine design (in part) was provided to USQ from Oxford University and specifically Professor Richard Stone. The design is Professor Stone's from the early 1990's and consists of various general arrangement and detail drawings mostly of the crankcase and counter balance shafts required to counter the various shaking forces produced. The package also included what appeared to be detail drawings of parts from existing stationary engines. This was concluded from the presence of part numbers on the drawings. It is possible these parts were intended for use in the original optical access engine design.

It needs to be mentioned that the design package acquired was incomplete and unverified. Numerous drawings were missing/incomplete eg: bowditch piston, upper cylinder, cylinder head and associated parts to name a few. Also there was no complete document transmittal, design calculations or material specifications.

1.4 The Project Aim

The project task is to transform this incomplete package into a working design. Something the University can invest in for their engine laboratory and proceed to manufacture. This would include producing general arrangement drawings, detail or workshop drawings, design calculations verifying the design, material specifications (included in the drawings) and virtual simulation / finite element analysis to supplement the calculations.

To summarize the aim: Take the existing design as provided and transform it into a complete and usable design package.

1.5 Scope of Work

A shortened itemised list of contributions made to this project is given below,

1. Review & interpret the original design package provided by Prof. Stone
2. Model the existing design in 3D CAD software (Autodesk Inventor)

3. Determine the parts requiring design verification
4. Determine the parts missing from the package provided and parts requiring redesign due being designed by engine manufactures with tooling and production facilities unavailable to USQ.
5. Design the required parts: Bowditch piston, wrist pin, crankshaft, conrod, big end bearing, little end bearing, upper cylinder.
6. Perform engine balancing calculations and virtual simulation to verify
7. Design verify the required remaining parts
8. Produce assembly and detail drawings for manufacture.

Extensive research was required to perform a number of these tasks, knowing what features were typical in an optical access engine needed detailed research as did the materials used for certain parts (PTFE rings, Quartz widows etc.)

As mentioned a number of components were missing from the provided package and required designing. Designing these, the verification of existing and new components along with engine balancing and drawing production were the main contributions made.

An important requirement of this design and thus a design constraint was the final design needs to be produced in a general workshop. Complex manufacturing processes could not be part of the production requirements and therefore final design needed to account for this.

1.6 Conclusion

To date this project has involved engine research with a focus on existing optical access engines, research into engine balancing specifically single cylinder engines, mechanical component design with a focus on designing for infinite life (fatigue) and plain bearing design with a focus on hydrodynamic bearing conditions.

Extensive material research has been performed to select both suitable and available materials for each designed component factoring in the workshop limitations.

All this has led to the complete design of the engine bottom end up to the top of the piston and barrel. The crankshaft and conrod was redesigned using the original drawings as a guide but increased in strength and modified to better suit fabrication in a general machine shop. The piston and upper cylinder have been designed from scratch utilizing extensive research into materials, optical access designs and clearances.

Lastly all remaining parts were utilized from the design package and only minor modifications performed as a result of verification calculations, balancing requirements and parts / materials availability.

The focus of this dissertation will be discussing the new component design, existing component verification, engine balancing and design considerations / limitations. The literature review will outline other optical access engines, their place in industry, features, manufacturers and where the USQ optical access engine fits in.

It was originally hoped to reach the point of design where the head from an existing engine would be married up to the cylinder block of the new design. Unfortunately time constraints have caused this project to fall just short of this point and it is hoped that another student will continue the design pass this point.

A set of detail drawings and design calculations can be found in the appendices.

Chapter 2

Literature Review

The automotive industry has been utilizing optical access engines for a number of decades as a tool to assist in finding solutions to their development problems and meet ever changing emission specifications. Universities conducting research and education have also been using optical access engines for several years now to study the combustion process and educate students.

Engine and combustion research is driven by the need to improve existing designs, manufacturers are pressured by the market and governing bodies to improve emissions and fuel economy.

If the global increase in car sales continues with global sales topping 80 million in 2013 up 4.2% on 2012. (CNBC, 2014) and 92% of consumers rate fuel efficiency a top priority when purchasing a new car (KPMG, 2014). A continuing focus on engine research by manufactures seems a logical outcome to meet the ever tightening emission controls imposed by governments countering pollution due to an increase in the number of vehicles and the demands of consumers for more efficient vehicles. This is validated by the belief of 76% of automotive industry executives that internal combustion engine downsizing is a key issue for future development and 46% plan to invest in internal combustion engine downsizing more than any other power train technology investment. (KPMG, 2014)

To summarize –Legislative and consumer pressures are driving manufactures toward producing more efficient engines with complex control. Many of these new engines have benefited from optical research engines to develop these technologies. (Allen, J, et al., 2000).

This sets the scene as to the relevance of engine research for the domestic sales market alone, the place of optical access engines in engine research is wide spread. A number of papers can be read with respect to engine research using optical access engine on the SAE (Society of Automotive Engineers) digital library. Research topics include laser diagnostics and optical measurement techniques, characterization of combustion, piston temperatures, fuel sprays and fuel-air mixing to name a few in a variety of engine types eg: SIDI, CIDI, LPG engines and more.

Research into existing optical access engines has been explored in SAE Technical Papers (Carling et al, 1999, Catapano et al, 2011, Steeper et. al, 2000, Weinrotter et al. 2005, Aronsson et al. 2011, Liu et al. 2014) and more. Mechanical design & engine research has been explored in textbooks (Stone, 1999, Norton 2000, Budynas et al. 2012, Young et al. 2012).

2.1 What is an optical access engine?

As mentioned in the introduction an optical access engine is an engine where the combustion process can be observed from outside the engine. Unfortunately this description doesn't paint a clear picture as to what an optical access engine is and what types there are.

Through the reading of numerous engine research publications it has been derived that there are only four commonly and possibly only used methods of gaining optical access to the combustion chamber of an I.C.E. (Internal Combustion Engine).

1. Bowditch Piston (Transparent Crown)
2. Transparent Cylinder Liner (Optical Bore)
3. A Transparent Window in the Head
4. Endoscopic probe

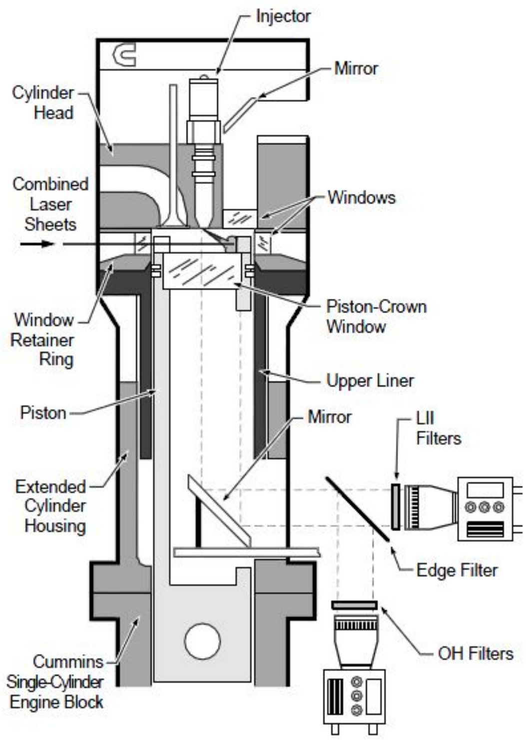


Figure: 2.1: Schematic diagram of a diesel optical access engine. Including Bowditch piston, 45° mirror, upper cylinder windows & optical access window in the cylinder head. (RW Carling et al., 1999, p2)

Although it is possible utilize all four methods on a single engine, modern engines using overhead valves & particularly multi-valve engines all but eliminates the use of the third method. Referring to 2.1 left you can see the top end of an optical access engine in cross section, the extended piston with a transparent crown is a Bowditch piston. The combustion in the chamber can be viewed using the 45° shown at the bottom of the piston. Optical access and laser application can also be achieved using the transparent window at the top of the cylinder. In most research laser diagnostics are applied through the top transparent sleeve and the imaging equipment receives through the bowditch piston and mirror arrangement. In this figure there is an example of an optical access window in the head however it's is far more common to apply an endoscopic probe through the head instead.

To paint a historical picture transparent cylinder heads (or at least heads containing a viewing window) were all that was in use up to the 1960(s), however were always limited by the fact that their size with respect to the cylinder was restricted by the engine valves (assuming an OHV engine) and where the head design was the focus of the research a new window need be fitted with every new cylinder head (Bowditch, F et al, 1958).

By far the most common arrangement is the bowditch piston and transparent upper cylinder liner, it is common to include an endoscopic probe for research purposes, papers by (Catapano, Sementa & Vaglieco, 2011) mention the use of endoscopic probes for optical research in their "Design for a multi- cylinder hi-performance engine GDI engine". Similarly (Kong, Ricart & Reitz, 1995) utilize an endoscopic probe to acquire luminous flame images from the combustion chamber for research into "In-cylinder diesel imaging compared with numerical computations".

The vast majority of engine research papers found in the SAE digital library utilizing optical access regardless of whether a petrol or diesel engine was the focus of the research used a combination of a bowditch piston and transparent upper cylinder liner. In no case where laser diagnostics were applied was a transparent upper cylinder liner/window not used.

2.2 Applications

Optical access engines are a tool for research and in the case of universities both research and education. This is clear by the fact that virtually all are utilized in research laboratories be it private research facilities such as the General Motors Collective Research Laboratories or a university laboratory. Their primary use within research is for the purpose of combustion research and engine behaviour.

General Motors Collective Research Laboratories claim to apply and develop optical diagnostics as a means to reveal the physical understanding and factors limiting the implementation of novel combustions modes such as Homogeneous Charge Compression Ignition (HCCI) & spray-guided direct-injection stratified charge. Such research and development is carried out by GM at their dedicated optical engines laboratory and in conjunction with the University of Michigan. (University of Michigan, 2009).

Many Universities choose to construct their own optical access engine (USQ for example) not only here in Australia but all over the world. A simple google search on the topic will find institutions such as Michigan State University, Melbourne University, Massachusetts Institute of Technology, Oxford University, Technische Universiteit Eindhoven to name only a few.

Numerous research papers have been written, specializing on various aspects of engine combustion research using optical access engines. A large number of these papers can be found in the SAE digital library. Such research has involved engine types such as petrol or diesel, turbo or naturally aspirated, direct injection or premixed, large bore or small, swept volumes of 0.3L to 2.5L.

Most research is performed on engine sizes (swept volume) of the intended target. This is to say that research on GDI (Gasoline Direct Injection) engines would be aimed at typical automotive engines (cars) and therefore the swept volume of the research engine would be around 0.5L. This would cover a 2.0L engine if a four cylinder and a 3.0L engine if six.

The Sandia National Laboratories constructed a DISI (Direct injection Spark Ignition) optical access engine for combustion research which was the focus of their paper “Characterization of combustion, piston temperatures, fuel sprays & fuel-air mixing in a DISI optical engine”. The engine they constructed possesses a cylinder swept volume of 0.565L.

Another engine constructed was by the Catapano, Sementa & Vaglieco of the Vienna University of Technology, their design is a four cylinder in-line engine of 1750cc total displacement giving an individual cylinder swept volume of 0.438L. Both engines were spark ignition petrol engines as it is intended our engine will be and the cylinder swept volume will be 0.458L when built.

2.3 Existing Optical Access Engine Designs.

As previously mentioned there are a great number of optical access engines in use in various research & educational laboratories. To give a better idea of what makes up an optical access engine particular designs and their details shall be discussed.

The “Design for a multi- cylinder hi-performance engine GDI engine” paper by Catapano, Sementa & Vaglieco, 2011 describes in detail their design, construction and operational results.

This team basically took an existing 4 cylinder 1.8L turbo GDI engine of undisclosed make and converted it into a complete multi-cylinder optical access engine. Where this design differs from most engine designs researched is that although multi-cylinder engines are often utilized in the construction of an optical access engine (for various reasons discussed later) they are not commonly converted so that all cylinders are optically accessible.

The designers Catapano, Sementa & Vaglieco state in their paper the design objectives of avoiding modifying the operational characteristics of the real engine, constructive simplicity, and the thermo-fluid dynamic behaviour of the real engine remaining unchanged thus avoiding the usual brevity of optical engine running tests as reasons for designing the engine in such a fashion.

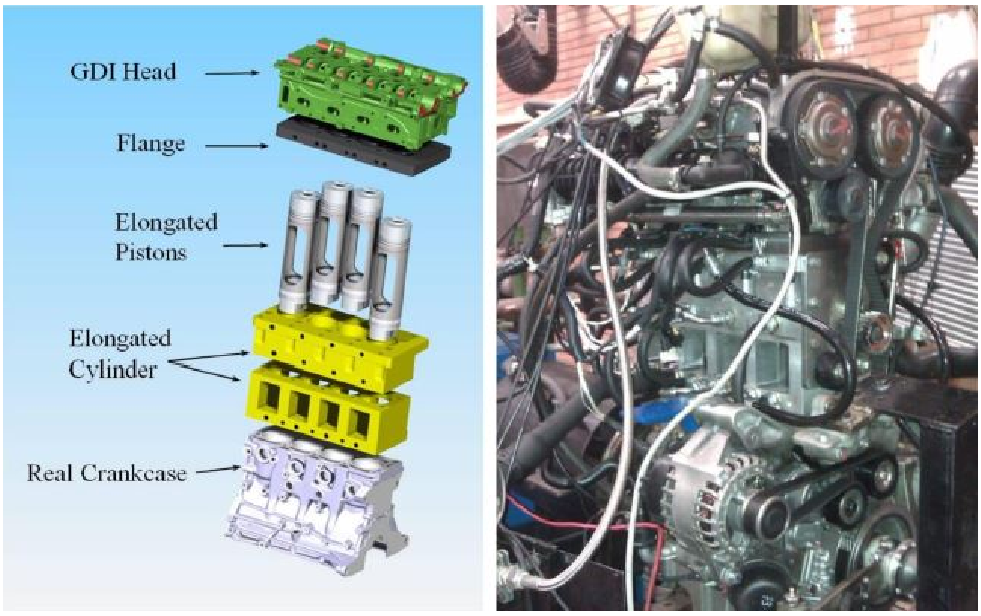


Figure 2.2: Left: Exploded model view of the multi-cyl optical research engine. Right: Actual engine on test bench in laboratory (Catapano F et al., 2011)

The above configuration of the engine is clear where the basic engine design has been modified as little as possible. Viewing of the combustion process and receipt of the laser images for analysis via the 45° mirror is through the square openings at the bottom of the elongated cylinder block. The laser sheeting is applied through quartz windows built into the flange and the endoscopic probe used as another optical access device is fitted to the GDI (not shown).

Another feature of this design, although not unique it is less common, is that engine powers itself as a typical engine does. With the exception of the optical access features (Bowditch piston, elongated cylinder, flange/window & probe) this engine is a typical car engine.

Many optical research engines do not run as engines but are turned over by a coupled drive usually an electric motor. The cylinder is then fired on alternate cycles, even every third or so cycle to control operating temperatures (skip fire routine). This approach was taken by (RR Steeper,et al., 2011) seen below and validated as an acceptable method for achieving suitable piston temperatures and combustion performance with respect to a real engine situation.

All optical research engine designs must address engine balancing as all engines generate shaking forces as a product of operation. Unbalanced engines generate excessive shaking forces that influence the research apparatus. The engine used in the Catapano, Sementa & Vaglieco design is a four cylinder in-line engine and thus is already balanced for the first order forces of inertia (forces in phase with engine RPM).

The engine design used by Richard R Steeper & Eric J Stevens of Sandia National Laboratories for the research paper “Characterization of combustion, piston temperatures, fuel sprays & fuel-air mixing in a DISI optical engine” seen in figure 2.3 was a single cylinder engine utilizing two balance shafts to counter the primary inertia force generated by the engine motion. These shafts were additionally weighted using tungsten alloy to counter the additional forces due to the heavier bowditch piston (RR Steeper,et al., 2011).

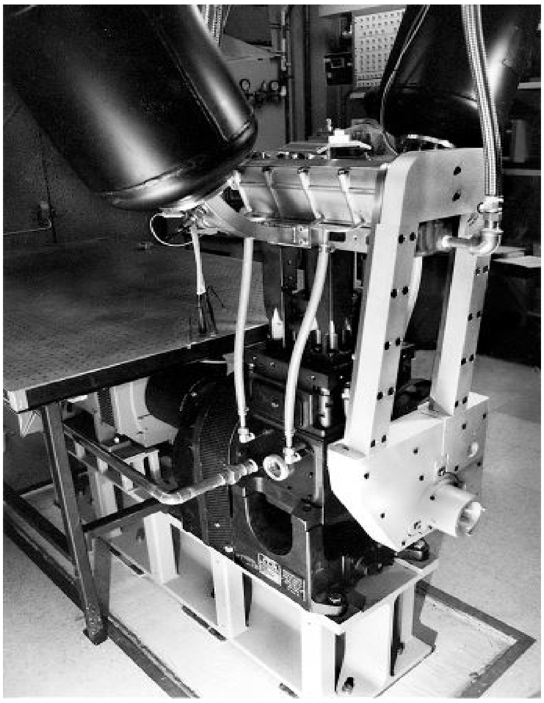


Figure 2.3: View of a single cylinder DISI optical research engine used by Sandia Laboratories. (RR Steeper et. al, 2000)

A common approach particularly by universities when constructing research engines is to take an existing multi-cylinder engine and remove the head of all but the optical access cylinder. The optical access cylinder is then constructed as intended for the experiments and the whole arrangement is driven by an electric motor. This provides effective balance against the shaking force of the engine turning over and reduces the amount of investment in designing and constructing the bottom end of the engine. This approach was employed by Maunoury, Duverger & Mokaddem (2002) in their research paper "Optical Investigation of Auto-Ignition in a small DI Diesel Engine" where a Peugeot 2.0L 4 cylinder diesel engine was used. Another team (Weinrotter et al. 2005) used an in-line six cylinder Scania D12 diesel engine for their research paper "Optical Diagnostics of Laser-Induced & Spark Plug-Assisted HCCI Combustion"

It is important to note that all designs employ high pressure transducers, thermocouples and rotary encoders to map cylinder pressure, temperature, visual imaging and laser diagnostics to the crank angle. Referring to the paper (Lui et al., 2014) on the effects of charge homogeneity & repeatability on particulates using the PLIF technique in an optical DISI engine an example can be seen of the diagnostics results being represented with respect to crankangle below.

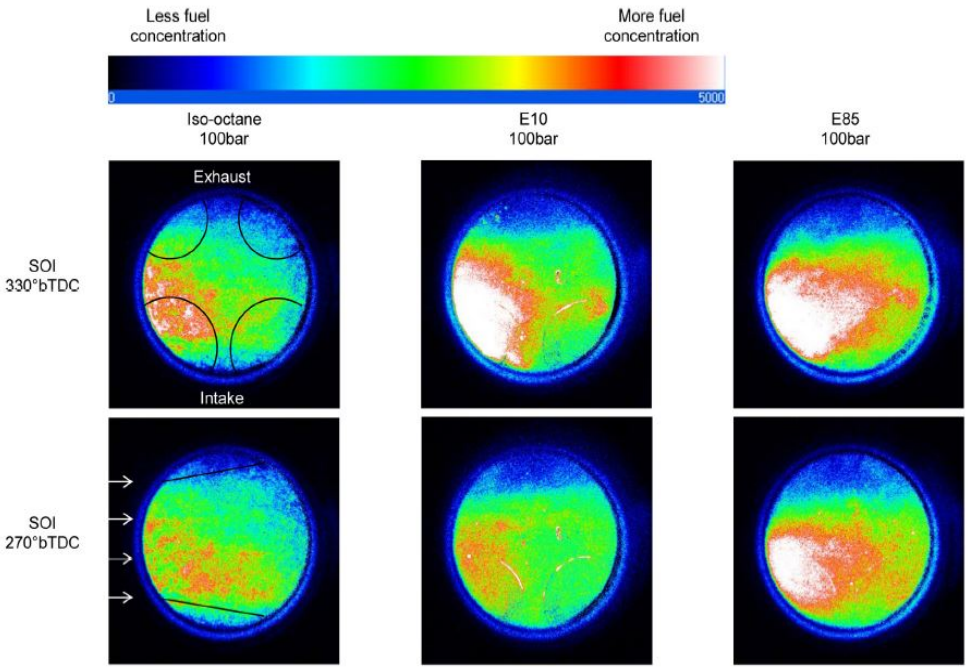


Figure 2.4: Results from single cylinder DISI optical research engine shown with respect to crank angle; engine used by Brunel University. (Lui et al., 2000)

To assist in gaining perspective as to what optical access engines are being constructed (ie: petrol/diesel, single-cyl/multi-cyl) both by OEM(s) such as Ricardo or institutions such as Brunel University, a table has been constructed consisting of the optical research engines reviewed in the process of researching this project.

It is important to note that most of the engines reviewed were not part of design papers but combustion research papers and the engine details were taken from the methodology or setup chapters of these papers. Also added to the list are the leading OEM of optical access engines products for perspective into what is being used, there are numerous laboratories using the Ricardo and Lotus packages for research and thus form part of the greater picture.

From the table 2.5 it can be seen that there is a wide mix of petrol and diesel engines in use for research. This is logical as both engines are in wide use in the automotive industry and it is therefore expected that they would be the focus of much research, especially considering 46% of automotive industry executives plan to invest in ICE downsizing more than any other power train research (KPMG,2014). The use of hydrogen and alternative fuels is in more common use with newer research engines, this can be attributed in part to increased knowledge of hydrogen combustion and alternative fuels use in newer combustion modes such as HCCI, Low Temperature Combustion & Controlled Auto-Ignition.

Another detail to note from table 2.5 is the use of the bowditch piston, virtually all optical research engines use this configuration & a large number employ the optical bore or an optical window. This can be attributed mostly to the use of laser diagnostics and in the case of the Sandia, Ricardo & Lotus engines as a means of viewing the injection phase & charge compression dynamics using hi-speed imaging.

Optical Access Engine Feature List

	Maker/Facility	Fuel	Fuel System	Piston	Endoscopic Probe	Optical Bore	Optical Head Window	Cyl(s)	Year	Balancing	Additional Details
1	Ricardo (Hydra)	All	GDI, IDI, DI	Bowditch	Optional	Yes	No	1	2014	Balance Shafts	Custom built to client specifications, (suitable for HCCI, PCCI,DISI,DICI), Advanced valve train technology available
2	Ricardo (Proteus)	All	GDI, IDI, DI	Bowditch	Optional	Yes	No	1	2014	Balance Shafts	Custom built to client specifications, (suitable for HCCI, PCCI,DISI,DICI), Advanced valve train technology available
3	Sandia	Alternative	DI	Bowditch	No	Yes	No	1	1999	Balance Shafts	
4	Sandia	Diesel	DI	Bowditch	No	Yes	Yes	1	1999	Balance Shafts	
5	Sandia	Petrol	GDI	Bowditch	No	Yes	No	1	2000	Balance Shafts	
6	MIT	Petrol	GDI	Aluminium Square	No	Yes	No	1	2001	Unknown	
7	University of Naples	Petrol	GDI	Bowditch	Yes	Window	No	4	2011	Multicylinder	
8	Peugeot/Citroen	Diesel	DI	Bowditch	No	Yes	No	1-4	2002	Multicylinder	
9	University of Technology Valencia	Diesel	DI	Standard	No	Window	No	1	2003	Unknown	Two Stroke
10	Vienna University of Technology	Diesel	IDI	Bowditch	No	Yes	No	1-6	2005	Multicylinder	HCCI
11	University College of London	Petrol , Hydrogen	DI, Port Injection	Bowditch	No	Yes	Yes	1	2009	Balance Shafts	Four stroke Lister, V8 Proto typehead
12	University of Wisconsin	Petrol Diesel Hybrid	DI	Standard	Yes (2)	No	No	1	2010	Unknown	PCCI engine (Premixed charge compression ignition
13	Lotus	All	GDI, IDI, DI	Bowditch	Optional	Yes	No	1	2014	Balance Shafts	Custom built to client specifications, (suitable for HCCI, PCCI,DISI,DICI), Advanced valve train technology available
14	IFP	Diesel	DI	Bowditch	No	Yes	No	1	2009	Unknown	
15	Brunel University	Petrol / Ethanol	GDI	Bowditch	No	Yes	No	1	2014	Balance Shafts	Ricardo Hydra bottom End

Table 2.5: Table of features covering various optical access engines. Taken from manufactures datasheets and SAE research papers.

2.4 The Bowditch Piston Arrangement

Referring back to Figure 1.1 (Introduction) and also Figure 2 (glossary) the transparent window crown, elongated body and skirt opening can be seen, these are the key features of the bowditch piston. Situating a mirror mounted to the barrel inside the piston cavity allows the viewing of the combustion process above the piston. As can be seen in the image below where the intake and exhaust valves can be seen in the mirror.

Figure 2.6: Left: Rear view of an optical access engine, bowditch piston and bowditch mirror. Valves are visible in the mirror. (RR Steeper et. al, 2000)

Right: Example of a bowditch piston, note the elongated opening and crown. (Catapano F et al., 2011)

The amount of elongation or additional length to the piston is a function of the size of the mirror used and the stroke of the engine, clearly the mirror is not allowed to foul on the operating piston. Additional length to the piston can also be a result of the crown being raised to prevent the rings from operating on the transparent liner as is the case shown above right taken from (Catapano F et al., 2011)

Some Bowditch piston and optical bore designs allow the rings to run in the quartz cylinder liner, this requires thick cylinder walls and sleeves longer than the sum of the engine stroke and depth of the rings from the piston crown so that the rings do not cross over the joint between the optical bore and the lower cylinder. This approach is quite common particularly where the users wish to view more than simply the combustion phase of engine operation. Examples of this approach can be found in figure 2.6 above and the Lotus engine below in figure 2.7.

When designing the piston arrangement allowance must be made for lubrication particularly in the upper cylinder region as no lubricating oil reaches the upper piston skirting. The highest friction occurs at the rings due to cylinder pressure assisting the rings in sealing by applying pressure in an outwards direction on the inside ring face. To deal with this in standard engines compression rings are made as thin as possible and cross hatching the cylinder bore allows small quantities of oil to remain and reduce the friction between the ring face and cylinder. In an optical access engine using a bowditch piston this is not possible as no oil reaches the upper cylinder.

To deal with this hi-temperature polymer rings are used, the plastic to metal or quartz friction co-efficient is quite low and any resulting wear occurs mostly on the rings which are easily replaceable. Another advantage to using polymer rings is reduced bore scuffing, this is important where a long optical bore is used to prevent reduction in optical access quality. Catapano F et al., 2011 used Carbon PTFE and Bronze PTFE as their ring materials whilst (Rosati, MF et al. 2009) used Torlon™ as the ring material.

2.5 Optical Access Engines (Cutting edge)

Who is at the cutting edge of optical access engine design and engine research is hard to know for certain as most companies guard their intellectual property tightly. However several companies offer optical access engine/research engine packages for sale to institutions wishing to perform research, such packages are certainly current technology by today's definition.

Two well-known market suppliers of Optical Research Engine packages are LOTUS and the Ricardo group. LOTUS offers a single cylinder research package named "SCORE" which stands for Single Cylinder Optical Research Engine, this package as seen in figure 2.7 can be used for a variety of optical diagnostics such as,

- Particle Image Velocimetry (PIV)
- Phase Doppler Anemometry (PDA)
- Laser Induced Fluorescence (LIF)
- Laser Induced Incandescence (LII)
- High Speed Imaging

And more….

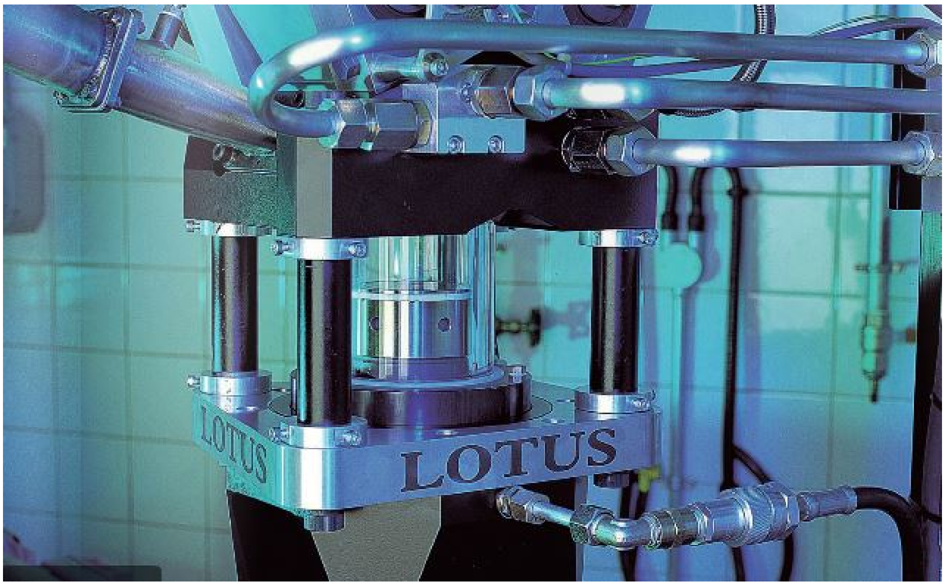


Figure 2.7: A view of the head & optical of the LOTUS SCORE. (Lotus,2013)

Most optical access engine are able to apply these diagnostics however the LOTUS package is able to operate at real engine loads and speeds up to 5000RPM. Another feature available in the SCORE package is AVT (Active Valve Train) which is a hydraulically actuated valve system utilising advanced electronic control to manipulate and thus test and research valve timing and depression effects on combustion.

The system was developed to support research into Low Temperature Combustion, Controlled Auto Ignition and Homogeneous Charge Compression Ignition (HCCI). All of which can be researched using the LOTUS SCORE system.

The RICARDO group offer their own packages of optical research engines, divided into two groups named Hydra & Proteus these research engines are available in a variety of sizes and feature options. The Hydra package is the lighter duty package available in petrol, diesel & alternative fuels with GDI, port injection & diesel DI or IDI. Bore sizes between 65mm & 110mm, the valve train is available in Camless hydraulic valve (HVA) which is similar to LOTUS AVT, variable lift or standard cam actuated to name a few features. Essentially this engine is suitable for typical automotive engine research. The Proteus package is the heavy duty package more aimed at the transport and marine engine research with bore size from 100mm to 150mm and peak pressures of 250bar.

Numerous research facilities including universities utilize the Ricardo engines for their laboratories, Ricardo claim their engines have contributed to 250 research papers in the last forty years. (Ricardo,2013).

What appears to set the Ricardo & Lotus engines apart from University laboratory engine setups are the peripherals and polished finish to the product. Operational features in general are superior to the typical University setup such as their engines can operate at full loads for research where many other setups need employ techniques such as skip firing to control temperatures. This may be erroneous as a research paper discussing an experiment is obliged to inform the reader of limitations, manufacturers almost never mention them in their marketing media.

2.6 The USQ optical access engine

To put the USQ engine into perspective or at least what has been designed at this stage refer back to table 2.5. The USQ engine is a single cylinder engine like most other engine setups reviewed; utilizing balance shafts to counter the operating inertia forces. As opposed to the slightly less common alternative of running one optical access cylinder in an operational or non-operational multi-cylinder engine arrangement for balance.

The USQ engine will be a bowditch piston type optical access arrangement, an optical bore will be included in the design for expected laser diagnostics. The use of endoscopic probes is unknown at the moment as the head arrangement is yet to be finalised.

Referring below to a flyer image for the Hydra light design marketed by Ricardo Group you can see similarities building between this product and the USQ engine. To get a better idea of this refer to the general arrangement drawing 4111-A001 found in Appendix C.

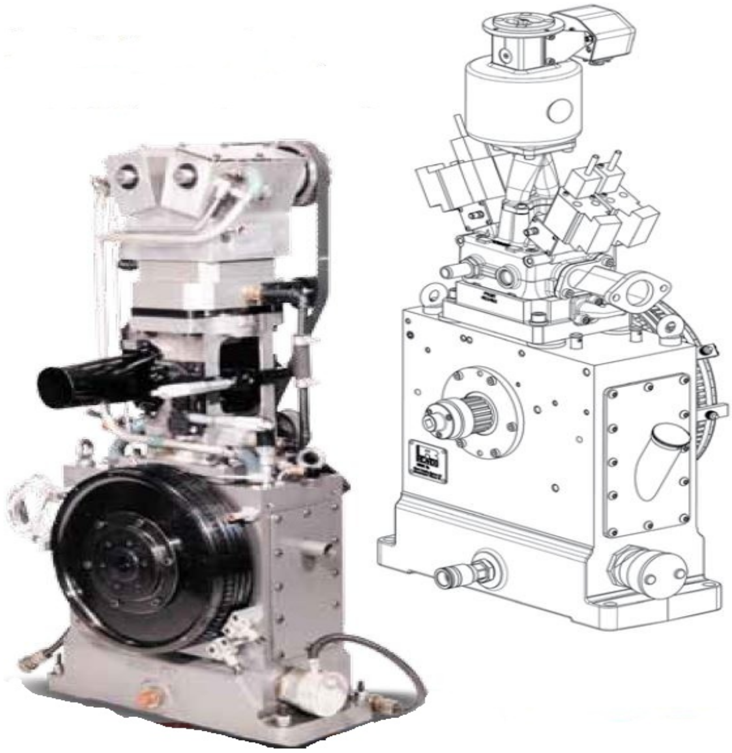


Figure 2.8: Ricardo Hydra Engine Image (Ricardo, 2013)

At this stage the USQ engine has been designed as an intended petrol engine however to allow flexibility in research down the track the piston, crankshaft & conrod are being designed to operate at higher than likely pressures to allow for the possibility of operation as a diesel engine, hybrid fuels engine etc. This hopefully will allow USQ to research more recent developments in combustion technology such as HCCI (Homogeneous Charge Compression Ignition), Low Temperature Combustion, PCCI (Premixed Charge Compression Ignition) and hopefully more.

2.7 Engine Design

This report is more of a design paper than a research paper however in order to design an optical access engine, research into what and optical access engine is, it's purpose and features, design pitfalls to avoid and specific mechanical design knowledge need be covered.

So far the literature review has covered the what, why & who of optical research engines but only briefly the how of their design. Most technical papers on the subject are more focused on the results of combustion research using these engines as opposed to detailing the research methodology used to design. It will be the focus of this paper to cover this missing content and provide insight into the design process. To design an engine be it an optical access I.C.E. or a typical I.C.E. research must be performed into the design of various components and the understanding of phenomenon relative to the specific machine.

Below is an abridged list of texts and papers utilized for research into the **design** of the USQ Optical Access Engine.

- Engine Balancing: Design of Machinery (Norton 2000) & Introduction to Internal Combustion Engines (Stone 1999)

- Mechanical Fatigue: Shigley's Mechanical Engineering Design (Budynas et al. 2012)

- Solid Mechanics: Roark's Formulas for Stress & Strail (Young et al. 2012)

- Crankshaft Design: Calculation of Crankshafts for Internal Combustion Engines (Paper) (Germanischer Lloyd, 2012)

- Piston Stresses: Impact of Mechanical Deformation due to Pressure, Mass & Thermal Forces on the In-cylinder Volume Trace in Optical Engines of

Bowditch Design (SAE Paper) (Aronsson, U, 2011)

• Materials:	Matweb.com website ; Bohler Uddeholm website; Capral Aluminium Catalogue; Dotmar Plastics website; Glacier Bearings DU catalogue; Heraeus Quartz datasheets, ASME Handbook, CP Carillo general catalogue.
• Small parts:	Blackwoods Catalogue; Bearing Services (BSC Catalogue; Unbrako Fasteners Catalogue
• Cylinder Pressures	Cylinder Pressure in a Spark Ignition Engine: A Computational Model (Kuo,PS , 1996); Perry's Chemical Engineers Handbook (2012)

Chapter 3

Methodology

This chapter covers the design considerations, procedures, tools & resources for the design of the USQ optical access engine as far the project scope covers. The results of work outlined in this chapter will be covered in the next section: Results/Discussion.

The specific aspects and components covered in this chapter will be,

- Engine Balancing: Theory & Design, Application of Virtual Balancing.
- Crankshaft Design: Cylinder Pressures, Application of Design Paper, FEA
- Conrod Design: Designing for Infinite Life (Fatigue), FEA.
- Bearing Design: Main Bearings, Big-End, Little End
- Piston Design: Design Considerations, Window Strength, FEA
- Various Design: Various Design Considerations not Covered

Overall Considerations

As mentioned in the introduction one of the design constraints for this project is for this engine to be fabricated in a typical engineering workshop, for example but not specifically the USQ fabrication workshop. Throughout this project and particularly during the modelling stage this consideration was one of the foremost factors influencing design decisions.

An example of this is when looking at the geometrical shaped of most parts an experienced tradesperson should be able to see that it can be made from common billet stock material and machined using a manual centre lathe & universal mill. Certain parts will be very much easier if machined in a three axis CNC mill however no major part should require anything more than this.

Material sizes have been kept to a minimum, the aim being to reduce material and machining costs.

Further to the interest of simplifying the design and solving certain parts issues it was decided at this stage to utilize the head off an existing engine. The head arrangement is probably the most complicated part of an engine and in using an existing head from an engine with a similar sized engine cylinder considerable time, money and design work can be saved.

After extensive research the engine selected was a Mitsubishi 4G93 engine, this engine possesses the same stroke as the original design by Prof. Stone and a bore size only one millimetre larger. By increasing the existing design bore size to the Mitsubishi engine's a number of parts can utilized; parts such as the oil rings, internal head components, head gaskets, fuel injection system, ignition system to name a few can be used in constructing the engine.

When making further design choices, if a part from the 4G93 is suitable it is not designed. Similarly or when a design decision or assumption needs to be made the Mitsubishi engine weighs considerably in the final outcome. One example of this is the ignition timing.

3.1 Engine Balancing

Due to the reciprocating manner of an internal combustion engine I.C.E. there are large forces at play during operation. These forces increase with respect to engine size (specifically piston or reciprocating mass) and engine RPM. It is important to reduce and counter these forces as much as practicable in order to prevent any adverse effect as a result of their presence.

The methods of dealing with the reciprocating forces in an I.C.E. are mass reduction and balancing. Both have been applied in the design of this engine, with respect to mass reduction the piston and conrod have been designed as light as possible with compromising structural integrity. It is the balancing that shall be the focus of this section.

3.1.1 Balancing Theory

The following sections cover the theory behind determining the inertia forces induced by the slider crank mechanism operating. This leads to how to counter these forces and thus balance the engine.

3.1.1.1 Slider Crank Kinematics

Below in figure 3.1 is the first design revision of the piston, conrod and crankshaft arrangement used in the project engine. It is this arrangement that was used to establish the theoretical model which will be explained here. The following theory for the piston displacement, velocity & acceleration is taken from (Introduction to Internal Combustion Engines 3rd ED., Richard Stone 1999).

The piston position or distance - X - is given by the equation….

$$x = r\cos\theta + l\cos\emptyset \qquad \text{Eq 3.1}$$

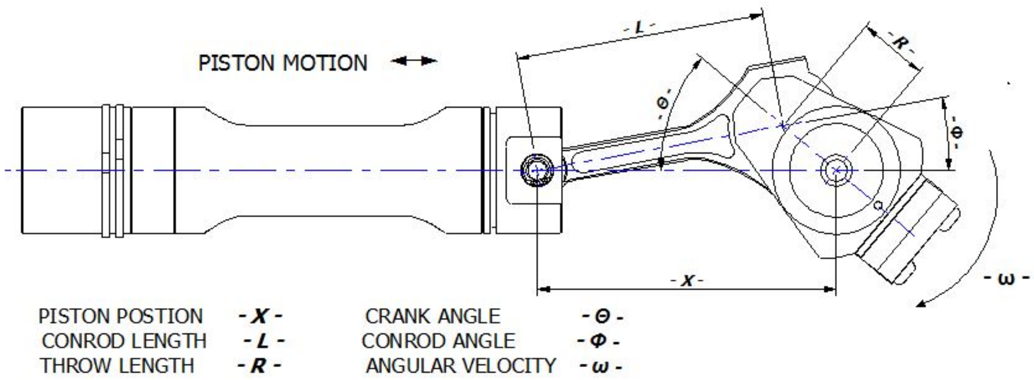


Figure: 3.1: Crankshaft, Bowditch Piston & Conrod Arrangement: Dimensioned for Kinematic Analysis

Note that….

$$r\sin\theta = l\sin\emptyset \qquad \text{Eq 3.2}$$

Remember that….

$$cos\emptyset = \sqrt{}(1 - sin^2\emptyset) \qquad \text{Eq 3.3}$$

Using equations 2.2 & 2.3 we can represent the piston position as …..

$$x = r(cos\theta + {}^{l}/_{r}\sqrt{\left\{1 - \left({}^{r}/_{l}\right)^2 sin^2\,\theta\right\}}) \qquad \text{Eq 3.4}$$

Using the Binomial Theorem to expand the square root term into ….

$x = r\{cos\theta + {}^{l}/_{r}\{1 - \frac{1}{2}(r/l)^2\, sin^2\theta - \frac{1}{8}(r/l)^4\, sin^4\theta + \cdots.$ Eq 3.5

The powers of sinθ can be expressed as….

$sin^2\theta = \frac{1}{2} - \frac{1}{2}cos2\theta$

$sin^4\theta = \frac{3}{8} - \frac{1}{2}cos2\theta + \frac{1}{8}cos4\theta$ Eq 3.6

Substituting Eq3.6 into 3.5 we get….

$x = r\{cos\theta + {}^{l}/_{r}\,[1 - \frac{1}{2}\left({}^{r}/_{l}\right)^2\left(\frac{1}{2} - \frac{1}{2}\,cos2\theta\right) - \frac{1}{8}\left({}^{r}/_{l}\right)^4\left(\frac{3}{8} - \frac{1}{2}\cos 2\theta + \frac{1}{8}cos4\theta\right)$ Eq 3.7

Since the conrod length or "l" must be twice the crank arm length or "r" plus allow clearance for the diameter of the big end bearing, piston skirting or barrel the ratio $(r/l)^2$ is invariably less than 0.1, which means that $(r/l)^4$ is an acceptable term to neglect.

Therefore an acceptable approximation of the piston position is…

$x \approx r\{cos\theta + {}^{l}/_{r}\left[1 - \frac{1}{2}\left({}^{r}/_{l}\right)^2\left(\frac{1}{2} - \frac{1}{2}cos2\theta\right)\right]\}$ Eq 3.8

Differentiating Eq 3.8 once with respect to time yields piston velocity twice yields acceleration given in the equations below…

$\dot{x} = -r\omega(sin\theta + \frac{1}{2}{}^{r}/_{l}\,sin2\theta)$ Eq 3.9

$\ddot{x} = -r\omega^2(cos\theta + {}^{r}/_{l}\,cos2\theta)$ Eq 3.10

Using the equation for acceleration the axial force developed in the engine due to the reciprocating mass can be expressed as…

$Fy_{rm} \approx m_{rm}r\omega^2(cos\theta + {}^{r}/_{l}\,cos2\theta)$ Eq 3.11

Where Fy_{rm} = vertical force due to the reciprocating mass

m_{rm} = equivalent reciprocating mass

ω = angular velocity, dθ/dt

r = crankshaft throw

l = conrod length (bearing centres)

$cos\theta$ = primary term

$cos2\theta$ = secondary term

θ = crank angle in degrees (0° = TDC)

Studying equation 3.11 you can see there is a primary force varying in amplitude with crankshaft rotation with the maximum amplitude governed by the reciprocating mass & angular velocity. This can be graphically expressed in a simple harmonic wave form with the maximum corresponding to crank angles of 0 & 180 degrees. (Refer figure 2.3)

There is a secondary force present corresponding to the second term in equation 3.11, this force varies in amplitude at twice the speed of the crankshaft and hence its maximum values correspond to the crank angles 0, 90, 180, & 270 degrees. (Remember the direction of the forces will change, this is denoted by a change in sign).

3.1.1.2 Equivalent Masses

To help clarify equation 3.11 and the graph below, the reciprocating masses needs to be clarified. "When calculating the engine balance, the connecting rod is treated as two masses concentrated at the centre of the big end and the centre of the little end." (Stone, 1999, p447).

Figure 3.2 below is the conrod taken from the first design revision, using this drawing it can be seen how the conrod can be described as two masses about a centre of gravity.

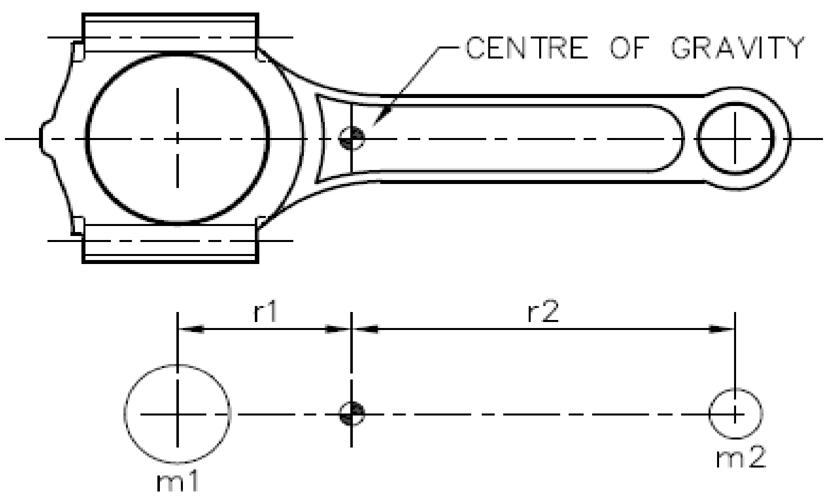


Figure: 3.2: The first connecting rod revision & its equivalent model.

The following theory is also taken from (Introduction to Internal Combustion Engines 3rd ED., Richard Stone 1999).

For equivalence…

$$m_{(conrod)} = m_1 + m_2 \qquad \text{Eq 3.12}$$

$$m_1 r_1 = m_2 r_2 \qquad \text{Eq 3.13}$$

The mass m_2 can be considered as part of the reciprocating mass giving m_{rm} as the sum of the piston mass and m_2. (Stone, 1999).

$$m_{rm} = m_p + m_2 \qquad \text{Eq 3.14}$$

Where	m_p	=	Piston mass
	m_2	=	Little end bearing mass

These equations and resulting equivalent masses when applied to the conrod allow the allocation of mass to the reciprocating mass component and thus equation 3.11. The remaining mass forms part of the crankshaft model and requires balancing through the crankshaft design.

Taking Eq 3.11, 3.14 and inputting the final masses for the piston & conrod along with the engine speed we get the following plot of the engine shaking forces (Vertical) in Newtons (N) with respect to crank angle (θ). In red is the total reciprocating force produced by the engine. In blue and black are the primary and secondary forces that make up the total force. The magnitude of the plotted forces varies with engine speed and should be ignored at this stage, what is important is the pitch and phase.

The plot below contains three traces where,

$$Fy_{rp} = Primary\ Reciprocating\ Forces = m_{rm} r \omega^2 cos\theta$$

$$Fy_{rs} = Secondary\ Reciprocating\ Forces = m_{rm} r \omega^2\ {}^{r}/_{l}\, cos2\theta$$

$$Fy_{rm} = Total\ Reciprocating\ Forces = Fy_{rp} + Fy_{rs} = Eq\ 3.11$$

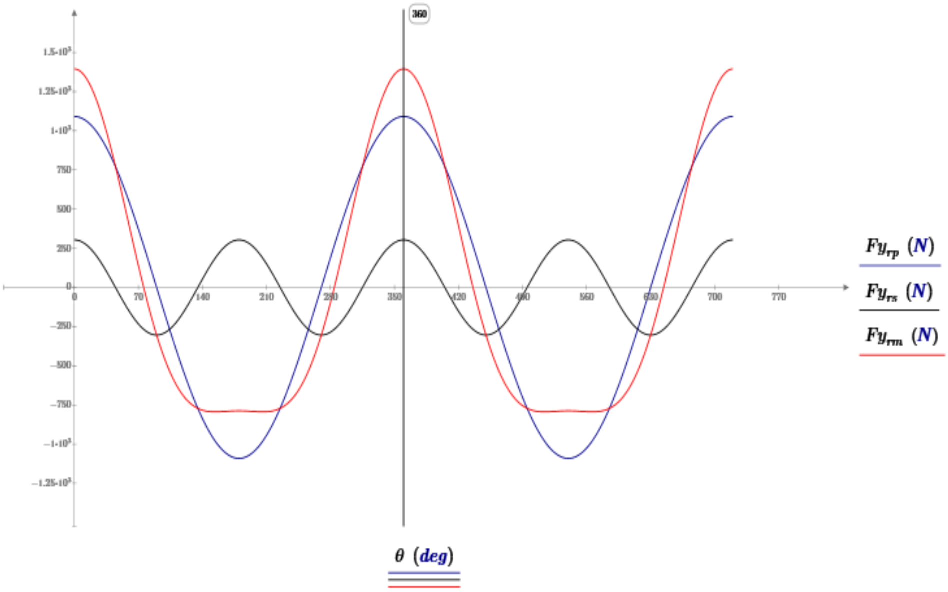


Figure: 3.3: Representation of Reciprocating Forces Inside the Optical Access Engine.
Note: Maximum forces occur at TDC (0°,360°,720°) & BDC (180°,540°)

It is important to note from the above equations and figure 3.3 that the total shaking or reciprocating forces are the sum of two sets of oscillating forces. It is these forces also known as shaking forces that are the focus of the balancing and will be discussed further in "Slider Crank Kinematics & Engine Balancing".

3.1.1.3 Shaking Torque

Another influence on the engines stability is the presence of shaking torque commonly known as inertia torque, inertia torque results from the action of the inertia forces at a moment arm. These arise due to the lateral component of the force driving the little end bearing component of the conrod plus the piston along its path by the distance of the little end bearing from the crankshaft centre. For a clearer picture refer to figure 3.4 below.

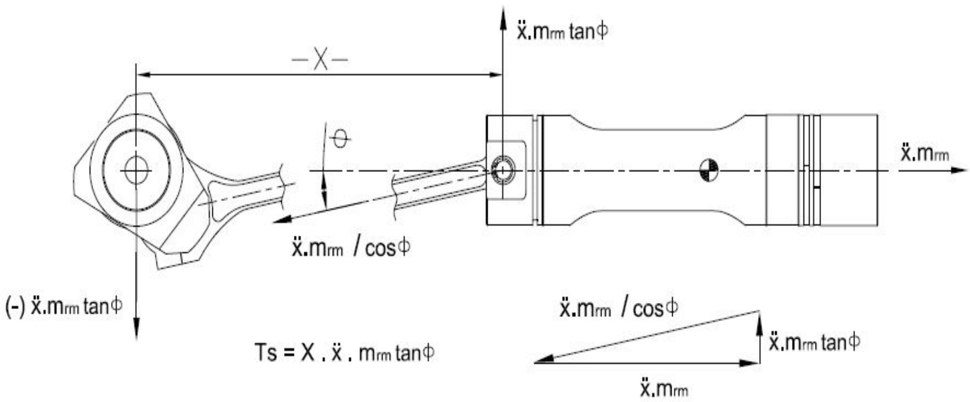


Figure: 3.4: Representation of Shaking Torque

Shaking torque can be represented with respect to crank angle as opposed to conrod as described in figure 3.4 above by the following equation (Norton,2000) ,

$$Ts = \frac{1}{2}m_b r^2 \omega^2 [\frac{r}{2l} sin\theta - sin2\theta - \frac{3r}{2l} sin3\theta] \qquad \text{Eq 3.15}$$

Where Ts = Shaking torque

Shaking torque has an average value of zero and thus contributes nothing to the net driving torque. (Norton,2000). Shaking torque is can be counteracted in a multicylinder engine however is not practicable to counter in a single cylinder engine. To reduce the shaking torque without affecting engine function the reciprocating mass must be reduced.

3.1.2 Slider Crank Kinematics & Engine Balance

Now that the shaking forces produced by the engine have been identified and broken down into components and magnitudes how to counter these forces will be discussed.

3.1.2.1 Shaking Forces

In a single cylinder engine as is the case here, the primary forces can be mostly cancelled out using a weighted crankshaft. This weight is known as the “bob weight” and is the common method of improving the balance of a single cylinder engine. The is weight is intended to provide a 180° out of phase sinusoidal varying force of mostly the same magnitude as the primary reciprocating force, it also balances the rotating force of the big end bearing & lower conrod (m_1).

However the bob weight does not counter the secondary forces of the reciprocating mass. What is required to counter the secondary forces is the introduction of another harmonic force at the same frequency but at a phase angle of 180° to the engine’s second order forces. The amplitude must match the secondary forces so as to perfectly counter and thus eliminate the engine shaking produced by the second order forces.

Although this sounds complicated it is quite simple, by introducing two opposite rotating balance shafts turning at twice the engine speed and timed at 180° to the engine the secondary forces, they both cancel out. (Stone 1999). Figure 3.5 below gives a simplified demonstration of this concept.

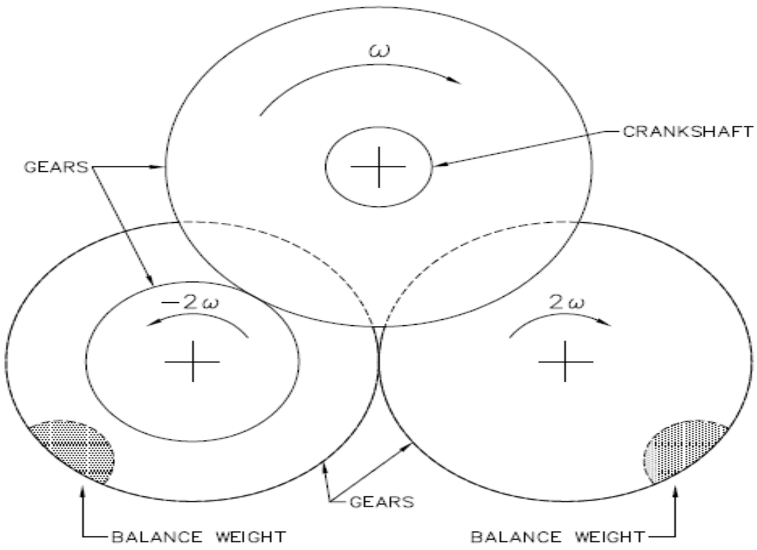


Figure: 3.5 Representation of Rotating Counter Balance Shafts (Stone,1999)

By employing a bob-weight and counter balance shafts the vertical shaking forces can be counter-acted and thus reduced to acceptable levels. Another advantage of employing bob-weights is the forces are reduced at the source and thus smaller crankshaft bearing may be utilised. However by employing a bob-weight you create an alternating force at 90° to the travel of the piston the magnitude of which is calculated using the equation below.

$$Fx_{bm} = m_b r_b \omega^2 \cos(\theta + 90) \qquad \text{Eq 3.16}$$

Where Fx_{bm} = force at 90° to the piston axis due to the bobweight

m_b = effective bob-weight mass

ω = angular velocity (dθ/dt) or engine speed

r_b = radius of the bob-weight COG for crankshaft centre

θ = crank angle (0° = TDC)

Giving a total alternating force at 90° to the piston travel of,

$$\sum Fx = \omega^2 \sin\theta \, [m_b r_b - m_{be} r] \qquad \text{Eq 3.17}$$

Where $\sin\theta$ = $\cos(\theta + 90)$

m_{be} = mass of the crankshaft & conrod big-end bearing

This is a useful trade off as usually the direction of the shaking forces is not of concern simply the magnitude (Norton,2001). Norton was considering the practicalities of the engine design ie: cost and complexity when he considered it in acceptable trade off. Norton also did not use the twin counterbalance shafts in his single cylinder end balancing analysis, similarly engine manufactures by employing only a bob-weight to balance their engines.

In the USQ optical engine design the additional cost of employing the counterbalance shafts to eliminate shaking forces is worth the investment, especially considering the potential improvements. It is actually worth balancing the crankshaft so that there are no alternating forces at 90° to the piston travel and employing four sets of counterbalance shafts. Two shafts in order to balance the primary forces and two shafts to balance the secondary forces. As they are in equal and opposite rotation only vertical forces are produced and in opposition to the engine shaking forces.

This design is demonstrated in the next section, please refer to figure 3.6.

It is unclear if Prof. Stone intended for the crankshaft to be statically balanced and have the balance shafts do all of the work countering the reciprocating mass. However in the interest of the best possible final result this is the approach taken. By simply increasing the mass of the bolt on bob-weights and reducing the primary balance weights more of the balancing workload can be performed at the crank. This adjustable approach was chosen to give operational flexibility to the end user.

3.1.2.2 Applying Balancing Theory to the Stone Engine Design

In the engine design by (Stone et al.) provided to USQ there was the addition of four balance shafts, two rotating at engine speed & two at twice the engine speed. As discussed in the previous section this is to counter the primary and secondary forces by means of equal magnitude and phase opposition.

Referring to figure 3.3 the trace of Fy_{rm} which represents the total shaking force as the sum of the primary shaking forces Fy_{rp} and the secondary shaking forces Fy_{rs} it can be seen by inspection that if the engine design was to provide equal and opposite forces then the total shaking forces would be reduced to zero. It is by this concept that our engine balance and thus the elimination of shaking forces (not torques) is realized.

Figure 3.6 below is a simplified drawing of the piston, crankshaft and balance shaft cluster in the engine design. These parts can be seen along with the rotational and directional orientations of the moving parts. Using the following equations and assumptions we can realize a balanced system with respect to shaking forces in all directions.

$$Fy_{pcb} = m_{pcb} \cdot \omega^2 \cdot r_{pcb} \qquad \text{Eq 3.18}$$

$$Fy_{scb} = m_{scb} \cdot (2\omega)^2 \cdot r_{scb} \qquad \text{Eq 3.19}$$

$$m_b r_b - m_{be} r = 0 \qquad \text{Eq 3.20}$$

Where Fy_{pcb} = Primary counter balance shaft force (vertical)

Fx_{scb} = Secondary counter balance shaft force (vertical)

m_p = Primary counter balance mass (combined)

m_s = Secondary counter balance mass (combined)

Assuming a perfect statically balanced crankshaft arrangement is achieved including the conrod big-end bearing mass thus achieving by design and test equation 3.20. The following equations can be realized by application of the correct counter balance weights and bob weights. It is this application that is the work of balancing the engine, the purpose of this section and one of the main design targets.

$$\sum Fy = Fy_{rm} + Fy_{pcb} + Fy_{scb} = 0 \qquad \text{Eq 3.21}$$

$$\sum Fx = \omega^2 \sin\theta \, [m_b r_b - m_{be} r] = 0 \qquad \text{Eq 3.22}$$

The application of these into a theoretical model along with the results shall be discussed further in the next chapter. To view the model please refer to Section 2.0 of Appendix B where a printout of the equations used in the mathematical software is presented. Please note the nomenclature is not exactly the same as that used here however it is close and understandable.

The forces derived from Eq 3.11, 3.18 & 3.19 above are also useful with respect to determining bearing loads on both the main bearings and the balance shaft bearings, the calculations used for determining bearing life and capacity can be found in later sections of this chapter as well as Section 4.2 of Appendix B.

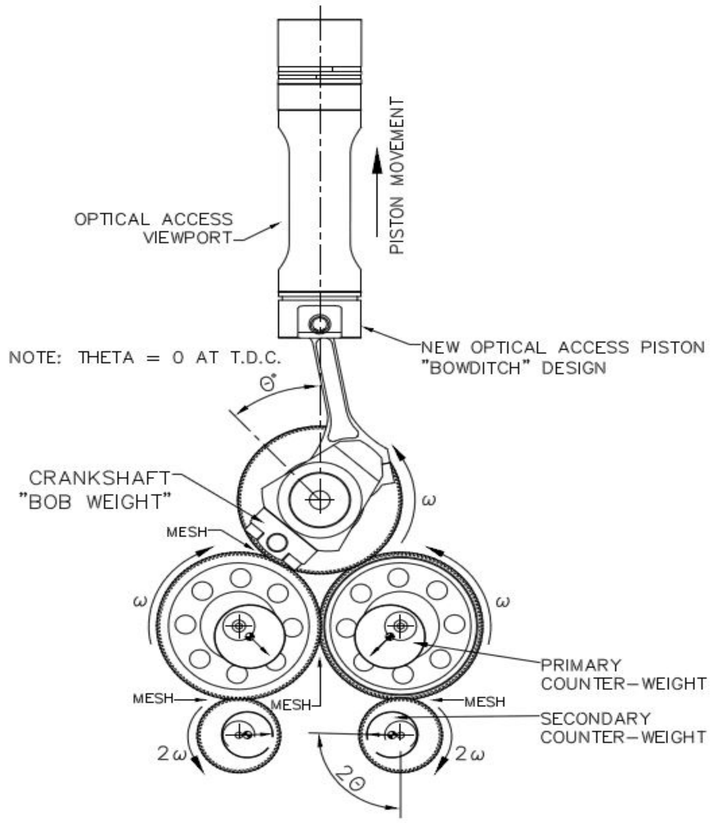


Figure: 3.6: Crank & Balance Shaft Arrangement for this Project (Revision A)

The application of the theory discussed is carried out frequently in the design and component verification of this engine. Situations such determining crankshaft stresses, conrod stresses, wrist pin bending as well as the bearings mentioned above are a few examples of these. Careful calculation and application of this theory will play an essential part in the successful design of our optical access engine.

3.1.3 Virtual Balancing

To give confidence to the design by verifying the theory given in the earlier sections the engine model will be been turned over using mechanical virtual simulation software. The software package used specifically is Autodesk Mechanical Simulator which is part of the Inventor design suite used to solid model the engine and create the drawings.

Although much of the process of setting up the simulation is automated when the same software is used to create the solid model as run the simulation, it is still important to review the constraints, degrees of freedom, standard joints, grounded components, component masses and operational parameters.

To help explain what some of the above terms mean, please refer to the simulation model below.

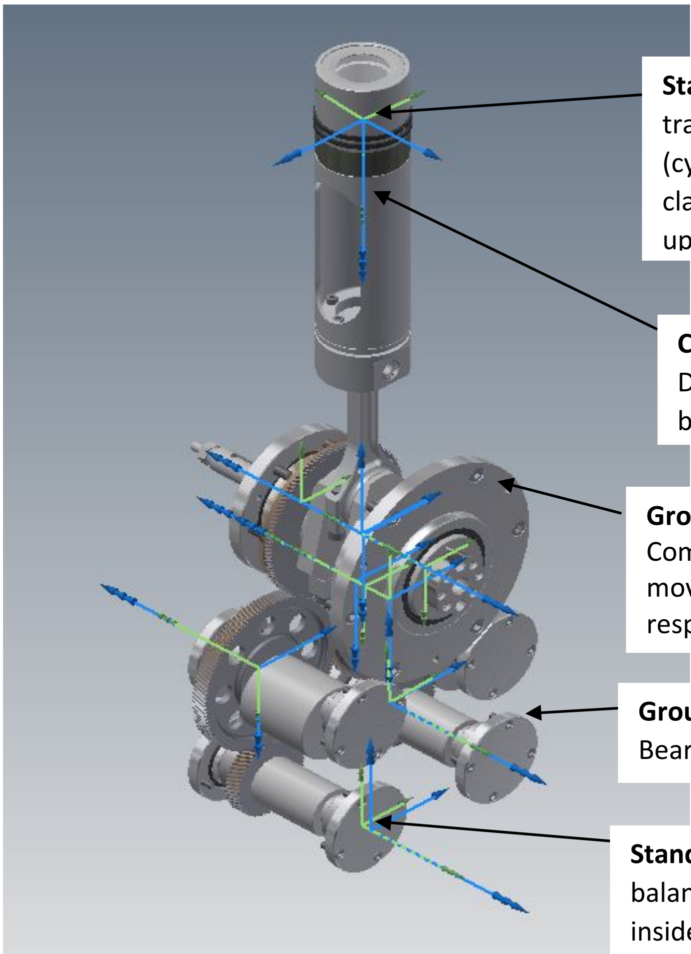


Figure: 3.7: Crank & Balance Shaft Arrangement within the Dynamic Simulation Environment

The software outputs reaction forces for every joint in the coordinates shown by the vectors in blue. A single arrow head denotes the local x-axis, two and three heads the y & z axes respectively. These forces are outputted with respect to chosen time steps and thus all data can be exported to an excel spreadsheet for further analysis.

Once all of the constraints, joints, drive speeds, part masses etc. were set the model is essentially run like engine would be. It important to note that the model is only simulating the “turning over” of the engine and no dynamics of combustion and the associated gas pressures are included. This is considered acceptable from a balancing perspective as the gas forces cancel each other out within the engine and thus do not contribute to the shaking forces of the engine. Please note that they do contribute to the shaking torque of the engine however it is not practical to counter these in a single cylinder engine.

There were two simulations run once the model was setup and checked for errors. The first simulation was the existing Stone et al. design with the first and therefore only roughly balanced crankshaft design along with the first bowditch piston design. The purpose of this first simulation was to compare the results with the theoretical model results. Should both results line up then the theoretical model and virtual model is validated. Design modifications to finish balancing the engine can be made with confidence.

The results of the initial simulation and theoretical model can be found in the next chapter along with the final balanced results for both methods.

3.2 Crankshaft Design

In a typical automotive, transport or marine engine applications the crankshaft loads are well known as is the required life of the crankshaft. Most engines spend many years in service often well exceeding a decade and not all that uncommon for an engine's life to exceed two decades. In the transport & marine industry engines often operate for more than 24hrs at a time so it stands to reason that engine crankshafts are designed for infinite life.

The definition of infinite life is arguable amongst different texts and varies between materials. For the purpose of this project the definition of infinite life will be taken from Shigleys, 2012 as exceeding 10^7 cycles.

The required life of this optical access engine's crankshaft is unknown as the expected usage of the engine is unknown, therefore the crankshaft will be designed for infinite life. The hand calculation methodology is taken from a German design standard published by Germanischer Lloyd, 2012 for crankshaft design in marine diesel engines. Although this engine is expected to be a spark ignition petrol engine the possibility of its use in HCCI research or small CI diesel research is there and not to be overlooked. The relevance of the calculations to a petrol engine is clear and this paper was deemed very technically sound and thus suitable for this purpose. The design standard is quite conservative and compliance will result in a very dimensionally acceptable crankshaft.

The hand calculation method discussed herein will be complimented by an F.E.A. on the crankshaft to validate both methods. The F.E.A. will be discussed after the hand calculation method.

The purpose of this section is to demonstrate the methodology applied to the design of the crankshaft and some of the considerations made.

3.2.1 Gas Pressure

Amongst other requirements the design paper required the determination of the conrod forces over the duration of one cycle, specifically to find the maximum and minimum bending and radial stresses in the crankshaft. This means the gas pressure with respect to time or crank angle needs to be found along with the inertia forces determined in the balancing section.

A paper submitted to MIT (Kuo,1996) covering the determination of cylinder pressures in a petrol engine with respect to crank angle was utilized in performing the calculations. The method is based on a constant volume (Oto Cycle) to determine the gas pressure during the cycle. This model does not allow for heat release to occur (ie: losses) or a change in combustion chamber volume during the burn phase.

Other approximations include the use of a single polytropic constant for each phase, (the constant would in reality change with respect to temperature and pressure). The pressure rise is based on the chemical energy release of the fuel as opposed to the mass of fuel actually burned.(Heywood,1988) There are better methods available to determine the maximum cylinder pressure such as the "heat release method" however this approach is simpler and more conservative.

Finding the gas pressure with respect to time/crank angle for the entire cycle (two revolutions) was deemed unnecessary as the design paper specifically requires the determination of the maximum and minimum stresses. To save analysis time a realistic assumption will be made that the outer limiting stresses occur in the compression, burn & expansion phases and therefore the gas pressures have been calculated over these phases only..

The equations and methodology for determining the cylinder gas pressure described herein are taken from (Kuo, 1996) and based on the assumptions,

- $\gamma = 1.48$ (Expansion); $\gamma = 1.3$ (Compression); $\gamma = 1.25$ (Combustion);
- $\phi_s = 13$ (Stoichiometric Ratio)
- $\phi_e = 0.9$ (Equivalence Ratio)
- $a_x = 5$; $m_x = 2$ (Burn constants used in the Weibe Function)
- 12° BTDC (Ignition Timing)
- 24° (Burn Duration)
- $C_{fuel} = 0.9$ (Burnt Fuel Constant)

The cylinder pressure at all times with the exception of at the burn phase is determined using the polytropic equation…

$$PV^{\gamma} = constant \qquad \text{Eq 3.23}$$

Where	P	=	Final State Pressure
	V	=	Final State Volume
	γ	=	Polytropic Exponent

This equation allows the calculation of the pressure at any time in the engine cycle based on the knowledge of the cylinder volume at the point of interest, the initial pressure, initial volume & the polytropic constant (given in the assumptions). The initial pressure and volume are used in the calculation of the constant be this at IVC or the end of combustion.

The next equation is the McCuiston, Lavoie & Kauffman (MLK) model used to determine the cylinder pressure with respect to crank angle. The MLK model is represented by the equation below.

$$Xb = \frac{PV^{\gamma} - P_o V_o^{\gamma}}{P_f V_f^{\gamma} - P_o V_o^{\gamma}} \qquad \text{Eq 3.24}$$

Where	Xb	=	Mass burn fraction
	P	=	Pressure corresponding to burn fraction
	V	=	Volume corresponding to burn fraction
	P_o	=	Pressure at start of combustion
	V_o	=	Volume at start of combustion
	P_f	=	Pressure at the end of combustion
	V_f	=	Volume at the end of combustion
	γ	=	Polytropic Constant

The next function is named the Weibe function (Kuo, 1996) and models the combustion with respect to crank angle.

$$Xb = 1 - e^{\left(\frac{-a_x(\theta-\theta_o)}{\theta_\Delta}\right)^{m_x+1}} \qquad \text{Eq 3.25}$$

Where	Xb	=	Mass burn fraction
	Θ	=	Crank angle in radians
	θ_Δ	=	Burn duration in radians
	θ_o	=	Ignition start angle in radians
	a_x , b_x	=	Burn Constants

The mass burn fraction given by the equation is the proportion of burnt fuel with respect to crank angle starting at 0% at ignition and finishing at 100% after the burn duration. Plotting this function gives a visual representation of the rate of combustion in an engine. Figure 3.8 shown below is the plot of the burn fraction with respect to crank angle in the theoretical model used for this engine. Note the burn starts at 348° or 12° B.T.D.C. which was taken from the ignition timing setting of a 4G93 Mitsubishi engine (4G9x Engine Manual, 2001). The burn duration is taken as the symmetrical range about TDC (Kou, 1996) which in this case equals 24° of crank angle.

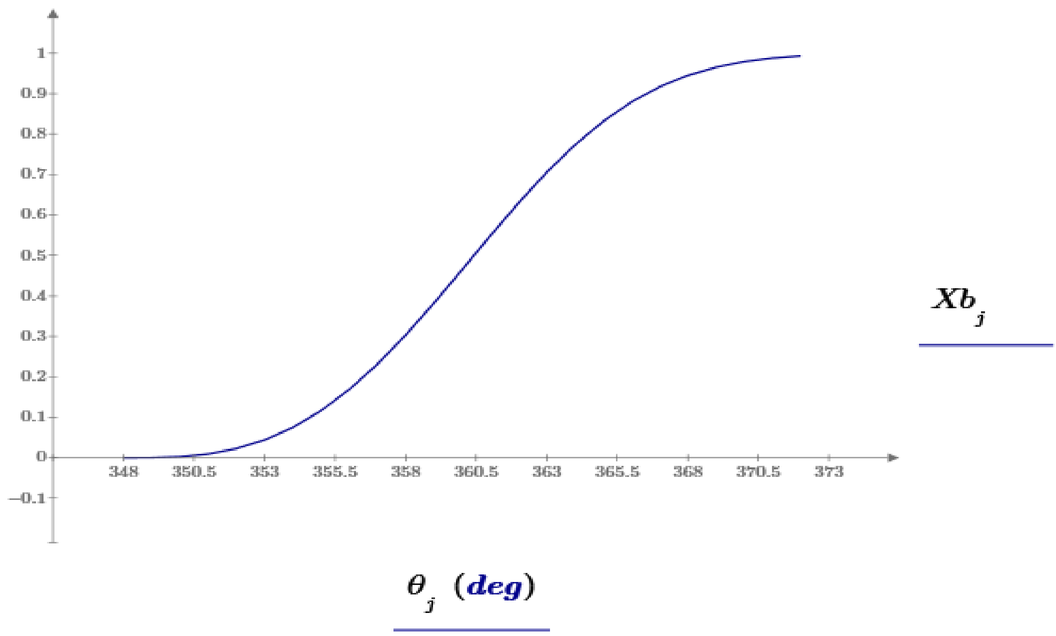


Figure: 3.8: Mass burn fraction during the burn phase 348° to 372° CA (12°BTDC to 12°ATDC)

Using the Weibe function to determine the mass burn fraction and rearranging the MLK model equation to give pressure in terms of the initial pressure and volume, final pressure and volume and the mass burn fraction the engine's cylinder pressure with respect to crank angle can be plotted for the burn phase. Using the polytropic equation Eq 3.22 as first mentioned the cylinder pressure during the compression phase and expansion phase can also be plotted.

An equation is now needed to complete the method of determining cylinder pressure. The following equations are based on the Otto cycle (heat addition at constant volume) and provides the cylinder pressure at the end of combustion. This equation assumes all heat is added at the maximum pre-ignition cylinder pressure and at a constant volume. The reality is the volume changes slightly over the combustion/burn phase and along with minor heat losses means the actual final pressure is below the calculated figure. However for the purpose of engine component design this is a conservative approach and therefore its use is quite justifiable.

$$T_f = \frac{C_{fuel} \cdot M_{fuel} \cdot Q_{fuel}}{(M_{fuel} + M_{air}) Cv_{air}} + T_o \qquad \text{Eq 3.26}$$

Where
- T_f = Final gas temperature at the end of combustion
- C_{fuel} = Unburnt fuel allowance
- M_{fuel} = Fuel mass in the charge
- M_{air} = Air mass in the charge
- Cv_{air} = Specific heat value of air at constant volume
- Q_{fuel} = Heating value of the fuel
- T_o = Initial temperature of the charge gas (pre-spark)

Taking the final temperature found and using the ideal gas law the pressure at the end of combustion can be found.

$$P_f = \frac{P_o V_o T_f}{T_o V_f} \qquad \text{Eq 3.27}$$

Where P_F = Final cylinder pressure

P_o = Initial Pressure

V_o = Initial Cylinder Volume

V_f = Final Cylinder Volume

To summarize, the polytropic equation is used to find the pressure in the compression phase and the pressure at the point before ignition. The pressure and the end of combustion is then calculated using the above two equations. Using the pressure at the beginning and end of the burn phase there is sufficient information to apply the MLK model and Weibe function. These models will give the pressure in the cylinder during the burn phase with respect to the crank angle. Once these have been established re-application of the polytropic equation using the pressure and volume at the end of combustion will determine the cylinder pressure with respect to crank angle over the expansion phase.

For a plot of the theoretical cylinder pressure over the crank angles 180° - 540° refer to the results section in the next chapter. To review both the detailed calculations and plots of cylinder pressure vs crank angle refer to Section 3.0 of appendix b.

3.2.2 Conrod Forces

With the cylinder pressure found using the method discussed in the previous section the conrod forces can be determined with respect to crank angle. Using the following equations to determine the conrod angle and piston (gas) force, a plot of the axial force in the conrod with respect to crank angle can be produced.

It is important to note that the piston (gas) force is not the only forces acting on the conrod at any given instant. Inertial forces calculated in the balancing section must be accounted for when determining the axial force in the conrod and hence the applied force to the crankpin with respect to crank angle.

Referring back to Eq. 3.2 and Figure 3.1 in the balancing section…

$$r \sin\theta = l \sin\emptyset \qquad \text{Eq 3.28}$$

Using this equation to find Ø (conrod angle) we get…

$$\emptyset = \sin^{-1}\left(\frac{r}{l}\, sin\theta\right) \qquad \text{Eq 3.29}$$

The piston (gas) force is described using…

$$F_{g_\theta} = (\pi D_p^2 P_\theta)/4 \qquad \text{Eq 3.30}$$

Where F_{g_θ} = Piston gas force at a given crank angle

P_θ = Cylinder pressure at a given crank angle

D_p = Piston Diameter

Representing the forces acting on the piston in a free body diagram we can easily derive an equation for the conrod forces with respect to crank angle. Figure 3.9 below represents the forces acting on the parts themselves and the equation for the force in the conrod.

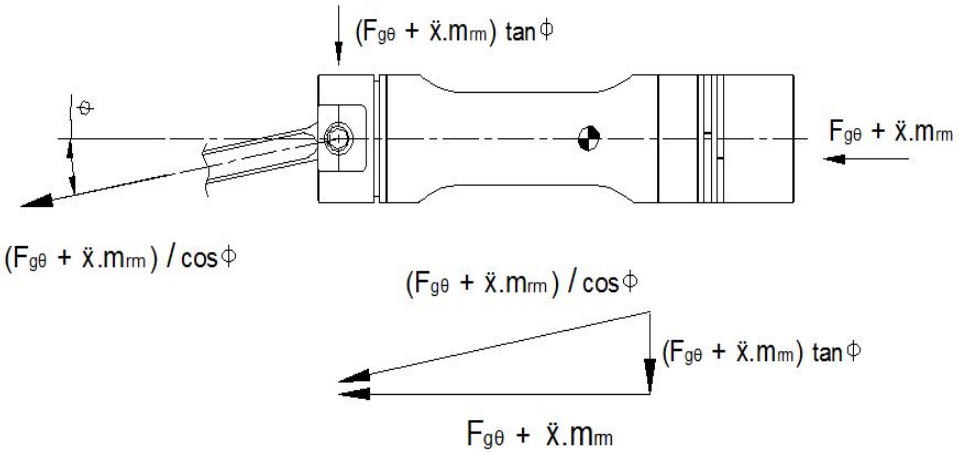


Figure: 3.9: Piston & Conrod Arrangement with the Acting Forces; Beneath a FBD of the Forces.

From the FBD in Figure 3.9 above and Eq. 3.11 the equation for the conrod force with respect to crank angle is,

$$F_{c\theta} = (F_{g_\theta} + Fy_{rm})/\cos(\theta) \qquad \text{Eq 3.31}$$

Where $F_{c\theta}$ = force through the conrod at a given crank angle and cylinder pressure.

The force through the conrod Eq. 3.31 above is represented in terms of the gas pressure and the inertia forces. The arrangement is similar to that shown in the balancing section where the inertia torque was discussed. A component of the inertia torque is included in Figure 3.9, it should be noted that the gas pressure adds to the value of the shaking torque. Unfortunately these forces cannot be countered and must be dealt with by other means.

Taking the value of the conrod force with respect to crank angle the stresses in the crankshaft can now be calculated. The conrod forces calculated will also be used to validate and possibly refine the conrod design in the next section.

3.2.3 Application of Design Paper

The exact methodology for the crankshaft design is set out in the design paper by (Germanischer Lloyd, 2012), detailed calculations of the application of this methodology can be found in Section 4.1 of Appendix B. For brevity only the main points of the procedure and principles of calculations will be discussed here.

The design of crankshafts using this standard is based on an evaluation of safety against fatigue in the highly stressed areas. The calculation is also based on the assumption that the areas exposed to highest stresses are the following. (Germanischer Lloyd, 2012)

- Fillet transitions between the journal and web and the crankpin and web
- Outlets of crankpin oil bores.

These assumptions are validated using the F.E.A. methods discussed next with results shown in the next chapter.

The calculation of the crankshaft strength consists of calculating the nominal alternating stresses (bending and torsion) which when multiplied by the appropriate stress concentration factors (SCF) and combined as appropriate using Von Mises Criterion result in an equivalent alternating stress in each respectively calculated location. This equivalent stress is then compared with the infinite life fatigue strength of the selected material (4340 in this case). This comparison determines the dimensional adequacy of the crankshaft design.

The first step in calculating the stresses is determining the bending moments, applied torque and axial forces acting on the web. In the figure below the reactions, load points and lines of action for the crankshaft bending are shown. Also given are the necessary dimensions used to calculate the various bending moments and reactions. Other dimensions given are for determining the section properties and stress concentration factors used to calculate individual stresses at critical points.

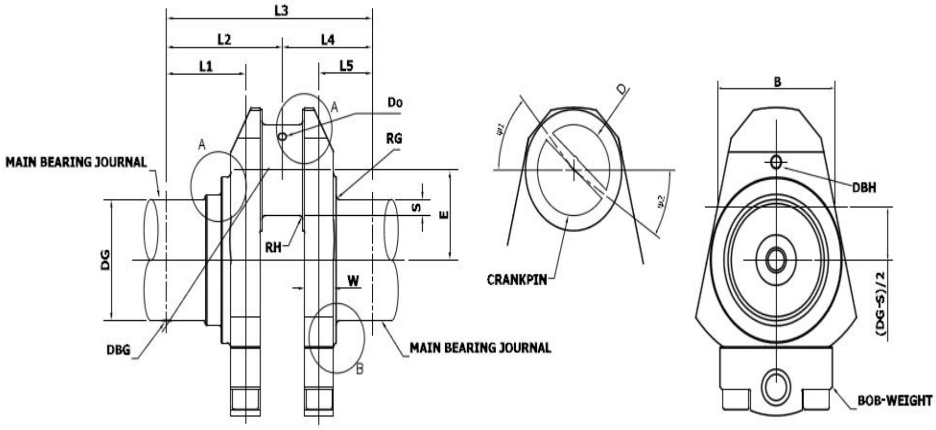


Figure: 3.10: Dimensioned crankshaft used for design calculations (refer to appendix B for dimensions, descriptions and detailed calculations)

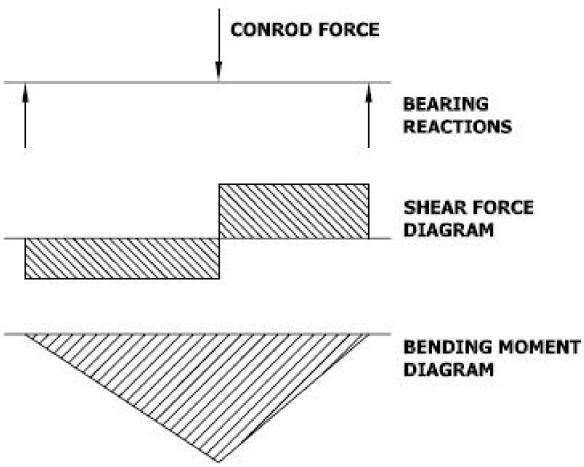


Figure: 3.11: Force / reaction model, Shear Force Diagram & Bending Moment Diagram (all indicative only)

The first step in the crankshaft design is to calculate the maximum & minimum bending moments and radial forces acting in the web, this is ultimately to find the maximum alternating stresses occurring located at the top of the main bearing journal fillet) and bottom of crankpin journal fillet (tension/compression only).

To do this the calculated conrod forces with respect to crank angle determined in the previous section are applied at the centre of the crankpin journal. Over the period of the compression to expansion phases (180° BTDC to 180°ATDC) the bending moment and axial forces in the web are calculated using the equations below.

$$Q_{rf\theta} = F_{c_\theta} \cos\varphi \, \frac{L2}{L3} \qquad \text{Eq 3.32}$$

Where $Q_{rf\theta}$ = The axial force acting in the web at angle θ

φ = The angle between the conrod and the crankarm

$\frac{L2}{L3}$ = Ratio of the maximum load bearing web

$$M_{brf\theta} = F_{c_\theta} \cos\varphi \, \frac{L4.L1}{L3} \qquad \text{Eq 3.33}$$

Where M_{brf} = Bending moment in web due to radial forces at θ

$\frac{L4.L1}{L3}$ = Ratio of dimensions to find the bending moment.

Note the axial forces and bending moments calculated in the web are strictly the radial component of the conrod forces. The bending moments in the web due to the tangential forces occur significantly out of phase to the radial moments and the resultant stresses are considerably less due to the higher section modulus of the web in the relevant plane of bending.

Next to be calculated is the bending moment in the crankpin with respect to the oil bore, as mentioned at the beginning of this section the oil bore is one of the points of greatest stress (assumed) and thus is the location of interest. Although the bending moment in the crankpin will be highest at the outside of the section aligned with the conrod axis the SCF relating to the oil bore geometry easily accounts for the assumption that the highest stresses will occur at the outlet.

To calculate the moment at the oil bore the following equation applies,

$$M_{bo\theta} = \left(F_{c_\theta} \cos\varphi \tfrac{L2.L4}{L3}\right) \sin\psi + \left(F_{c_\theta} \cos\varphi \tfrac{L2.L4}{L3}\right) \cos\psi \qquad \text{Eq 3.34}$$

Where M_{bo} = Bending moment at the respective oil bore

ψ = Angle of the oil bore outlet to the crank arm (Refer Figure 3.10)

3.2.3.1 Alternating Stress

Once all of the bending moments and axial forces have been calculated with respect to crank angle, the next step according to the design standard being followed is the calculation of the alternating stress occurring at the points of interest. This involves the application of the bending moments just determined and the relative section properties. Below is a series of equations used to calculate the required stresses.

$$\sigma_{bfn} = \frac{\frac{1}{2}\left[M_{brf(max)} - M_{brf(min)}\right]}{W_{eqw}} \qquad \text{Eq 3.35}$$

Where σ_{bfn} = Alternating bending stress in the web

W_{eqw} = Section modulus of the web

$$\sigma_{qfn} = \frac{\frac{1}{2}\left[Q_{rf(max)} - Q_{rf(min)}\right]}{B.W} \qquad \text{Eq 3.36}$$

Where σ_{qfn} = Alternating radial stress in the web

$B.W$ = Cross sectional area of the web

$$\sigma_{bon} = \frac{\frac{1}{2}\left[M_{bo(max)} - M_{bo(min)}\right]}{W_e} \qquad \text{Eq 3.37}$$

Where σ_{bon} = Alternating bending stress at the oil bore

W_e = Section modulus of the crankpin

An additional stress that needs to be calculated at this stage is the torsional stress occurring in the crankshaft. The following equations determine this,

$$M_{t\theta} = F_{c_\theta} \sin\varphi \; E \qquad \text{Eq 3.38}$$

$$\tau_n = \frac{\pm\frac{1}{2}\left[M_{t(max)} - M_{t(min)}\right]}{W_p} \qquad \text{Eq 3.39}$$

Where $M_{t\theta}$ = Torsion in the crankshaft at crank angle θ

W_p = Section modulus of the main bearing journal

τ_n = Torsional stress in the main bearing journal

3.2.3.2 Stress concentration factors

Once all of the alternating stresses have been calculated using the above equations, Stress Concentration Factors (SCF) need to be applied and certain stresses need to be combined for points of interest where multiple stresses occur concurrently.

The preceding equations for stress have made no allowances for geometric irregularities, such irregularities are called stress raisers and the degree to which they affect a component's stress value is dependent of the geometry of the body. A stress concentration factor (SCF) is used to relate the actual maximum stress to the nominal stress at a given discontinuity. (Shigleys, 2012)

The SCF(s) used to determine the maximum alternating stresses at the various points on the crankshaft were taken from the design standard using very lengthy equations based on empirical data resulting from extensive testing.

For detailed calculations of the stress concentration factors refer to the design paper listed in the bibliography.

The stress concentration factors used in the crankshaft design are listed below,

α_b = Crankpin Fillet Bending Stress Factor

α_t = Crankpin Fillet Torsion Stress Factor

β_b = Journal Fillet Bending Stress Factor

β_t = Journal Fillet Torsion Stress Factor

β_q = Journal Fillet Radial Compression Stress Factor

γ_b = Crankpin Oil Bore Bending Stress Factor

γ_t = Crankpin Oil Bore Torsion Stress Factor

Applying the stress concentration factors to their respective nominal alternating stresses is demonstrated below using the following equations.

$$\sigma_{bh} = \pm(\alpha_b \,.\, \sigma_{bfn}) \quad \text{Eq 3.40}$$

$$\sigma_{bg} = \pm(\beta_b \,.\, \sigma_{bfn} + \beta_q \,.\, \sigma_{qfn}) \quad \text{Eq 3.41}$$

$$\sigma_{bo} = \pm(\gamma_b \,.\, \sigma_{bon}) \quad \text{Eq 3.42}$$

$$\tau_h = \pm(\alpha_t \,.\, \tau_n) \quad \text{Eq 3.43}$$

$$\tau_g = \pm(\beta_t \,.\, \tau_n) \quad \text{Eq 3.44}$$

$$\sigma_{to} = \pm(\gamma_t \,.\, \tau_n) \quad \text{Eq 3.45}$$

Where σ_{bh} = Alternating bending stress in crankpin fillets

σ_{bg} = Alternating bending stress in bearing journal fillets

σ_{bo} = Alternating bending stress in crankpin oil bore

τ_h = Alternating torsional stress in crankpin fillets

τ_g = Alternating torsional stress in bearing journal fillets

σ_{to} = Alternating torsional stress in crankpin oil bore

3.2.3.3 Equivalent Stress.

Once the factored alternating stresses are calculated they are combined into equivalent alternating stresses using the von mises criterion. This step reduces all stress equations down to three equivalent stresses, one for each of the respective points of concern mentioned previously (oil bore, crankpin & main bearing journal).

The equations to calculate the equivalent alternating stress by location are listed below.

$$\sigma_{vcp} = \pm\sqrt{(\sigma_{bh}{}^2 + 3\tau_h{}^2)}$$ Eq 3.46

$$\sigma_{vj} = \pm\sqrt{(\sigma_{bg}{}^2 + 3\tau_g{}^2)}$$ Eq 3.47

$$\sigma_{vbo} = \pm\frac{1}{3}\sigma_{bo}\left[1 + 2\cdot\sqrt{1 + \frac{9}{4}\left(\frac{\sigma_{to}}{\sigma_{bo}}\right)}\right]$$ Eq 3.48

Where σ_{vcp} = Equivalent alternating stress in crankpin fillet

σ_{vj} = Equivalent alternating stress in bearing journal fillet

σ_{vbo} = Equivalent alternating stress in crankpin oil bore

3.2.3.4 Fatigue Strength

The fatigue strength is to be understood here as the value of equivalent alternating stress (von mises) which a crankshaft can permanently withstand at the most highly stressed points. This method is different to some methods given in textbooks where the mean and alternating stresses are calculated then factored appropriately to cover their geometric irregularities. Once the factored stresses are known the values are interpreted using fatigue models and determined if a failure is likely to occur.

In this method a maximum allowable alternating stress is calculated based on material properties and dimensional values local to the point of interest. This results in a different value for fatigue strength for the oil bore, crankpin fillet and main bearing journal fillet.

The equations used to calculate fatigue strength by location are listed below,

$$\sigma_{dwcp} = \pm K\,(0.42\sigma_b + 39.3)\cdot(0.264 + 1.073\cdot D^{-0.2} + \frac{785 - \sigma_b}{4900} + \frac{196}{\sigma_b}\cdot\sqrt{\frac{1}{Rx}})$$ Eq 3.49

$$\sigma_{dwbg} = \pm K\ (0.42\sigma_b + 39.3)\ \cdot (0.264 + 1.073 \cdot\ Dg^{-0.2} + \frac{785-\sigma_b}{4900} + \frac{196}{\sigma_b} \cdot \sqrt{\frac{1}{Rg}}) \qquad \text{Eq 3.50}$$

Where σ_{dwcp} = Fatigue strength for the crank pin

σ_{dwcp} = Fatigue strength for the main bearing

K = Manufacturing factor

σ_b = Minimum material tensile strength

R_x = Rh in the fillet area; Do/2 in the oil bore area

3.2.3.5 Acceptability Criterion

The crankshaft will be considered dimensionally should the ratio of fatigue strength to equivalent stress be equal to or greater than 1.15. This comparison needs to be carried out for the oil bore, crankpin fillet and main bearing journal fillet. The comparison is based on the formula,

$$Q = \frac{\sigma_{dw}}{\sigma_v} \qquad \text{Eq 3.51}$$

Where Q = Acceptability Factor

σ_{dw} = Fatigue strength

σ_v = Equivalent stress

Adequate dimensioning of the crankshaft is achieved when the smallest acceptability factor is,

$$Q \geq 1.15 \qquad \text{Eq 3.52}$$

The final acceptability factor can be found in the detailed calculations given in Section 4.0 of Appendix B. The results will be discussed in the next chapter.

3.2.4 Additional Design Checks (FEA)

To assist in validating the design, additional design checks have been performed to compliment the application of the design standard. These additional checks include maximum stress calculation and comparison to yield strength, along with finite element analysis (FEA) of the crankshaft in multiple positions to validate/check the calculations in the standard.

The yield strength checks are discussed briefly in the results section with more detailed calculations found in Section 4.0 of Appendix B. The FEA results will be discussed in detail in the next chapter. Below in figure 3.12 is an example of a load case applied to the crankshaft in F.E.A. Only the model, mesh and load glyph are shown, the support constraints are difficult to see but are positioned where the centre of the main bearings would normally lie.

Example Load Case:

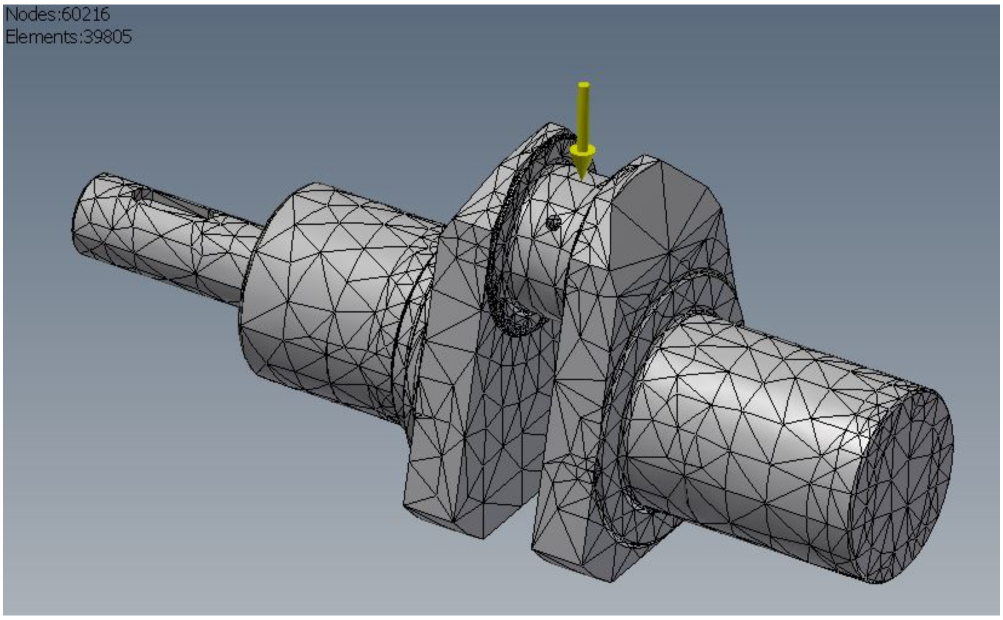


Figure: 3.12: Crankshaft FEA load case (8°ATDC, 10.511MPa cylinder pressure)

Desc: Max Cylinder Pressure

Position: 8° ATDC

Load: 10.511 MPa (applied via the conrod)

Loading Location: Bearing load distribution across crank pin journal aligned with conrod.

Constraints:

1. Fixed constraint at flywheel side main bearing centre (free rotation about perpendicular)
2. Frictionless constraint at encoder side main bearing centre

3. Rotationally constrained

Model Components:

1. Crankshaft

Mesh Elements:

1. Tetrahedronal Elements

Total Elements: 39805 Elements

Element Size: 0.1 (avg) of component size

Applying a number of these load cases such as maximum cylinder pressure (shown), maximum inertia force, maximum torque etc. the maximum and minimum stresses can be found in the crankshaft.

From there stress values can be applied to determine the fatigue stresses and resulting design factors of safety with respect to fatigue. Additionally the maximum stresses in the journal fillets can be checked against yield stress limits to ensure the component doesn't fail early in its operation. The methodology of fatigue analysis is covered in the next section applied to the conrod.

Additionally materials selection and deflection values can be selected and checked for the component. Detailed results of the F.E.A. and hand calculations are discussed in the next chapter.

3.3 Conrod Design

One of the simpler main components to analyse was the conrod. Although subject to high loads due to the conservative cylinder design pressure calculated and inertia forces resulting from a piston mass exceeding 2.0 kg the conrod design is still relatively simple.

This design discussion does not cover the big end or little end bearings as these are discussed separately in the section on bearings.

During an operating cycle the conrod is subjected to both tensile and compressive forces due to the reciprocating motion of the piston and the gas pressures used to drive the crankshaft. This creates fluctuating stresses inside the conrod that require fatigue analysis, as there is no design paper available for the conrod, Shigleys shall be used to determine the dimensional acceptability of the conrod design.

Fatigue and yield analysis will be the main design consideration here as the big-end bearing size is determined in the crankshaft and bearing design. Similarly the case for the little end bearing with the wrist pin being the deciding component.

The first design revision of the conrod will be based closely on the design contained in the original drawing package provided to USQ. The only changes being the increase in the little end bearing diameter based on preliminary wrist pin bending calculations and the modification of the big end bearing bolting arrangement for simplicity and reduction of stress raisers around the bolts. The bolting arrangement is based on racing engine conrod design used in lancers and similar vehicle engines.

3.3.1 Stress Analysis

The application of Shigleys will mostly cover the stresses occurring in the little end bearing eye where curved beam analysis is required due to the geometry. The formulas used to calculate the stress in the curved beam are numerous as the neutral axis needs to be found, the moments calculated and the stresses determined for the top of the eyelet as well as the sides. Below is the general formula used to determine the stress, detailed calculations can be found in Section 4.4 of Appendix B.

$$\sigma_{le} = \frac{M_C \cdot (\frac{h_{eye}}{2} + e)}{h_{eye} \cdot w_{eye} \cdot e \cdot (\frac{R_C}{2})} \quad \text{Eq 3.53}$$

Where σ_{le} = Stress in the little end bearing eye

M_c = Bending moment in the little end bearing eye

h_{eye} = Eyelet wall thickness

w_{eye} = Eyelet width

e = Distance between section centroid and neutral axis

R_c = Distance between neutral axis and extreme fibres

To verify the Shigleys based calculations and to cover other areas not able to be verified by hand an F.E.A. analysis over a number of load cases will be performed on the conrod with the results applied to the fatigue analysis discussed below. The results and any possible design revisions will be discussed in the next chapter.

3.3.2 Fatigue Analysis

To perform a fatigue analysis for the fluctuating loads the first steps are to calculate the alternating stress and the mean stress using the equations below,

$$\sigma_m = \frac{1}{2}(\sigma_{max} + \sigma_{min}) \qquad \text{Eq 3.54}$$

$$\sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{min}) \qquad \text{Eq 3.55}$$

Where

σ_m = Mean stress at a point in the conrod over one cycle.

σ_a = Alternating stress at a point in the conrod over one cycle.

σ_{max} = Maximum stress at the point of interest during a cycle

σ_{min} = Minimum stress at the point of interest during a cycle

Since the conrod undergoes fluctuating loads of both tension and compression, it is not difficult to see that the dominating stresses will vary with respect to position in the conrod. During the engine cycle the conrod is being either pushed against or pulled by the piston. The compressive (pushing) forces pass through the bottom of the little end, along the pillar and into the crankpin by the top of the big end. At no time do any forces pass through the top of the little end or the bottom of the big end, thus these areas of the conrod do not undergo any stress when the conrod is in compression. However during the exhaust and intake phase particularly at top dead centre the conrod is in tension and thus the sections A, B & C are in tension (refer figure below). This means that during a cycle section “A” stresses cycle from tension to compression however sections B & C are only in tension or zero stress.

The application of the equations 3.54 & 3.55 above remain the same however the final required fatigue analysis may differ.

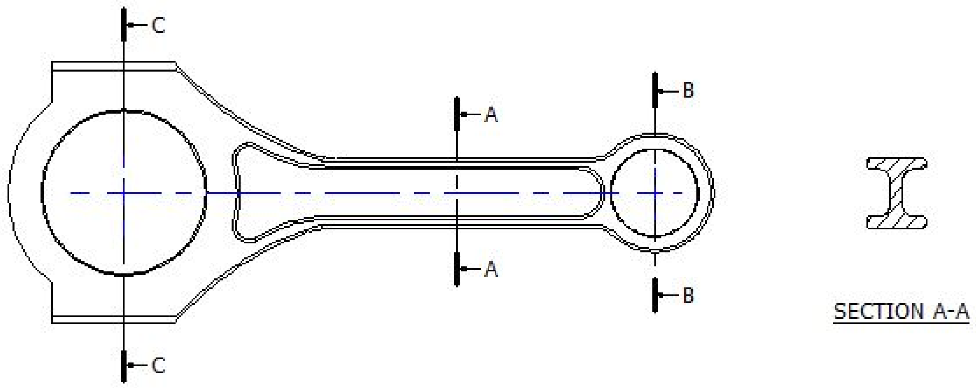


Figure: 3.13: Conrod Design Revision A

Once the stresses have been determined it is important to check if the mean stress is in tension or compression, should the mean stress be compressive there is no need to develop any fatigue criteria (Shigley,2012). Checking to see if the maximum stress does not exceed the yield limit and the alternating stress does not exceed the endurance limit is all that is required with respect to fatigue. The yield stress is a material property and referring to the material data sheet is all that need be performed. The endurance limit however requires calculation and is based on the operating conditions, certain geometries and the reliability required. The formula to calculate the endurance limit can be found below.

$$Se = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_e' \qquad \text{Eq 3.56}$$

Where Se = Endurance limit (MPa)

k_a = Surface condition modification factor

k_b = Size modification factor

k_c = Load modification factor

k_d = Temperature modification factor

k_e = Miscellaneous effects modification factor

S_e' = Rotary beam test specimen endurance limit. (MPa)

Should the alternating stress exceed the endurance limit, fatigue failure is likely to occur in less than 10^6 cycles. Refer to Shigleys,2012 for how to determine each of the above factors.

For points on the conrod where the mean and stress is tensile will require a fatigue analysis to determine if the conrod will fail. The first step will be to determine if the part will fail due to overloading, this can be determined by the equation below,

$$\sigma_{max} < \sigma_{y\ 4340} \qquad \text{Eq 3.57}$$

Where $\sigma_{y\ 4340}$ = Yield stress of the conrod material

Assuming the maximum stress is less than the yield stress the fatigue factor of safety needs to be calculated. The equation for this is taken from (Shigleys,2012, table 6-7, p307) and is the ratio of the combined alternating and mean stresses vector to the projected failure vector along the same path (refer to Shigleys Example 6-10, p308 for more detail).

$$nf = \frac{1}{2} \cdot \left(\frac{Sut}{\sigma_m}\right)^2 \cdot \left(\frac{\sigma_a}{Se}\right) \left[-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_m \cdot Se}{Sut \cdot \sigma_a}\right)^2}\right] \qquad \text{Eq 3.58}$$

Where nf = Fatigue factor of safety

Sut = Minimum ultimate stress limit of the material

For the conrod to possess an infinite service life, "nf" will need to be > 1 and preferably greater than 2 as fatigue analysis is not precise and thus requires the use of good margins of safety particularly if the consequence of failure is high.

3.3.3 Buckling Analysis

As an additional design check the force required to buckle the conrod under compression will be checked to ensure the conrod will not fail on the first cycle. The following equation is used to determine this and is taken from (Roark's, 2012).

$$F_{c(max)} = K_{yy} \cdot \frac{(\pi \cdot E_{4340} \cdot I_{yy})}{L_c} \qquad \text{Eq 3.59}$$

Where K_{yy} = Conditions constant (Roark's,2012, Table 15.1)

E_{4340} = Elastic modulus of the conrod material (4340).

I_{yy} = 2^{nd} Moment of area for the respective direction

L_c = Conrod length between fixities.

Detailed calculations of the above can be found in Section 4.4 of Appendix B and discussed in detail in the next chapter.

3.3.4 F.E.A.

Details and discussion of the F.E.A. performed will also be given in the next chapter. For an example of the applied F.E.A. refer to the figure below.

This figure represents the solid model, mesh, load glyph and direction for the load case of "maximum compressive load". The same solid body and mesh are used in the next load case of "maximum tensile load" (not shown).

Example Load Case:

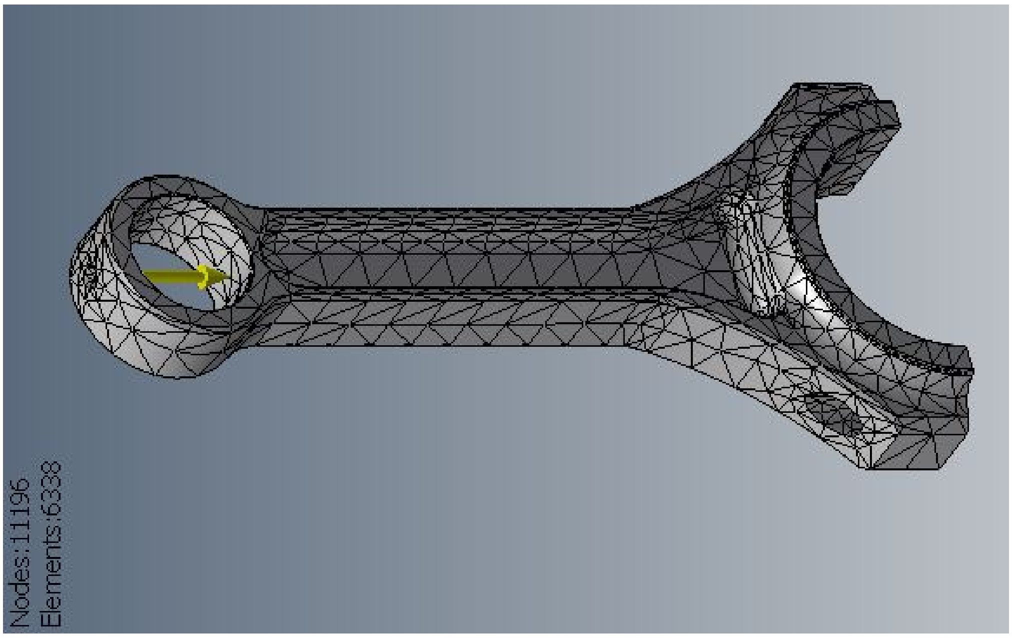


Figure 3.14: **Conrod FEA load case , 52.3kN (max compressive force),**

Desc: Max Cylinder Pressure

Position: 8° ATDC

Load: 52.3 kN

Loading Location: Bottom of little end bearing eyelet aligned with pillar axis

Constraints:

1. Frictionless pin constraint at the big end bearing
2. Rotationally constrained

Model Components:

1. Conrod (no bearing cap)

Mesh Elements:

1. Tetrahedronal Elements

Total Elements: 6338 Elements

Element Size: 0.05 (avg) of component size

The results of these load cases (ie: signed von-mises stresses) are the stress values used in the fatigue analysis discussed above.

3.4 Bearing Design

There is a variety of bearings used in this engine design which can be divided into two categories, plain & rolling element. The work required for our project here is the validation of the original bearings specified in the design package provided.

The main considerations will be maximum capacity & bearing life for the rolling element bearings used on the counter balance shafts. For the plain bearings such as the main bearings and conrod bearings the primary consideration will be the operational mode (ie: Hydrodynamic or Boundary Lubrication)

3.4.1 Main Bearings

The main bearings were a specified component in the package provided by Oxford University's Prof. Stone so the work required is to validate the design and modify if necessary. The bearings specified were a modified DU bearing (lead PTFE); made by GB bearing company and to be modified by the fabricators of the engine.

Modifications included an oil port to allow the supply of engine oil to the bearing and an oil groove to allow the distribution of the oil across the journal. The purpose of this is to turn the DU bearing which is a plain bearing into a hydrodynamic bearing. Thus allowing greater surface speeds and therefore engine RPM plus load capacity and thus a stronger bearing arrangement.

Before any work begins the existing design provided by Prof. Stone was adjusted slightly to the nearest metric size from the imperial size specified. The next change was a small increase in width due to the off the shelf availability of that particular length bearing. Otherwise the original design remained unmodified.

To validate the design it was chosen to calculate if the bearings could carry the maximum load produced by the engine without bottoming on the bearings shell. As a further check using the DU bearing catalogue the plain bearing was design checked to see if it could operate without an oil supply and thus in a boundary lubrication state (ie: residual oil remaining only).

The equation to calculate this is called the PU factor (also known as pressure velocity),

$$PU = P \cdot U \qquad \text{Eq 3.60}$$

Where P = Bearing pressure in (MPa)

U = Journal relative speed (m/s)

The acceptable boundary lubricated PU factor for an oil lubricated DU bearing is not less than,

$PU =$ 3.5 MPa.m/s (intermittent operation)

The actual PU factor will be discussed in the next chapter and detailed calculation can be found in Section 4.2 of Appendix B.

3.4.1.1 Hydrodynamic design check

The first design verification will be to ensure the bearing goes into hydrodynamic mode during operation. This will be determined by the oil film thickness occurring at full load (maximum cylinder pressure), should the oil film reduce to below the calculated minimum the bearing will be considered as failing under full load and not be dimensionally acceptable. The following equations and procedure are taken from (Shigleys,2012).

To determine the oil film thickness the Sommerfeld number must be determined, the equation for this is,

$$S = \left(\frac{r}{c}\right)^2 \cdot \left(\frac{\mu \cdot N}{P}\right) \qquad \text{Eq 3.61}$$

Where	S	= Sommerfeld Number (Dimensionless)
	r	= Journal radius (mm)
	c	= Bearing radial clearance (mm)
	µ	= Lubricant dynamic viscosity (Pa.s)
	N	= Engine Speed in Rev/s
	P	= Nominal Bearing Pressure

To calculate the nominal pressure "P" we use the formula below,

$$P = \frac{W}{2 \cdot r \cdot l} \qquad \text{Eq 3.62}$$

Where	W	= Maximum bearing load.
	l	= Bearing length (mm)

Once the Sommerfeld number is calculated we apply the chart (Shigleys, 2012, Figure 12-16, p636) to determine the minimum film thickness ratio and thus the minimum film thickness. Addition details on this chart are the minimum friction point and maximum load point, the optimum design aimed for is to achieve a value that lies between these two limits.

The clearance determined from the application of Shigley's Figure 12-16 must satisfy the equation below for the minimum allowable oil film thickness.

$$h_o \geq 0.00508 + 0.00004 \cdot D \qquad \text{Eq 3.63}$$

Where h_o = Minimum allowable oil film thickness.

D = Bearing journal diameter (mm)

In doing so the bearing is considered to operating in full hydrodynamic mode whilst under full operating load. This check needs to be performed at a range of engine speeds to ensure the bearing doesn't bottom out at low speeds.

3.4.2 Big-End Bearing Design

The big end bearing was not included in the design package provided and therefore will need to be designed from scratch. Research into these bearing found a range of materials used the most common being white metal or babbit. Another material used among racing engines is aluminium bronze for it high strength and resulting pressure velocity which is essential should the bearing go into boundary lubrication mode under adverse conditions.

Aluminium bronze was selected for the initial design as it can be easily machined into the required shape from bar stock and is freely available. A white metal bearing would require fabricating a backing shell and finding an engineering company to plate the shell with babbit.

For the big end bearing design check will use a similar methodology to the main bearings with one significant difference. The big end bearing relative velocities (bearing velocity with respect to journal velocity) will require determination.

Using the instantaneous centre method it can be found that the maximum angular velocity of the conrod is at TDC (& BDC). At this point (and BDC **only**) the conrod angular velocity can be defined as,

$$\omega_{co} = \frac{r}{l}\omega_{cr} \qquad \text{Eq 3.64}$$

Where ω_{co} = Conrod angular velocity

ω_{cr} = Crankshaft angular velocity

At TDC & BDC where the conrod force approaches a relative maximum the conrod/crankshaft relative angular velocity approaches a maximum or minimum depending on whether it is at TDC or BDC respectively. It is here that we shall determine the dimensional adequacy of the big-end bearing.

The relative angular velocity between the crankpin, big-end bearing and the load at these points is,

$$\omega' = \omega_{cr} \pm \omega_{co} \pm \omega_l \qquad \text{Eq 3.65}$$

Where ω' = Relative angular velocity

ω_l = Angular velocity of the load

The angular velocity of the load in this analysis shall be ignored for simplicity, once the relative angular velocity is found it can be substituted into Eq 3.61 and the process continued same as the main bearing. For the results of the conrod analysis and the required dimensions of the big end bearing refer to the next chapter.

3.4.3 Little End Bearing Design

Since the wrist pin remains stationary the relative velocity between pin and little end bearing is harmonic or oscillating and not very high. As a result of this it is assumed the bearing will function as a plain bearing and the contact surfaces will interact in boundary lubrication.

Using sintered bronze as the bearing material the design verification will involve checking the pressure velocity and compressive stresses do not exceed the material's maximum. Firstly we will determine the maximum compressive stress in the bearing. Using the following equation (Shigleys,2012) for maximum pressure across the journal diameter we can determine if the maximum compressive stress is exceeded and also apply this value to determine the pressure velocity.

$$P_{max} = \frac{4 \cdot F_{c(max)}}{\pi \cdot 2 \cdot r \cdot l} \qquad \text{Eq 3.66}$$

Where P_{max} = Maximum pressure across bearing diameter (MPa)

$F_{c(max)}$ = Maximum conrod force (N)

So at TDC during the burn phase where cylinder pressure is nearly at a maximum and the conrod angular velocity is a maximum the highest pressure velocity (or approximately so) on the little end bearing can be found using the following,

$$PV = \omega_{co} \cdot r_{wp} \cdot P_{max} \qquad \text{Eq 3.67}$$

Where PV = Maximum pressure velocity (MPa.m/s)

r_{wp} = Wrist pin radius (m)

This is a simplified approach to the little end bearing design however if the bearing material meets these conditions it will operate comfortably. For the analysis results and discussion refer to Section 4.2 of Appendix B and the next chapter.

3.4.4 Counter Balance Shafts

The bearings on the counter balance shaft have also been specified in the Stone design package. Like the main bearings the counter balance shaft bearings will verified for load capacity and in addition their expected operating life will be determined.

All calculations have been taken from the SKF bearing general catalogue and thus are verified by the manufacturer.

The gear/pinion side bearings are specified in the detail drawings as 6205 deep groove ball bearings and the floating side bearings are N205 roller bearings. This arrangement suits speeds of up to 5000 rpm in oil lubrication and temperatures up 120°C according to the SKF catalogue. These parameters are within the engine operating range however does limit the engine to 2500 RPM running speed.

To calculate the load on the bearings we must determine the force created by the balance masses when rotating. The design speed of the engine is 2500 RPM and the force can be can calculated using the following formula

$$F_{bs} = M_b \cdot \omega^2 \cdot r \qquad \text{Eq 3.68}$$

Where F_{bs} = Balance mass radial force

M_b = Balance Mass

ω^2 = Rotating Speed (Rad/sec)

r = Mass offset distance

Note that this force has already been calculated as part of the balancing calculations for both primary and secondary balance shafts.

Once the forces have been determined the bearing lives can be calculated. Using the following formula the L90 bearing life can be calculated.

$$L_{90} = \frac{10^6}{60 \cdot RPM}\left(\frac{C}{Fbs}\right)^{\frac{10}{3}} \qquad \text{Eq 3.69}$$

Where L_{90} = Bearing life in hours before the onset of wear.

The detailed calculations can be found in Section 4.2.1 of Appendix B and the discussion of results can be found in the next chapter.

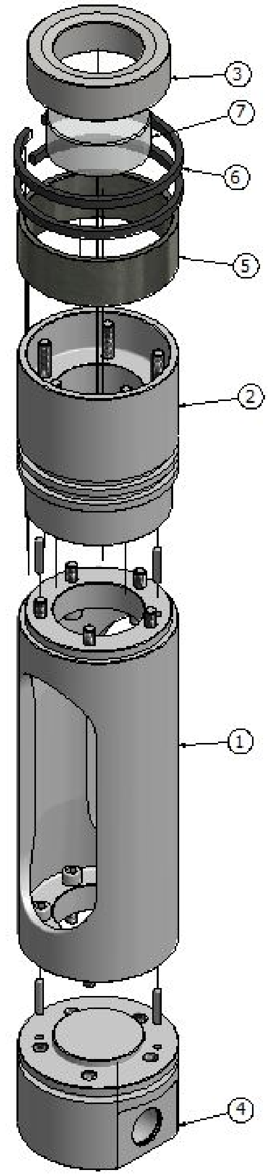
3.5 Piston Design

Modern optical access engines nearly always use a bowditch piston as part of the optical access components, this is certainly the plan for the USQ engine as previously mentioned. With this design consideration at a conclusion the finer details such as dimensions, materials and general construction need be sorted.

The piston is an essential part of the engine design and unfortunately one of the parts with the least available literature on design. To deal with this the design plan is to utilize FEA extensively to validate the design with respect to strength. The functionality component of the design does not require calculation as the interaction of the piston with its mating parts is a dimensional issue not one of strength or fatigue. Therefore functionality shall only be briefly discussed before the details of the FEA.

Below is an isometric figure of the piston assembly broken into parts, as discussed the piston is a bowditch type piston having an elongated body opened in the middle for mirror placement and a hollow head for a transparent crown.

Material selection and functionality design proved to be the most complex aspect of the piston design. Extensive research was performed with respect to several parts and in the end certain details required thoughtful assumption in the design and acceptance that redesign may be required at a later date.



An example of this is the decision to make the piston in three parts with the window being retained by a collar fastened from underneath the piston head. This was to eliminate any fasters from the combustion chamber and give a piston crown with fewer irregularities. This however was at the sacrifice of 2-4mm of window diameter which may not have been worthwhile in the mind of the end user.

Another example of design assumption is the lowering of the rings and guide sleeve to below the optical bore depth. The fact that rings crossing between the metal sleeve and optical bore when operating would cause rapid damage is clear. However low friction P.T.F.E. composite rings should be able to operate on the transparent bore permanently so the decision ended up being more about the optical bore size than a piston design issue. The simpler and probably cheaper design was a short optical bore with the rings lowered as shown. Once again the end user may have preferred the alternative approach and redesign may be required.

Figure 3.15: Bowditch piston model (refer to glossary for part names)

3.5.1 Piston material:

Research into the piston material was extensive with the best sources proving to be the ASME Handbook Vol 2 and piston manufacturer catalogues such as CP Carillo & Wiseco. The two materials found to be most commonly used are aluminium alloys 2618-T6 for high performance and 4032-T6 for general engines. Either material would be acceptable in this design.

The availability of these materials in Australia in billet stock may prove to be a significant challenge. It is possible USQ may need to cast the material for machining from existing pistons and perform materials testing in their laboratories to ensure the final alloy meet the requirements.

It is also important to note that regardless of what material alloy is used the piston will require final heat treatment before finish machining.

3.5.2 Rings and Sleeve Material:

Unlike typical I.C. engines the majority of the piston will not be lubricated. Oil lubrication must stop below the piston extension to prevent leakage. As a result the upper piston must be guided using lubricant free materials, additionally the rings must also run dry and therefore need to be suitable for lubricant free operation. Additionally these materials and particularly the rings need to be heat resistant.

Various SAE research papers such as (Catapano F et al., 2011) discuss the use of various P.T.F.E. composite materials. The final selections made were Tetron C for the rings (P.T.F.E. /carbon composite) and Tetron B for the guide sleeve (P.T.F.E. / bronze composite). Both materials have a continuous operating temperature of 260°C and a 3-4hrs operating temperature of 310°C.

An important note regarding operation is the rings will only have a total life of 3-4hrs at 310°C not accounting for friction wear. It is expected the rings will require frequent replacement due to the harsh operating conditions.

3.5.3 Window Material:

Through extensive research it was found the only material used for the optical access components (both the piston window and optical bore) was quartz or specifically fused silica. Some literature loosely termed the material as glass however reading further in, the material was later specified as quartz glass. This material is available in various grades categorized by composition and production process. Certain grades possess superior optical properties and most grades possess similar mechanical properties. The material will require specialist machining and polishing as drawn plate is not available in the material thickness required. Final material selection will be subject to the required optical properties and availability; Heraeus Quarzglas HSQ-300 has been selected initially for calculations.

The maximum stress in the window and thus a thickness can be calculated using the following equations (Roark's,2012),

$$\sigma_{max} = \frac{6 \cdot M_c}{t^2} \qquad \text{Eq 3.67}$$

$$M_c = \frac{q \cdot a^2 \cdot (3+v)}{16} \qquad \text{Eq 3.68}$$

Where	σ_{max}	=	Max stress in the window
	M_c	=	Bending moment per unit length
	t	=	Plate thickness
	a	=	Plate radius
	v	=	Poisson's ratio
	q	=	Applied uniform pressure

3.5.4 FEA (Load cases):

Utilizing the forces calculated during the balancing and crankshaft design at least two load cases shall be establish to apply to the piston assembly and test for adverse results. Of interest will be the existence of stress concentration points for the purpose of fatigue analysis, such points will be where fractures propagate from. Also of particular interest will be the length reduction or "Squish" of the piston due to the applied cylinder pressures and additional length over a typical piston.

Another outcome of interest with be the lateral deflection of the piston skirts particularly around the centre of the extension piece. Should this deflection exceed the allowed clearance in the cylinder liners the piston will bind against the cylinder walls causing damage to the piston and appreciable operating resistance.

3.5.4.1 Load Case One

Desc: Max Cylinder Pressure

Position: 8° ATDC

Load: 10.511 MPa

Loading Location: Uniformly across piston crown.

Constraints:

1. Frictionless cylinder wall at lower piston skirting
2. Frictionless cylinder wall on guide bush
3. Fixed wrist pin bore

Assembly Components:

1. Lower Piston
2. Piston Extension
3. Piston Head
4. Quartz Window Crown
5. Optical Window Collar
6. Guide Bush/Collar

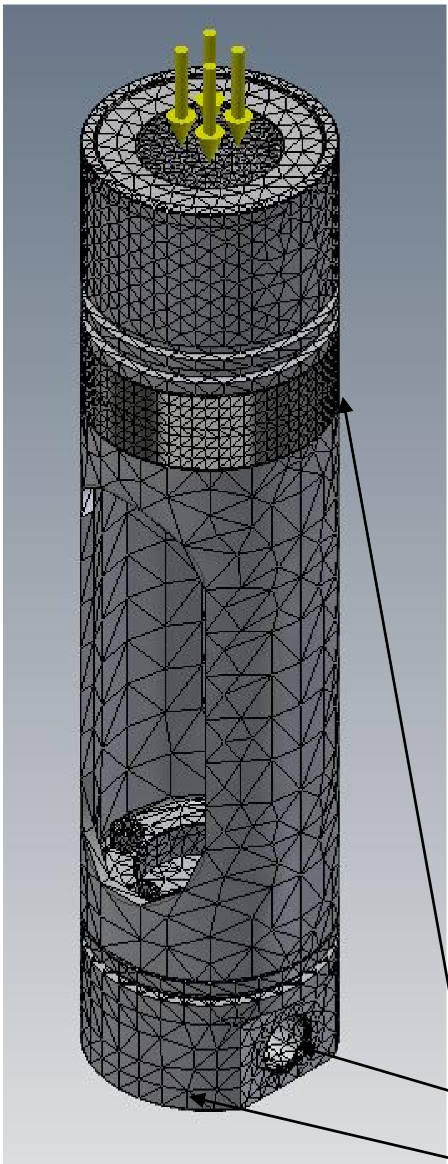


Figure 3.16: Bowditch piston FEA model (Load case one)

Mesh Elements:

1. Tetrahedronal Elements (Most Common)
2. Quadrahedronal Elements (Least Common)

Total Elements: 83139 Elements

Element Size: 0.1 (avg) of component size

Total Nodes: 138330 Nodes

3.5.4.2 Load Case Two

Desc: Max Inertia Force

Position: 360° ATDC

Load: 10.81 kN

Loading Location: Lower half of wrist pin bore

Constraints:

1. Frictionless cylinder wall at lower piston skirting
2. Frictionless cylinder wall on guide bush
3. Fixed piston crown

Assembly Components:

1. Lower Piston
2. Piston Extension
3. Piston Head
4. Quartz Window Crown
5. Optical Window Collar
6. Guide Bush/Collar

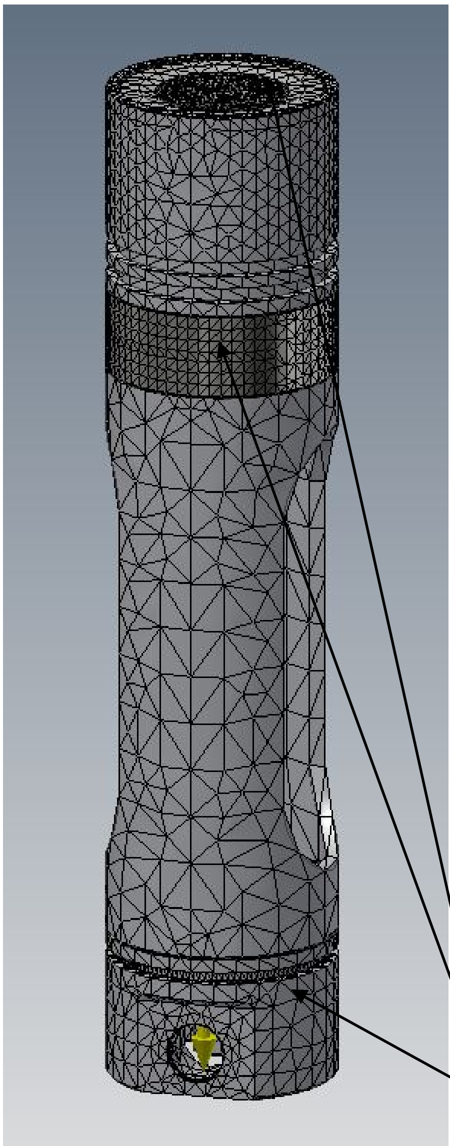


Figure 3.17: Bowditch piston FEA model (Load case two)

Mesh Elements:

1. Tetrahedronal Elements (Most Common)
2. Quadrahedronal Elements (Least Common)

Total Elements: 83549 Elements

Element Size: 0.1 (avg) of component size

Total Nodes: 140483 Nodes

The results of the piston analysis can be found in the next chapter, for detailed calculations and results refer to appendix…

3.6 Flywheel Design

Final flywheel design requires knowledge of the load to give efficient and effective design, since the operation details of the engine are as yet unknown with respect to skip firing, load testing etc. it has been decided to use the original 50kg Stone designed flywheel and assume that it will be sufficient to store and release energy during the power and dwell phases of the engine cycle.

Should the engine be operated in a skip fire routine and hence be driven by an electric motor a flywheel will most likely prove unnecessary or at least not require the same moment of inertia as the existing.

3.7 Optical Bore Design

It is expected that the optical bore will require further design revision based on feedback from USQ as to what features they would like in their final engine design. Another influencing factor is the requirements of the diagnostic equipment to be used on the engine, something the University will play a role in.

Probably the main design influence on the optical bore will come from the cylinder head and its intended use. Parts such as head gaskets and head bolts will have control over the optical bore dimensions as will the cooling circuits of the head to name a few.

It was decided that the optical bore cannot be properly designed until the final engine head was selected, assuming a commercial head is to be used. The parts that make up the optical bore have been modelled and drawn to invite feedback from the end users.

It is expected that the next designer will adjust these designs to suit both the University's needs and the cylinder head.

3.8 Drafting and Communication

Part of the design methodology is the application of technical drawings to deliver the end design. As mentioned previously drawings are provided in the appendix, specifically Appendix C for key assembly and part drawings and Appendix D for the complete design.

To ensure the drawings can be interpreted by all stakeholders (academic staff, engineering trade staff, contractors, suppliers, students and external interested parties in the event of publication) the drawings need be produced to a standard. The relevant Australian Standard for technical drafting in this project is AS1100.101 General Principles & AS1100.201 Mechanical engineering. These codes will be used as reference material with respect to drawing layout, dimensioning, geometrical tolerancing, surface finish notation etc. all drawings so far have been produced to these standards.

Another Australian standard to be used in the production of these drawings is AS1654.1 (ISO system of limits and fits) for use in controlling part sizings and fits. This standard will be particularly necessary when the dimensioning phase of the drafting takes place. This is to ensure all parts not only fit and operate together but all replacement parts if required will mate to their respective parts.

Chapter 4

Results & Discussion

This chapter covers the results and discussion of the theoretical and virtual analysis discussed in the previous chapter. The results for the engine balancing, crankshaft design, piston design, conrod design bearing calculations etc. Discussion shall include the results of any analysis performed and reiterations of calculations, designs and simulations should any first round of results prove unsatisfactory.

As mentioned previously the focus of this paper is not to design an engine as this has already been performed by Prof. Stone of Oxford University. This dissertation focuses more on design verification of the engine for the intended application.

At the end of this chapter the reader will know how balanced the engine is, the final stresses in the major components, any design changes made as a result of failed parts and the engine's operating limits.

Specific work includes:

- Engine Balancing: Simulation vs Theory Results, Crankshaft Balancing Results, Final Balancing & Results.
- Crankshaft: Design Paper & FEA Results for Applied Load Cases & Discussion, Redesign & Results
- Conrod: Hand Calculation & FEA Results for Applied Load Cases, Redesign & Results.
- Bearings Results of calculations (oil film thickness, bearing life)
- Piston: F.E.A. Results from Applied Loads Cases, Redesign & Results.

The engine balancing shall be broken into two sections found at the beginning and end of this chapter. The first balancing discussion will cover the theoretical engine operation force calculation results and the virtual operation results. In the final balancing section the required balance masses will be given using the theory established so far then validated using virtual simulation. The reason for dividing this section in two is that essential components such as the piston and crankshaft may require design modifications and thus mass changes may result. For continuity in this report the methodology, results and discussion will attempt to follow the same order as the design process itself.

4.1 Engine Balancing Results (Simulation vs Theory)

As mentioned in the previous chapter the piston/crank mechanism and balance shaft cluster was turned over in a virtual simulation environment. The reaction forces between the crankshaft and main bearings along with the reactions between the balance shafts and their respective bearings were recorded with respect to time and plotted. Two simulations were run, firstly an initial run on the draft design as mentioned in the previous chapter to validate both the simulation results and the theory. Secondly a simulation on the final design after all design modifications were complete and balanced using the theoretical model. The initial run will be discussed here and the final run discussed at the end of this chapter.

4.1.1 Slider-Crank Simulation

Referring back to Figure 3.3 in the previous section it can be seen that the plot below for the vertical forces [Y] is the inverse and hence reaction plot of Eq 3.11. Note that it is not an exact inverse plot of Eq 3.11 but very close as these are the virtual simulation results. Their similarities to the theoretical results will be discussed below after Figure 4.2.

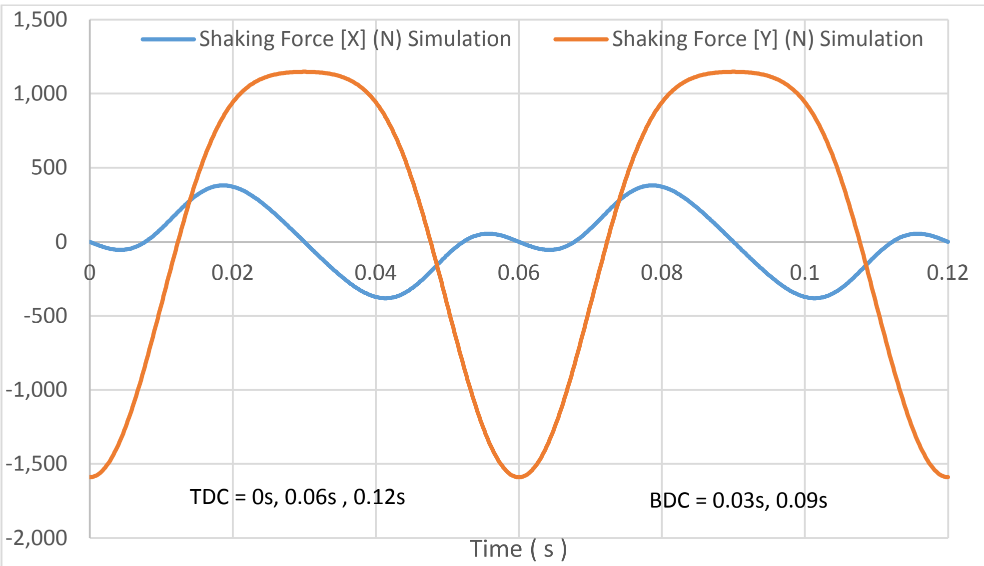


Figure 4.1: First run virtual simulation plot (Main Bearings Reaction Forces: 1000RPM)

Not given in any plot previously is the horizontal forces [X] show above. This plot is the horizontal reaction forces acting at the main bearings. These results will be discussed later as the sum of the inertia torque force and the force due to the imbalance in the crankshaft.

The horizontal force component of the shaking torque represented in Figure 3.4 cannot be removed or countered due to the slider-crank geometry. Its equal and opposite force acting through the little end bearing means that there is no contribution to net shaking force.

Since we can define the horizontal forces acting on the main bearings by components we can validate the theory using the simulation and reduce its magnitude as best as possible.

Below is the combined plots of the simulation plot above and the calculated main bearing reactions. Using Eq 3.11 to define and plot the vertical shaking forces and thus the vertical bearing reactions we get Theoretical Force [Y] below. The sum of Eq 3.17 for the crankshaft out of balance & Eq 3.15 divided by the piston position we get the plot Theoretical Force [X] below.

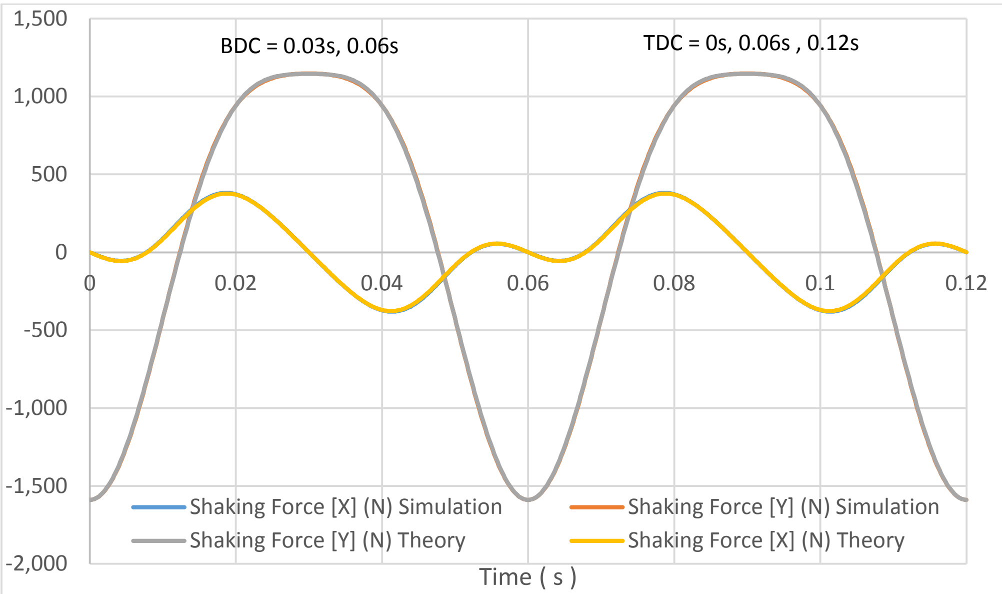


Figure 4.2: First run theoretical plot + simulation plot (Main Bearings Reaction Forces: 1000RPM)

It is clear to see that the differences between the theoretical results and the virtual simulation are indistinguishable on a plot of this size. Indeed the differences are difficult to notice even on a much larger plot.

The plot below is the differences between the simulation results and the theoretical results for both the horizontal [X] forces and the vertical forces [Y]. It is important to note that the difference plot is not an error plot; an error from the true values could occur in both data. For example if the crankshaft in the simulation model is not precisely aligned with TDC at 0 seconds then all simulation results will be slightly out of phase to the true values. Similarly an incorrect mass entered into the theoretical model will alter the magnitudes of the given values.

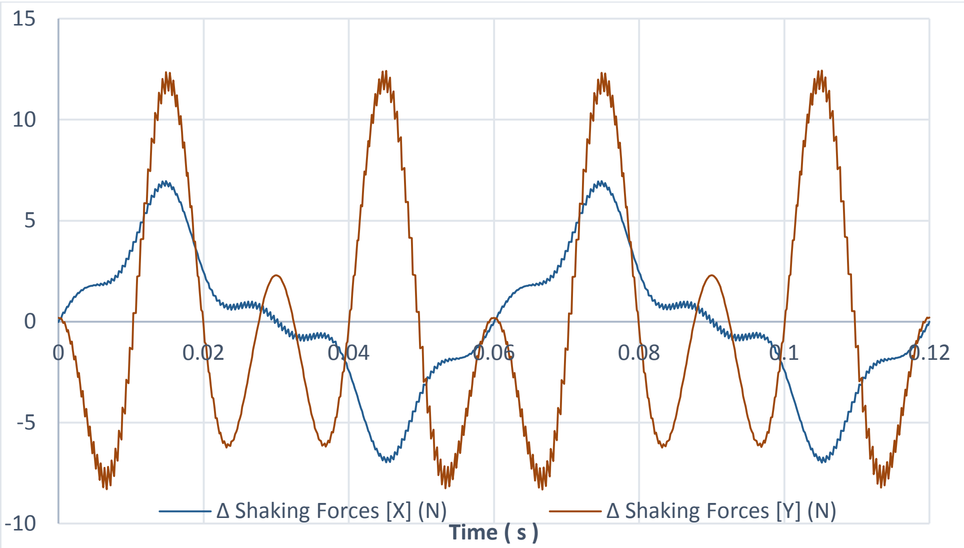


Figure 4.3: First run error/difference plot (Main Bearings Reaction Forces : 1000RPM)

The above plots in Figure 4.3 shows the presence of first order and second order frequencies in the difference plot. It is likely from this that differences in results occur in the reciprocating mass results, however it is difficult to determine which results if not both are the cause of these differences and will not be investigated here. It is likely though that a slight error in the reciprocating mass used in the theoretical model has occurred. An inaccuracy in rotating mass results (crankshaft assembly) could also be present and would affect both the horizontal & vertical forces but only at the primary frequency.

The fact that the plots 4.2 & 4.3 are in phase (think of the second order forces for [Y]) although opposite in magnitude at times suggests that the errors or a sizeable portion of the errors as implied in the previous paragraph are a result of incorrect masses in one of the models. Small differences due to setup flaws in the simulation model cannot be ruled out though.

4.1.2 Simulation vs Theoretical Conclusions:

The overlay plots in Figure 4.2 firmly validate both the theory and simulations giving confidence to the future results of the final balanced engine. The differences plot in figure 4.3 although demonstrates differences of both first order and second order frequency, it also demonstrates the difference magnitudes to be very low. With a maximum difference of 3% of the reciprocating force [Y] between theory and simulation occurring at 90° & 270° from TDC and a maximum difference of <0.8% with respect to the total maximum reciprocating force. This means the use of either method to calculate and correct the engine balance is quite acceptable. It is intended that the theoretical model be used to determine the required design changes needed for balancing and the simulation to validate such results once the changes to the model have been made.

4.1.3 Balance shafts:

Due to the simplicity of the balance shafts their simulation and theoretical results will not be discussed here. Their application and results in the final balancing will be discussed at the end of this chapter.

4.2 Crankshaft Design Results

As discussed in chapter three the crankshaft design was analysed using two methodologies, hand calculations using a design paper specifically drafted for crankshaft design and a finite element analysis based approach using the determined stresses to perform a traditional fatigue analysis from established texts (ie: Shigleys,2012).

The results for the design paper approach shall be given first followed by the FEA approach. Once both results have been given any required changes to the crankshaft design will be made and their results discussed.

4.2.1 Design Paper Results

The following table covers the stresses determined using the methodology laid out in the design paper (Germanischer Lloyd, 2012). All stresses are Von-mises stresses taken from the same listed location.

Location	Max Stress σ_{max}	Min Stress σ_{min}	Alt Stress σ_m	Mean Stress σ_a	Factor Q
Crankpin Jrn. Fillet	619MPa	-130MPa	374MPa	244MPa	1.057
Crankpin Oil bore	17.1MPa	-442MPa	229MPa	-212MPA	1.712
Main Bearing Jrn. Fillet	199MPa	-572MPa	385MPa	-186MPa	0.98

Table 4.1: Crankshaft stress hand calculation results (Design revision A)

The calculations performed to determine these results can be found in Section 4.1 of Appendix B. The results show that the crankshaft is under considerable stress during the burn phase particularly during 0°ATDC and 20°ATDC with the maximum stress occurring in the crankpin journal fillet at 8°ATDC. This corresponds to the maximum cylinder pressure of 10.511MPa occurring at 8°ATDC.

Simply put the maximum stress is occurring at the stress raiser in the crankpin journal when the bending moment is at a maximum.

The second highest stress occurs in the main bearing journal fillet at the same crank angle. What needs to be noticed about these results is that the highest stresses are geometrically opposing each other in the crankshaft web (crank arm). From this it can be reasoned that the bending moment occurring in the web is the main contributor to the maximum stresses with the respective journal fillets acting as stress raisers. Confirmation of this can be sort by viewing Eq 3.40 & Eq 3.41 in chapter three and Appendix B.

4.2.2 Finite Element Analysis Results

A number of load cases were tried to find the maximum stresses occurring the crankshaft as discussed in the previous chapter. Taking the maximum cylinder pressure, the resulting conrod force and the position at which this occurs the first load case was performed using the FEA software. An image of the stressed model for load case one can be seen in the figure below.

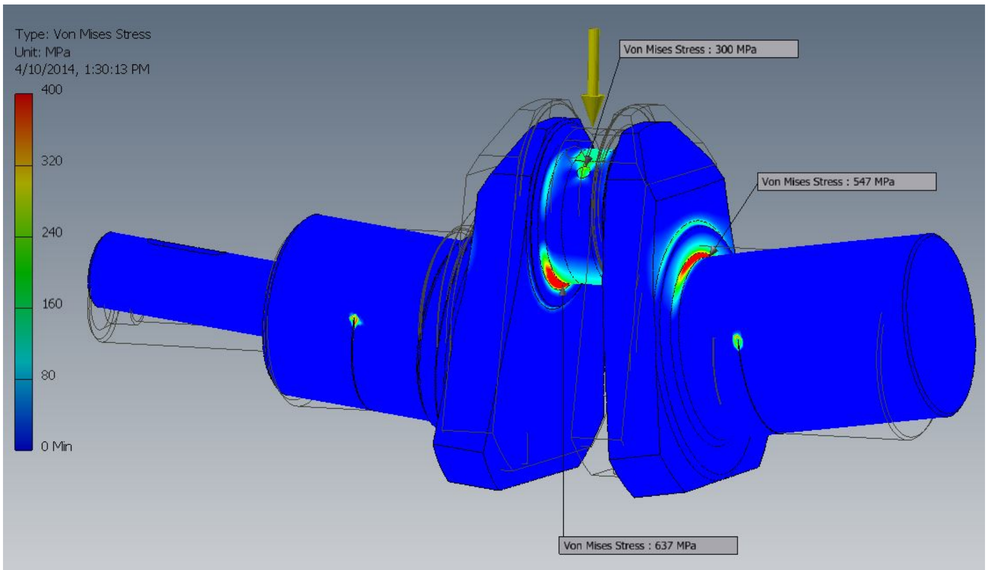


Figure 4.4: Crankshaft stress FEA load case one (8°ATDC, 10.511MPa cylinder pressure (Design revision A)

The resulting stresses are given in table 4.2 below however it can be seen that the assumptions given by the design paper are validated with respect to the maximum stresses occurring in the crankpin journal, crankpin oil bore and main bearing journal fillets. The stresses occurring in the centre of the main bearing journals need to be ignored as these represent the model support points. Such points correspond to supports in a free body diagram and do not exist in reality as the journals are supported by 40mm wide bearings.

The next load case is where the minimum stresses were found to occur, or more specifically the maximum opposing stresses to the maximum stresses found in load case one. This load case is found to occur at 360°ATDC where no cylinder pressure is present to counter the maximum inertia forces due to the piston mass and engine speed. An image of the stressed model for load case three can be seen in the figure below.

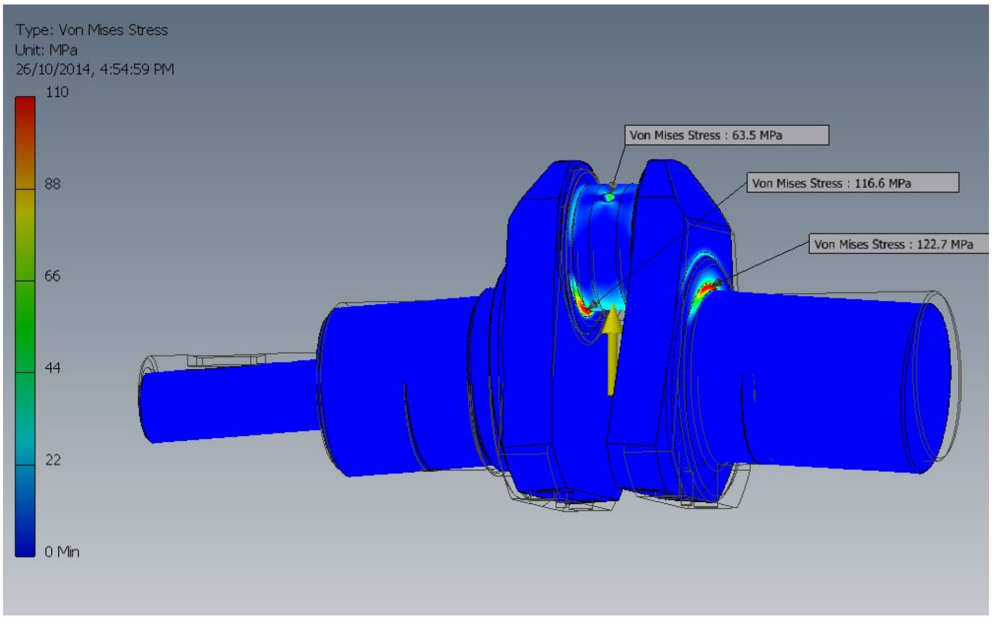


Figure 4.5: Crankshaft stress FEA load case three (360°ATDC , 10.81kN vertical force) (Design revision A)

It can be seen the same locations contain the maximum stresses as load case one however the signs have all changed (eg: tension to compression or vice versa).

The following table covers the stresses determined using finite element analysis (FEA), all stresses are Von-Mises stresses taken from the same location probe. (eg: the exact same point in the same location). This gives more exact results as opposed to the maximum and minimum results across the same location.

Location	Max Stress σ_{max}	Min Stress σ_{min}	Alt Stress σ_m	Mean Stress σ_a
Crankpin Journal Fillet	637MPa	-117MPa	377MPa	260MPa
Crankpin Oil bore	63MPa	-300MPa	182MPa	-118MPa
Main Bearing Journal Fillet	123MPa	-547MPa	335MPA	-212MPa

Table 4.2: Crankshaft stress FEA results (Design revision A)

Applying the above results for the crankpin journal to equation 3.58 from the conrod design section in chapter three shown below,

$$nf = \frac{1}{2} \cdot \left(\frac{Sut}{\sigma_m}\right)^2 \cdot \left(\frac{\sigma_a}{Se}\right)\left[-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_m \cdot Se}{Sut \cdot \sigma_a}\right)^2}\right] \qquad \text{Eq 3.58}$$

We get the result,

$$nf = 0.928$$

Which is a little below the acceptability factor determined using the design paper (GL, 2012). Equation 3.58 uses the ASME elliptic for fatigue failure criterion and is one of the less conservative methods of fatigue analysis, this doesn't mean it is unaccepted method in fact it is the method emphasized by Shigley's.

4.2.3 Discussion of Design Revision A Results

The first result to discuss is that using both methods the crankshaft failed the fatigue analysis for infinite life with mostly similar results for maximum/minimum stresses. This indicates that both methods are valid and either would be acceptable for use here. Since the FEA approach relies on fewer assumptions and is more conservative in determining its factor of safety it will be used for any further analysis required. This is not to say the design paper approach relied on any wild assumptions, it did however rely on a number of geometric assumptions that were mostly but not perfectly met by this design.

Of interest was the closeness of results between the acceptability factor for the crankpin journal in the design paper approach and the factor of safety determined using the FEA approach and ASME elliptic for fatigue. These are essentially the same thing determined using different approaches. This give further confidence in both methods and their respective results.

The next and most important result to discuss is the maximum tensile stress occurring in the crankpin journal fillet. Both methods determined this to be the maximum stress location in the crankshaft. Reiterated below is the determined value and the percentage of maximum yield stress these results give.

Location / Method	**Max Stress** σ_{max}	**Yield Stress** (4340@20°C)	**% of** σ_y	**Yield Stress** (4340@200°C)	**% of** σ_y
Crankpin Journal Fillet (Design Paper)	619 MPa	710 MPa	87.2%	510 MPa	121%
Crankpin Journal Fillet (Finite Element)	637 MPa	710 MPa	89.7%	510 MPa	125%

Table 4.3: Crankshaft stress results with respect to yield stress (Design revision A)

It can be seen that the maximum stresses approach dangerously close to yield for room temperature and well past for elevated temperatures. It should be noted here that the given yield strengths are for heat treated material states provided by the manufacturer based on a 60mm quenched and tempered specimen. Our crankshaft is sized larger and is intended to be nitrided, however, regardless of this, these figures are considered to be the best available and still quite applicable.

Also it is expected that the maximum crankshaft temperature will not actually reach 200°C but for conservative design this is the figure that will be used.

Expecting the FEA result to be the most accurate it is still unwise to design the engine to operate at approximately 90% of the crankshaft yield and to design above the operating temperature yield is simply not done. Also since the fatigue factor of safety is below one the design will eventually fail in fatigue at the crank pin journal.

The conclusion is to redesign the crankshaft with the maximum stress occurring below the operating temperature yield stress limit and with a fatigue safety factor above 1.15. No further results of design revision A will be discussed at this point.

4.2.4 Further Results (Redesign / Design Revision B).

As mentioned above the decision is to deal with the stresses exceeding the material yield & fatigue limits by redesigning the crankshaft. Since the maximum stresses occur in the journal fillets the most appropriate response would be to increase the fillet radii and thus reduce the stress concentration at these points.

An additional response chosen is to increase the crankpin journal diameter, it is expected this will also aid in the reduction of stress in the crankpin journal fillets and aid in countering the lost big end bearing area due to increasing the fillet radii.

Below is the image of the stressed model for load case four. Here the position and load for load case one is repeated using the new crankshaft geometry.

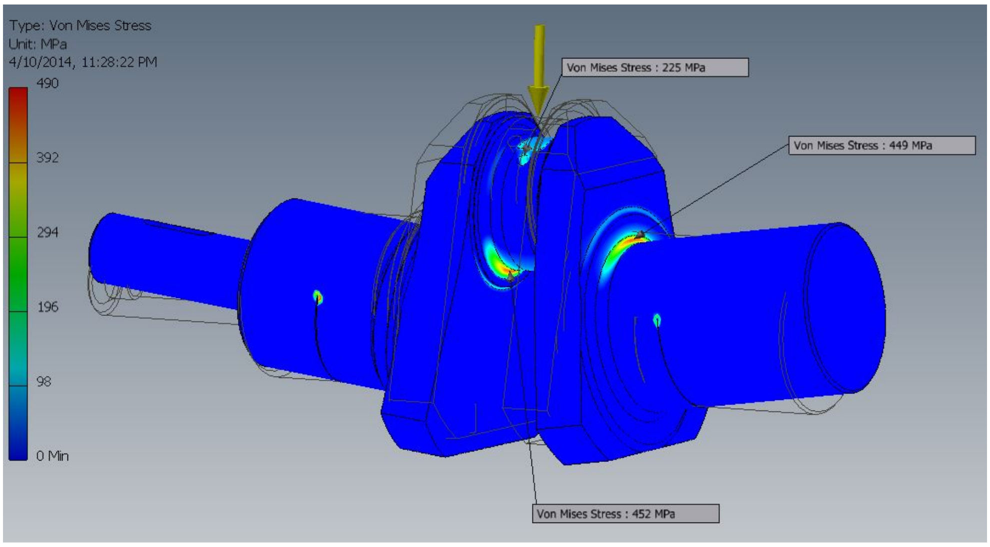


Figure 4.6: Crankshaft stress FEA load case four (8°ATDC, 10.511MPa cylinder pressure (Design revision B)

The reduction in stress at the journal fillets is clearly significant with a nearly 30% reduction in stress (approx. 185MPa) at the crankpin journal simply by increasing the crankpin diameter from 45mm to 48mm and the fillet radii from 2mm to 3.2mm.

Similarly an increase in the main bearing journal fillet greatly reduced the maximum stress there.

Location / Method	Max Stress σ_{max}	Yield Stress σ_y (4340 @ 20°C)	% of σ_y	Yield Stress σ_y (4340 @ 200°C)	% of σ_y
Crankpin Journal Fillet (F.E.A.)	452 MPa	710 MPa	63.7%	510 MPa	88.6%

Table 4.4: Crankshaft stress (Crankpin journal fillet) with respect to material yield stress (Design revision B)

Listing the results from Figure 4.6 in Table 4.4 above for the new geometry we see an improvement in the maximum value for stress with respect to operating temperature yield strength. Although the value the sits close to the 90% of yield at 200°C, the maximum design pressure (10MPa) is less than the load case pressure (10.51MPa) and the operating temperature is not expected to be 200°C. These facts combined justify acceptance of the 452MPa maximum equivalent stress value.

Below in Table 4.5 are the results of the FEA over all load cases for the revised crankshaft design. These figures will be used in the fatigue analysis.

Location	Max Stress σ_{max}	Min Stress σ_{min}	Alternating Stress σ_a	Mean Stress σ_m
Crankpin Journal Fillet	452MPa	-87MPa	269MPa	183MPa
Crankpin Oil bore	51.1MPa	-225MPa	138 MPa	-87MPa
Main Bearing Journal Fillet	96.6MPa	-449MPa	273MPA	-176MPa

Table 4.5: Crankshaft stress FEA results (Design revision B)

Reapplying Eq 3.58 to the revised crankshaft design we get the new factor of safety for fatigue at the crankpin journal,

$$nf = 1.301$$

Since the yield stresses have been addressed and the factor of fatigue safety improved we shall now introduce further results to validate this design or determine the need for additional improvements.

Next we apply equation 3.56 from the conrod design section in chapter 3 for the endurance limit of this material applied to the crankshaft,

$$Se = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_e' \qquad \text{Eq 3.56}$$

The values for ka, kb etc. can be found in Section 4.1 of Appendix B for which we get the result,

$$Se = 371.3\text{MPa}$$

So far where FEA method has been applied to fatigue analysis we have only addressed the crankpin journal fillet. The reason for this is that not only is this journal fillet the point of highest stress it is also the only point where the mean stress is in tension. For fatigue analysis where the region in question is in tension the appropriate criterion (ie: ASME, Goodman etc.) needs to be applied. However when the mean stress is in compression only the following need be verified (Shigleys, 2012),

$$\sigma_a \leq Se \qquad \text{Eq 4.1}$$

$$\sigma_{max/min} \leq \sigma_y \qquad \text{Eq 4.2}$$

To validate the design we need to ensure that these are meet, utilising the values given in Tables 4.4 & 4.5 it can be seen that this has been achieved.

4.2.5 Final Discussion

On completion of the redesign and subsequent stress analysis / fatigue analysis the results determine that the new crankshaft design is fit for permanent operation under the following conditions,

1. 10 MPa maximum cylinder pressure
2. 200°C maximum operating temperature.
3. 2500 RPM maximum engine speed.

It would be possible to raise the above limits however careful planning, consideration and risk assessment should be exercised.

These limits are set as mildly conservative limits to give confidence to the operators that the machine will not prematurely fail.

4.3 Conrod Design Results

Like the crankshaft design, when designing the conrod there is a strong focus on fatigue analysis as mentioned in the methodology. Another important point of design methodology mentioned in the previous chapter is the procedure of using two analysis methods wherever possible. In this case as across the whole project where possible the two methods are hand calculations and computer based analysis. Specifically with respect to the conrod, crankshaft and piston the computer based analysis is finite element analysis FEA.

The first results presented will be for the little end bearing eyelet hand calculations. Next will be the FEA approach to yield and fatigue, finally a buckling analysis of the conrod pillar will be given using hand calculations from established texts. All of the above is mentioned in chapter three.

4.3.1 Little end bearing eyelet

Below is the general equation Eq 3.53 used to determine the stresses in the conrod little end bearing. Variations of this equation were used to determine stresses at various locations in the eyelet.

$$\sigma_{le} = \frac{M_c \cdot (\frac{h_{eye}}{2} + e)}{h_{eye} \cdot w_{eye} \cdot e \cdot (\frac{R_c}{2})} \qquad \text{Eq 3.53}$$

The results from the hand calculations for maximum stress in the little end bearing eyelet are as follows,

$$\sigma_{le\_top} = 402.4MPa$$

$$\sigma_{le\_side} = 391.8MPa$$

$$\sigma_{le\_top\_factored} = 523.2MPa$$

Note that the minimum stresses are not listed and taken as zero. During the operation of the engine there is no point where the upper half of the eyelet goes into compression and therefore the minimum stress is considered to be zero.

Detailed calculations are given in Section 4.4 of Appendix B including moment calculations and section properties. The above results are checked against the F.E.A. results to validate both.

4.3.1.1 FEA results

Two load cases were covered for the conrod, the first is maximum inertia acting on the conrod with a focus on the little end bearing eyelet. The second is the maximum compressive force acting on the conrod with a focus on the I-beam pillar and little end bearing bottom face.

Shown below is the FEA model of the little end bearing eyelet in load case one,

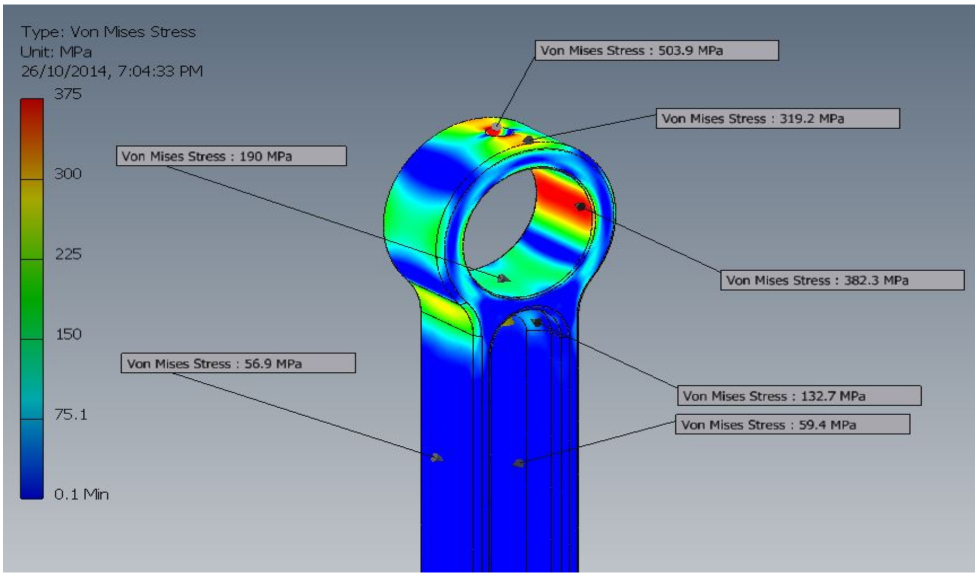


Figure 4.7: Conrod stress FEA load case one (360°ATDC, 10.81kN Tensile force from maximum inertia (Design revision A)

It can be seen that the areas of high stress match the hand calculations within approx. 5%. These results are quite close and not only give confidence to both sets of results but allows the use of the hand calculations in determining a design solution to the high stress values.

The peak stress values are of concern, as can be seen in the tabled results below (4.6 & 4.7) the stress at the little end bearing oil port exceeds the yield limit.

Shown next is the F.E.A. model of the conrod in load case two where the maximum compressive load is applied,

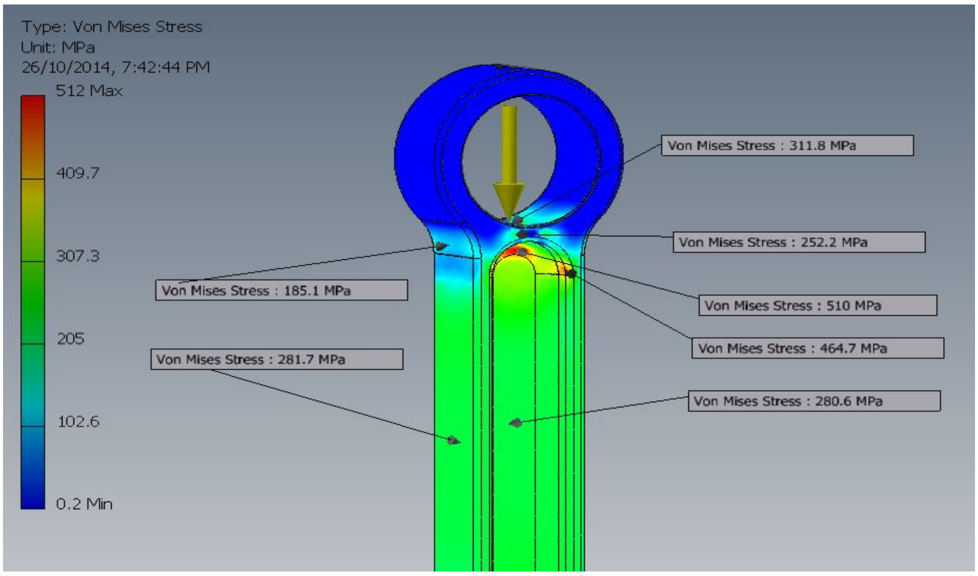


Figure 4.8: Conrod stress FEA load case two (8°ATDC, 52.3kN compressive force from maximum cylinder pressure (Design revision A)

No formal hand calculations were used to determine the compressive stress in the pillar as it was deemed a simple situation. However the designer did perform a quick check using a hand calculator and the equation of force divided by area and determined the F.E.A. results in the centre of the pillar to be within 2%.

As was the case with the crankshaft design the FEA results have been selected to perform the fatigue analysis and all further redesign analysis. As before, the reasons are that the F.E.A. relies on fewer assumptions and will deliver the needed final results in less time.

4.3.1.2 Fatigue Analysis

Using the same approach to fatigue analysis as the crankshaft and discussed in chapter three we table the F.E.A. results of both load cases below,

Location	**Max Stress** σ_{max}	**Min Stress** σ_{min}	**Alternating Stress** σ_a	**Mean Stress** σ_m
Eyelet oil inlet	504MPa	0 MPa	252 MPa	252 MPa
Web radius below eyelet	132MPa	-510MPa	321 MPa	-189 MPa
Conrod pillar web	59.4MPa	-282MPa	170.6 MPA	-111 MPa

Table 4.6: Conrod FEA stress results (Design revision A)

Firstly applying equation 3.58 from the chapter 3 shown below to the eyelet oil bore which is the only location of high stress where the mean stress is in tension,

$$nf = \frac{1}{2} \cdot \left(\frac{Sut}{\sigma_m}\right)^2 \cdot \left(\frac{\sigma_a}{Se}\right)\left[-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_m \cdot Se}{Sut \cdot \sigma_a}\right)^2}\right] \qquad \text{Eq 3.58}$$

We get the result,

$$nf = 1.14$$

Which is acceptable however not a great result considering the point in question exceeds the minimum infinite life criterion by only 14%. Continuing the fatigue analysis the endurance limit needs to be determined and compared to the results in Table 4.6. Equation 3.54 from chapter 3 for the endurance limit of this material is applied to the conrod,

$$Se = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_e' \qquad \text{Eq 3.56}$$

The values for ka, kb etc. can be found in Section 4.4 of Appendix B for which we get the results,

$Se = 312.8\ \mathrm{MPa}$ For the eyelet

$Se = 314.6\ \mathrm{MPa}$ For the pillar

Taking results from Table 4.6 above, results for the endurance limit immediately above and applying equations 4.1 & 4.2 below it can be claimed that the conrod is not quite fit for infinite life.

$$\sigma_a \leq Se \qquad \text{Eq 4.1}$$

$$\sigma_{max/min} \leq \sigma_y \qquad \text{Eq 4.2}$$

Referring to the table below the limits are exceed at the following points by the given amounts.

Location / Method	**Max Stress** σ_{max}	**% of Yield Stress** σ_y (510MPa)	**Alt Stress** - σ_a	**% of Endurance Limit** - σ_y
Eyelet oil inlet (FEA)	504 MPa	98.8%	252 MPa	80.1%
Web radius below eyelet	-510 MPa	100.%	321 MPa	102%

Table 4.7: Conrod eyelet stress load case two with respect to yield stress (Design revision A)

Both points either meet or brush the yield limits which in many design standards is not allowed (AS3990 for example), in no way is plastic design intended to be introduced here. In the case of the region below the eyelet both the yield limit and endurance limit are met.

It is decided since the conrod will clearly fail at operating temperature and in fatigue to address these concerns by redesigning the conrod pillar and little end bearing eyelet. The results of the redesign will be addressed after the results of the buckling analysis are presented.

4.3.2 Buckling

The methodology for buckling analysis was discussed in the previous chapter and is based around the general equation,

$$F_{c(max)} = K_{yy} \cdot \frac{(\pi \cdot E_{4340} \cdot I_{yy})}{L_c} \qquad \text{Eq 3.59}$$

Applying this equation to the conrod in both directions we get the results,

$F_{c(max)x} = 1047kN$ Buckling at 90° to the bearing rotation

$F_{c(max)y} = 865.2kN$ Buckling in line to the bearing rotation

This results in a buckling "factor of safety" of the following,

$$sf_y = F_c / F_{c(max)y} = 16.54 \qquad \text{Eq 4.3}$$

$$sf_x = F_c / F_{c(max)x} = 20.02 \qquad \text{Eq 4.4}$$

These results are to be expected since the conrod is quite short and well constrained in the weak direction of the conrod pillar. An increased factor of safety to say above “20” would be the designer’s choice since the material will operate at an elevated temperature and receive rapidly applied loads. However this is only the discretion of the designer and with respect to the elevated temperature is not based on any documented research findings or known established texts.

4.3.3 Further Results (redesign).

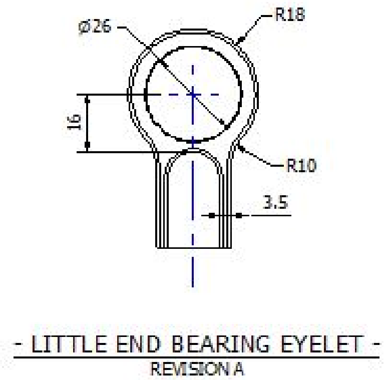
The first design modification performed was to increase the material around the conrod eye from 5mm to 6mm giving a greater section depth to what is essentially a curved beam bearing a parabolic distributed load. The next modification was to increase the width of the eyelet from 18mm to 20mm. The aim of this change is to improve the bearing surface stresses in the bottom of the eyelet and bush.

Using the hand calculations to quickly determine the effectiveness of a small increase in the material it was found only an additional 1mm radially was required to reduce the average stress by more than 100MPa.

The next modification was to lower the pillar recess by approximately 6mm to increase the material immediately below the little end bearing bore allowing better force distribution into the pillar. In effect what was performed was a geometry improvement to reduce the local stress raisers.

Finally the flanges of the pillar I-section were increased in thickness by 0.5mm to increase the cross-sectional area and thus decrease the compressive stresses by approximately 7%. Another aim of the flange increase was to improve the buckling factors of safety slightly.

The design modifications are given below,



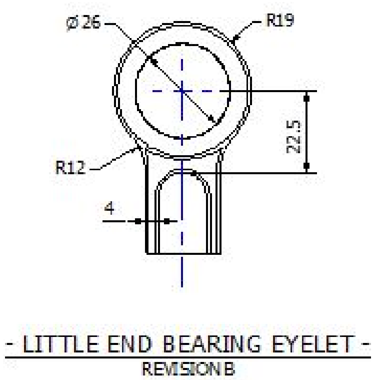


Figure 4.9: Conrod eyelet modifications, Left: Before , Right: After

Referring to the F.E.A. models below it can be seen that the desired stress reductions were achieved. The peak stress in the sides of the little end bearing eyelet has reduced by approximately 120MPa and the stress at the oil inlet at the top of the eyelet by approximately 185MPa.

Stresses at the web below the eyelet have also reduced by nearly 150MPa giving a total reduction in the maximum percentage yield stress at the eyelet and web radius of 36.9% & 29% respectively.

Additionally reductions in the alternating stresses gave improvements in the percentage of endurance limit values. All of this adds to the factor of safety in the design for only 28grams of additional mass.

Below can be seen the eyelet of the conrod in load case one and load case two where the stress distributions remain the same but the magnitude of stress is decidedly less.

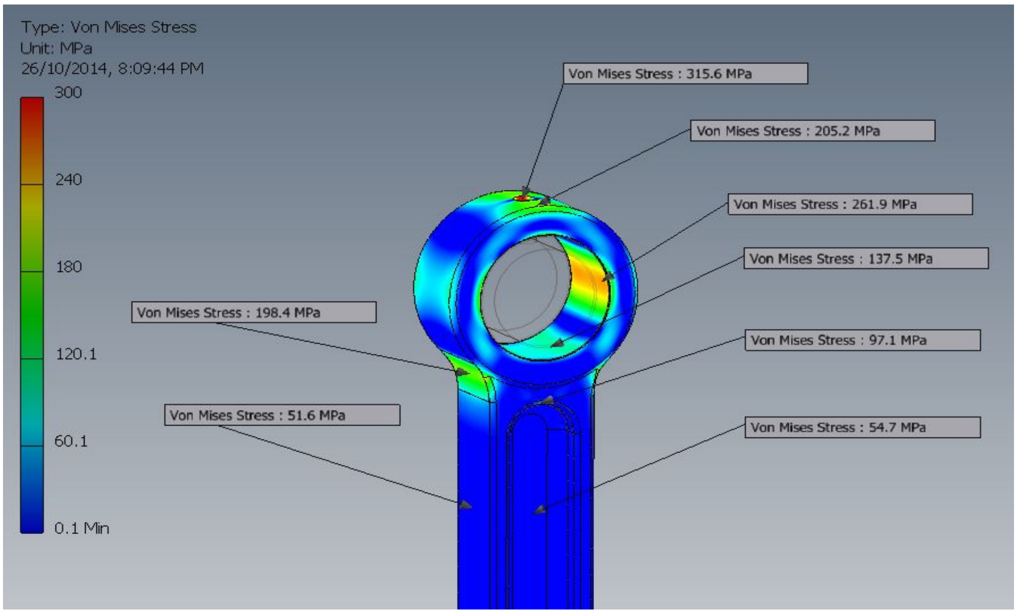


Figure 4.10: Conrod stress FEA load case one (360°ATDC, 10.81kN tensile force from maximum inertia (Design revision B)

Referring to figure 4.11 below it is worth noticing the stress distribution beneath the eyelet due to the increase in distance between the eyelet and pillar recess. The forces travel into the flanges more smoothly (without marked direction change) than did previously. This results in the large reduction of stress in the pillar without a serious change in cross sectional area.

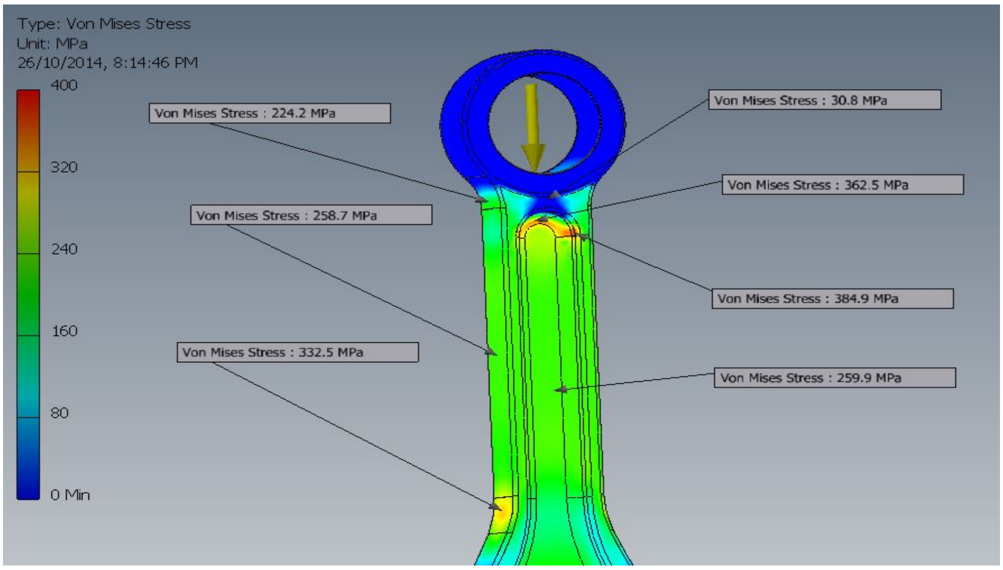


Figure 4.11: Conrod stress FEA load case two (8°ATDC , 52.3kN compressive force from maximum cylinder pressure (Design revision B)

Tabling the results from the F.E.A. of Revision B below we see an appreciable improvement in all stress values. As expected the smallest improvement was in the pillar due to the minor increase in cross sectional area.

Location	**Max Stress** σ_{max}	**Min Stress** σ_{min}	**Alternating Stress** σ_a	**Mean Stress** σ_m
Eyelet oil inlet	316MPa	0 MPa	157 MPa	157 MPa
Web radius below eyelet	97 MPa	-363MPa	231 MPa	-133 MPa
Conrod pillar web	54.7 MPa	-260MPa	157 MPA	-103 MPa

Table 4.8: Conrod FEA stress results (Design revision B)

The improvements in the percentage of yield stress and endurance limits can be seen below when compared to table 4.7. With the maximum stresses around 70% and below of the operating temperature yield point and the same for the endurance limit the revised design is a substantial improvement.

Location / Method	**Max Stress** σ_{max}	**% of Yield** - (510MPa)	**Alternating Stress** - σ_a	**% of Endurance Limit** - σ_y
Eyelet oil inlet (FEA)	316 MPa	61.9%	157 MPa	50.2%
Web radius below eyelet	-363 MPa	71%	231 MPa	73.4%

Table 4.9: Conrod FEA stress results with respect to yield strengths (Design revision B)

Applying equation 3.58 to the values in Table 4.9 we get the new fatigue factor of safety for the conrod eyelet,

$$nf = 1.82$$

Updating the critical buckling forces and resulting factors of safety we notice only a small improvement in the buckling, particularly in the plane of the bearing rotation. This not too surprising when considering the end fixities of a double pinned column give a no reduction to the effective length.

$F_{c(max)x} = 1203\ kN$ Buckling at 90° to the bearing rotation

$F_{c(max)y} = 914.3\ kN$ Buckling in line to the bearing rotation

These results give a buckling "factor of safety" of the following

$$sf_y = F_c/F_{c(max)y} = 17.48$$

$$sf_x = F_c/F_{c(max)x} = 22.99$$

4.3.4 Discussion of Results

The reduction in peak stresses to 71% of the operating temperature yield strength from 100% was the most important outcome of the design modifications. The additional gain in the fatigue factor of safety due to the improvement in peak tensile stress at the oil inlet port is another significant gain.

The endurance limit did account for the conrod surface to be machined not ground giving a lower endurance limit than the crankshaft of the same material, this was an essential allowance as imperfections will be left on the conrod surface after machining due to the awkward geometry of the part. Additional allowances were made for the differing sections between the eyelet and the pillar and the nature of the loads applied, again necessary due to the heavy workload of this component.

The care taken and methods applied to the conrod design give confidence that this component will be fit for infinite working life at the design limits for speed, cylinder pressure and operating temperature.

4.4 Bearing Design Results

The bearing design unlike other solid components cannot utilize freely available software to perform operational analyses. Software is available but beyond the budget of this project. Traditional methods using hand calculations and bearing tables are the most applicable resource available in this project. Specifically Shigley's was the main text utilized for the bearing analysis with reference to (Stolarski, 1990) to re-check check the film thickness in the main bearings.

As mentioned in chapter three the manufacture's handbook for the main bearings (Glacier) was consulted for both the hydrodynamic operating ranges and boundary lubrication limits (PU factor).

4.4.1 Main Bearings

Application of the equation for the Sommerfield number Eq3.61 on the main bearings can be seen below,

$$S = \left(\frac{r}{c}\right)^2 \cdot \left(\frac{\mu \cdot N}{P}\right) \qquad \text{Eq 3.61}$$

Using the bearing dimensions, idling speed of the engine (800 R.P.M.) and maximum design speed (2500 R.P.M.), mean bearing clearance, maximum cylinder pressure and the properties of SAE30 engine oil the resulting Sommerfield numbers are given below.

$$S_{800} = 0.184$$

$$S_{2500} = 0.575$$

Applying these values to Figure 4.12 below and using the result h_{0}/c we determine the main bearing operating clearances to be,

$$h_{0(800)} = 0.0234\text{mm}$$

$$h_{0(2500)} = 0.039\text{mm}$$

Using EQ 3.63 the minimum allowable main bearing film thickness is,

$$h_{0(min)} = 0.00688\text{mm} \qquad \text{Eq 3.63}$$

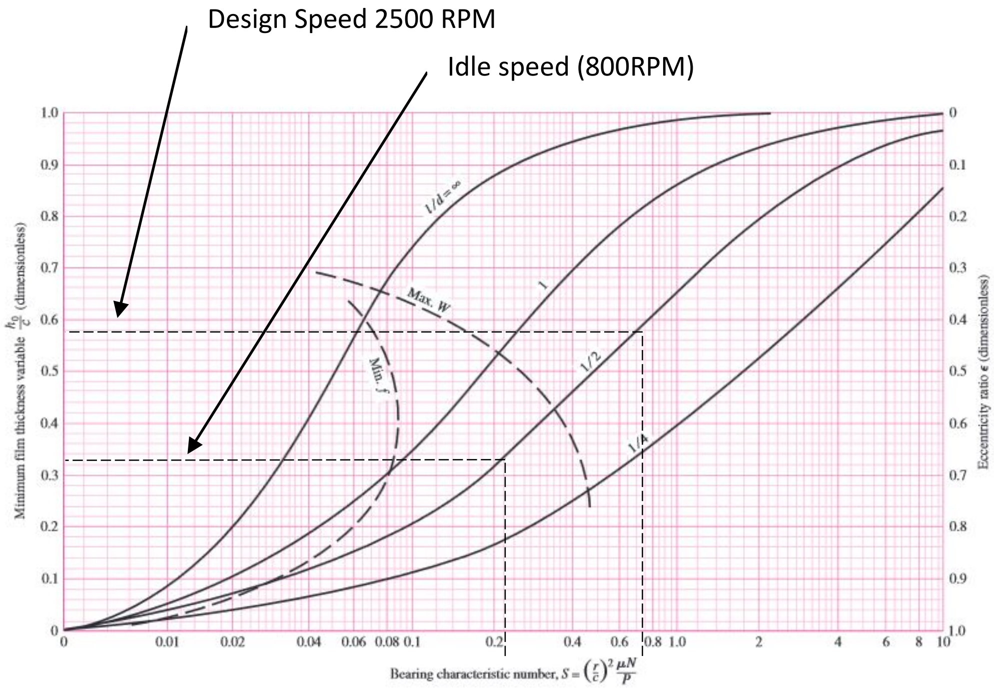


Figure 4.12: Chart for minimum film thickness variable and eccentricity ratio (Shigley,2012)

From these numbers it is deemed that the main bearing arrangement has the capacity to carry the maximum load created by the gas pressures even at idle speed. Detailed calculations found in Section 4.2 of Appendix B and used in conjuction with the charts on page 25 of the DU bearing catalogue demonstrate the main bearings capacity at start up lie just within the bearing's design capacity. This is under the assumption that the cylinder pressures are very small under 60RPM and rise linearly to a maximum at some point after 300RPM.

An important limiting factor in a plain bearing is the temperature rise of the oil in the bearing as it passes through the gap between the journal and bore. One of the design checks performed during the bearing analysis was to calculate this temperature. The result of which can be found below,

$$\Delta T = 3847°\text{C}$$

This temperature rise was calculated at idle speed for maximum cylinder pressure. It is important to remember that the maximum load only occurs for an instant and the peak load region is only about 25° of crank angle in duration. This essentially means that the bearing and oil temperature will not reach anywhere near this temperature. It does mean however that a lot of energy will be put into the oil and that the bearings may run hot and require close monitoring during commissioning. Exact bearing temperature was not possible with the resources available.

4.4.2 Big End Bearing

As mentioned the methodology for the big end bearing design is much the same as the main bearings. Once the appropriate relative velocities have been calculated the procedure is the same. Applying equation 3.61 to the big end bearing we get the Sommerfield numbers of,

$$S_{800} = 0.037$$

$$S_{2500} = 0.116$$

Applying these values to Figure 4.12 above and using the result h_0/c we determine the main bearing operating clearances to be,

$$h_{0(800)} = 0.00715\text{mm}$$

$$h_{0(250)} = 0.0143\text{mm}$$

Using equation 3.63 we determine the minimum allowable big end bearing film thickness to be,

$$h_{0(min)} = 0.007\text{mm} \qquad \text{Eq 3.63}$$

From the figures immediately above it can be read that we are operating on the very limit of oil films when the engine is at idle speed. Since the figures are calculated at maximum cylinder pressure regardless of engine speed and it is expected that the cylinder pressure will be lower at idle, these figures are conservative.

By using aluminium bronze for the bearing should the hydrodynamic mode fail the bearing will cope.

4.4.3 Little End Bearing

The little end bearing as mentioned in the methodology is considered a plain bearing and not to operate in a hydrodynamic mode. For this reason the analysis will be based on maximum compressive strength of the material and the allowable pressure velocity of the material.

Applying equation 3.66 from chapter three below,

$$P_{max} = \frac{4 \cdot F_{c(max)}}{\pi \cdot 2 \cdot r \cdot l} \qquad \text{Eq 3.66}$$

We get,

$$P_{max} = 181.05\ MPa$$

Unfortunately the sintered bronze SAE841 originally selected for the little end bearing fails at less than half of this compressive stress. Therefore the pressure velocity will not be calculated for this material and a new material needs to be selected for the little end bearing.

The material 954 aluminium bronze was selected for the little end bearing due to the following properties,

$$P_{max} = 265\ MPa$$

$$T_{max} = 260\ °C$$

The material 954 Al. Bronze is a hard, abrasive resistant bearing material whose properties remain good at elevated temperatures. This material is used by some performance conrod manufactures as part of their product line. The pressure velocity of this material is unavailable from the material suppliers however it is expected to be quite high.

4.4.4 Counter Balance Shaft Bearings

Like the main bearings the counter balance shaft bearings are part of the existing design, therefore the purpose of the analysis is to check their capacity. Using the original design balance weights the capacity of the bearings was checked using equation 3.66 and found to be quite suitable.

Since the crankshaft was redesigned to be statically balanced and thus eliminate the horizontal shaking forces on the engine the new counter balance weights required are heavier. The increase in mass has contributed to a serious decrease in the bearing life.

The primary & secondary counter balance shaft bearing lives based on the masses required to counter design revision B at 2500 R.P.M. are given below,

$$L_{90\_Primary} = \frac{10^6}{60 \cdot RPM}\left(\frac{C\_6205}{Fbs}\right)^{\frac{10}{3}}$$

$$L_{90\_Primary} = 23.54\ hr$$

$$L_{90\_Secondary} = \frac{10^6}{60 \cdot RPM}\left(\frac{C\_6205}{Fbs}\right)^{\frac{10}{3}}$$

$$L_{90\_Secondary} = 846.3\ hr$$

Note that the lower capacity bearing results have been given, the results of which are quite poor especially in the case of the primary balance shafts. Although the life increases dramatically with respect to any reduction in engine speed (eg: from 846hrs at 2500rpm to 4500+ hrs at 2000rpm) a life of 23.4hrs for the primary shaft bearing is unacceptable. Especially considering the required balance mass may increase with further design iterations.

It is necessary as a result of these figures that the primary balance shaft bearings be changed to higher capacity bearings. It would be prudent to ensure the bearings can exceed the engine maximum design speed of 2500rpm to prevent reduction in the maximum design speed.

Choosing a recommended arrangement from the SKF literature (SKF general catalogue,2011) the bearings 22205 (spherical roller) and C2205 (toroidal roller) were selected. This arrangement has the highest radial load capacity without modifying any of the shaft or housing diameters. The bearings possess a rated speed of 7000rpm and the C2205 bearing allows an internal axial float of 2.8mm much the same as the original N205 bearing.

The new life calculation results for the bearings is,

$$L_{90\_Primary} = 627.5\ hrs$$

$$L_{90\_Secondary} = 22390\ hrs$$

The L90 life of a bearing means that 90% of bearings will only **begin** to wear after this period of time. The engine is not likely operate at its maximum for lengthy periods of time so the designer accepts the above life figures. By inspection of the design it can be seen that the bearings are not difficult to replace giving another reason to accept the above L90 life. To reduce costs the end user may wish to down size the secondary balance shaft bearings, this would be quite acceptable and the only reason the current designer has not done so was simplicity.

4.5 Piston Design Results

It was mentioned in chapter three that the main method of analysis for the piston design was the use of FEA. This was due to the difficult geometry of the piston and the absence of any known design papers or standards.

Much of the gas pressure calculation performed for the crankshaft design was used in the FEA models for the piston. The pressures, forces and constraints have been laid out in detail in chapter 3 and will not be reiterated here.

We shall discuss in detail here the results of interest to the engine design.

4.5.1 Window Design

Taking the cylinder pressure from the crankshaft design, material properties for HSQ-300 quartz, current widow dimensions design revision A and equations from (Roarks, 2012) mentioned in chapter three the results for the window design are,

$t = 10mm$ (Window thickness)

$a = 46.5mm/2$ (Window radius)

$q = 10.511\ MPa$ (Cylinder Pressure)

$$M_c = \frac{q \cdot a^2 \cdot (3+v)}{16} \qquad \text{Eq 3.68}$$

$$M_c = \ 1.13kNm/m$$

$$\sigma_{max} = \frac{6 \cdot M_c}{t^2} \qquad \text{Eq 3.67}$$

$$\sigma_{max} = 67.543\ MPa$$

Referring to the materials data sheets found in Appendix E it can be seen that the maximum bending stress for HSQ 300 is 67MPa and tensile stress 50MPa. This results in simple component failure and needs to be addressed.

By iteration utilizing the MathCAD software used to perform the calculations given in Section 4.5 of Appendix B, greater plate thicknesses were trialled to meet the stress limits. The result was the selection of 20mm plate based on the final bending stress and suitable value for deflection.

The results for the final dimensions are given below,

$t = 20mm$ (Window thickness)

$$\sigma_{max} = \frac{6 \cdot M_c}{t^2} \quad \text{Eq 3.67}$$

$$\sigma_{max} = 16.89\ MPa$$

$$\Delta_{max} = \frac{-q \cdot a^2 \cdot (5+v)}{64 \cdot D_c \cdot (1+v)} \quad \text{Eq 4. 05}$$

$$\Delta_{max} = .004\ mm$$

These calculated results give a factor of safety of 3.9 for bending stress and a deflection of 1 in 11750 which is very small. Clearly bending stress was the true limiting factor and a conservative approach was required as data for the mechanical properties of HSQ-300 at elevated temperatures was unavailable.

4.5.2 Materials Selection

The materials given in chapter three remain as the selected materials for construction. Refer to the detail drawings in Appendix D and materials data sheets given in Appendix E for specific material information and application.

4.5.3 Finite Element Analysis Results

The primary load case given in chapter 3 was the piston under maximum cylinder pressure (10.51MPa). The second load case is where the piston is at top dead centre between the exhaust and intake strokes, the acting forces at this point are the inertia forces.

The third load case (not given in the methodology) is the maximum conrod force or more specifically the conrod force developed under maximum cylinder pressure acting on the piston. Here the forces are the equal and opposite of the pressure forces in load case one but due to geometric and software application constraints requires a stand-alone load case.

Shown below is the FEA model for load case two, note the stresses occurring in the wrist pin bore.

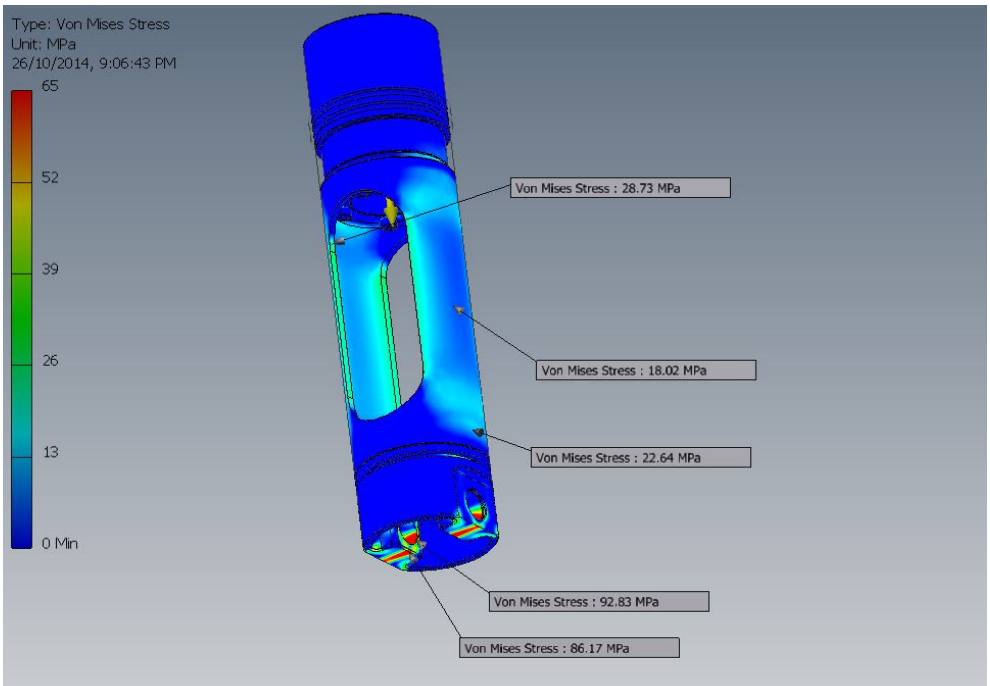


Figure 4.13: Piston stress FEA load case two (10.81kN load at 360° ATDC from maximum inertia (Design revision A)

It is the results of these load cases analysed using F.E.A. that we shall present first the maximum, minimum, mean and alternating stress. Further application of the fatigue methods shown in the crankshaft and conrod sections previously will be performed to validate the design or determine required modifications.

As previous we are interested in the maximum or minimum (compressive) stress with respect to yield stress and the alternating stress with respect to the material's endurance limit.

Shown below is the FEA model for load case three,

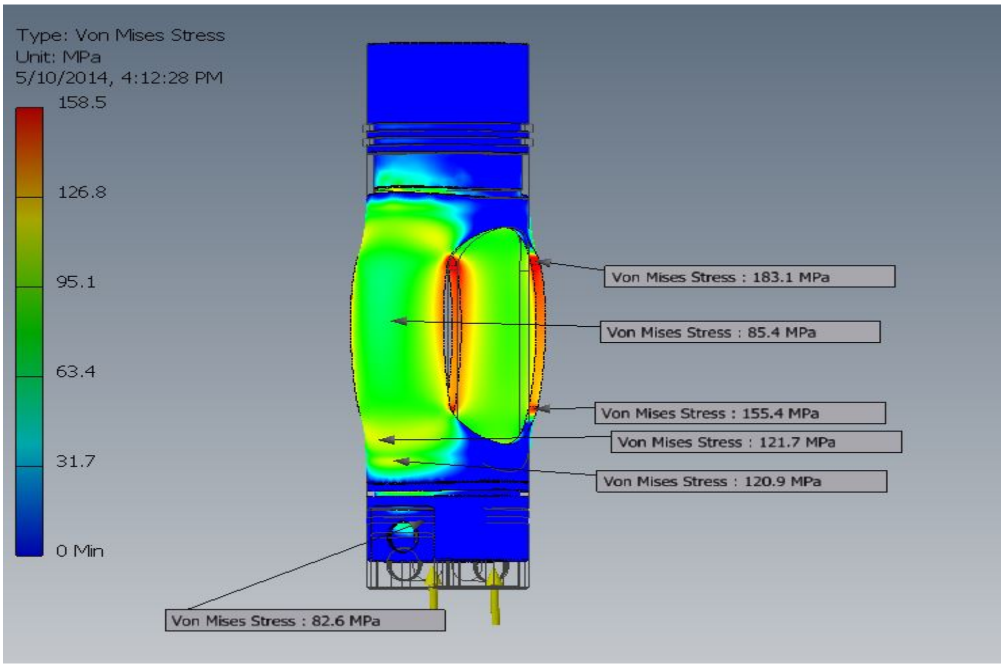


Figure 4.14: Piston stress FEA load case three (10.511MPa cylinder pressure at 8° ATDC (Design revision A)

Gathering the F.E.A. stress results from all three load cases and tabulating the pertinent values we get the table shown below.

Location	**Max Stress** σ_{max}	**Min Stress** σ_{min}	**Alternating Stress** σ_m	**Mean Stress** σ_a
Upper skirting opening	28.7 MPa	-183.1 MPa	106 MPa	-77.2 MPa
Lower skirting centre	22.6 MPa	-121.7 MPa	72.2 MPa	-49.6 MPa
Lower wrist pin bore	92.8 MPa	-16 MPa	54.4 MPA	38.4 MPa

Table 4.8: Piston stress FEA results (Design revision A)

It can been seen from the results in the above table that the predominant stress type in the piston is compressive. Since the only positive mean stress is in the wrist pin bore only there will any fatigue criterion be applied. Applying equation 3.58 to the wrist pin bore results we get,

$$nf = 3.416 \qquad \text{Eq 3.58}$$

Which is quite a satisfactory result for fatigue.

Next applying equation 3.54 from the conrod design section in chapter 3 for the endurance limit of this material applied to the piston,

$$Se = k_a \cdot k_b \cdot k_c \cdot k_d \cdot k_e \cdot k_f \cdot S_e' \qquad \text{Eq 3.54}$$

The values for ka, kb etc. can be found in appendix b for which we get the result,

$$Se = 142.86 \text{ MPa}$$

As previously mentioned when the mean stress is in compression, only the following need be verified,

$$\sigma_a \leq Se$$

$$\sigma_{max/min} \leq \sigma_y$$

To validate the design we need to ensure that these are meet, utilising the values given in Table 4.8 and Appendix E it can be seen that this has been achieved although the alternating stress is a little high in the upper skirting.

The final design verification required will be a deflection check of the skirting and crown to ensure there is no adverse piston deformation. Shown below is the FEA model for load case three,

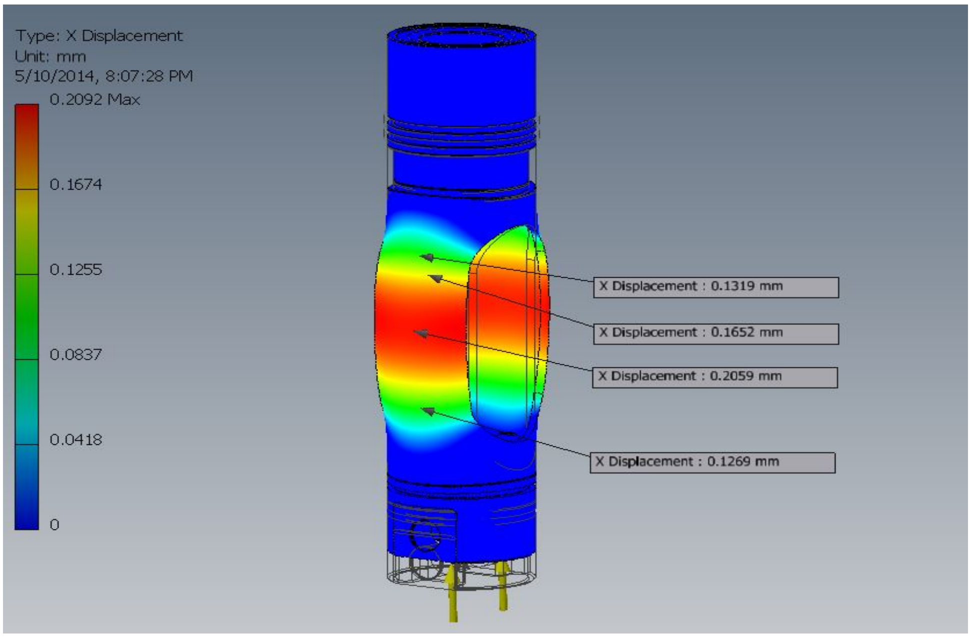


Figure 4.15: Load case three deflection results (10.511MPa at 8°ATDC (Design revision A)

From the model above showing maximum compression the greatest deflection is given in red & occurring at the centre of the extension skirting. The deflection shown is horizontal only acting along the wrist pin bore axis and therefore doesn't show the effects of piston squish common in bowditch pistons.

From the model the value of maximum horizontal deflection is given below,

$$\Delta_m = 0.2092mm$$

This results in a total diameter change at the skirting of,

$$\Delta_\emptyset = 2 \cdot \Delta_m = 0.4184mm$$

The value for skirting deflection is of concern as the point of maximum deflection is enclosed in the cylinder bore when the piston undergoes maximum cylinder pressure. Although there is clearance in the cylinder there is simply not enough to prevent the skirting binding against the sleeve and causing damage.

The solution is to redesign the piston extension so that the deflection is reduced sufficiently.

4.5.4 Redesign & Further Results

In order to reduce the deflection in the skirts it was decided to stiffen the piston extension in the region of greatest deflection by means of increasing the skirting thickness and slightly reducing the window size. In short by increasing the moment of area through the skirts the resulting deflection per unit force will reduce. Unfortunately this solution is at the expense of increasing the reciprocating mass which has serious effects to the engine operation.

The final decision was to partially stiffen the skirting to reduce the deflection in both the skirts and the piston length, this also produced the effect of reducing the compressive and bending stress occurring in the piston skirts.

The other half of the solution was to reduce the extension diameter enough to counter the deflection distance radially. This leaves the points of greatest radial deflection still lying within the bore diameter envelope. Below are the section views of both revisions to allow comparison.

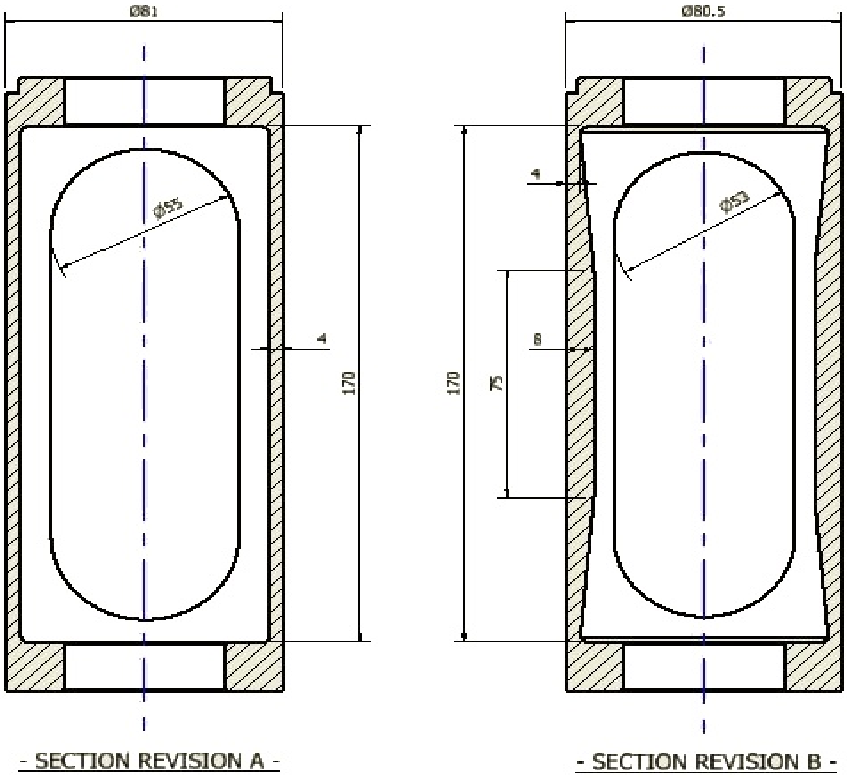


Figure 4.16: Modifications to piston extension (Left: Before modifications, Right: After modifications)

Shown below is the F.E.A. model for load case three using the revised piston extension, note the visible increase in wall thickness.

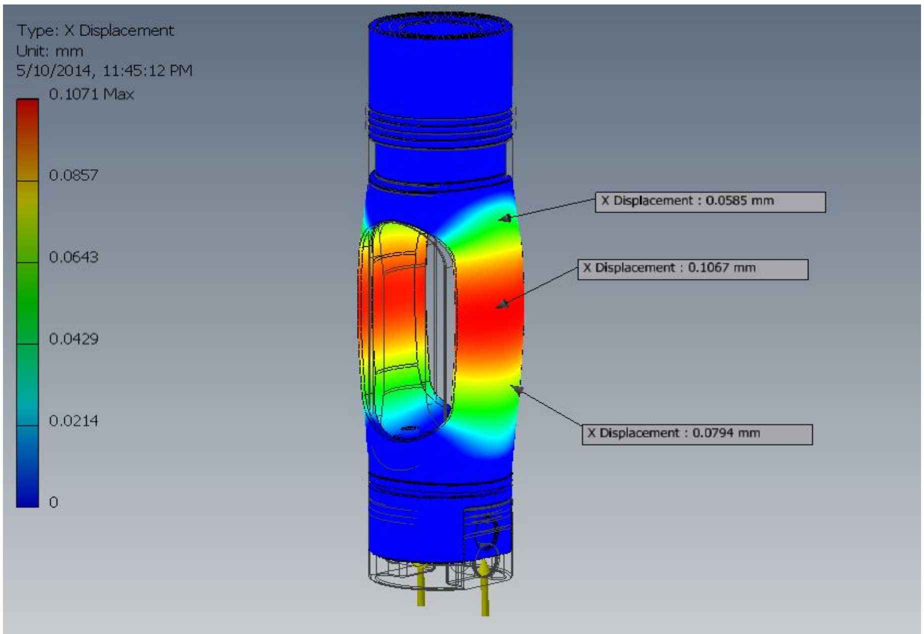


Figure 4.17: Piston stress FEA load case three (10.511MPa cylinder pressure at 8°ATDC (Design revision B)

From the FEA results the value of maximum horizontal deflection is given below,

$$\Delta_m = 0.1067mm$$

This results in a total diameter change at the skirting of,

$$\Delta_\emptyset = 2 \cdot \Delta_m = 0.2134mm$$

With a reduction in diameter of 0.5mm in the piston skirting there is sufficient clearance with little lost in the way of strength and what loss of strength there was is more than compensated by the increase in wall thickness.

On further inspection of the design calculations it was determined that the bolts retaining the extension to the lower piston body were undergoing considerable load for their size. Five M6 bolts and more importantly five M6 thread inserts into the aluminium were exposed to the inertia force of the top half of the piston. This equated to over 7.5kN total or 1.5kN per insert; high tensile bolts could easily carry this force even over the design life of the engine however the thread inserts would be testing their limits.

For the sake of approx. 100 grams of additional mass various reinforcements were made to the lower piston to accommodate M8 fasters and inserts at a slightly deeper engagement length. Additionally the material was increased around the wrist pin bore for added good measure.

To demonstrate the reduction in stresses resulting from the design changes see below to the revised combined load case stresses table.

Location	**Max Stress** σ_{max}	**Min Stress** σ_{min}	**Alternating Stress** σ_m	**Mean Stress** σ_a
Upper skirting opening	26.7 MPa	-144 MPa	85.25 MPa	-58.6MPa
Lower skirting centre	17.7 MPa	-97 MPa	57.4 MPa	-39.7 MPa
Lower wrist pin bore	71.2 MPa	-5MPa	38.1 MPA	33.1 MPa

Table 4.9: Piston stress FEA results (Design revision B)

Clearly a favourable reduction in stresses and deflection have occurred completing the piston design revisions. The adjustment to the piston design came at the price of an additional 230grams, and the modifications to the quartz window increased the piston mass by 150 grams mostly in the stainless retaining ring. This gives a final piston assembly mass of 2432 grams, higher than originally expected.

4.6 Final Balancing

Once all engine component design was complete and thus the final masses of each part known, the final engine balancing was able to take place. Inputting all new part masses and appropriate centre of gravity into the theoretical model the total vertical and horizontal shaking forces were able to be determined. Once known these force can be countered which essentially is balancing as previously stated.

4.6.1 Crankshaft Balancing

As mentioned in the previous chapter complete engine balancing can be realized with respect to all shaking forces. Specifically referring to the crankshaft component of the balancing Eq 3.22 and precisely Eq 3.20 needs to be met to achieve this. These equations are given again below for convenience,

$$\sum Fx = \omega^2 \sin\theta \, [m_b r_b - m_{be} r] = 0 \qquad \text{Eq 3.22}$$

$$m_b r_b - m_{be} r = 0 \qquad \text{Eq 3.20}$$

To achieve a balanced crankshaft & thus to eliminate the horizontal "shaking" forces Eq 3.20 needs to be met. By inspection it can be seen that in achieving Eq 3.20 we meet Eq3.22 and thus horizontally balance the engine.

Below in Figure 4.18 the crankshaft out of balance forces of before and after balancing are plotted. Please note that the initial horizontal forces were for the first crankshaft design and only roughly balanced to suit the "expected" conrod.

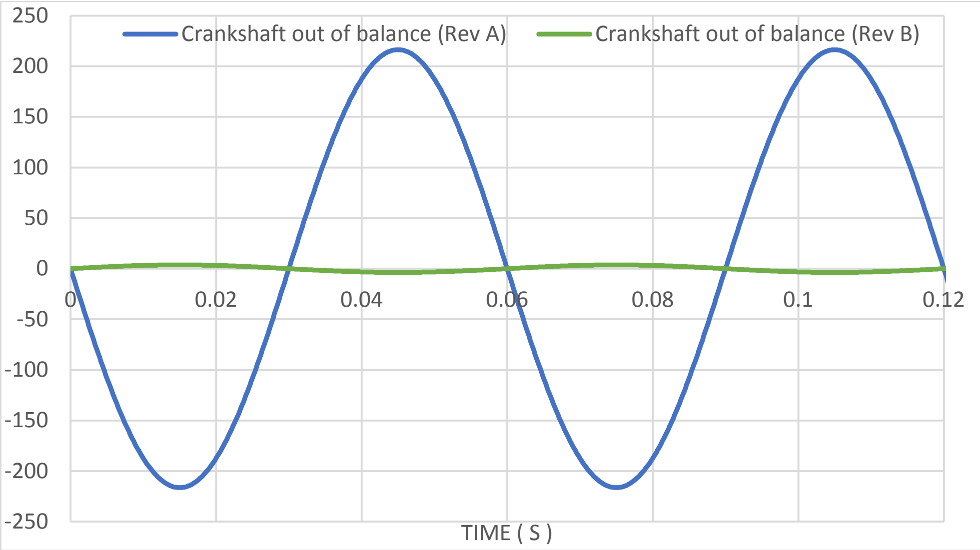


Figure 4.18: Crankshaft out of balance forces at 1000RPM (Design Revision A Vs Balanced Design Revision B)

The final results can be seen in green and show the crankshaft producing virtually no shaking forces.

Keep in mind that the forces plotted also act in the vertical direction at 90° out of phase to the horizontal forces and thus contribute to the vertical shaking forces. By adding balance weights as shown in figure 4.19 to balance the crankshaft including the lower conrod (not shown) the vertical shaking forces are reduced by more than 200N at only 1000 RPM.

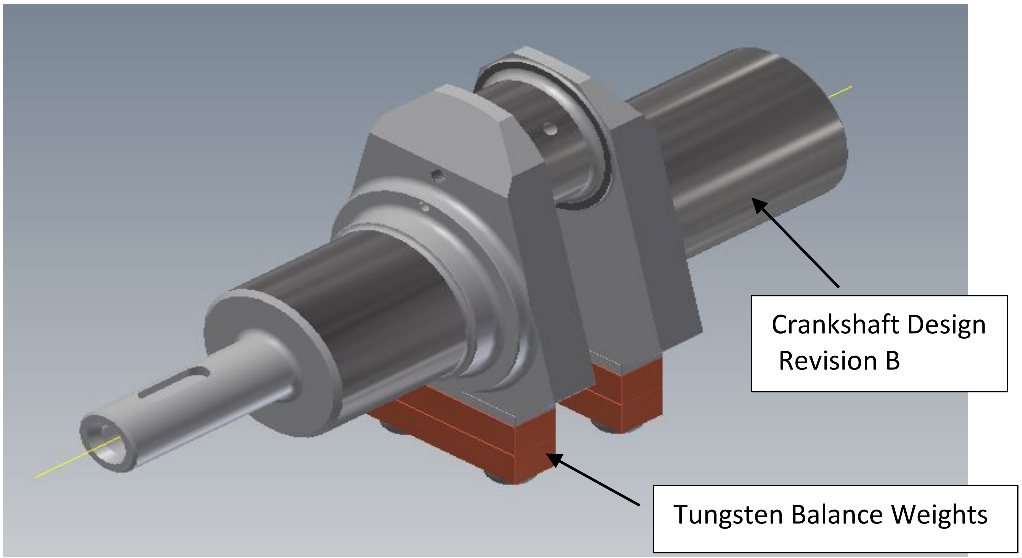


Figure 4.19: Crankshaft assembly, balanced design revision B

The values in Figure 4.18 above are calculated results, comparison of simulated and theoretical results for the balanced crankshaft are plotted below in Figure 4.20. The plots below show a slight phase shift between the two results. It is thought this is due to the actual centre of gravity being slightly sideways of the rotating axis. This can be explained by the offset oil bore drilled through the crankpin which does not pass through the crankpin centre.

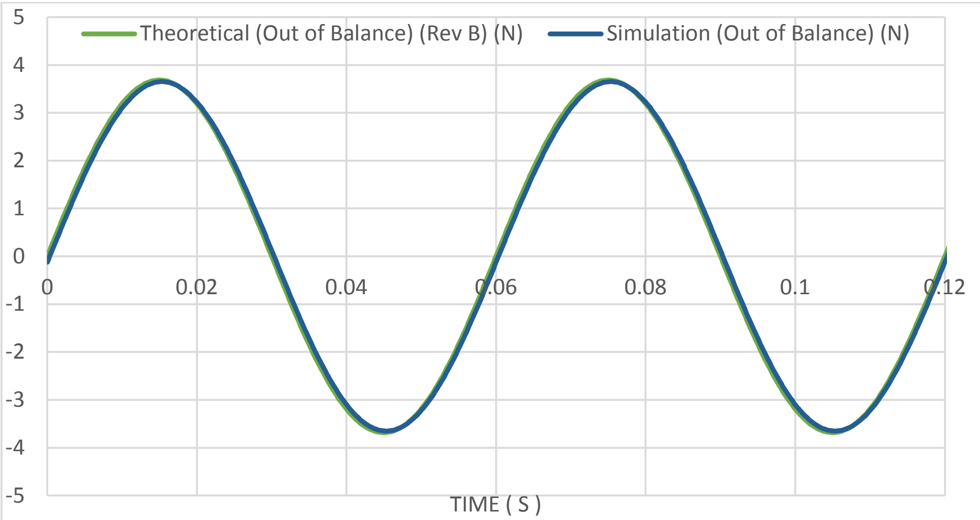


Figure 4.20: Crankshaft horizontal shaking force at 1000rpm (Virtual Vs Theoretical)

The effect of the reduction in vertical forces due to the now balanced crankshaft is discussed briefly in the next section.

4.6.2 Final Shaking Forces

Below is a plot of the final forces (theoretical) acting at the main bearings much the same as was given in Figure 4.1 minus the theoretical/virtual comparison. An important detail to note is that although the reciprocating mass has increased by approximately 380grams the maximum inertia force increased by slightly less than 80N at 1000RPM . This is due to the static balancing of the crankshaft discussed previously in this section. If the crankshaft is balanced it will not contribute any forces to the overall horizontal or vertical shaking forces. This can also be seen in the reduced horizontal reaction forces acting at the main bearings when compared to Figures 4.1 or 4.2.

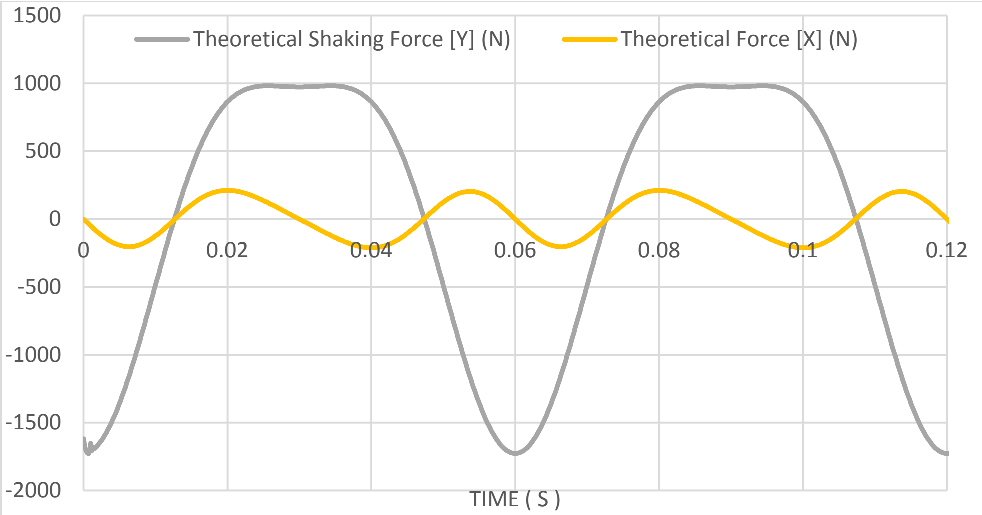


Figure 4.21: Engine shaking forces at 1000rpm (Theoretical vertical & Theoretical horizontal forces)

As a result of the balanced crankshaft the above horizontal plotted force is almost entirely the force component of the shaking torque discussed in the previous chapter. As demonstrated in Figure 3.4 there is an equal and opposite force acting through the piston skirting and therefore does not contribute to the engine shaking forces. This means the plot [X] in Figure 4.21 does not represent the horizontal shaking forces.

Breaking down the final vertical shaking forces into their respective primary and secondary components as previously demonstrated in figure 3.3 for the first design revision we can establish the required countering forces needed to balance the engine.

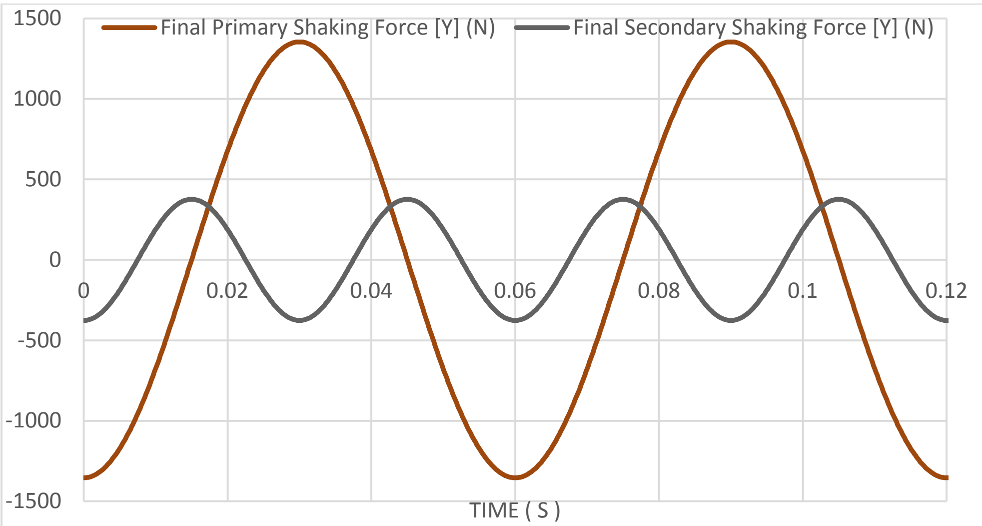


Figure 4.21: Breakdown of final vertical shaking forces at 1000rpm (Primary & Secondary forces)

As mentioned in the previous chapter by utilising Eq 3.18 & Eq 3.19 and the maximum forces plotted in Figure 4.21 above the required balance masses can be found. The results of which can be seen below.

$$Fy_{pcb} = m_{pcb} \cdot \omega^2 \cdot r_{pcb} = 1353.7N$$

$$\therefore\ m_{pcb} = 6.1698kg$$

$$Fy_{scb} = m_{scb} \cdot (2\omega)^2 \cdot r_{scb} = 376N$$

$$\therefore m_{scb} = 1.993kg$$

Two new balance weights each for the primary and secondary arrangements at half the masses given above were modelled for the virtual simulation as seen in figure 3.6. The result was inverse plots of the traces above found below in figure 4.22, in the interest of validating results both the theoretical and virtual simulation results have been included below.

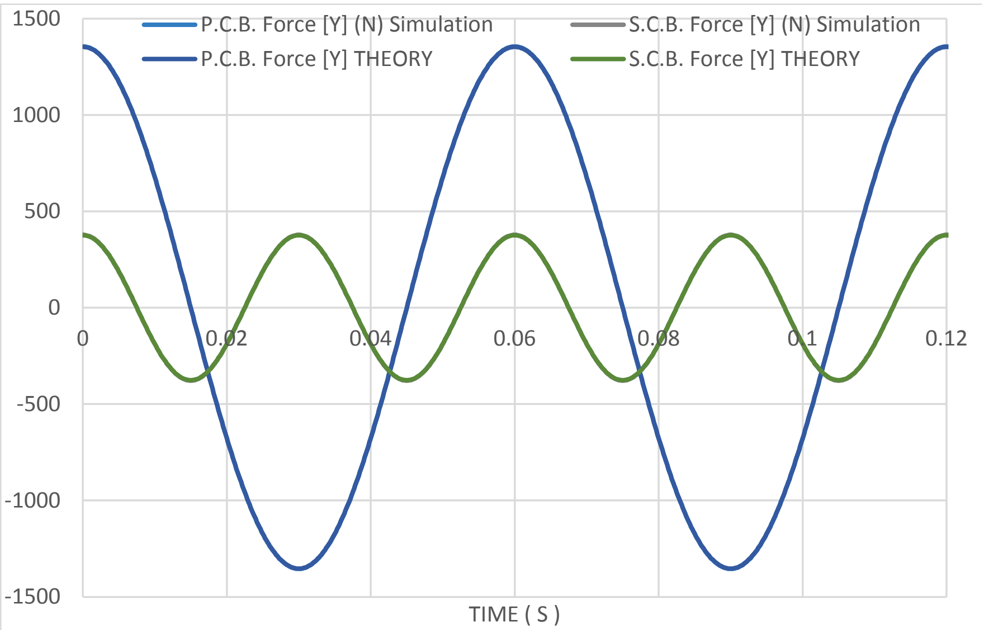


Figure 4.22: Vertical forces generated by the counter balance shafts at final balance mass and 1000rpm. (Virtual and Theoretical)

As previously seen the difference between the virtual results and the theoretical results cannot be discerned on a plot of this size which is a favourable outcome and consistent with previous comparisons.

Summing the vertical forces produced by the slider crank mechanism and the balance shafts we get a plot of the effective shaking forces produced by the engine.

In the first plot below the theoretical and simulation results are plotted together, the small differences between the virtual and theoretical results of previous comparisons is more visually pronounced here.

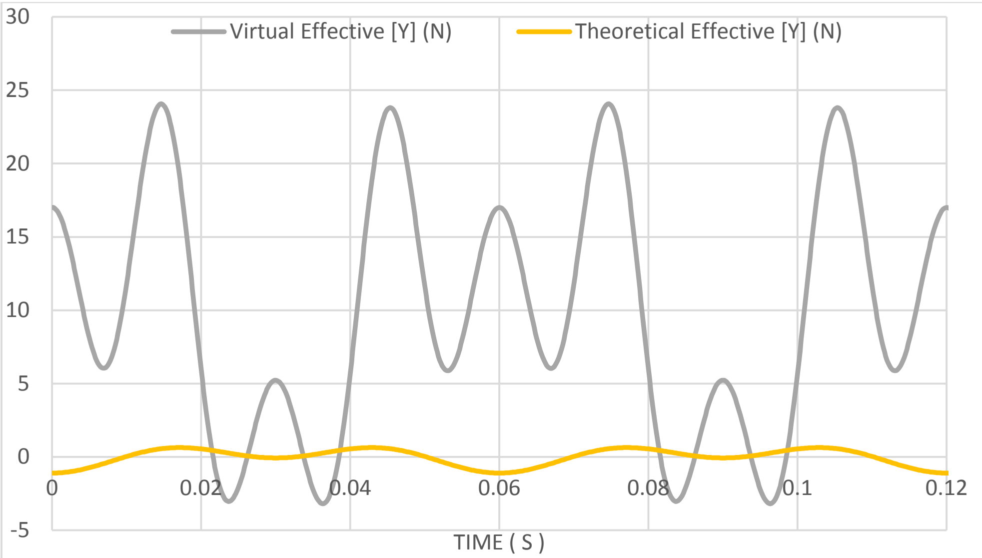


Figure 4.23: Total engine shaking forces at 1000rpm. (Virtual Vs Theoretical)

This result is not unexpected and is consistent with the differences plot of figure 4.3 taken from the first design revision results. To explain this further the slight increase in the difference is expected to be largely due to the fact that engine parts have been modelled as close as dimensionally practicable for a machine shop to fabricate. As such some of the masses and dimensions used in the theoretical model are not precisely the same as the simulation. Keep in mind that the values used in either method are still quite close but not often better than 0.1mm or 1gram.

That noted the final figures given by the simulation are still quite good and less than 25N of shaking force at 1000rpm is a well-balanced engine. The final fabricated engine should expect vertical shaking forces on the first test run not too much greater than the virtual results assuming the fabricators follow the detail drawings closely.

Given below is the same plot run at 2500 rpm engine speed which is the maximum design speed of the engine. These results too are expected, the forces generated by the engine are proportionate to the square of the engine speed and thus increase dramatically as the engine revs. At maximum speed the virtual simulation which like at 1000rpm can be taken as the most probable result gives less than 100N shaking force.

Too put these results into better perspective the maximum out of balance force (shaking force) is still only 0.9% of the total generated shaking force.

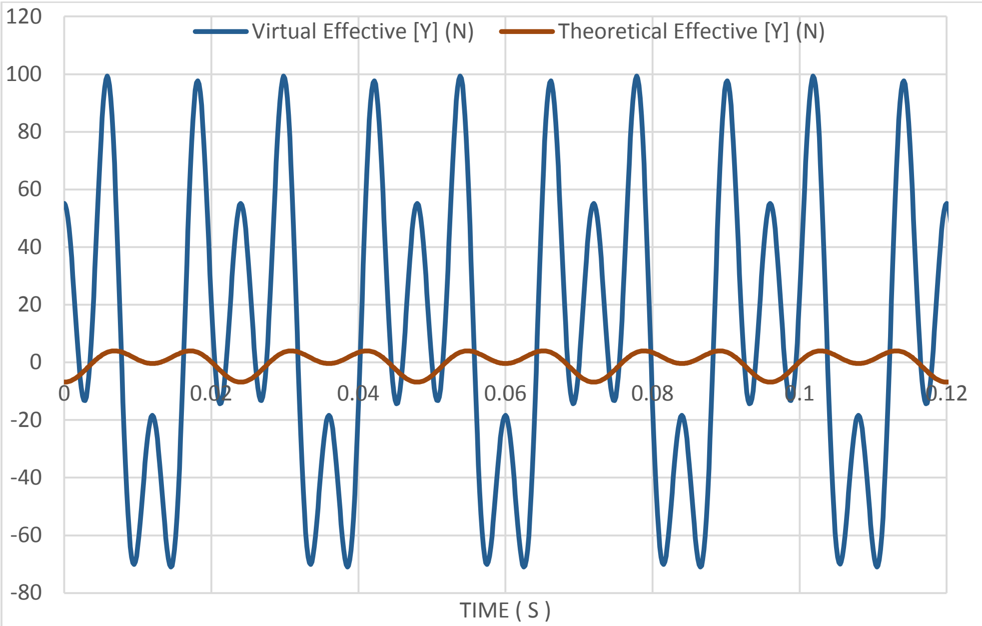


Figure 4.24 : Total engine shaking forces at 2500rpm. (Virtual and Theoretical)

Like at 1000rpm the end user should expect actual shaking forces to be not too far from the simulated results given above.

4.7 Conclusion of Results

Now that all analysis is concluded the results can be summarized with respect to the engine.

Key engine components have been rigorously design checked using wherever possible more than one method to give greater confidence to matching results and to alert of any mistakes when results diverge. Differences greater than ten percent were considered to be worth further investigation, fortunately few results exceeded this and the mistakes were easily corrected.

Several components have been design checked but escaped mention above in the interest of controlling the document size. To name few,

- Wrist pin
- Lower barrel main bolts and inserts

- Crankshaft bob-weight fasters
- Upper barrel tie rods
- Piston crown retaining bolts
- Counter balance shafts

The engine limitations with respect to key component needs to be briefly summarized.

The crankshaft is the limiting component with respect to cylinder pressure, although the part has been designed for infinite life at 10MPa cylinder pressure and 2500RPM this operating state is close to the limit of design and should not be pressed without careful consideration.

The conrod and wrist pin arrangement are the limiting parts with respect to engine speed. The crankshaft can handle over and above 2500RPM however the wrist pin and conrod eyelet are undergoing great stress at this speed.

With respect to temperature the rings are likely to be the first parts to fail under thermal load and care should be taken to ensure they don’t get too hot. After the rings the engine parts such as the piston, conrod and crankshaft need to be watched with respect to temperature. Crankshaft and conrod strength will drop rapidly after 200°C and the parts were design with only 200°C in mind with respect to yield strength.

Chapter 5

Conclusions

5.1 Project Compliance

Overall project specification compliance is considered to be high. The specification was drafted with the intent of working through the programme step by step in a near chronological order. The final outcome did not quite match this intent however the main targets were still met.

The original design review was the first point performed as per the first specification point; out of chronological order the materials research was performed next (point six) as many materials list in the original design we not available in Australia or at least not under the same name.

Research into balancing was performed as per point two of the specification utilizing the original engine designer’s (Prof. Richard Stone) own textbook. This theory was applied to the design both to validate the virtual simulation and further the balancing work as per point three.

The engine was modeled in Autodesk Inventor as per point five of the specification. This model was used and refined in the virtual simulations for balancing and FEA for stress based design. The model after the necessary refinements resulting from the analyses was used to produce the detail drawings found in appendix c as per specification points 7, 8 ,9 & 10.

To summarize the project aim was achieved with all key specification points fulfilled. Additional points to be fulfilled if time permitted were not attempted due to time constraints. The original project specification is contained in appendix A.

5.2 Materials Availability

During discussions with USQ it was agreed to keep the design materials to those which can be easily sourced from local suppliers. Great care was taken to achieve this target and with the exception of the quartz used for the optical bore and piston crown plus the aluminium alloy needed to make the piston this was achieved.

All alloy steels used in the design can be sourced from Bohler or one of their stockists such as Blackwoods. All aluminium except the piston material is produced here in Australia by Capral and freely available.

All bearings and small parts such as dowels, circlips and fasteners can be sources from any bearing supplier such as BSC or CBC. Blackwoods also stocks all of the parts and fasteners. The balancing material SD-170 otherwise known as Mallory Metal should be able to source through an engine rebuilder or balancing workshop.

5.3 Final Balancing Results

As can be seen in the body of this report a great amount of work was performed to balance this engine. The final results on the theoretical side were pleasing with than two Newtons of total shaking force at the maximum design speed of 2500rpm. The shaking force were found to reduce proportionately to the square of the engine speed as the speed reduces.

The virtual simulations results were also quite good with less than 100 Newtons (99.3N) of shaking force at 2500rpm. To put this into perspective this is less than 10 kilograms of load at the worst case. The virtual simulation also showed a reduction in shaking force proportionate to the square of the engine speed and maintained a magnitude directly proportionate to the theoretical results. This demonstrates the difference in results is largely due to the modelling methods paralleling realistic machine shop production.

Although no research papers read on optical access engines made mention of the magnitude of shaking force produced in their respective engines, it is unlikely the engine in this project would be any worse than those used. All single cylinder research engine balancing arrangements mentioned only noted two balance shafts.

This leads to the expectation that the primary forces were balanced at the crankshaft and the secondary forces were countered by the balance shafts. This method is quite sound in simplicity and effectiveness but is limited in how much of the primary forces can countered before creating excessive lateral shaking forces.

If this method is the normal approach better results can be expected from the USQ engine. Should other single cylinder designs employ four balance shafts like this engine similar results can be expected. Like always the final result will depend heavily on production quality and care taken during commissioning.

5.4 Temperature Limitations

Like all engines optical access engines such the one being designed here are subject to maximum operating temperatures. Along with this maximum allowable temperature there is a threshold temperature and duration for which the engine can operate.

The main limitation of this engine with respect to temperature is the ring and piston sleeve materials. The materials used for the rings is "Tetron C" and the sleeve is "Tetron B". Both are low friction PTFE / Metal polymer composites. Both have a limiting 3-4hr temperatures of 310°C and continuous operating temperatures 260°C. This means that the final engine operating setup will need to monitor piston temperatures or monitor ring wear and expect to replace the rings and sleeve frequently.

Although not mentioned in any technical papers read on the subject of optical access engines, it is expected that this procedure of regular replacement of the compression rings is quite normal. One particular paper by (Rosati et al. ,2009) acknowledges the harsh high temperature environment with the absence of lubrication and the suitability of materials like the PTFE composite used here and the need to replace periodically.

5.5 Operating Limitations

Component stress due to engine operation must be kept under control to prevent wear and breakage. Certain components will wear rapidly if operated at too high a speed such the bearings, other parts such as the crankshaft will fatigue if the cylinder pressure is allowed exceed the design pressure. Like any normal engine these limitations can be considered as the "redline". Should this engine need to be operated beyond this limit either redesign or engine wear will have to be accepted.

The limitations are summarized below,

- The maximum allowable piston temperature is 310°C for 3hrs or 250°C continuous.
- The maximum continuous engine speed is 2500RPM, short duration operation may reach 3000RPM however care should be taken particularly when the engine is hot.
- A maximum cylinder pressure of 10MPa is allowed for continuous operation, this pressure may be exceeded for short periods however like the engine speed care should be taken and preferably when the engine temperature in not too hot.
- The engine oil should be kept clean and at a minimum viscosity of SAE30. This is to maintain the oil film in the big end bearing at idling (750RPM).

Future Work

It would be desirable for USQ if they take up this design to fashion this optical access engine with features found on research engines of a world class standard. Although the real “cutting edge” features of optical research engines is in the diagnostic tools, some of which have already been mentioned in the literature review. By constructing a strong and versatile engine as the basis for a variety of combustion modes and respective diagnostics the engine of this project could perform as well as any optical research engine found in world class research laboratories such as Sandia Labs and General Motors Optical Research Laboratories.

The diagnostic tools required to make a research engine setup will not be discussed as this is outside the scope of this project. However below is a number of recommendations for any future design work on the engine itself that would be useful in making it a world class optical research engine.

Larger Optical Bore: Redesigning the optical bore to be larger and thus give greater viewing area would be in line with some of the more advanced and recent optical access engine designs (eg: The LOTUS S.C.O.R.E. , Ricardo’s Hydra and Proteus, The engines in use at Sandia Laboratories etc.)

This was considered by the current designer however it is difficult to design the optical bore without the intended cylinder head. It is for this reason the optical bore although modelled and drawn was left out of the design analysis and should not be considered as part of the current design.

Removable Optical Bore: Should the optical bore be redesigned it would be advisable to design it in such a way that is it not more than a couple of minutes work to remove. In doing so the piston’s wear components (rings and sleeve) can be easily replaced as can the bore itself in the event of requiring replacement.

Actuated Cylinder Head: A common feature among optical access engines is an actuated cylinder head (Raise/Lower). In the operation of the engine soot and other particles cover the piston window and optical bore. These parts thus require frequent cleaning to ensure good optical diagnostics. A cylinder head that can be raised to access the inside for cleaning would be a very useful feature.

Across several research papers taken from the SAE digital library this feature was noted. More commonly amongst university research setups than other facilities and more common again among newer constructed research engines.

Every paper that made mention of this feature described it as an essential or extremely useful feature.

Interchangeable Cylinder Heads: Since this engine was constructed to withstand a fairly high cylinder pressure (10MPa or 1450psi) it may be worth considering designing the top end to fit both a commercially available petrol engine head and a diesel engine head. This would be a useful feature for research and education and not commonly found in research engines. The Ricardo Hydra engine can be used in this fashion if the additional parts are purchased.

The existing engine design was slightly modified to match a 4G93 mitsubishi engine. If a commercially produced diesel engine with a similar bore and stroke exists then modifying the head and the necessary parts to fit would be all that is required.

Convex Piston Window: Designing a convex piston window for the bowditch piston to increase the viewing angle would be advantageous for the end users. Many optical research engines both commercially available and constructed in universities possess this feature. The current window was designed with a suitable safety margin so a slight reduction in thickness to accommodate this is allowable if required.

VF Controlled AC / DC Drive Motor: Many optical research engines employ a skip fire routine to control operating temperatures. This extends the life of the piston and specifically the rings through reduced surface temperatures. Should the university intend to run the engine in this fashion this feature will require designing.

It is expected the drive motor will be variable speed to give flexibility in operation. Numerous research papers in the SAE digital library make mention of the applicability of the skip fire routine and the basic arrangement with respect to the engine.

All Metal Sleeve: To run the engine in what is called a full thermodynamic cycle requires in many designs the removal of the optical bore. Although LOTUS and Ricardo claim to be able to run their engines at full load it unclear whether that is with the optical bore in place. Many research papers mentioned having to replace the optical bore with a full metal sleeve to run the engine out of the skip fire routine and in full thermodynamic cycle.

It would be prudent to determine if the engine requires this feature and if so include this feature in the next design.

Hydraulic Valve Timing (HVT) or Variable Valve Actuation (VVA): The most advanced optical research engines have features which allow the timing and depression of the valves. This gives operational control over the engine which has relevance in certain research areas. Currently LOTUS offer this feature called (HVT) and Ricardo have this feature option available under the abbreviation (VVA).

Design and construction of this feature would most likely be a project in itself, features such as these are quite mechanically advanced and would require a great deal of research, analysis and time.

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Appendices

The University of Southern Queensland

Faculty of Engineering & Surveying

ENG4111/ENG4112 Research Project
Project Specification

For: Kevin Bruce Dray *(Student No: 0011120331)*

Topic: Optical Access Engine Development

Supervisor: Professor David Buttsworth

Sponsorship: None

Project Aim: Develop an existing engine design into an optical access engine design to the point of full workshop drawings for manufacture.

Programme Issue: **(Version 1- 12th March, 2014)**

1. Review the 1994 Oxford University optical access engine design as provided to USQ and interpreted/adjusted by the previous project student.
2. Research & review the theory associated with the balancing of the piston, conrod & crank arrangement (literature review)
3. Apply learnt theory of mechanical balancing to determine initial engine component design details
4. Perform initial (back of the envelope) mechanical design calculations for various mechanical parts.
5. Develop a preliminary optical access engine model using Autodesk Inventor.
6. Perform materials research and selection for all model parts. (commercially acquired parts not included)
7. Research the use of Autodesk Mechanical Simulator to create/develop a model of operation simulation.
8. Refine the design based on simulation results
9. Perform secondary mechanical simulation using Autodesk mechanical simulator to determine final virtual balancing results.
10. Create detailed workshop drawings of the final engine design to AS1100
11. Write an academic dissertation on the project.

If time permits the project will include the following:

12. Determine engine peripherals details in conjunction with USQ. (ie: ignition system, stator (magneto), carburetor etc.)
13. Gather quotes for supply of various engine components.

Agreed: ___________ Kevin Dray *(Student)* ____________ David Buttsworth *(Supervisor)*

Examiner/Co-Examiner ______________________

Appendix B: Design Calculations

Engine Design Calculations Performed Using PTC MathCAD

Note: Nomenclature differs from that used in report body

The University of Southern Queensland

Project: Optical Access Engine Development
Subject: Eng4111/4112 Research Project

Student: Kevin Dray
Student No: 0011120331

Start Date: 07-06-2014

Revision A : 02-07-2014
Revision B : 17-09-2014
Revision C : 30-10-2014

Sections: (1.0) - Inputs & Initial Calculations
(2.0) - Engine balancing
(3.0) - Gas pressure & combustion
(4.0) - Engine component design.

Section 1.0 - Initial Inputs & Calculations

1.1 - Mass Inputs:

$m_p := 2.432\ kg$	[Piston Mass (Mp)]
$m_c := 0.882\ kg$	[Conrod Mass (Mc)]
$m_{wp} := 0.114\ kg$	[Wrist Pin Mass (Mwp)]
$m_{crankshaft} := 7.55\ kg$	[Crankshaft Mass]
$m_{pcb} := 2 \cdot 3.085\ kg = 6.17\ kg$	[Primary Balance Mass (Mpcb)]
$m_{scb} := 2 \cdot .997 \cdot kg = 1.994\ kg$	[Secondary Balance Mass (Mscb)]
$m_{flywheel} := 55\ kg$	

1.2 - Length Inputs:

$L_{th} := 44.5\ mm$	[Throw Length (Lth)]
$L_{co} := 160\ mm$	[Conrod Length (Lco)]
$L_{pbo} := 20\ mm$	[Primary Balance Offset (Lpbo)]
$L_{sbo} := 4.3\ mm$	[Secondary Balance Offset (Lsbo)]
$L_{cwo} := 3.923\ mm$	[Crankshaft Weight Offset (Lcwo)]
$L_{cro} := 118.68\ mm$	[Conrod COG to Little End Bearing Dist.]
$L_{bore} := 81\ mm$	[Bore Diameter]
$L_{brg\_out\phi} := 52\ mm$	[22205; C2205 bearing outside diameter]
$L_{brg\_ins\phi} := 25\ mm$	[22205; C2205 bearing inside diameter]

1.3 - Miscellaneous Inputs:

$RPM := 2500$ [Engine RPM (RPM)]

$i := 0 .. 720$ [Range variable representing 0-720 degrees crank angle]

$\mu := 0.02$ [Friction co-efficient between lubricated parts]

$\mu_{rolling\_brg} := 0.002$ [Friction co-efficient of rolling element bearing]

$\mu_{oil} := 0.11\ N \cdot \frac{s}{m^2}$ [Absolute viscosity of SAE30 engine oil at 40 °C]

$\rho_{oil} := 880\ \frac{kg}{m^3}$ [Density of Engine oil]

$g := 9.81\ \frac{m}{s^2}$ [Gravitational constant]

1.4 - Material Inputs:

$\sigma_{b\_4340} := 1005\ MPa$ [Ultimate Tensile Strength of 4340 -Heat Treated]

$\sigma_{y\_4340} := 710\ MPa$ [Yeild Strength of 4340 -Heat Treated]

$\sigma_{y\_200\_4340} := 510\ MPa$ [200 °C Yeild Strength of 4340 -Heat Treated]

$\sigma_{b\_4032AL} := 379\ MPa$ [Ultimate Tensile Strength of 4340 -Heat Treated]

$\sigma_{y\_4032AL} := 317\ MPa$ [Yeild Strength of 4340 -Heat Treated]

$E_{steel} := 210 \cdot 10^9\ Pa$ [Modulus of Elasticity - Steel]

$E_{4032AL} := 78.6 \cdot 10^9\ Pa$ [Modulus of Elasticity - 4032 Aluminium]

1.5 - Initial Calculations

$$\theta_i := 2 \cdot i \cdot \frac{\pi}{360}$$ [Crank Angle Vector]

$$\phi_i := \operatorname{asin}\left(\frac{L_{th}}{L_{co}} \cdot \sin\left(2 \cdot i \cdot \frac{\pi}{360}\right)\right)$$ [Conrod Angle Vector]

$$\omega := \frac{RPM \cdot 2 \cdot \pi}{60}\ s^{-1} = 261.799\ \frac{1}{s}$$ [Angular velocity]

$$x_{piston} := L_{th} \cdot \left(\cos(\theta) + \frac{L_{co}}{L_{th}} \left(1 - 0.5 \left(\frac{L_{th}}{L_{co}}\right)^2 (0.5 - 0.5 \cos(2 \cdot \theta))\right)\right)$$

[Piston displacement W.R.T. crank angle]

$$v_{piston} := -L_{th} \cdot \omega \cdot \left(\sin(\theta) + 0.5 \left(\frac{L_{th}}{L_{co}}\right) \sin(2 \cdot \theta)\right)$$

[Piston velocity W.R.T. crank angle]

$$A_{bore} := L_{bore}^{\ 2} \cdot \frac{\pi}{4} = \left(5.153 \cdot 10^{3}\right)\ mm^{2}$$ [Cylinder Cross Sectional Area]

Section 2.0 - Balancing Calculations

$$m_{rm} := m_c \cdot \frac{\left(L_{co} - L_{cro}\right)}{L_{co}} + m_{wp} + m_p = 2.774\ kg$$ [Reciprocating Mass]

$$m_{ro} := m_c \cdot \frac{L_{cro}}{L_{co}} + m_{crankshaft} = 8.204\ kg$$ [Rotating Mass]

$$L_{ro} := \frac{\left(\left(m_{crankshaft} \cdot L_{cwo}\right) - \left(m_c \cdot \frac{L_{cro}}{L_{co}} \cdot L_{th}\right)\right)}{m_{ro}}$$ [Rotating Mass Offset from Crank Centre]

$$a_{piston} := L_{th} \cdot \omega^{2} \cdot \left(\cos(\theta) + \left(\frac{L_{th}}{L_{co}}\right) \cos(2 \cdot \theta)\right)$$ [Piston acceleration W.R.T. crank angle]

$$Fy_{rm} := m_{rm} \cdot a_{piston}$$ [Vertical reciprocating forces W.R.T. crank angle]

$$Fy_{crankshaft} := -\left(m_{ro} \cdot \left|L_{ro}\right| \cdot \omega^{2} \cdot \cos(\theta + \pi)\right)$$ [Vertical crankshaft forces W.R.T. crank angle]

$$Fx_{crankshaft} := m_{ro} \cdot L_{ro} \cdot \omega^{2} \cdot \sin(\theta + \pi)$$ [Horizontal crankshaft forces W.R.T. crank angle]

$$Fy_{pcb} := m_{pcb} \cdot L_{pbo} \cdot \omega^{2} \cdot \cos(\theta + \pi)$$ [Vertical primary counterbalance forces W.R.T. crank angle]

$$Fy_{scb} := m_{scb} \cdot L_{sbo} \cdot (2 \cdot \omega)^{2} \cdot \cos(2 \cdot \theta + \pi)$$ [Vertical secondary counterbalance forces W.R.T. crank angle]

$$Fy_{ca} := Fy_{rm} + Fy_{crankshaft} - \left(m_{flywheel} \cdot g\right)$$ [Vertical forces acting through the crankshaft]

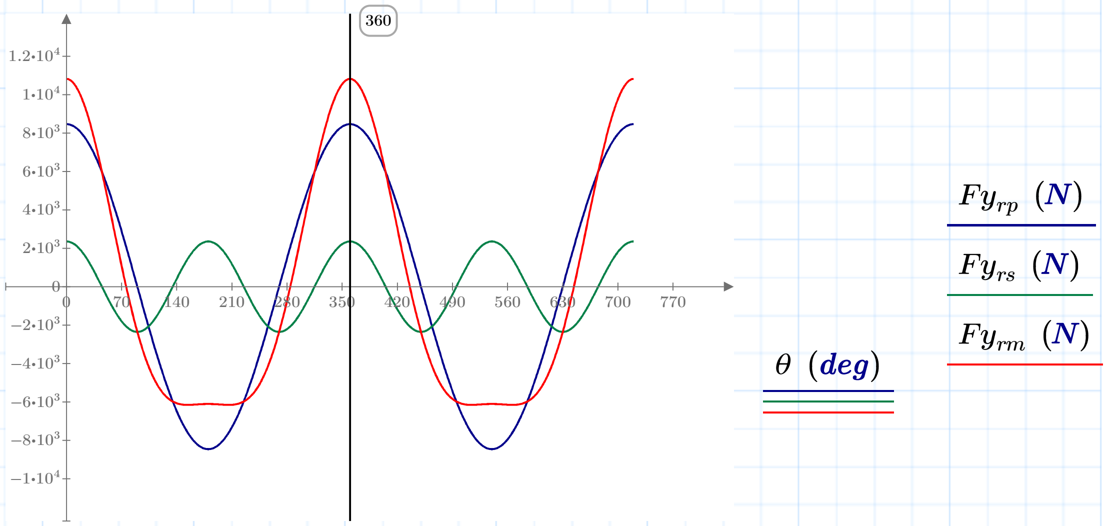
$$Fy_{total} := -\left(Fy_{rm} + Fy_{pcb} + Fy_{scb}\right)$$ [Total vertical shaking forces acting on the engine body]

$$Fy_{rp} := m_{rm} \cdot L_{th} \cdot \omega^{2} \cdot \cos(\theta)$$ [Vertical primary reciprocating forces]

$$Fy_{rs} := m_{rm} \cdot L_{th} \cdot \omega^{2} \cdot \left(\left(\frac{L_{th}}{L_{co}}\right) \cos(2 \cdot \theta)\right)$$ [Vertical secondary reciprocating forces]

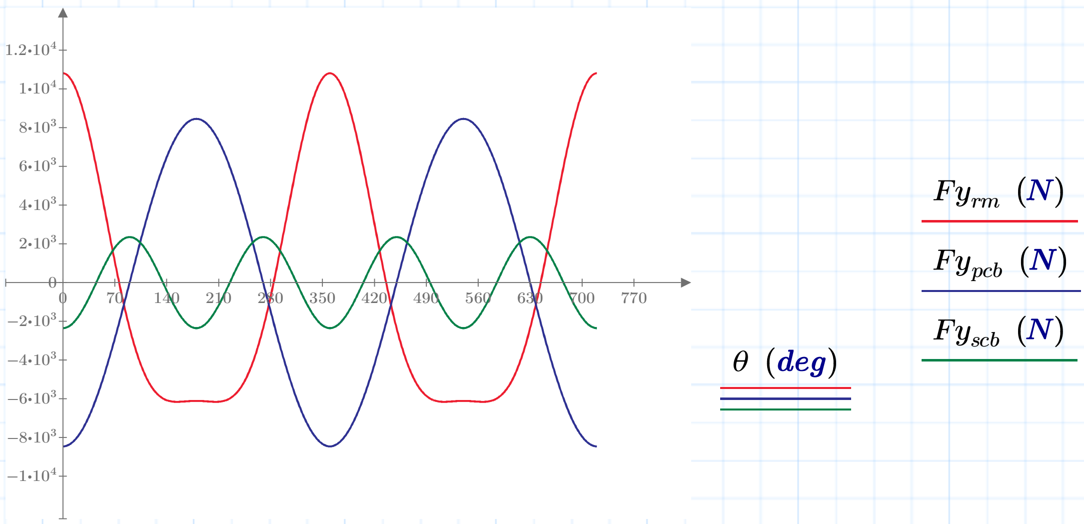
Reciprocating Forces Breakdown - Total, Primary & Secondary

(Vertical Shaking Force vs Crank Angle)



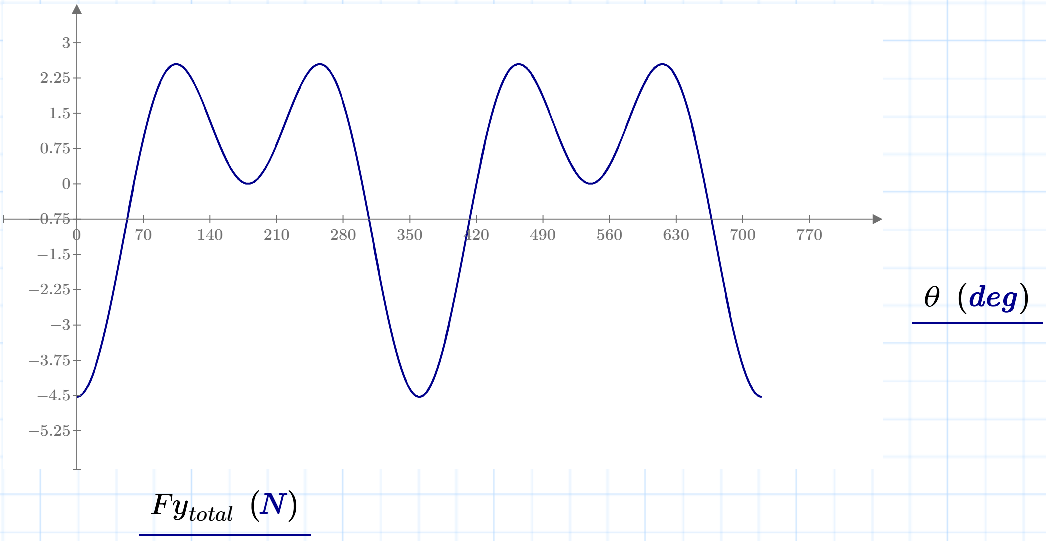
Shaking Forces Breakdown - Reciprocating, Primary Counterbalance & Secondary Counter Balance

(Vertical Shaking Force vs Crank Angle)



Final Shaking Forces - Sum of Reciprocating forces, Primary & Secondary Counter Balance Shaft Forces

(Vertical Shaking Force vs Crank Angle)



Section 3.0 - Gas Pressure Calculations

3.1 - Inputs:

$\gamma_{expansion} := 1.48$	[Adiabatic Constant during expansion]
$\gamma_{comp} := 1.3$	[Adiabatic Constant during compression]
$\gamma_{burn} := 1.25$	[Adiabatic Constant during combustion]
$Q_{hv} := 44 \cdot 10^{6} \ \dfrac{J}{kg}$	[Heating Value of Fuel]
$C_{fuel} := 0.90$	[Burnt fuel Constant]
$T_{air} := 298 \ K$	[Ambient Air Temp]
$V_{Chamber} := 0.05 \ L$	[Combustion Chamber Volume]
$\rho_{air} := 1.177 \ \dfrac{kg}{m^{3}}$	[Density of Air at 25degrees]
$S_{ratio} := 13$	[Stoichiometric Ratio]
$Cv_{air} := 1087 \ \dfrac{J}{kg \cdot K}$	[Specific Heat of Air at Constant Volume]
$\phi := 0.9$	[Equivalence Ratio]
$P_{air} := 101.3 \ kPa$	[Air Pressure]
$a_x := 5$	[Real burn constant -Weibe Function]
$m_x := 2$	[Real burn constant -Weibe Function]
$\theta_{\Delta} := 0.209 \cdot 2$	[Combustion start angle BTDC]
$\theta_o := 2 \ \pi - \left(\dfrac{\theta_{\Delta}}{2}\right) = 6.074$	[Combustion period in radians]
$j := 348 \, .. \, 372$	[Burn phase range matrix]
$\chi_j := 2 \cdot j \cdot \dfrac{\pi}{360}$	[Burn phase range matrix in C.A. Radians]
$k := 180 \, .. \, 348$	[Compression phase range matrix]
$\chi_k := 2 \cdot k \cdot \dfrac{\pi}{360}$	[Compression phase range matrix in C.A. Radians]
$l := 372 \, .. \, 540$	[Expansion phase range matrix]
$\chi_l := 2 \cdot l \cdot \dfrac{\pi}{360}$	[Compression phase range matrix in C.A. Radians]

3.2 - Gas Pressure Calculations

$$V_{CylVol} := L_{bore}{}^{2} \cdot \frac{\pi}{4} \cdot 2 \cdot L_{th}$$ [Cylinder Volume]

$$m_{cylair} := V_{CylVol} \cdot \rho_{air}$$ [Mass of air in cylinder]

$$m_{fuel} := \frac{m_{cylair}}{S_{ratio}} \cdot \phi$$ [Mass of Fuel in mixture]

$$T_{spark} := T_{air} \cdot \left(\frac{V_{CylVol}}{V_{Chamber}}\right)^{\gamma_{comp}-1}$$ [Charge Temp before ignition]

$$T_{burn\_end} := \frac{C_{fuel} \cdot m_{fuel} \cdot Q_{hv}}{\left(m_{fuel}+m_{cylair}\right) \cdot Cv_{air}} + T_{spark}$$ [Temperature at end of burn]

$$P_{burn\_end} := \frac{P_{air} \cdot \left(V_{CylVol}+V_{Chamber}\right) \cdot T_{burn\_end}}{\left(T_{air} \cdot V_{Chamber}\right)}$$ [Pressure at end of burn]

$$V_{\theta} := V_{Chamber} - \left(L_{bore}{}^{2} \cdot \frac{\pi}{4} \cdot \left(x_{piston} - \left(L_{co}+L_{th}\right)\right)\right)$$ [Chamber Volume wrt Crank Angle]

$$Xb_{j} := 1 - e^{\left(-a_x \left(\frac{\left(2 \cdot j \cdot \frac{\pi}{360}\right) - \theta_o}{\theta_{\Delta}}\right)^{m_x+1}\right)}$$ [Burn Fraction wrt Crank Angle]

$$x_{piston\_compression_k} := L_{th} \cdot \left(\cos\left(\chi_k\right) + \frac{L_{co}}{L_{th}} \left(1 - 0.5 \left(\frac{L_{th}}{L_{co}}\right)^{2} \left(0.5 - 0.5 \cos\left(2 \cdot \chi_k\right)\right)\right)\right)$$

[Piston position wrt Crank Angle in compression phase]

$$x_{piston\_burn\_start} := L_{th} \cdot \left(\cos\left(\theta_o\right) + \frac{L_{co}}{L_{th}} \left(1 - 0.5 \left(\frac{L_{th}}{L_{co}}\right)^{2} \left(0.5 - 0.5 \cos\left(2 \cdot \left(\theta_o\right)\right)\right)\right)\right)$$

[Piston position at burn start 12deg $BTDC$]

$$x_{piston\_burn_j} := L_{th} \cdot \left(\cos\left(\chi_j\right) + \frac{L_{co}}{L_{th}} \left(1 - 0.5 \left(\frac{L_{th}}{L_{co}}\right)^{2} \left(0.5 - 0.5 \cos\left(2 \cdot \chi_j\right)\right)\right)\right)$$

[Piston position during burn phase]

$$x_{piston\_burn\_end} := L_{th} \cdot \left(\cos\left(\theta_o + \theta_\Delta\right) + \frac{L_{co}}{L_{th}} \left(1 - 0.5 \left(\frac{L_{th}}{L_{co}} \right)^2 \left(0.5 - 0.5 \cos\left(2 \cdot \left(\theta_o + \theta_\Delta\right)\right)\right) \right) \right)$$

[Piston position at burn finish 12deg $ATDC$]

$$x_{piston\_expansion_l} := L_{th} \cdot \left(\cos\left(\chi_l\right) + \frac{L_{co}}{L_{th}} \left(1 - 0.5 \left(\frac{L_{th}}{L_{co}} \right)^2 \left(0.5 - 0.5 \cos\left(2 \cdot \chi_l\right)\right) \right) \right)$$

[Piston position wrt Crank Angle in expansion phase]

$$V_{compression_k} := V_{Chamber} - \left(L_{bore}^{\ 2} \cdot \frac{\pi}{4} \cdot \left(x_{piston\_compression_k} - \left(L_{co} + L_{th}\right) \right) \right)$$

[Chamber volume during compression phase]

$$V_{burn\_start} := V_{Chamber} - \left(L_{bore}^{\ 2} \cdot \frac{\pi}{4} \cdot \left(x_{piston\_burn\_start} - \left(L_{co} + L_{th}\right) \right) \right)$$

[Chamber volume at burn start 12deg $BTDC$]

$$V_{burn_j} := V_{Chamber} - \left(L_{bore}^{\ 2} \cdot \frac{\pi}{4} \cdot \left(x_{piston\_burn_j} - \left(L_{co} + L_{th}\right) \right) \right)$$

[Chamber volume during burn phase]

$$V_{burn\_end} := V_{Chamber} - \left(L_{bore}^{\ 2} \cdot \frac{\pi}{4} \cdot \left(x_{piston\_burn\_end} - \left(L_{co} + L_{th}\right) \right) \right)$$

[Chamber volume at burn finish 12deg $ATDC$]

$$V_{expansion_l} := V_{Chamber} - \left(L_{bore}^{\ 2} \cdot \frac{\pi}{4} \cdot \left(x_{piston\_expansion_l} - \left(L_{co} + L_{th}\right) \right) \right)$$

[Chamber volume during expansion phase]

$$P_{compression_k} := \frac{\left(P_{air} \cdot \left(V_{CylVol} + V_{Chamber}\right)^{\gamma_{comp}} \right)}{\left(V_{compression_k} \right)^{\gamma_{comp}}}$$

[Chamber pressure wrt to crank angle in compression phase]

$$P_{burn\_start} := \frac{\left(P_{air} \cdot \left(V_{CylVol} + V_{Chamber}\right)^{\gamma_{comp}} \right)}{\left(V_{burn\_start} \right)^{\gamma_{comp}}}$$

[Chamber pressure at burn start]

$$P_{burn_j} := \frac{\left(Xb_j \cdot \left(P_{burn\_end} \cdot V_{burn\_end}{}^{\gamma_{burn}} - P_{burn\_start} \cdot V_{burn\_start}{}^{\gamma_{burn}}\right) + P_{burn\_start} \cdot V_{burn\_start}{}^{\gamma_{burn}}\right)}{\left(V_{burn_j}\right)^{\gamma_{burn}}}$$

[Chamber pressure wrt to crank angle in burn phase]

$$P_{burn\_end} := \frac{P_{air} \cdot \left(V_{CylVol} + V_{Chamber}\right) \cdot T_{burn\_end}}{\left(T_{air} \cdot V_{Chamber}\right)}$$

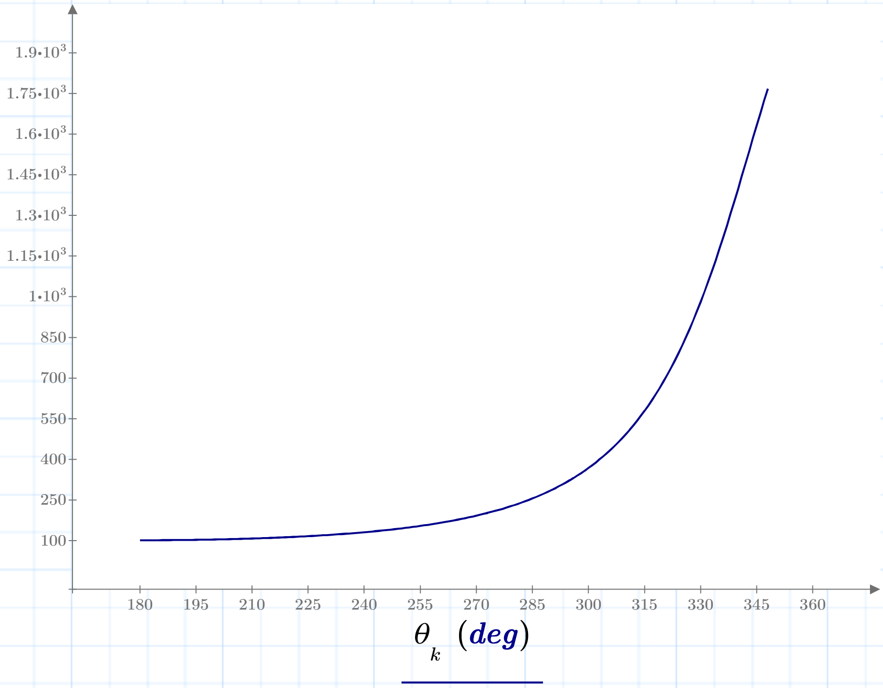
[Chamber pressure at burn end]

$$P_{expansion_l} := \frac{\left(P_{burn\_end} \cdot V_{burn\_end}{}^{\gamma_{expansion}}\right)}{\left(V_{expansion_l}\right)^{\gamma_{expansion}}}$$

[Chamber pressure wrt to crank angle in expansion phase]

Compression phase plot 180° BTDC to 12° BTDC

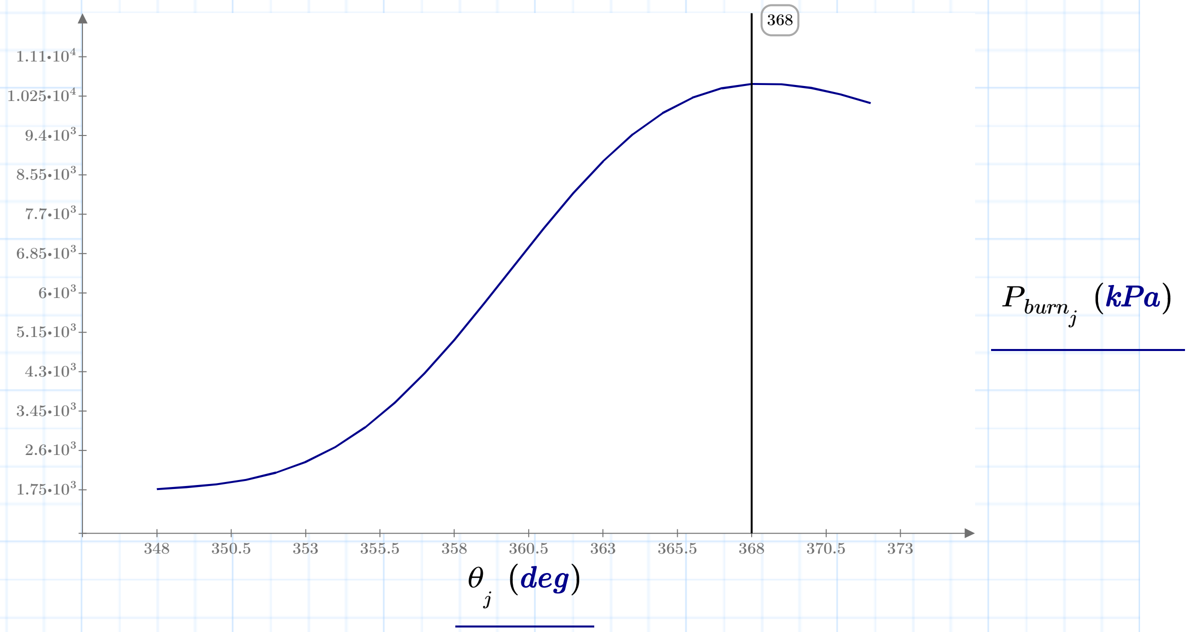
(Cylinder pressure vs crank angle)



$P_{compression_k}$ (kPa)

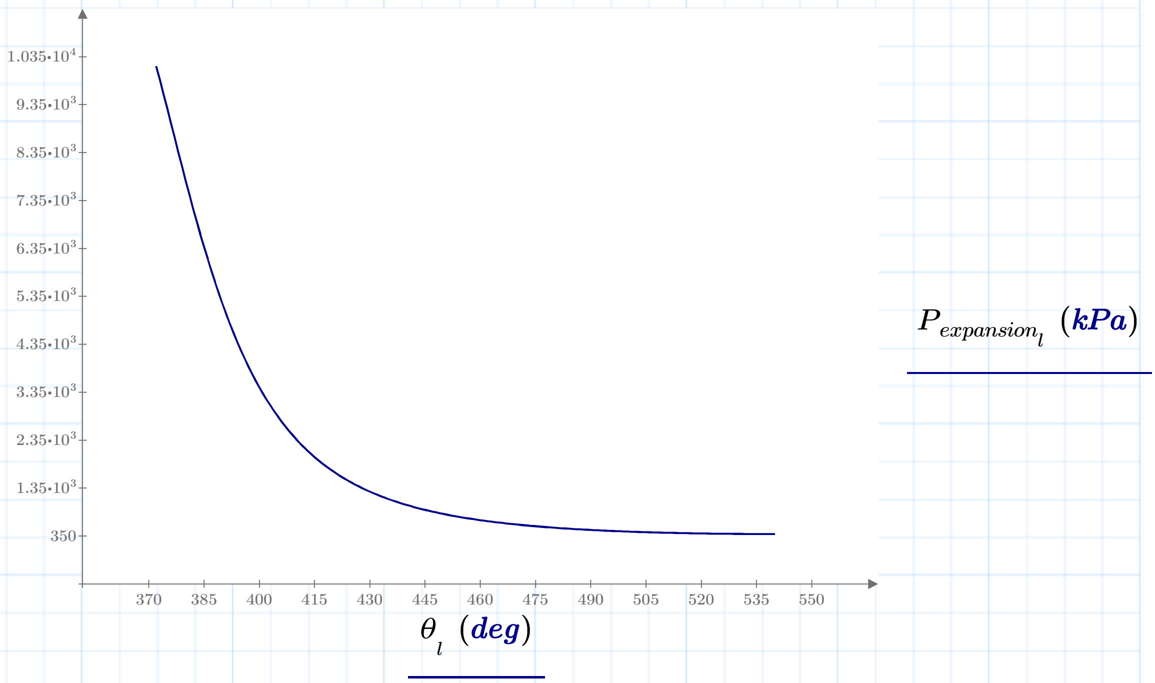
Burn phase plot 12° BTDC to 12° ATDC

(Cylinder pressure vs crank angle)



Expansion phase plot 12° ATDC to 180° ATDC

(Cylinder pressure vs crank angle)



Section 4.0 - Engine Component Design

Design calculations

1. Crankshaft Design (GL Design Paper, FEA Results and Fatigue Analysis)
2. Bearing Design (Balance Shaft, Main, Big End & Little End)
3. Counter Balance Shaft Design
4. Conrod Design
5. Piston Design
6. Wrist Pin Design

4.1 - Crankshaft Design

GL Design Paper Method

4.1.1 - Crankshaft Dimensions

$S := 8\ mm$	[Overlap dist.]
$B := 73.12\ mm$	[Web width]
$W := 19\ mm$	[Web thickness]
$DG := 60\ mm$	[Main bearing dia]
$D := 45\ mm$	[Crank pin dia]
$Rg := 2\ mm$	[Main bearing journal fillet]
$Rh := 2\ mm$	[Crank pin journal fillet]
$Do := 4.5\ mm$	[Oil bore dia]
$Dbg := 6\ mm$	[Main bearing oil bore dia]
$Dbh := 6\ mm$	[Crank pin axial oil bore dia]
$E := 44.5\ mm$	[Crankshaft Throw]
$L_1 := 49\ mm$	[See Figure]
$L_2 := 72\ mm$	[See Figure]
$L_3 := 128\ mm$	[See Figure]
$L_4 := 56\ mm$	[See Figure]
$L_5 := 33\ mm$	[See Figure]
$K := 1$	[Four Stroke Constant]

4.1.2 - Crankshaft Section Properties

$I_{x\_crankpin} := 198418.9\ mm^4$	[Crank Pin Moment of Inertia x-x]
$I_{y\_crankpin} := 164598.9\ mm^4$	[Crank Pin Moment of Inertia y-y]
$x_1 := -15.1\ mm$	[Oil outlet #1 x-position]
$x_2 := 18.94\ mm$	[Oil outlet #2 x-position]
$y_1 := 16.68\ mm$	[Oil outlet #1 y-position]
$y_2 := -11.73\ mm$	[Oil outlet #2 y-position]

$\psi_1 := 48\ deg$ [Oil outlet #1 angular position]

$\psi_2 := 32\ deg$ [Oil outlet #2 angular position]

$$W_{eqw} := \frac{(B \cdot W^2)}{6} = (4.399 \cdot 10^3)\ mm^3$$ [Web section modulus]

$$W_{main\_brg} := \frac{\pi}{16} \cdot \left(\frac{DG^4 - Dbg^4}{DG}\right)$$ [Main bearing journal polar section modulus]

$$W_{ecp\_no\_oil\_bore} := \frac{\pi}{32} \cdot \left(\frac{D^4 - Dbh^4}{D}\right) = (8.943 \cdot 10^3)\ mm^3$$ [Crankpin section modulus]

$$W_{ecp\_oil\_bore\_min} := \frac{(2.\ I_{y\_crankpin})}{D} = (7.316 \cdot 10^3)\ mm^3$$ [Crankpin section modulus through oil bore]

$$\lambda_{z\_crankpin} := \frac{W_{ecp\_oil\_bore\_min}}{W_{ecp\_no\_oil\_bore}} = 0.818$$ [Crankpin section modulus ratio]

4.1.3 - Slider/Crank Trigonometry Calculations

$$\phi_k := \operatorname{asin}\left(\frac{L_{th}}{L_{co}} \cdot \sin\left(2 \cdot k \cdot \frac{\pi}{360}\right)\right) \qquad \theta_k := 2 \cdot k \cdot \frac{\pi}{360}$$

[Crank & conrod angle during compression]

$$\phi_j := \operatorname{asin}\left(\frac{L_{th}}{L_{co}} \cdot \sin\left(2 \cdot j \cdot \frac{\pi}{360}\right)\right) \qquad \theta_j := 2 \cdot j \cdot \frac{\pi}{360}$$

[Crank & conrod angle during burn phase]

$$\phi_l := \operatorname{asin}\left(\frac{L_{th}}{L_{co}} \cdot \sin\left(2 \cdot l \cdot \frac{\pi}{360}\right)\right) \qquad \theta_l := 2 \cdot l \cdot \frac{\pi}{360}$$

[Crank & conrod angle during expansion]

$$\varphi_{comp_k} := \pi + \phi_k - (2 \cdot \pi - \theta_k)$$ [Conrod to Crank Arm angle during compression]

$$\varphi_{burn_j} := \pi - \left|\phi_j\right| - \left(\left|\theta_j - 2.\ \pi\right|\right)$$ [Conrod to Crank Arm angle during burn]

$$\varphi_{exp_l} := \pi - \phi_l - (\theta_l - 2.\ \pi)$$ [Conrod to Crank Arm angle during expansion]

4.1.4 - Conrod Load Calculations

$$a_{piston_k} := L_{th} \cdot \omega^2 \cdot \left(\cos\left(\theta_k\right) + \left(\frac{L_{th}}{L_{co}}\right) \cos\left(2 \cdot \theta_k\right) \right)$$

[Piston acceleration W.R.T. crank angle during compression phase]

$$Fy_{rm_k} := m_{rm} \cdot a_{piston_k}$$

[Vertical reciprocating forces W.R.T. crank angle during compression phase]

$$F_{c_k} := \frac{\left(\left(\left(P_{compression_k} - P_{air}\right) \cdot A_{bore}\right) - Fy_{rm_k}\right)}{\cos\left(\phi_k\right)}$$

[Conrod force W.R.T. crank angle during compression phase]

$$a_{piston_j} := L_{th} \cdot \omega^2 \cdot \left(\cos\left(\theta_j\right) + \left(\frac{L_{th}}{L_{co}}\right) \cos\left(2 \cdot \theta_j\right) \right)$$

[Piston acceleration W.R.T. crank angle during burn phase]

$$Fy_{rm_j} := m_{rm} \cdot a_{piston_j}$$

[Vertical reciprocating forces W.R.T. crank angle during burn phase]

$$F_{c_j} := \frac{\left(\left(\left(P_{burn_j} - P_{air}\right) \cdot A_{bore}\right) - Fy_{rm_j}\right)}{\cos\left(\phi_j\right)}$$

[Conrod force W.R.T. crank angle during burn phase]

$$a_{piston_l} := L_{th} \cdot \omega^2 \cdot \left(\cos\left(\theta_l\right) + \left(\frac{L_{th}}{L_{co}}\right) \cos\left(2 \cdot \theta_l\right) \right)$$

[Piston acceleration W.R.T. crank angle during expansion phase]

$$Fy_{rm_l} := m_{rm} \cdot a_{piston_l}$$

[Vertical reciprocating forces W.R.T. crank angle during expansion phase]

$$F_{c_l} := \frac{\left(\left(\left(P_{expansion_l} - P_{air}\right) \cdot A_{bore}\right) - Fy_{rm_l}\right)}{\cos\left(\phi_l\right)}$$

[Conrod force W.R.T. crank angle during expansion phase]

$$F_{r_k} := F_{c_k} \cdot \cos\left(\varphi_{comp_k}\right)$$

[Radial Forces acting on crankshaft WRT (tension/comp)] [Compression Phase]

$$F_{r_j} := F_{c_j} \cdot \cos\left(\varphi_{burn_j}\right)$$

[Radial Forces acting on crankshaft WRT (tension/comp)] [Burn Phase]

$$F_{r_l} := F_{c_l} \cdot \cos\left(\varphi_{exp_l}\right)$$

[Radial Forces acting on crankshaft WRT (tension/comp)] [Expansion Phase]

$$F_{r\_TDC} := \max\left(Fy_{rm}\right)$$

[Radial Force acting on crankshaft at TDC exhaust/intake]

$$F_{t_k} := F_{c_k} \cdot \sin\left(\varphi_{comp_k}\right)$$

[Tangential Forces acting on crankshaft WRT (tension/comp)] [Compression Phase]

$$F_{t_j} := F_{c_j} \cdot \sin\left(\varphi_{burn_j}\right)$$

[Tangential Forces acting on crankshaft WRT (tension/comp)] [Burn Phase]

$$F_{t_l} := F_{c_l} \cdot \sin\left(\varphi_{exp_l}\right)$$

[Tangential Forces acting on crankshaft WRT (tension/comp)] [Expansion Phase]

$$M_{t_k} := F_{t_k} \cdot E$$

[Produced Torque [Compression Phase]]

$$M_{t_j} := F_{t_j} \cdot E$$

[Produced Torque [Burn Phase]]

$$M_{t_l} := F_{t_l} \cdot E$$

[Produced Torque [Expansion Phase]]

4.1.5 - Crankshaft Bending Stress Calculations

4.1.5.1 - Bending Moments

$$M_{bro_k} := -F_{r_k} \cdot \frac{L_2 \cdot L_4}{L_3}$$

[Bending moment of the radial component of the crank shaft force (crank pin) [Compression Phase]

$$M_{bro_j} := -F_{r_j} \cdot \frac{L_2 \cdot L_4}{L_3}$$

[Bending moment of the radial component of the crank shaft force (crank pin) [Burn Phase]

$$M_{bro_l} := -F_{r_l} \cdot \frac{L_2 \cdot L_4}{L_3}$$

[Bending moment of the radial component of the crank shaft force (crank pin) [Expansion Phase]

$$M_{bro\_TDC} := \max\left(Fy_{rm}\right) \cdot \frac{L_2 \cdot L_4}{L_3}$$

[Bending moment of the radial component of the crank shaft force (crank pin) [TDC Exhaust/intake]

$$M_{bto_k} := -\left(F_{t_k} \cdot \frac{L_2 \cdot L_4}{L_3}\right)$$

[Bending moment of the tangential component of the crank shaft force (crank pin) [Compression Phase]

$$M_{bto_j} := F_{t_j} \cdot \frac{L_2 \cdot L_4}{L_3}$$

[Bending moment of the tangential component of the crank shaft force (crank pin) [Burn Phase]

$$M_{bto_l} := F_{t_l} \cdot \frac{L_2 \cdot L_4}{L_3}$$

[Bending moment of the tangetial component of the crank shaft force (crank pin) [Expansion Phase]

$$M_{brf_k} := F_{r_k} \cdot \frac{L_4 \cdot L_1}{L_3}$$

[Bending moment of the radial component of the crank shaft force (Web) [Compression Phase]

$$M_{brf_j} := F_{r_j} \cdot \frac{L_4 \cdot L_1}{L_3}$$

[Bending moment of the radial component of the crank shaft force (Web) [Burn Phase]

$$M_{brf_l} := F_{r_l} \cdot \frac{L_4 \cdot L_1}{L_3}$$

[Bending moment of the radial component of the crank shaft force (Web) [Expansion Phase]

$$M_{brf\_TDC} := -\max\left(Fy_{rm}\right) \cdot \frac{L_4 \cdot L_1}{L_3}$$

[Bending moment of the radial component of the crank shaft force (Web) [TDC Exhaust/intake]

4.1.5.2 - Bending Stress at the Oil Bore

$$\sigma_{bon1_k} := \frac{-\left(M_{bto_k} \cdot \cos\left(\psi_1\right) + M_{bro_k} \cdot \sin\left(\psi_1\right)\right)}{W_{ecp\_no\_oil\_bore}}$$

[Bending stress at crankpin oil hole #1] [Compression Phase]

$$\sigma_{bon2_k} := \frac{\left(M_{bto_k} \cdot \cos\left(\psi_2\right) + M_{bro_k} \cdot \sin\left(\psi_2\right)\right)}{W_{ecp\_no\_oil\_bore}}$$

[Bending stress at crankpin oil hole #2] [Compression Phase]

$$\sigma_{bo1_j} := \frac{-\left(M_{bto_j} \cdot \cos\left(\psi_1\right) + M_{bro_j} \cdot \sin\left(\psi_1\right)\right)}{W_{ecp\_no\_oil\_bore}}$$

[Bending stress at crankpin oil hole #1] [Burn Phase]

$$\sigma_{bo2_j} := \frac{\left(M_{bto_j} \cdot \cos\left(\psi_2\right) + M_{bro_j} \cdot \sin\left(\psi_2\right)\right)}{W_{ecp\_no\_oil\_bore}}$$

[Bending stress at crankpin oil hole #2] [Burn Phase]

$$\sigma_{bo1_l} := \frac{-\left(M_{bto_l} \cdot \cos\left(\psi_1\right) + M_{bro_l} \cdot \sin\left(\psi_1\right)\right)}{W_{ecp\_no\_oil\_bore}}$$

[Bending stress at crankpin oil hole #1] [Expansion Phase]

$$\sigma_{bo2_l} := \frac{\left(M_{bto_l} \cdot \cos\left(\psi_2\right) + M_{bro_l} \cdot \sin\left(\psi_2\right)\right)}{W_{ecp\_no\_oil\_bore}}$$ [Bending stress at crankpin oil hole #2] [Expansion Phase]

4.1.5.3 - Bending Stresses for Fatigue Analysis

(a) - Oil Bore

$\sigma_{bo1\_max} := 5.97\ MPa$ [Crankpin Oil Hole #1 Max Stress (uncorrected) at BDC] (540deg)

$\sigma_{bo1\_min} := -159.5\ MPa$ [Crankpin Oil Hole #1 Min Stress (uncorrected) at 10deg ATDC] (370deg)

$\sigma_{bo2\_max} := 130.8\ MPa$ [Crankpin Oil Hole #2 Max Stress (uncorrected) at 12deg ATDC] (372deg)

$\sigma_{bo2\_min} := -4.26\ MPa$ [Crankpin Oil Hole #2 Min Stress (uncorrected) at BDC] (540deg)

$$\sigma_{bon} := \frac{\left(\sigma_{bo1\_max} - \sigma_{bo1\_min}\right)}{2} = 82.735\ MPa \text{ “+/−”}$$ [Oil bore alternating stress]

$$\sigma_{bo\_mean} := \frac{\left(\sigma_{bo1\_max} + \sigma_{bo1\_min}\right)}{2} = -76.765\ MPa$$ [Oil bore mean stress]

(b) - Centre of Web

$M_{brf\_max} := 1097\ N \cdot m$ **8deg ATDC** [Max bending moment centre of web]
$M_{brf\_min} := -231.8\ N \cdot m$ **360deg ATDC** [Min bending moment centre of web]

$$M_{brfn} := \frac{\left(M_{brf\_max} - M_{brf\_min}\right)}{2} \text{ “+/−”}$$ [Alternating bending moment centre of web]

$M_{brfn} = 664.4\ N \cdot m$ “+/−”

$$M_{brf\_mean} := \frac{\left(M_{brf\_max} + M_{brf\_min}\right)}{2}$$ [Mean bending moment centre of web]

$M_{brf\_mean} = 432.6\ N \cdot m$

$$\sigma_{bfn} := \frac{M_{brfn}}{W_{eqw}} = 151.021\ MPa$$ [Max alternating stress centre of web]

$$\sigma_{bf\_max} := \frac{M_{brf\_mean} + M_{brfn}}{W_{eqw}} = 249.353\ MPa$$ [Max bending moment centre of web]

$$\sigma_{bf\_min} := \frac{M_{brf\_mean} - M_{brfn}}{W_{eqw}} = -52.689\ MPa$$ [Min bending moment centre of web]

(c) - Radial Stress in Web

$F_{rf\_max} := F_{r\_TDC} = \left(1.081 \cdot 10^{4}\right)\ N$ **360deg ATDC** [Max radial force acting on web]

$F_{rf\_min} := -51190\ N$ **8deg ATDC** [Min radial force acting on web]

$$F_{rn} := \frac{\left(F_{rf\_max} - F_{rf\_min}\right)}{2} = \left(3.1 \cdot 10^{4}\right)\ N$$ "+/−" [Alternating radial force acting on web]

$$F_{rf\_mean} := \frac{\left(F_{rf\_max} + F_{rf\_min}\right)}{2} = -2.019 \cdot 10^{4}\ N$$ [Mean radial force acting on web]

$$\sigma_{rfn} := \frac{F_{rn}}{W \cdot B} = 22.315\ MPa$$ "+/−" [Alternating radial stress in web]

$$\sigma_{rfn\_max} := \frac{F_{rf\_mean} + F_{rn}}{W \cdot B} = 7.783\ MPa$$ [Max radial stress in web]

$$\sigma_{rfn\_min} := \frac{F_{rf\_mean} - F_{rn}}{W \cdot B} = -36.846\ MPa$$ [Min radial stress in web]

(d) - Crank Torsion

$M_{t\_max} := 742.3\ N \cdot m$ @22deg ATDC [Max driving torque]

$M_{t\_min} := -104.5\ N \cdot m$ @21deg BTDC [Max slowing torque]

$$M_{tn} := \frac{\left(M_{t\_max} - M_{t\_min}\right)}{2} = 423.4\ N \cdot m$$ "+/−" [Alternating torque]

$$M_{t\_mean} := \frac{\left(M_{t\_max} + M_{t\_min}\right)}{2} = 318.9\ N \cdot m$$ [Mean torque]

$$\tau_{mbj} := \frac{M_{tn}}{W_{main\_brg}} = 9.984\ MPa$$ [Alternating torsion stress in main bearing]

$$\tau_{cpj} := \frac{M_{tn}}{W_{ecp\_no\_oil\_bore}} = 47.342\ MPa$$ "+/−" [Alternating torsion stress in crankpin]

$$\tau_{cpj\_max} := \frac{M_{t\_mean} + M_{tn}}{W_{ecp\_no\_oil\_bore}} = 83 \; MPa$$ [Max main bearing torsion stress]

$$\tau_{cpj\_min} := \frac{M_{t\_mean} - M_{tn}}{W_{ecp\_no\_oil\_bore}} = -11.685 \; MPa$$ [Min main bearing torsion stress]

$$\tau_{mbj\_max} := \frac{M_{t\_mean} + M_{tn}}{W_{main\_brg}} = 17.504 \; MPa$$ [Max main bearing torsion stress]

$$\tau_{mbj\_min} := \frac{M_{t\_mean} - M_{tn}}{W_{main\_brg}} = -2.464 \; MPa$$ [Min main bearing torsion stress]

4.1.6 - Crankshaft Fatigue Analysis (Design Paper Method)

4.1.6.1 - Stress Concentration Factors

$\alpha_b := 2.47$ [SCF: Crankpin fillet bending]

$\alpha_t := 2.018$ [SCF: Crankpin Torsion]

$\beta_b := 2.211$ [SCF: Main Bearing Fillet bending]

$\beta_q := 2.229$ [SCF: Main Bearing Fillet bending]

$\beta_t := 2.113$ [SCF: Main Bearing Fillet Torsion]

$\gamma_b := 2.77$ [SCF: Crankpin oil bore bending]

$\gamma_t := 3.703$ [SCF: Crankpin oil bore torsion]

4.1.6.2 - Factored Stress Values

$\sigma_{bh} := \alpha_b \cdot \sigma_{bfn} = 373.022 \; MPa$ [Factored crankpin fillet alternating bending stress]

$\sigma_{bg} := \beta_b \cdot \sigma_{bfn} + \beta_q \cdot \sigma_{rfn} = 383.647 \; MPa$ [Factored main bearing journal fillet alternating bending stress]

$\sigma_{bo} := \gamma_b \cdot \sigma_{bon} = 229.176 \; MPa$ [Factored crankpin oil bore alternating bending stress]

$\tau_g := (\beta_t \cdot \tau_{mbj}) = 21.096\ MPa$ [Factored main bearing alternating torsional stress]

$\tau_h := (\alpha_t \cdot \tau_{mbj}) = 20.148\ MPa$ [Factored crankpin fillet alternating torsional stress]

$\sigma_{to} := (\gamma_t \cdot \tau_{cpj}) = 175.309\ MPa$ [Factored crankpin oil bore alternating torsional stress]

$\sigma_{to} := 1\ MPa$ [Corrected oil bore torsion for single cylinder engine]

4.1.6.3 - Factored Equivalent Stress (Von-Mises)

$\sigma_{v\_cp\_fillet} := \sqrt{\left(\sigma_{bh}^{\ 2} + 3 \cdot \tau_h^{\ 2}\right)} = 374.651\ MPa$ [Factored von-mises alternating stress in crankpin fillet]

$\sigma_{v\_mb\_fillet} := \sqrt{\left(\sigma_{bg}^{\ 2} + 3 \cdot \tau_g^{\ 2}\right)} = 385.383\ MPa$ [Factored von]-mises alternating stress in main bearing fillet]

$\sigma_{v\_bo} := \frac{\sigma_{bo}}{3} \cdot \left(1 + 2 \cdot \sqrt{1 + \frac{9}{4} \cdot \left(\frac{\sigma_{to}}{\sigma_{bo}}\right)^2}\right) = 229.179\ MPa$ [Factored von-mises alternating stress in crankpin oil bore]

4.1.6.4 - Allowable Fatigue Stress

$$\sigma_{dw\_cpj} := K \cdot (.42 \cdot \sigma_{b\_4340} + 39.3\ MPa) \cdot \left(0.264 + 1.073 \cdot \left(\frac{D}{mm}\right)^{-0.2} + \frac{\left(785 - \left(\frac{\sigma_{b\_4340}}{MPa}\right)\right)}{4900} + \frac{196}{\left(\frac{(\sigma_{b\_4340})}{MPa}\right)} \cdot \sqrt{\frac{1}{\left(\frac{Rh}{mm}\right)}}\right)$$

[Maximum allowable fatigue stress in crankpin fillet]

$$\sigma_{dw\_bo} := K \cdot (.42 \cdot \sigma_{b\_4340} + 39.3\ MPa) \cdot \left(0.264 + 1.073 \cdot \left(\frac{D}{mm}\right)^{-0.2} + \frac{\left(785 - \left(\frac{\sigma_{b\_4340}}{MPa}\right)\right)}{4900} + \frac{196}{\left(\frac{(\sigma_{b\_4340})}{MPa}\right)} \cdot \sqrt{\frac{1}{\left(\frac{\left(\frac{Do}{2}\right)}{mm}\right)}}\right)$$

[Maximum allowable fatigue stress in crankpin oil bore]

$$\sigma_{dw\_mbj} := K \cdot \left(.42 \cdot \sigma_{b\_4340} + 39.3\ MPa\right) \cdot \left(0.264 + 1.073 \cdot \left(\frac{DG}{mm}\right)^{-0.2} + \frac{\left(785 - \left(\frac{\sigma_{b\_4340}}{MPa}\right)\right)}{4900} + \frac{196}{\left(\frac{\left(\sigma_{b\_4340}\right)}{MPa}\right)} \cdot \sqrt{\frac{1}{\left(\frac{Rg}{mm}\right)}}\right)$$

[Maximum allowable fatigue stress in main bearing fillet]

4.1.7 - Crankshaft Dimensional Acceptability

$$Q_{cpj} := \frac{\sigma_{dw\_cpj}}{\sigma_{v\_cp\_fillet}} = 1.057$$ [Crankpin journal fillet acceptability factor]

$$Q_{bo} := \frac{\sigma_{dw\_bo}}{\sigma_{v\_bo}} = 1.712$$ [Crankpin oil bore acceptability factor]

$$Q_{mbj} := \frac{\sigma_{dw\_mbj}}{\sigma_{v\_mb\_fillet}} = 0.994$$ [Main bearing journal fillet acceptibility factor]

F.E.A. Design Method

4.1.8 - Crankshaft FEA Results (Design Revision A)

$\sigma_{cp\_fillet\_max} := 637\ MPa$ [Crank pin journal fillet max stress]

$\sigma_{cp\_fillet\_min} := -116.6\ MPa$ [Crank pin journal fillet min stress]

$$\sigma_{alt\_a\_a\_cpj} := \frac{\sigma_{cp\_fillet\_max} - \sigma_{cp\_fillet\_min}}{2} = 376.8\ MPa$$ "+/−"
[Journal alternating stress]

$$\sigma_{mean\_a\_a\_cpj} := \frac{\sigma_{cp\_fillet\_max} + \sigma_{cp\_fillet\_min}}{2} = 260.2\ MPa$$ [Journal mean stress]

$\sigma_{oil\_bore\_max} := 63.5\ MPa$ [Crank pin oil bore max stress]

$\sigma_{oil\_bore\_min} := -300\ MPa$ [Crank pin oil bore min stress]

$$\sigma_{alt\_a\_a\_bo} := \frac{\sigma_{oil\_bore\_max} - \sigma_{oil\_bore\_min}}{2} = 181.75\ MPa$$ "+/−"
[Oil bore alternating stress]

$$\sigma_{mean\_a\_a\_bo} := \frac{\sigma_{oil\_bore\_max} + \sigma_{oil\_bore\_min}}{2} = -118.25\ MPa$$ [Oil bore mean stress]

$\sigma_{mainbrg\_fillet\_max} := 122.7\ MPa$ [Main journal fillet max stress]

$\sigma_{mainbrg\_fillet\_min} := -547\ MPa$ [Main journal fillet min stress]

$$\sigma_{alt\_a\_a\_mbj} := \frac{\sigma_{mainbrg\_fillet\_max} - \sigma_{mainbrg\_fillet\_min}}{2} = 334.85\ MPa$$ "+/−"

[Main journal alternating stress]

$$\sigma_{mean\_a\_a\_mbj} := \frac{\sigma_{mainbrg\_fillet\_max} + \sigma_{mainbrg\_fillet\_min}}{2} = -212.15\ MPa$$

[Main journal mean stress]

$\Delta_{crankpin\_max\_8ATDC} := 0.124\ mm$ [Crankpin vertical displacement]

$\Delta_{web\_max} := 0.274\ mm$ [Web horizontal displacement]

4.1.9 - Crankshaft FEA results (Design Revision B)

$\sigma_{cp\_fillet\_max} := 452\ MPa$ [Crank pin journal fillet max stress]

$\sigma_{cp\_fillet\_min} := -86.9\ MPa$ [Crank pin journal fillet min stress]

$$\sigma_{alt\_a\_a\_cpj} := \frac{\sigma_{cp\_fillet\_max} - \sigma_{cp\_fillet\_min}}{2} = 269.45\ MPa$$ "+/−"
[Journal alternating stress]

$$\sigma_{mean\_a\_a\_cpj} := \frac{\sigma_{cp\_fillet\_max} + \sigma_{cp\_fillet\_min}}{2} = 182.55\ MPa$$ [Journal mean stress]

$\sigma_{oil\_bore\_max} := 51.1\ MPa$ [Crank pin oil bore max stress]

$\sigma_{oil\_bore\_min} := -225\ MPa$ [Crank pin oil bore min stress]

$$\sigma_{alt\_a\_a\_bo} := \frac{\sigma_{oil\_bore\_max} - \sigma_{oil\_bore\_min}}{2} = 138.05\ MPa$$ "+/−"

[Oil bore alternating stress]

$$\sigma_{mean\_a\_a\_bo} := \frac{\sigma_{oil\_bore\_max} + \sigma_{oil\_bore\_min}}{2} = -86.95\ MPa$$ [Oil bore mean stress]

$\sigma_{mainbrg\_fillet\_max} := 96.6\ MPa$ [Main journal fillet max stress]

$\sigma_{mainbrg\_fillet\_min} := -449\ MPa$ [Main journal fillet min stress]

$$\sigma_{alt\_a\_a\_mbj} := \frac{\sigma_{mainbrg\_fillet\_max} - \sigma_{mainbrg\_fillet\_min}}{2} = 272.8\ MPa$$ "+/−"

[Main journal alternating stress]

$$\sigma_{mean\_a\_a\_mbj} := \frac{\sigma_{mainbrg\_fillet\_max} + \sigma_{mainbrg\_fillet\_min}}{2} = -176.2\ MPa$$

[Main journal mean stress]

$\Delta_{crankpin\_max\_8ATDC} := 0.109\ mm$ [Crankpin vertical displacement]

$\Delta_{web\_max} := 0.234\ mm$ [Web horizontal displacement]

4.1.10 - Crankshaft Fatigue Analysis (Design Revision B)

4.1.10.1 - Endurance Limit Coefficients

$$k_a := 1.58 \cdot \left(\frac{\sigma_{b\_4340}}{MPa}\right)^{-.085} = 0.878$$ [Surface Factor: Shigley's Eq 6-19 & Table 6-2]

$$k_b := 1.24 \cdot \left(\frac{D}{mm}\right)^{-0.107} = 0.825$$ [Size Factor: Shigley's Eq 6-20]

$k_{d\_200} := 1.02$ [Temperature Factor: Shigley's Table 6-4]

$k_{e\_50} := 1$ [Reliability Factor: Shigley's Table 6-5]

4.1.10.2 - Endurance Limit

$$Se_{\_4340} := \frac{\sigma_{b\_4340}}{2} \cdot k_a \cdot k_b \cdot k_{d\_200} \cdot k_{e\_50} = 371.31\ MPa$$

[Material Endurance Limit]

$$nf_{\_cpj} := 0.5 \cdot \left(\frac{\sigma_{b\_4340}}{\sigma_{mean\_a\_a\_cpj}}\right)^2 \cdot \left(\frac{\sigma_{alt\_a\_a\_cpj}}{Se_{\_4340}}\right) \cdot \left(-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_{mean\_a\_a\_cpj} \cdot Se_{\_4340}}{\sigma_{alt\_a\_a\_cpj} \cdot \sigma_{b\_4340}}\right)^2}\right)$$

$$nf_{\_cpj} = 1.301$$

[Fatigue Factor of Safety for crankpin journal]

Note: Other points on the crankshaft do not require fatigue analysis other than ensuring any local alternating stress in less than the local endurance limit (Shigley's,2012)

4.2 - Bearing Design

Bearing Life & Bearing Loads Calculations

1. Rolling bearing life calculations
2. Main bearing calculations
3. Big end bearing calculations
4. Little end bearing calculations

4.2.1 - Rolling Bearing Calculations (Counterbalance Shaft Bearings)

$$RPM_{top} := 2500\ \frac{rev}{min} = 261.799\ \frac{rad}{s}$$

[Engine design speed as at 28/9/14]

4.2.1.1 - Bearing Details

$$C_{22205} := 49\ kN$$ [Dynamic Load Rating (22205 bearing)]

$$C_{c2205} := 50\ kN$$ [Dynamic Load Rating (c2205 bearing)]

4.2.1.2 - Bearing Life Calculations

$$F_{pcb} := \frac{m_{pcb}}{2} \cdot \left(RPM_{top}\right)^2 \cdot L_{pbo} = 4.229\ kN$$

[Radial Force From Primary Counter Weight]

$$F_{scb} := \frac{m_{scb}}{2} \cdot \left(2 \cdot RPM_{top}\right)^2 \cdot L_{sbo} = 1.175\ kN$$

[Radial Force from Secondary Counter Weight]

$$BL_{22205} := \frac{10^6}{60 \cdot RPM_{top}} \left(\frac{C_{22205}}{\frac{F_{pcb}}{2}} \right)^{\left(\frac{10}{(3)}\right)} = 627.465\ hr$$

[Bearing Life in Hours (22205 Bearing) on the PCB shaft]

$$BL_{N205ECP} := \frac{10^6}{60 \cdot 2 \cdot RPM_{top}} \left(\frac{C_{22205}}{\frac{F_{scb}}{2}} \right)^{\left(\frac{10}{(3)}\right)} = \left(2.239 \cdot 10^4\right)\ hr$$

[Bearing Life in Hours (22205 Bearing) on the SCB shaft]

4.2.2 - Main bearing calculations

$$N_{idle} := 13.33\ \frac{rev}{s} = 799.8\ \frac{rev}{min}$$

$$N_{design\_speed} := 13.33\ \frac{rev}{s} = 799.8\ \frac{rev}{min}$$

4.2.2.1 - Main bearing details

$L_{\phi\_bearing\_flywheel} := 60\ mm$ [Flywheel side main bearing diameter]

$L_{\phi\_bearing\_encoder} := 60\ mm$ [Encoder side main bearing diameter]

$L_{bearing\_flywheel} := 40\ mm$ [Flywheel side main bearing length]

$L_{bearing\_encoder} := 40\ mm$ [Flywheel side main bearing length]

$c_{radial\_clearance} := 0.065\ mm$

$fr_{on}c\_idle := 5$

$\varepsilon := 0.62$

$P_{idle} := 120\ kPa$

4.2.2.2 - Main bearing calculations

$$P := \left| \frac{\frac{F_{rf\_min}}{2}}{L_{\phi\_bearing\_flywheel} \cdot L_{bearing\_flywheel}} \right| = 10.665\ MPa$$

[Nominal main bearing pressure]

$$S_{sommerfield\_no\_mainbrg} := \left(\frac{\frac{L_{\phi\_bearing\_flywheel}}{2}}{c_{radial\_clearance}} \right)^2 \cdot \left(\frac{\mu_{oil} \cdot N_{design\_speed}}{P} \right) = 0.184$$

[Sommerfield Number]

$$PU_{main\_brg} := P \cdot \frac{L_{\phi\_bearing\_flywheel}}{2} \cdot N_{design\_speed} = 26.796 \ \frac{MPa \cdot m}{s}$$

[Nominal main bearing pressure velocity]

$$h_o := 0.00508 \ mm + 0.00004 \cdot D = 0.00688 \ mm$$

[Minimum bearing clearance req'd under full load]

$$\Delta T := \frac{978000000 \cdot fr_{on}c\_idle \cdot S_{sommerfield\_no\_mainbrg} \cdot \left| \left(\frac{F_{rf\_min}}{2 \cdot kN} \right)^2 \right|}{\left(1 + 1.5 \cdot \varepsilon^2\right) \cdot \frac{P_{idle}}{kPa} \cdot \left(\frac{DG}{2 \cdot mm} \right)^4} = 3.847 \cdot 10^3$$

[Temperature rise in oil at instant of max cylinder pressure at design speed]

4.2.3 - Big-End Bearing Calculations

4.2.3.1 - Big-End Bearing Details

$L_{\phi\_BEbearing} := 48 \ mm$ [Big-end bearing diameter]

$L_{BEbearing} := 22.6 \ mm$ [Big-end bearing length]

$c_{radial\_clearance} := 0.055 \ mm$ [Big-end bearing radial clearance]

4.2.3.2 - Bearing Calculations

$$P_{BEbearing} := \frac{\left| F_{rf\_min} \right|}{L_{\phi\_BEbearing} \cdot L_{BEbearing}} = 47.188 \ MPa$$

[Nominal BE bearing pressure]

$$S_{sommerfield\_no\_BEbrg} := \left(\frac{\frac{L_{\phi\_BEbearing}}{2}}{c_{radial\_clearance}} \right)^2 \cdot \left(\frac{\mu_{oil} \cdot N_{design\_speed}}{P_{BEbearing}} \right) = 0.037$$

[Sommerfield Number]

$$PU_{BE\_brg} := P_{BEbearing} \cdot \frac{L_{\phi\_BEbearing}}{2} \cdot N_{design\_speed} = 94.854 \ \frac{MPa \cdot m}{s}$$

[Nominal BE bearing pressure velocity]

$$h_o := 0.00508 \ mm + 0.00004 \cdot L_{\phi\_BEbearing} = 0.007 \ mm$$

[Minimum bearing clearance req'd under full load]

4.2.4 - Little-End Bearing Calculations

$L_{\phi\_LEbearing} := 20 \ mm$ [LE bearing Diameter]

$L_{LEbearing} := 20 \ mm$ [LE bearing Width]

$$P_{LE\_max} := \frac{4}{\pi} \cdot \frac{\left| F_{rf\_min} \right|}{L_{\phi\_LEbearing} \cdot L_{LEbearing}} = 162.943 \ MPa$$

[LE bearing max pressure]

$$PU_{LE\_brg} := P_{LE\_max} \cdot L_{\phi\_LEbearing} \cdot \frac{L_{th}}{L_{co}} \cdot N_{design\_speed} = 75.913 \ \frac{MPa \cdot m}{s}$$

[Nominal BE bearing pressure velocity]

4.3 - Counterbalance Shaft Design

$L_{w\_scb} := 29.3 \ mm \cdot 2$ [Total rotating diameter SCB shaft]

$L_{w\_pcb} := 60 \ mm \cdot 2$ [Total rotating diameter PCB shaft]

4.3.1 - Viscous Drag Power Loss Calculations:

Taken from Perry's Chemical Engineers Handbook

$$N_{re\_scb} := \frac{L_{w\_scb}^{\ 2} \cdot 2 \cdot RPM \cdot \rho_{oil}}{\mu_{oil} \cdot 60 \ s} = 2.289 \cdot 10^3$$

[Secondary Counterbalance Shaft Reynolds Number]

$$N_{re\_pcb} := \frac{L_{w\_pcb}^{\ 2} \cdot RPM \cdot \rho_{oil}}{\mu_{oil} \cdot 60 \ s} = 4.8 \cdot 10^3$$

[Primary Counterbalance Shaft Reynolds Number]

$Np_{cb} := .45$

[Power number with respect to Reynolds Number, taken from Fig: 18-17 Perry Chemical Engineering Handbook]

$$P_{pcb\_vdl} := Np_{cb} \cdot \rho_{oil} \cdot \left(\frac{RPM}{60\ s}\right)^3 \cdot L_{w\_pcb}{}^5 = 712.8\ W$$

[Power loss due to rotating primary balance weight in oil]

$$P_{scb\_vdl} := Np_{cb} \cdot \rho_{oil} \cdot \left(\frac{2 \cdot RPM}{60\ s}\right)^3 \cdot L_{w\_scb}{}^5 = 158.358\ W$$

[Power loss due to rotating secondary balance weight in oil]

4.3.2 - Bearing Friction Power Loss Calculations:

$$T_{friction\_pcb} := F_{pcb} \cdot \mu_{rolling\_brg} \cdot \frac{L_{brg\_out\phi}}{2}$$

$$P_{friction\_pcb} := T_{friction\_pcb} \cdot \omega = 57.57\ W$$ [Power loss due to bearing friction]

$$T_{friction\_scb} := F_{scb} \cdot \mu_{rolling\_brg} \cdot \frac{L_{brg\_out\phi}}{2}$$

$$P_{friction\_scb} := T_{friction\_scb} \cdot 2 \cdot \omega = 32.001\ W$$ [Power loss due to bearing friction]

4.3.3 - Total Balance Shaft Power Losses

$$P_{loss\_pcb} := P_{pcb\_vdl} + P_{friction\_pcb} = 770.37\ W$$ [Total power loss due to friction and fluid losses]

$$P_{loss\_scb} := P_{scb\_vdl} + P_{friction\_scb} = 190.359\ W$$ [Total power loss due to friction and fluid losses]

Counterbalance Arrangement Calculations

Power losses in counterbalance shafts due to viscous drag & bearing losses are negligible, as a result no gear design verification will be performed and the additional bearing frictional losses due to drive reactions shall be ignored.

Only minimum shaft size will be calculated in the primary shaft. Bearing analysis has already been performed.

4.3.4 - Minimum Shaft Size

$L_{load\_point} := 21\ mm + 5\ mm$ [Force load point in shaft]

$M_q := \frac{F_{pcb}}{2} \cdot L_{load\_point} = 54.975\ N \cdot m$ [Max bending moment in PCB shaft]

$T_{pcb\_shaft} := \frac{P_{loss\_pcb}}{\omega} = 2.943\ N \cdot m$ [Max drive torque in PCB shaft]

$Ks := 1.15$ [Shaft Size Factor SAA-HB6]

$Fs := 2$ [Shaft Analysis Factor of Safety]

$K := 1$ [Stress concentration factor for shaft geometry (=1 due to straight shaft)]

$\sigma_{b\_4140} := 930\ MPa$ [Ultimate tensile strength of shaft material]

$Se_{\_4140} := 0.45 \cdot \sigma_{b\_4140}$ [Endurance limit of shaft material]

$$D_{pcb} := \left(\left(\frac{10^4\ Fs}{Se_{\_4140}} \right) \cdot \left(\left(Ks \cdot K \cdot (M_q) \cdot 10^{-3} \right)^2 + \frac{3}{16} \cdot \left(\left((1 + Ks \cdot K) \cdot (T_{pcb\_shaft}) \right) \cdot 10^{-3} \right)^2 \right)^{\frac{1}{2}} \right)^{\frac{1}{3}}$$

$D_{pcb} = 14.461\ mm$ [Minimum shaft size]

4.4 - Conrod Design

$F_{c\_max} := 52310\ N$ [Max conrod compressive force]

4.4.1 - Conrod design calculations

Refer to Figure 3.11

$Ic_{xx} := 3714\ mm^4$ [Conrod moment of area]

$Ic_{yy} := 11293\ mm^4$ [Conrod moment of area]

$Rx := 4.31\ mm$ [Radius of gyration]

$Ry := 7.52\ mm$ [Radius of gyration]

4.4.1.1 - Buckling in the Weak Direction of the Section

End Fixities: Fixed/Fixed

$K_{1\_xx} := 4$ [Buckling constant Roarks Table 15.5 -1e]

$$F_{\_buckling\_xx} := K_{1\_xx} \cdot \frac{\left(\pi^2 \cdot E_{steel} \cdot Ic_{xx}\right)}{L_{co}^{\ 2}} = \left(1.203 \cdot 10^3\right)\ kN$$ [Req'd Buckling Force]

$$Factor\_of\_Safety\_xx := \frac{F_{\_buckling\_xx}}{F_{c\_max}} = 22.993$$ [Buckling factor of safety]

4.4.1.2 - Buckling in the Strong Direction of the Section

End Fixities: Pinned/Pinned

$K_{1\_yy} := 1$ [Buckling constant Roarks Table 15.5 -1b]

$$F_{\_buckling\_yy} := K_{1\_yy} \cdot \frac{\left(\pi^2 \cdot E_{steel} \cdot Ic_{yy}\right)}{L_{co}^{\ 2}} = 914.299\ kN$$ [Req'd Buckling Force]

$$Factor\_of\_Safety\_yy := \frac{F_{\_buckling\_yy}}{F_{c\_max}} = 17.478$$ [Buckling factor of safety]

4.4.1.3 - Stress around the little end bearing (Design Revision B)

(a) - Eyelet Dimensions

$A_{cnrd\_leb\_bb} := 177.7\ mm^2$ [Section b-b Area]
$h_{sect} := 6\ mm$ [Eyelet section depth]
$w_{sect} := L_{LEbearing} = 20\ mm$ [Eyelet section width]
$d_{LE} := 26\ mm$ [Conrod LE bearing OD diameter]

(b) - Section & Load Calculations

$$R_{cnrd\_le} := \frac{\left(d_{LE} + h_{sect}\right)}{2} = 16\ mm$$ [Eyelet centroidal radius]

$$e := R_{cnrd\_le} - \frac{h_{sect}}{\ln\left(\frac{\left(\frac{d_{LE}}{2} + h_{sect}\right)}{\frac{d_{LE}}{2}}\right)} = 0.189\ mm$$

[Neutral axis offset]

$$Fy_{rm\_max} := \max\left(Fy_{rm}\right) = \left(1.081 \cdot 10^{4}\right)\ N$$

[Maximum inertia force]

$$Fy_{rm\_min} := 0\ N$$

[Minimum inertia force]

$$\sigma_{bb\_min} := 0\ MPa$$

[Minimum stress b-b]

$$\theta_{cnrd\_bb} := \frac{\pi}{4}\ rad$$

[Load point angle from centre axis]

$$scf_{\_eyelet} := 1.3$$

[Stress concentration factor]

$$L_{tm} := \left|\frac{Fy_{rm\_max}}{2}\right| \cdot R_{cnrd\_le} \cdot \left(\sin\left(\frac{\pi}{2}\right) - \sin\left(\theta_{cnrd\_bb}\right)\right)$$

[Curved beam load term]

$$k_2 := 1 - \frac{e}{R_{cnrd\_le}} = 0.988$$

[Curved beam constant]

(c) - Moment & Stress Calculations

$$M_{crnd\_leb\_yy\_C} := \frac{-\left|\frac{Fy_{rm\_max}}{2}\right| \cdot R_{cnrd\_le}}{\pi} \cdot \left(\left(\sin\left(\theta_{cnrd\_bb}\right) \cdot \left(\sin\left(\theta_{cnrd\_bb}\right) - \pi + \theta_{cnrd\_bb}\right)\right) + k_2 \cdot \left(1 + \cos\left(\theta_{cnrd\_bb}\right)\right)\right)$$

[Moment at the top of eyelet]

$$N_a := \frac{-\left|\frac{Fy_{rm\_max}}{2}\right| \cdot \left(\sin\left(\theta_{cnrd\_bb}\right)\right)^2}{\pi} = -860.463\ N$$

[Hoop load]

$$M_{crnd\_leb\_yy} := M_{crnd\_leb\_yy\_C} - N_a \cdot R_{cnrd\_le} \cdot \left(1 - \cos\left(\theta_{cnrd\_bb}\right)\right) + L_{tm} = 15.028\ N \cdot m$$

[Moment at 90deg to conrod axis]

$$\sigma_{i\_90°} := \frac{\left(M_{crnd\_leb\_yy} \cdot \left(\frac{h_{sect}}{2} + e\right)\right)}{h_{sect} \cdot w_{sect} \cdot e \cdot \left(\frac{R_{cnrd\_le}}{2}\right)} = 263.745\ MPa$$

[Inside stress in eyelet at 90deg to conrod axis]

$$\sigma_{o\_90°} := \frac{\left(\left|M_{crnd\_leb\_yy}\right| \cdot \left(\frac{h_{sect}}{2} - e\right)\right)}{h_{sect} \cdot w_{sect} \cdot e \cdot \left(\frac{R_{cnrd\_le}}{2} + h_{sect}\right)} = 132.821\ MPa$$

[Outside stress in eyelet at 90deg to conrod axis]

$$\sigma_{o\_top} := \frac{\left(\left|M_{crnd\_leb\_yy\_C}\right| \cdot \left(\frac{h_{sect}}{2} + e\right)\right)}{h_{sect} \cdot w_{sect} \cdot e \cdot \left(\frac{R_{cnrd\_le}}{2}\right)} = 251.693\ MPa$$

[Outside stress in eyelet at top]

$$\sigma_{i\_top} := \frac{\left(\left|M_{crnd\_leb\_yy\_C}\right| \cdot \left(\frac{h_{sect}}{2} - e\right)\right)}{h_{sect} \cdot w_{sect} \cdot e \cdot \left(\frac{R_{cnrd\_le}}{2} + h_{sect}\right)} = 126.752\ MPa$$

[Inside stress in eyelet at top]

(d) - Factored Stresses

$$\sigma_{a\_eye} := \frac{scf_{\_eyelet} \cdot \sigma_{i\_90°} - \sigma_{bb\_min}}{2} = 171.434\ MPa$$ [Factored alternating stress in eyelet]

$$\sigma_{m\_eye} := \frac{scf_{\_eyelet} \cdot \sigma_{i\_90°} + \sigma_{bb\_min}}{2} = 171.434\ MPa$$ [Factored mean stress in eyelet]

$$\sigma_{max\_eye} := scf_{\_eyelet} \cdot \sigma_{i\_90°} = 342.868\ MPa$$ [Factored max stress in eyelet]

4.4.2 - Conrod FEA results (Design Revision B)

$\sigma_{a\_a\_max} := 54.7\ MPa$ [Max stress in pillar]

$\sigma_{a\_a\_min} := -259.9\ MPa$ [Min stress in pillar]

$\sigma_{alt\_a\_a} := \dfrac{\sigma_{a\_a\_max} - \sigma_{a\_a\_min}}{2} = 157.3\ MPa$ [Alternating stress in pillar]

$\sigma_{mean\_a\_a} := \dfrac{\sigma_{a\_a\_max} + \sigma_{a\_a\_min}}{2} = -102.6\ MPa$ [Mean stress in pillar]

$\sigma_{b\_b\_max} := 315.6\ MPa$ [Max stress in eyelet]

$\sigma_{b\_b\_min} := 0\ MPa$ [Min stress in eyelet]

$\sigma_{alt\_b\_b} := \dfrac{\sigma_{b\_b\_max} - \sigma_{b\_b\_min}}{2} = 157.8\ MPa$ [Alternating stress in eyelet]

$\sigma_{mean\_b\_b} := \dfrac{\sigma_{b\_b\_max} + \sigma_{b\_b\_min}}{2} = 157.8\ MPa$ [Mean stress in eyelet]

$\sigma_{c\_c\_max} := 97.1\ MPa$ [Max stress in web radius below eyelet]

$\sigma_{c\_c\_min} := -363\ MPa$ [Min stress in web radius below eyelet]

$\sigma_{alt\_c\_c} := \dfrac{\sigma_{c\_c\_max} - \sigma_{c\_c\_min}}{2} = 230.05\ MPa$ [Alternating stress in web radius below eyelet]

$\sigma_{mean\_c\_c} := \dfrac{\sigma_{c\_c\_max} + \sigma_{c\_c\_min}}{2} = -132.95\ MPa$ [Mean stress in web radius below eyelet]

4.4.3 - Conrod Fatigue Analysis (Design Revision B)

4.4.3.1 - Endurance Limit Coefficients - Eyelet -

$k_a := 4.51 \cdot \left(\dfrac{\sigma_{b\_4340}}{MPa}\right)^{-.265} = 0.722$ [Surface Factor: Shigley's Eq 6-19 & Table 6-2]

$k_b := 1.24 \cdot \left(\dfrac{36\ mm}{mm}\right)^{-0.107} = 0.845$ [Size Factor: Shigley's Eq 6-20]

$k_{d\_200} := 1.02$ [Temperature Factor: Shigley's Table 6-4]

$k_{e\_50} := 1$ [Reliability Factor: Shigley's Table 6-5]

4.4.3.2 - Endurance Limit -Eyelet

$$Se_{\_4340} := \frac{\sigma_{b\_4340}}{2} \cdot k_a \cdot k_b \cdot k_{d\_200} \cdot k_{e\_50} = 312.778\ MPa$$ [Material Endurance Limit]

$$nf_{\_b\_b} := 0.5 \cdot \left(\frac{\sigma_{b\_4340}}{\sigma_{mean\_b\_b}}\right)^2 \cdot \left(\frac{\sigma_{alt\_b\_b}}{Se_{\_4340}}\right) \cdot \left(-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_{mean\_b\_b} \cdot Se_{\_4340}}{\sigma_{alt\_b\_b} \cdot \sigma_{b\_4340}}\right)^2}\right) = 1.82$$

[Fatigue Factor of Safety for crankpin journal]

4.4.3.3 - Conrod Pillar - Endurance limit

$$k_a := 4.51 \cdot \left(\frac{\sigma_{b\_4340}}{MPa}\right)^{-.265} = 0.722$$ [Surface Factor: Shigley's Eq 6-19 & Table 6-2]

$k_b := 1$ [Size Factor: Shigley's Eq 6-20]

$k_c := 0.85$ [Axial Load Factor Shigley's Eq 6-26]

$k_{d\_200} := 1.02$ [Temperature Factor: Shigley's Table 6-4]

$k_{e\_50} := 1$ [Reliability Factor: Shigley's Table 6-5]

$$Se_{\_4340} := \frac{\sigma_{b\_4340}}{2} \cdot k_a \cdot k_b \cdot k_c \cdot k_{d\_200} \cdot k_{e\_50} = 314.599\ MPa$$ [Material Endurance Limit]

4.5 - Piston Design

4.5.1 - Stress and Deflection in the Crown Window.

4.5.1.1 - Material & Section Inputs

$E_{quartz} := 72500\ MPa$ [Modulus of Elasticity of Quartz]
$t_{window} := 20\ mm$ [Window thickness]
$v_{quartz} := 0.17$ [Poissons ratio of Quartz]

$a_{window} := \frac{46.5}{2}\ mm$ [Window Radius]

$q := 10.511\ MPa$ [Distributed load on window]

4.5.1.2 - Material & Section Inputs

$$D_c := \frac{\left(E_{quartz} \cdot t_{window}{}^3\right)}{12 \cdot \left(1 - v_{quartz}{}^2\right)} = 49.772\ kN \cdot m$$ [Deflection Constant (Roark's, 2012)]

$$y_c := \frac{-q \cdot a_{window}{}^4 \cdot (5 + v_{quartz})}{64 \cdot D_c \cdot (1 + v_{quartz})} = -0.004\ mm$$ [Deflection at centre of window (Roark's,2012)]

$$M_c := \frac{q \cdot a_{window}{}^2 \cdot (3 + v_{quartz})}{16} = \left(1.126 \cdot 10^3\right) \frac{1}{m} \cdot N \cdot m$$ [Per unit bending moment (Roark's,2012)]

$$\sigma_{window\_max} := \frac{(6 \cdot M_c)}{t_{window}{}^2} = 16.886\ MPa$$ [Max stress in window (Roark's,2012)]

4.5.2 - Piston FEA results (Design Revision B)

$$\sigma_{upper\_skirting\_max} := 26.7\ MPa$$ [Max stress in upper skirting]

$$\sigma_{upper\_skirting\_min} := -143.8\ MPa$$ [Min stress in upper skirting]

$$\sigma_{alt\_upper\_skirting} := \frac{\sigma_{upper\_skirting\_max} - \sigma_{upper\_skirting\_min}}{2} = 85.25\ MPa$$ "+/−"

[Alternating stress in upper skirting]

$$\sigma_{mean\_upper\_skirting} := \frac{\sigma_{upper\_skirting\_max} + \sigma_{upper\_skirting\_min}}{2} = -58.55\ MPa$$

[Mean stress in upper skirting]

$$\sigma_{lower\_skirting\_max} := 17.7\ MPa$$ [Max stress in lower skirting]

$$\sigma_{lower\_skirting\_min} := -97\ MPa$$ [Min stress in lower skirting]

$$\sigma_{alt\_lower\_skirting} := \frac{\sigma_{lower\_skirting\_max} - \sigma_{lower\_skirting\_min}}{2} = 57.35\ MPa$$ "+/−"

[Alternating stress in lower skirting]

$$\sigma_{mean\_lower\_skirting} := \frac{\sigma_{lower\_skirting\_max} + \sigma_{lower\_skirting\_min}}{2} = -39.65\ MPa$$

[Mean stress in lower skirting]

$\sigma_{lower\_wristpin\_bore\_max} := 71.2\ MPa$ [Max stress in wrist pin bore]

$\sigma_{lower\_wristpin\_bore\_min} := -5\ MPa$ [Min stress in wrist pin bore]

$$\sigma_{alt\_lwpb} := \frac{\sigma_{lower\_wristpin\_bore\_max} - \sigma_{lower\_wristpin\_bore\_min}}{2} = 38.1\ MPa \quad \text{"+/−"}$$

[Alternating stress in wrist pin bore]

$$\sigma_{mean\_lwpb} := \frac{\sigma_{lower\_wristpin\_bore\_max} + \sigma_{lower\_wristpin\_bore\_min}}{2} = 33.1\ MPa$$

[Mean stress in wrist pin bore]

$\Delta_{skirting\_horizontal} := 0.1067\ mm$ [Bowing of the piston skirting]

$\Delta_{crown\_vertical} := 0.2273\ mm$ [Total piston length change]

Piston Fatigue Analysis (Design Revision B)

Endurance Limit Coefficients - Wrist pin bore -

$$k_a := 1.58 \cdot \left(\frac{\sigma_{b\_4032AL}}{MPa}\right)^{-.085} = 0.954$$

[Surface Factor: Shigley's Eq 6-19 & Table 6-2]

$$k_b := 1.24 \cdot \left(\frac{L_{bore}}{mm}\right)^{-0.107} = 0.775$$

[Size Factor: Shigley's Eq 6-20]

$k_{d\_200} := 1.02$ [Temperature Factor: Shigley's Table 6-4]

$k_{e\_50} := 1$ [Reliability Factor: Shigley's Table 6-5]

- Endurance limit -

$$Se_{\_4032AL} := \frac{\sigma_{b\_4032AL}}{2} \cdot k_a \cdot k_b \cdot k_{d\_200} \cdot k_{e\_50} = 142.855\ MPa$$

$$nf_{\_lwpb} := 0.5 \cdot \left(\frac{\sigma_{b\_4032AL}}{\sigma_{mean\_lwpb}}\right)^2 \cdot \left(\frac{\sigma_{alt\_lwpb}}{Se_{\_4032AL}}\right) \cdot \left(-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_{mean\_lwpb} \cdot Se_{\_4032AL}}{\sigma_{alt\_lwpb} \cdot \sigma_{b\_4032AL}}\right)^2}\right)$$

$nf_{\_lwpb} = 3.416$ [Fatigue Factor of Safety for Lower Wrist Pin Bore]

4.6 - Wrist Pin Design

4.6.1 - Stress & Deflection in the Wrist Pin

4.6.1.1 - Material & Section Inputs

$D_{wp} := 20\ mm$ [Wrist pin outside diameter]

$d_{wp} := 10\ mm$ [Wrist pin inside diameter]

$L_{wp\_eff} := 36\ mm$ [Wrist pin effective span]

$L_{wp\_load\_span} := w_{sect} - 6\ mm = 14\ mm$ [Load span]

$\sigma_{b\_4140} = 930\ MPa$ [Ultimate tensile strength of shaft material]

$Se_{\_4140} = 418.5\ MPa$ [Endurance limit of shaft material]

$I_{xx} := \left(D_{wp}^{\ 4} - d_{wp}^{\ 4}\right) \cdot \frac{\pi}{64}$ [Wrist pin moment of area]

$A_{wp} := \left(D_{wp}^{\ 2} - d_{wp}^{\ 2}\right) \cdot \frac{\pi}{4}$ [Wrist pin area]

4.6.1.2 - Bending & Shear Stresses

$$M_{wp\_max} := \frac{F_{c\_max}}{2} \cdot \left(\frac{L_{wp\_eff} - L_{wp\_load\_span}}{2}\right) = 287.705\ N \cdot m$$

[Max bending moment]

$$M_{wp\_min} := \frac{Fy_{rm\_max}}{2} \cdot \left(\frac{L_{wp\_eff} - L_{wp\_load\_span}}{2}\right) \cdot (-1) = -59.471\ N \cdot m$$

[Reversed max bending moment (labelled minimum)]

$$\sigma_{wp\_max} := \frac{\left(M_{wp\_max} \cdot \frac{D_{wp}}{2}\right)}{I_{xx}} = 390.739\ MPa$$ [Max bending stress]

$$\sigma_{wp\_min} := \frac{\left(M_{wp\_min} \cdot \frac{D_{wp}}{2}\right)}{I_{xx}} = -80.769\ MPa$$ [Min bending stress]

$$\tau_{wp\_max} := \frac{F_{c\_max}}{2 \cdot A_{wp}} = 111.005 \; \boldsymbol{MPa}$$

[Max shear stress]

$$\tau_{wp\_min} := \frac{Fy_{rm\_max}}{2 \cdot A_{wp}} \cdot (-1) = -22.946 \; \boldsymbol{MPa}$$

[Min shear stress]

$$\sigma_{vonmises\_wp\_max} := \left(\sigma_{wp\_max}{}^{2} + \left(3 \cdot \tau_{wp\_max}{}^{2}\right)\right)^{0.5} = 435.48 \; \boldsymbol{MPa}$$

[Max equivalent stress]

$$\sigma_{vonmises\_wp\_min} := \left(\sigma_{wp\_min}{}^{2} + \left(3 \cdot \tau_{wp\_min}{}^{2}\right)\right)^{0.5} \cdot (-1) = -90.017 \; \boldsymbol{MPa}$$

[Min equivalent stress]

4.6.1.3 - Wrist Pin Fatigue Analysis

$$\sigma_{alt\_wp} := \frac{\sigma_{vonmises\_wp\_max} - \sigma_{vonmises\_wp\_min}}{2} = 262.749 \; \boldsymbol{MPa}$$

$$\sigma_{mean\_wp} := \frac{\sigma_{vonmises\_wp\_max} + \sigma_{vonmises\_wp\_min}}{2} = 172.732 \; \boldsymbol{MPa}$$

$$nf := 0.5 \cdot \left(\frac{\sigma_{b\_4140}}{\sigma_{mean\_wp}}\right)^{2} \cdot \left(\frac{\sigma_{alt\_wp}}{Se_{\_4140}}\right) \cdot \left(-1 + \sqrt{1 + \left(\frac{2 \cdot \sigma_{mean\_wp} \cdot Se_{\_4140}}{\sigma_{alt\_wp} \cdot \sigma_{b\_4140}}\right)^{2}}\right) = 1.473$$

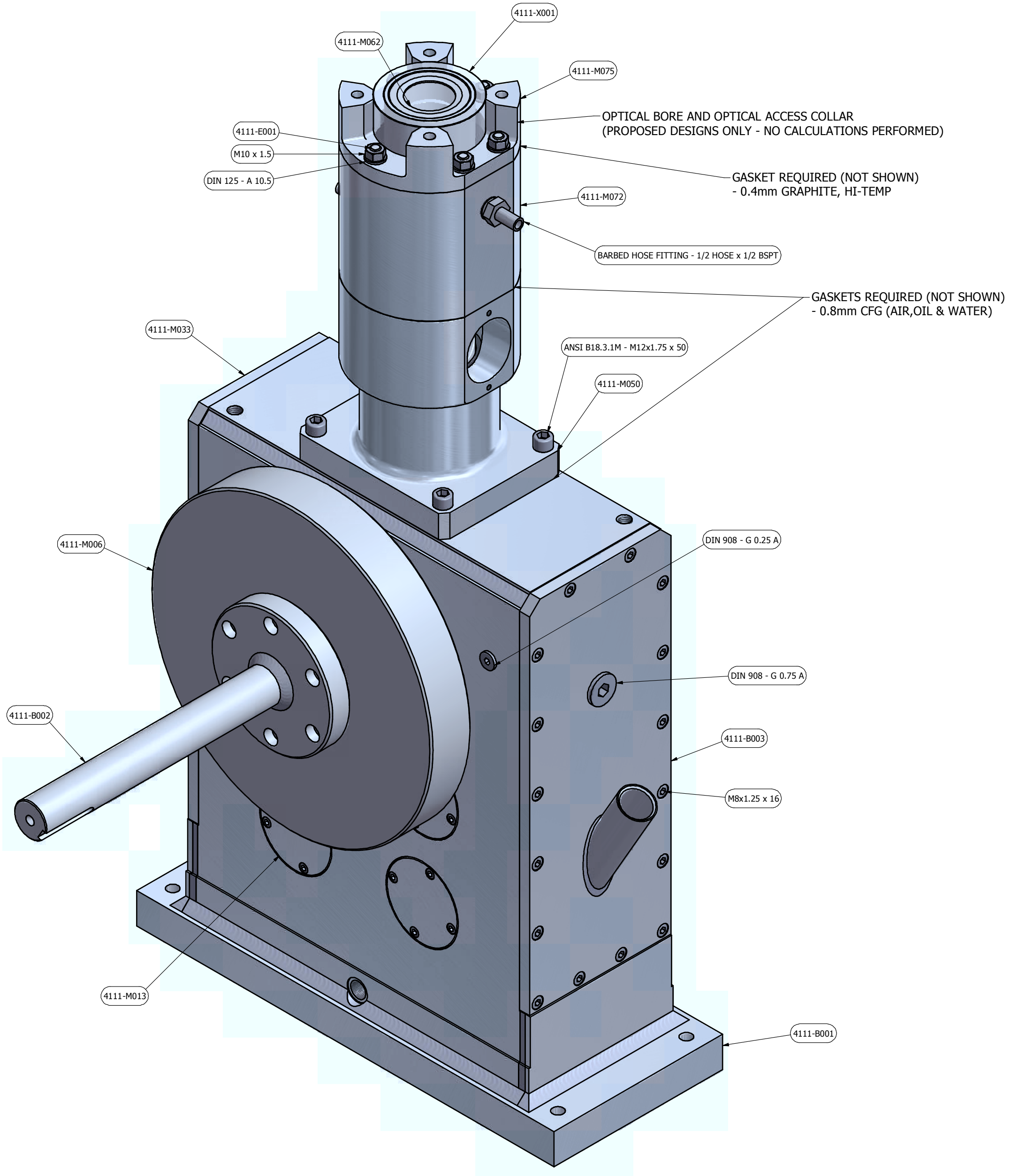
[Fatigue factor of safety for wristpin]

Appendix C: Selected Drawings (A4)

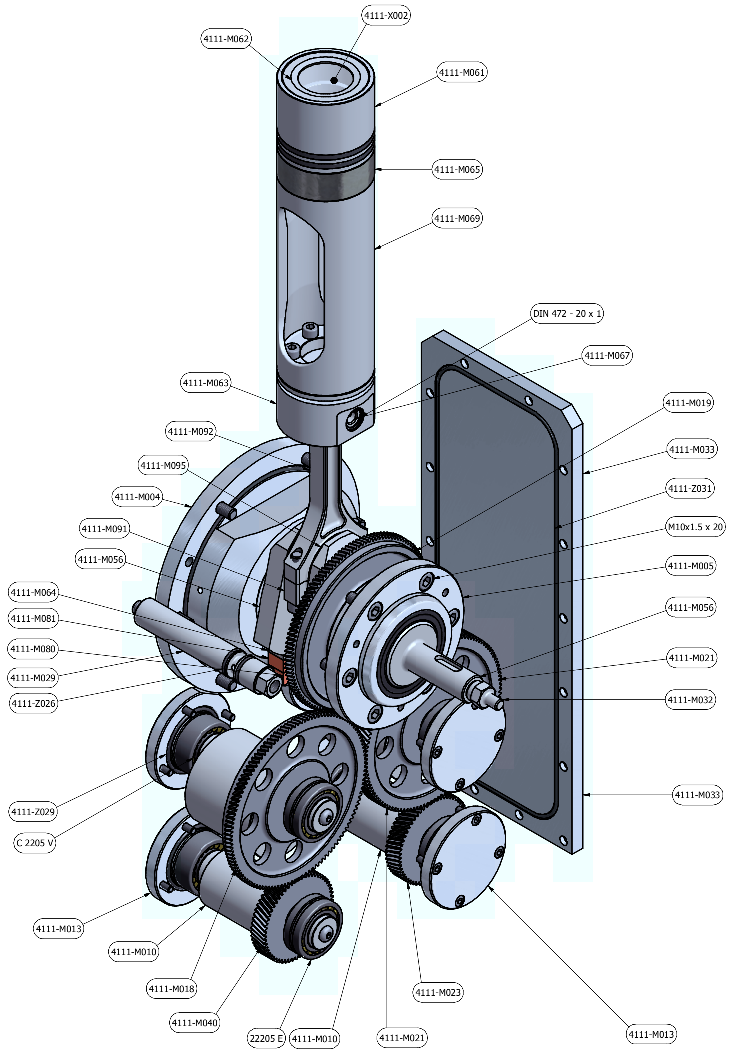
Miscellaneous Assembly and Detail Drawings

Note: Refer to Appendix D (attached separately in printed version) for full assembly and detail drawings.

DO NOT SCALE - IF IN DOUBT ASK.



4111-A001 - L.H.S. ISOMETRIC VIEW -



4111-A001 - R.H.S. ISOMETRIC VIEW -

NOTE: CRANKCASE,SIDECOVER & MISC REMOVED FOR INTERNAL VIEWING

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	29/04/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN: KBD	DATE: 29/04/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 29/04/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED:	DATE:	DRAWING DESCRIPTION: GENERAL ASSEMBLY & PRESENTATION
APPROVED:	DATE:	JOB No.: ENG4111/2 — DRAWING No.: 4111-A001 — 1 OF 3 — SIZE: A1 — REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Assemblies\4111-A001.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-A001.idw

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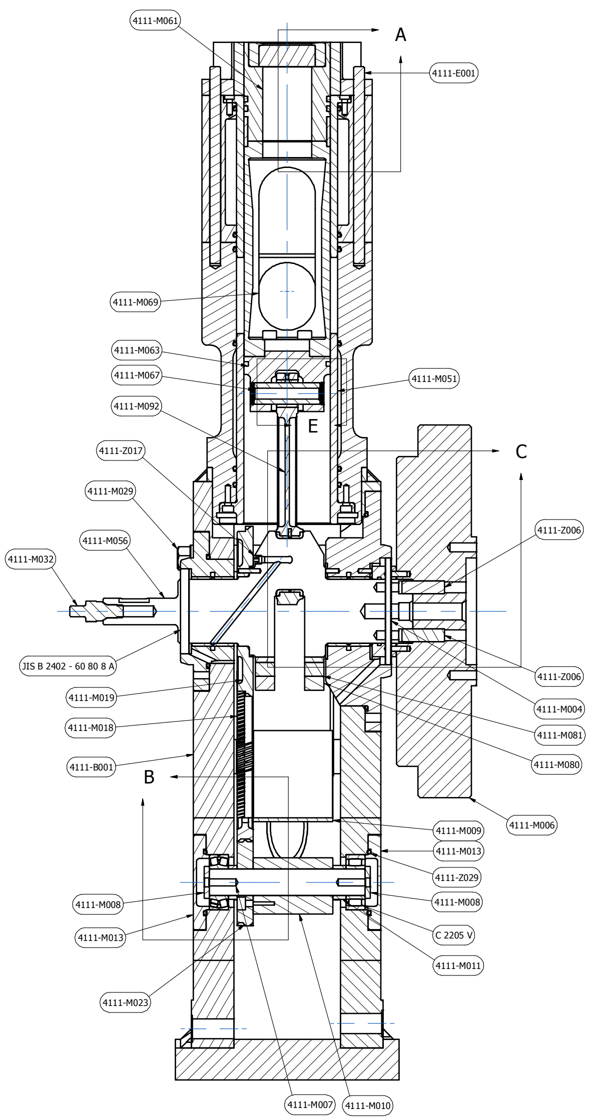
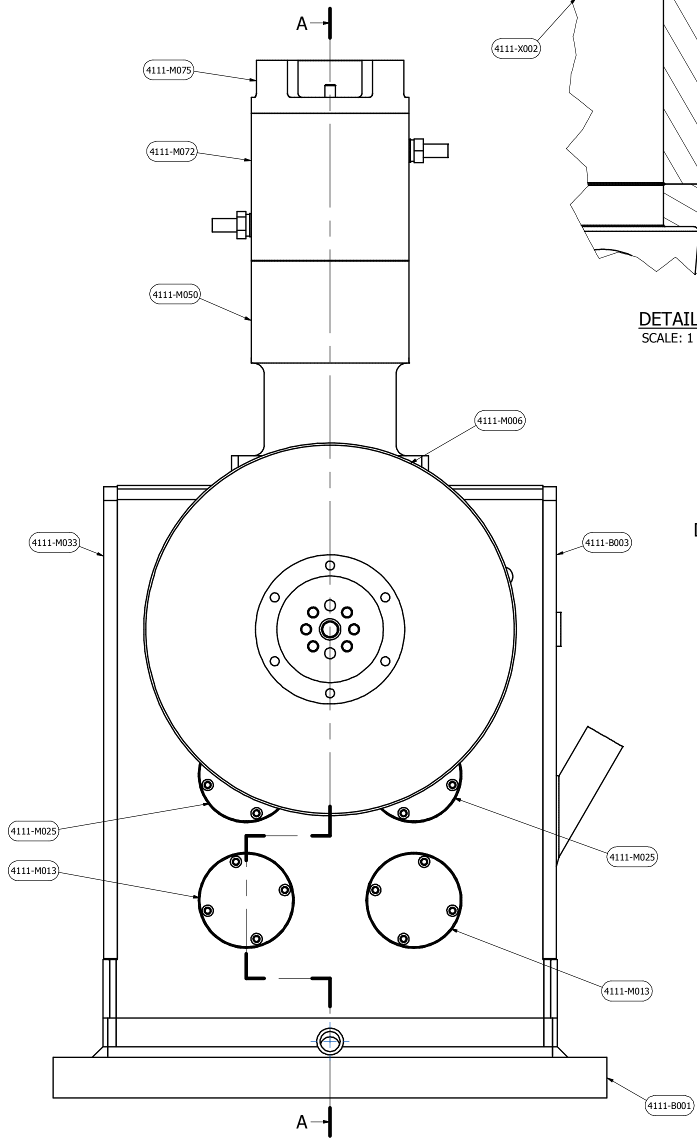
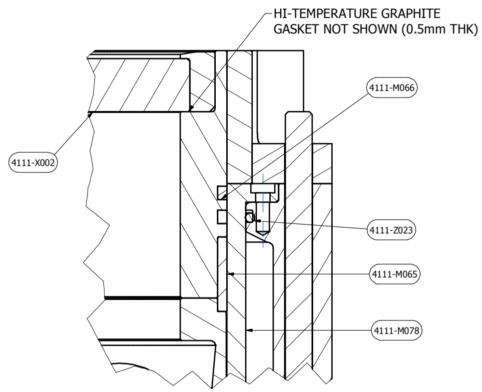


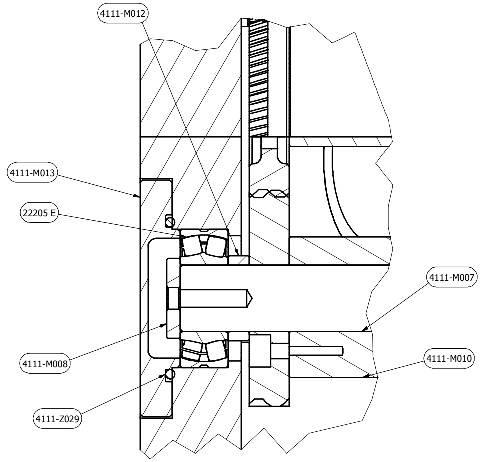
SECTION A-A
SCALE: 1 : 2.5



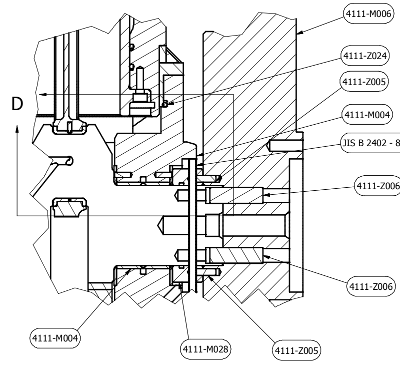
4111-A001 - FLYWHEEL SIDE VIEW -
SCALE: 1 : 2.5
NOTE: ALL FASTENERS SUPPRESSED



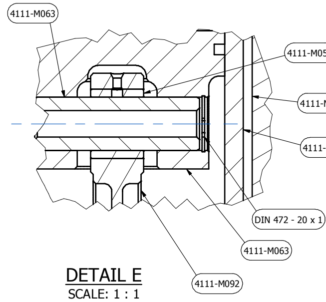
DETAIL A
SCALE: 1 : 1



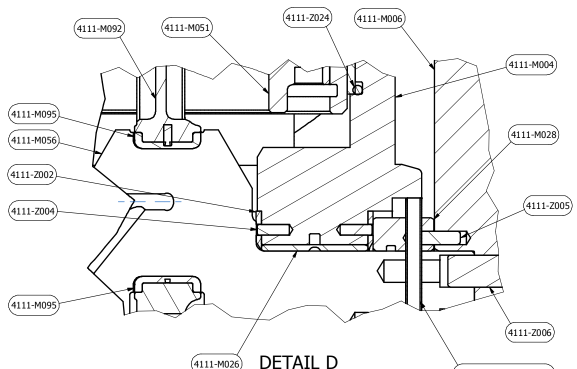
DETAIL B
SCALE: 1 : 1



DETAIL C
SCALE: 1 : 2



DETAIL E
SCALE: 1 : 1



DETAIL D
SCALE: 1 : 1

GENERAL TOLERANCES UNLESS INDICATED		ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE
PREPARATION AND ASSEMBLY	MACHINING	COPYRIGHT :- REPRODUCTION OF THIS DRAWING WHOLLY OR IN PART, IS PROHIBITED UNLESS WRITTEN PERMISSION IS OBTAINED FROM THE UNIVERSITY OF SOUTHERN QUEENSLAND (USQ)
< 25mm .... ±1.0mm	0 .... ±0.1mm	
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm	
>600mm .... ±2.0mm	0.00 .... ±0.01mm	
ANGLE .... ±1°	3.2	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DESIGNED: KBD	DATE: 29/04/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION: GENERAL ASSEMBLY & PRESENTATION
APPROVED	DATE	JOB No.: ENG4111/2 — DRAWING No.: 4111-A001 — 2 OF 3 — SIZE: A1 — REV.: 0

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PARTS LIST (PARTS PER ONE MAIN ASSEMBLY ONLY)

PART NUMBER	MATERIAL	DESCRIPTION	MASS	QTY	REFERENCE
22205 E	N/A	SRB, series CC/W33-Standard spherical roller bearings	0.05 kg	4	VENDOR SUPPLY
4111-B001	Aluminum 6061, Welded	MAIN CRANKCASE	52.72 kg	1	4111-B001
4111-B002	Steel, Mild, Welded	DRIVE ATTACHMENT	5.63 kg	1	4111-B002
4111-B003	Aluminum 6061, Welded	SIDE COVER (OIL FILL)	2.7 kg	1	4111-B003
4111-E001	AS1444 -4140	TIE ROD	0.04 kg	6	4111-E001
4111-M004	AS1865 -2011T6	FLYWHEEL SIDE MAIN BEARING HOUSING	3.41 kg	1	4111-M004
4111-M005	AS1865 -2011T6	ENCODER SIDE MAIN BEARING HOUSING	0.91 kg	1	4111-M005
4111-M006	AS3678 -GR250	FLYWHEEL	47.45 kg	1	4111-M006
4111-M007	AS1444 -4140	COUNTER BALANCE SHAFT	0.55 kg	4	4111-M007
4111-M008	AS1444 -1020	COUNTER BALANCE SHAFT WASHER	0.03 kg	8	4111-M008
4111-M009	AS1444 -1020	PRIMARY BALANCE WEIGHT	3.08 kg	2	4111-M009
4111-M010	AS1444 -1020	SECONDARY BALANCE WEIGHT	1 kg	2	4111-M010
4111-M011	AS1444 -1020	COUNTER BALANCE SHAFT SPACER LONG	0.03 kg	4	4111-M011
4111-M012	AS1444 -1020	COUNTER BALANCE SHAFT SPACER SHORT	0.02 kg	4	4111-M012
4111-M013	AS1865 -2011T6	BEARING CAP	0.16 kg	6	4111-M013
4111-M018	AS1444 EN36A	PRIMARY COUNTER BALANCE SHAFT GEAR	1.3 kg	1	4111-M018
4111-M019	AS1444 EN36A	CRANKSHAFT DRIVE PINION	1.38 kg	1	4111-M019
4111-M021	AS1444 EN36A	PRIMARY COUNTER BALANCE SHAFT GEAR	1.42 kg	1	4111-M021
4111-M023	AS1444 EN36A	SECONDARY COUNTER BALANCE SHAFT GEAR	0.5 kg	1	4111-M023
4111-M025	6061-T6	BEARING CAP (PRIMARY, FLYWHEEL SIDE)	0.15 kg	2	4111-M025
4111-M026	DU BEARING	6040DU - LEAD/PTFE BEARING (MODIFIED)	0.05 kg	2	4111-M026
4111-M028	AS1444 -1020	THRUST COLLAR	0.49 kg	1	4111-M028
4111-M029	AS1444 -1020	OIL SUPPLY FITTING	0.48 kg	1	4111-M029
4111-M032	AS1444 -4140	ENCODER ATTACHMENT	0.06 kg	1	4111-M032
4111-M033	AS1734 6061T6	CRANKCASE SIDE COVER	2.57 kg	1	4111-M033
4111-M040	AS1444 EN36A	SECONDARY COUNTER BALANCE SHAFT GEAR	0.5 kg	1	4111-M040
4111-M050	AS1866 6061T6	LOWER BARREL	6.47 kg	1	4111-M050
4111-M051	2P Cast Iron	LOWER CYLINDER SLEEVE	2.81 kg	1	4111-M051
4111-M056	AS1444 -4340	ENGINE CRANKSHAFT	6.45 kg	1	4111-M056
4111-M059	954 - AL. BRONZE	LITTLE END BEARING	0.03 kg	1	4111- M059
4111-M061	AS1866 4032-T6	PISTON HEAD	0.6 kg	1	4111-M061
4111-M062	SS-304 ROUND	OPTICAL WINDOW COLLAR	0.33 kg	1	4111-M062
4111-M063	AS1866 4032-T6	LOWER PISTON	0.39 kg	1	4111-M063
4111-M064	AS3678 -GR250	CRANK BALANCE WEIGHT	0.04 kg	2	4111-M064
4111-M065	Tetron B	SLIDING BUSH	0.09 kg	1	4111-M065
4111-M066	Tetron C	LOW FRICTION PISTON RINGS	0.01 kg	2	4111-M066
4111-M067	AS1444 -4340	WRIST PIN	0.11 kg	1	4111-M067
4111-M069	AS1866 4032-T6	PISTON EXTENSION	0.67 kg	1	4111-M069
4111-M072	AS1865 -2011T6	UPPER BARREL	3.61 kg	1	4111-M072
4111-M075	AS1865 -2011T6	OPTICAL ACCESS COLLAR	0.77 kg	1	4111-M075
4111-M076	AS1444 -4140	HEAD BOLT FERRULE	0.01 kg	4	4111-M076
4111-M078	2P Cast Iron	UPPER CYLINDER SLEEVE	2.29 kg	1	4111-M078
4111-M080	SD170	CRANK BALANCE WEIGHT	0.22 kg	2	4111-M080
4111-M081	SD170	CRANK BALANCE WEIGHT	0.22 kg	2	4111-M081
4111-M091	AS1444 -4340	CONROD BIG END SADDLE	0.19 kg	1	4111-M091
4111-M092	AS1444 -4340	CONROD	0.48 kg	1	4111-M092
4111-M095	954 - AL. BRONZE	BIG END BEARING SHELL	0.05 kg	2	4111-M095
4111-X001	FUSED SILICA (QUARTZ)	OPTICAL BORE	0.3 kg	1	4111-X001
4111-X002	FUSED SILICA (QUARTZ)	OPTICAL ACCESS WINDOW	0.1 kg	1	4111-X002
4111-Z002	DU BEARING	WC60DU - LEAD/PTFE THRUST BEARING (MODIFIED)	0.04 kg	2	4111-Z002
4111-Z004	MILD STEEL	4x12 MACHINE DOWEL	0 kg	4	VENDOR SUPPLY
4111-Z005	MILD STEEL	5 x 20 MACHINE DOWEL	0 kg	4	VENDOR SUPPLY
4111-Z006	MILD STEEL	12 x 20 MACHINE DOWEL	0.04 kg	2	VENDOR SUPPLY
4111-Z010	MILD STEEL	1/4" BSPT PRESSURE PLUG	0.01 kg	2	VENDOR SUPPLY
4111-Z015	MILD STEEL	2.5 x 12 MACHINE DOWEL	0 kg	2	VENDOR SUPPLY
4111-Z017	MILD STEEL	1/16" BSPT PRESSURE PLUG	0 kg	1	VENDOR SUPPLY
4111-Z022	MILD STEEL	4 x 20 MACHINE DOWEL	0 kg	5	VENDOR SUPPLY
4111-Z023	Rubber	#-240 (3-3/4" x 1/8") -N7 NITRILE O-RING	0 kg	6	VENDOR SUPPLY

PARTS LIST (PARTS PER ONE MAIN ASSEMBLY ONLY)

PART NUMBER	MATERIAL	DESCRIPTION	MASS	QT	REFERENCE
4111-Z024	Rubber	#-241 (3-7/8" x 1/8") -N7 NITRILE O-RING	0 kg	1	VENDOR SUPPLY
4111-Z025	Rubber	#-017 (11/16" x 1/16) -N7 NITRILE O-RING	0 kg	1	VENDOR SUPPLY
4111-Z026	Rubber	#-117 (13/16" x 3/32) -N7 NITRILE O-RING	0 kg	2	VENDOR SUPPLY
4111-Z028	Rubber	#-241 (3-7/8" x 1/8") -N7 NITRILE O-RING	0 kg	1	VENDOR SUPPLY
4111-Z029	Rubber	#-227 (2-1/8 x 1/8") -N7 NITRILE O-RING	0 kg	8	VENDOR SUPPLY
4111-Z031	Rubber	(1/8") -N7 NITRILE O-RING (CUSTOM)	0 kg	2	VENDOR SUPPLY
4111-Z032	MILD STEEL	4 x 20 MACHINE DOWEL	0 kg	1	VENDOR SUPPLY
4111-Z033	MILD STEEL	4 x 35 MACHINE DOWEL	0 kg	4	VENDOR SUPPLY
ANSI B18.3.1M - M6x1 x 25	Steel, Mild	Socket Head Cap Screw - Metric	0 kg	5	VENDOR SUPPLY
ANSI B18.3.1M - M6x1 x 70	Steel, Mild	Socket Head Cap Screw - Metric	0 kg	5	VENDOR SUPPLY
ANSI B18.3.4M - M10 x 1.5 x 40	Steel, Mild	Socket Button Head Cap Screw - Metric	0 kg	4	VENDOR SUPPLY
ANSI B18.3.4M - M8 x 1.25 x 16	Steel, Mild	Socket Button Head Cap Screw - Metric	0 kg	8	VENDOR SUPPLY
BARBED HOSE FITTING - 1/2 HOSE x 1/2 BSPT	Stainless Steel, Austenitic	HOSE FITTING (COOLANT)	0.1 kg	2	VENDOR SUPPLY
C 2205 V	N/A	CARB, cylindrical bore-Toroidal roller bearings	0 kg	4	VENDOR SUPPLY
DIN 125 - A 10.5	Steel, Mild	Flat Washer	0 kg	6	VENDOR SUPPLY
DIN 472 - 20 x 1	Steel, Mild	Spring Retaining Ring	0 kg	2	VENDOR SUPPLY
DIN 6912 - M5 x 12	Steel, Mild	METRIC LOW HEAD CAP SCREW	0 kg	10	VENDOR SUPPLY
DIN 908 - G 0.25 A	Steel, Mild	METRIC LOW HEAD CAP SCREW	0 kg	1	VENDOR SUPPLY
DIN 908 - G 0.5 A	Steel, Mild	METRIC LOW HEAD CAP SCREW	0 kg	1	VENDOR SUPPLY
DIN 908 - G 0.75 A	Steel, Mild	METRIC LOW HEAD CAP SCREW	0.1 kg	1	VENDOR SUPPLY
JIS B 2402 - 60 80 8 A	Rubber	Oil seals - Spring loaded, metal cased, with dust-lip DM	0 kg	1	VENDOR SUPPLY
JIS B 2402 - 85 100 6 A	Rubber	Oil seals - Spring loaded, non-spring, rubber covered G	0 kg	1	VENDOR SUPPLY
M10 x 1.5	Steel, Mild	Hex Nut	0 kg	6	VENDOR SUPPLY
M10x1.5 x 20	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	12	VENDOR SUPPLY
M10x1.5 x 30	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	6	VENDOR SUPPLY
M10x1.5 x 45	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	2	VENDOR SUPPLY
M10x1.5 x 70	Steel, Mild	Broached Socket Head Cap Screw - Metric	0.1 kg	6	VENDOR SUPPLY
M6x1 x 16	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	32	VENDOR SUPPLY
M6x1 x 20	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	6	VENDOR SUPPLY
M8x1.25 x 16	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	36	VENDOR SUPPLY
ANSI B18.3.1M - M12x1.75 x 50	Steel, Mild	Broached Socket Head Cap Screw - Metric	0.1 kg	4	VENDOR SUPPLY
ANSI B18.3.1M - M8x1.25 x 30	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	5	VENDOR SUPPLY

TIGHTENING TORQUE

FASTENERS ENGAGED INTO THREAD INSERTS ARE TO BE TORQUED TO THE FOLLOWING SETTINGS:

- M6 =(3.5 N.m.) - M8 =(8.5 N.m.)
- M10 =(17 N.m.) - M12 =(40 N.m.)

ALL REMAINING FASTENERS TO BE TORQUED TO THE FOLLOWING SETTINGS UNLESS NOTED OTHERWISE:
(DO NOT TORQUE FASTENERS SCREWED DIRECTLY INTO ALLOY)

- M6 =(9.0 N.m.) - M10 =(44 N.m.)
- M8 =(22 N.m.) - M12 =(77 N.m.)

NOTES:

GENERAL

1. ALL SHARP EDGES TO BE REMOVED
2. ALL TIGHT CORNERS / RADII TO BE IDENTIFIED AND ADDRESSED.
3. ALL PARTS TO FIT TOGETHER WITHOUT FORCING EXCEPT WHERE TOLERANCED SO.
4. ALL FASTENERS TO BE TORQUED TO THE SETTINGS PROVIDED IN THE APPROPRIATE TABLES
5. ALL FASTENERS TO BE LUBRICATED USING ANTI-SEIZE LUBRICANT BEFORE ASSEMBLY EXCEPT WHERE NOTED TO USE A THREAD LOCKING COMPOUND
6. ALL MATING PARTS TO BE LIGHTLY OILED BEFORE ASSEMBLY
7. ALL GASKETS TO BE ASSEMBLED DRY
8. ALL O-RINGS TO BE LUBRICATED USING SILCONE GREASE BEFORE ASSEMBLY
9. ENGINE TO BE FILED WITH SAE 30 ENGINE OIL AS MINIMUM, SAE 40 ALSO APPROPRIATE.
10. ALL PARTS TO BE ASSEMBLED FREE OF DIRT AND UNWANTED CONTAMINANTS

SURFACE TREAT

- HEAT SURFACE (DO NOT SOAK!) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION. HERE-IN KNOWN AS "BLUED"
- ALL EXPOSED FERROUS PARTS TO BE LIGHTLY OILED TO PREVENT SURFACE CORROSION

HEAT TREATMENT

4340:
NORMALIZE: 850-880°C, FOR 1hr MIN, AIR COOL.
HARDEN: 830-860°C , OIL QUENCH
TEMPER: 540-680°C HOLD FOR 1 hr MIN, AT TEMPERATURE , AIR COOL. (REFER BOHLER TEMPERATURE CHART) HERE-IN KNOWN AS "T" CONDITION
NITIRIDE: NITRIDE DEPTH TO 0.5mm UNLESS NOTED OTHERWISE

EN36A:
NORMALIZE: 850-880°C, FOR 1hr MIN, AIR COOL.
ANNEAL: 650°-700°C , COOL SLOWLY IN CONTROLLED FURNACE
CASE HARDEN: HERE-IN KNOWN AS "CH" CONDITION, MINIMUM DEPTH 0.5mm, PROCESS TO BE DETERMINED BY HEAT TREATMENT PROVIDER.

CONROD

1. NORMALIZE AS SPECIFIED IN HEAT TREATMENT BEFORE MACHINING.
2. ROUGH MACHINE TO 3mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
3. DIMENSIONS SHOWN THUS: TO BE FINISH MACHINED WITH CAP ASSEMBLED TO CONNECTING ROD.
4. FIXINGS (BOLTS) TO BE LIGHTLY OILED & TORQUED TO 55Nm FOR MACHINING THE BIG END BEARING.
5. CONROD ASSEMBLIES TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A021 +/- 0.03kg, +/- 0.05mm
6. CONROD AND SHELL TO BE "BLUED" AS NOTED IN SURFACE TREATMENT

CRANKSHAFT

1. NORMALIZE AS SPECIFIED IN HEAT TREATMENT BEFORE MACHINING.
2. ROUGH MACHINE TO 3mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
3. FINAL CRANKSHAFT ASSSEMBLY TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A010 +/- 0.03kg, +/- 0.05mm
4. CRANKSHAFT TO BE NITRIDED AFTER FINAL MACHINING.
5. CRANKSHAFT TO BE "BLUED" AS NOTED IN SURFACE TREATMENT

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

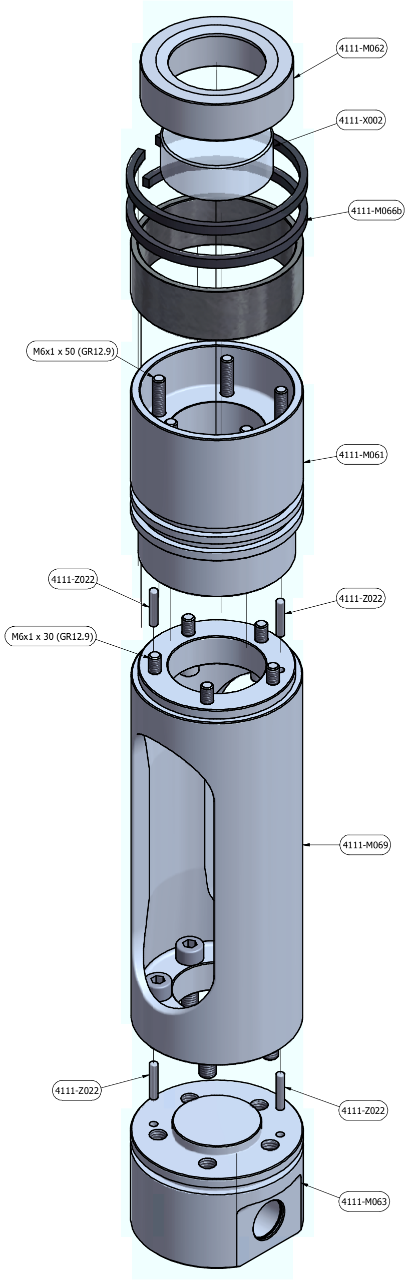
ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: N/A		THE UNIVERSITY OF SOUTHERN QUEENSLAND.	
DRAWN: KBD	DATE: 29/04/2014	CLIENT	
DESIGNED: KBD	DATE: 29/04/2014	DESCRIPTION	
CHECKED	DATE	DRAWING DESCRIPTION: GENERAL ASSEMBLY & PRESENTATION	
APPROVED	DATE	JOB No.	DRAWING No.: 4111-A001 — 3 OF 3 — SIZE: A1 — REV.: 0

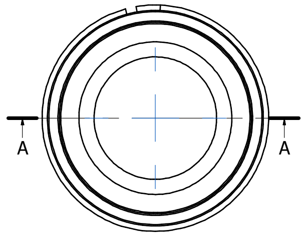
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-A001.idw

DO NOT SCALE - IF IN DOUBT ASK.



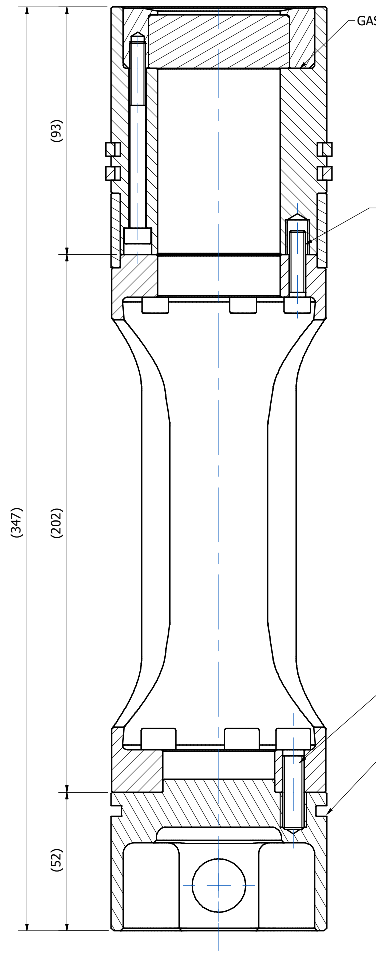
4111-A003 - EXPLODED VIEW -

SCALE: 1 : 1.25



4111-A003 - TOP VIEW -

SCALE: 1 : 1



GASKET NOT SHOWN

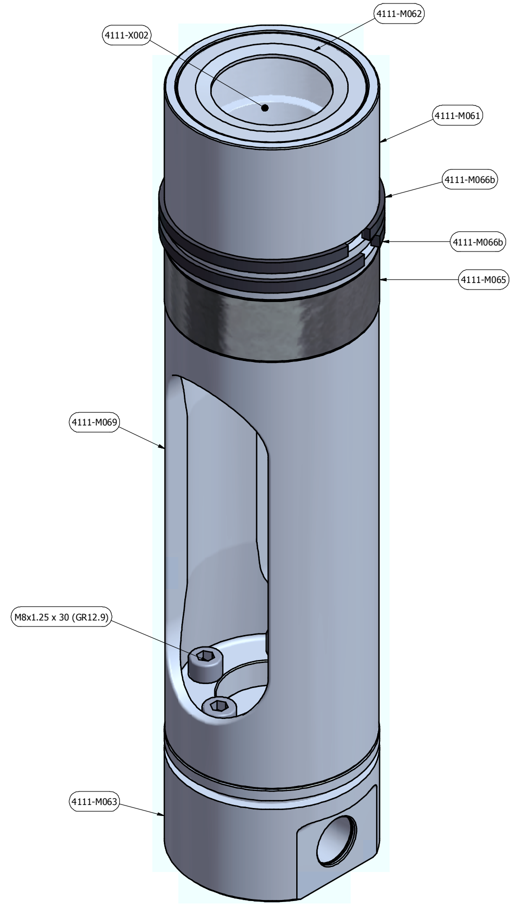
IMPORTANT:
MIN. 9mm EFFECTIVE THREAD ENGAGEMENT REQ'D
(TRIM LONGER BOLT TO SUIT IF NECESSARY)

IMPORTANT:
MIN. 12mm EFFECTIVE THREAD ENGAGEMENT REQ'D
(TRIM LONGER BOLT TO SUIT IF NECESSARY)

IMPORTANT:
- LOWER RING GROOVE DETAILS TO BE DETERMINED FROM SELECTED PART.
- OIL RETURN GALLERIES MUST BE DRILLED AS PART OF FINAL MACHINING

SECTION A-A

SCALE: 1 : 1



NOTE: 4111-066B & 4111-066 ARE SAME

4111-A003 - ISOMETRIC VIEW -

SCALE: 1 : 1

NOTES:

1. FASTENERS ENGAGED INTO THREAD INSERTS TO BE TORQUED TO THE FOLLOWING SETTINGS:
- M6 =(3.0 N.m.)
- M8 =(8.0 N.m.)
- M10 =(10.5 N.m.)
2. ALL FASTENERS MUST BE LUBRICATED BEFORE ASSEMBLY.
3. ALL INSERTS TO BE INSTALLED IN ACCORDANCE WITH "HELICOIL" PROCEDURES & SPECIFICATIONS.
3. ALL PARTS TO FIT TOGETHER FREELY WITHOUT FORCING
4. MAXIMUM ALLOWABLE RUN-OUT OF ANY PISTON SEGMENT WITH RESPECT TO ANOTHER IS 0.03mm
5. PISTON EXTERNAL SURFACES AND RING GROOVES TO BE FREE OF BURRS & SHARP EDGES.

PART NUMBER	DESCRIPTION	QTY	MASS	REFERENCE
M6x1 x 30 (GR12.9)	Broached Socket Head Cap Screw - Metric	5	0 kg	VENDOR SUPPLY
M6x1 x 50 (GR12.9)	Broached Socket Head Cap Screw - Metric	5	0 kg	VENDOR SUPPLY
4111-M061	PISTON HEAD	1	0.6 kg	4111-M061
4111-M062	OPTICAL WINDOW COLLAR	1	0.3 kg	4111-M062
4111-M063	LOWER PISTON	1	0.4 kg	4111-M063
4111-M065	SLIDING BUSH	1	0.1 kg	4111-M065
4111-M066b	LOW FRICTION PISTON RINGS	2	0 kg	4111-M066
4111-M069	PISTON EXTENSION	1	0.7 kg	4111-M069
4111-X002	OPTICAL ACCESS WINDOW	1	0.1 kg	4111-X002
4111-Z022	4 x 20 MACHINE DOWEL	4	0 kg	VENDOR SUPPLY
M8x1.25 x 30 (GR12.9)	Broached Socket Head Cap Screw - Metric	5	0 kg	VENDOR SUPPLY

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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SCALE: AS SHOWN			
DRAWN: KBD	DATE: 10/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 10/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	BOWDITCH PISTON ASSEMBLY
APPROVED	DATE	JOB No.: ENG4111/2	DRAWING No.: 4111-A003 1 OF 1, SIZE: A1, REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-A003.iam
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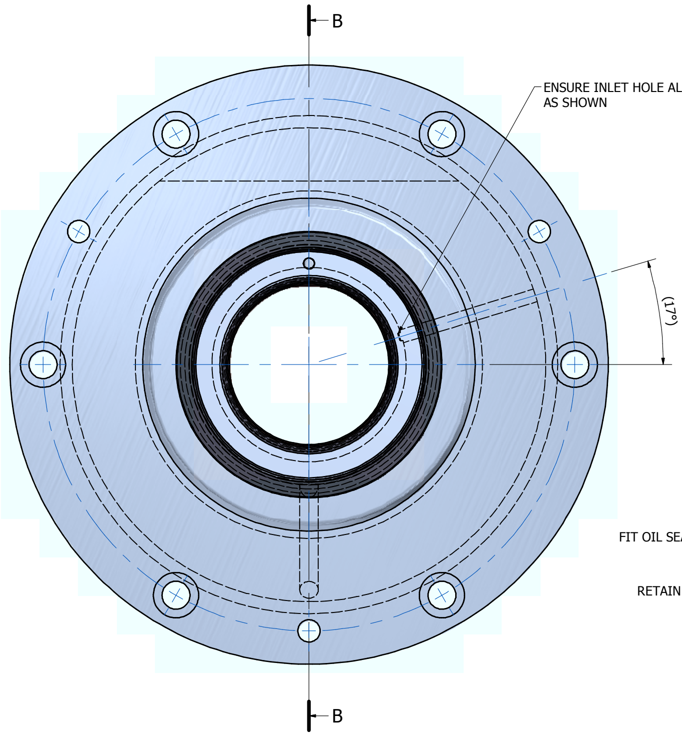
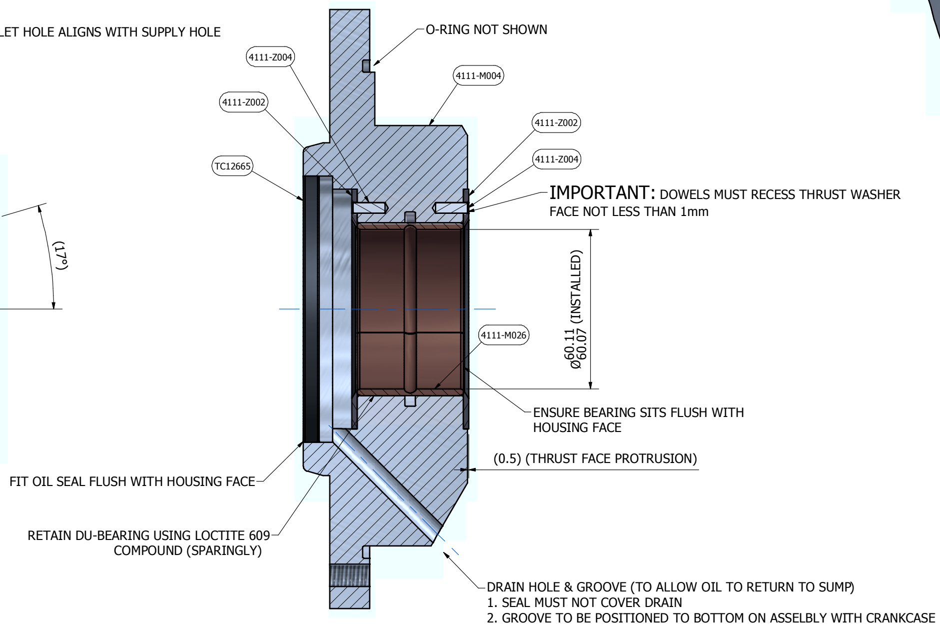
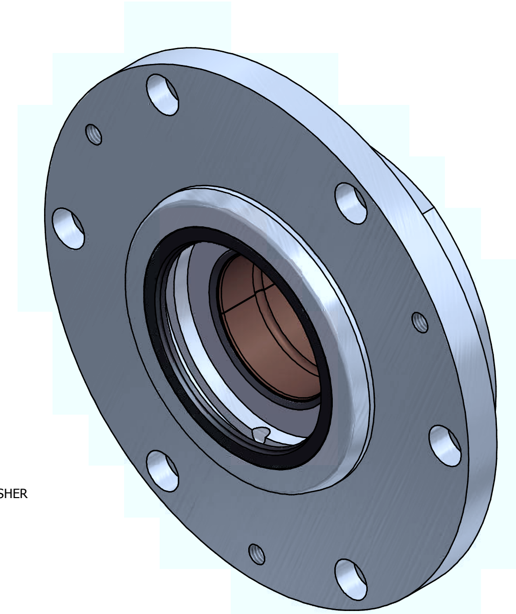


4111-A006 - ASSEMBLY -
SCALE: 1 : 1



SECTION B-B
SCALE: 1 : 1



4111-A006 - ISOMETRIC VIEW -
SCALE: 1 : 1

4111-M004	FLYWHEEL SIDE MAIN BEARING HOUSING	1	3.4 kg	4111-M004
4111-M026	6040DU - LEAD/PTFE BEARING (MODIFIED)	1	0.1 kg	4111-M026
4111-Z002	WC60DU - LEAD/PTFE THRUST BEARING (MODIFIED)	2	0 kg	4111-Z002
4111-Z004	4x12 MACHINE DOWEL	2	0 kg	VENDOR SUPPLY
TC12665	100x80x10 Oil seal - Spring loaded, metal cased, with dust-lip DM	1	0 kg	VENDOR SUPPLY
PART NUMBER	DESCRIPTION	QTY	MASS	KEYWORDS

PARTS LIST

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	10/08/2014	
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West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN: KBD	DATE: 22/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 22/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION: FLYWHEEL SIDE MAIN BEARING
APPROVED	DATE	JOB No. ENG4111/2 · DRAWING No. 4111-A006 · 1 OF 1 · SIZE A1 · REV. 0

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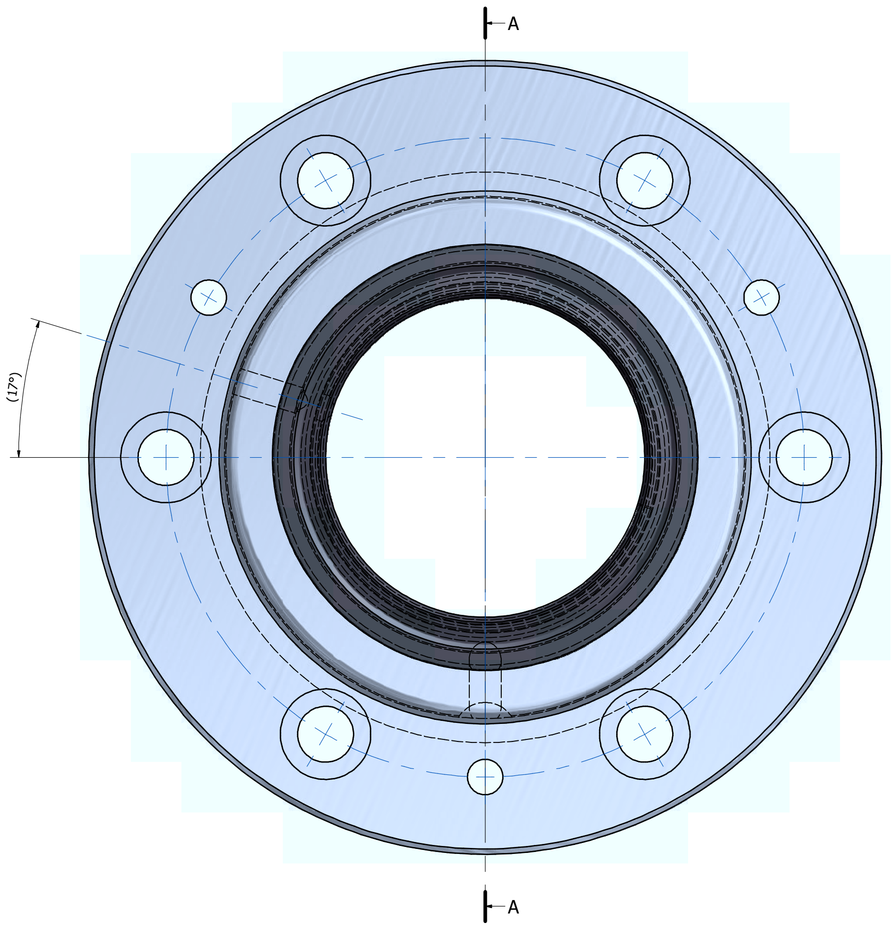
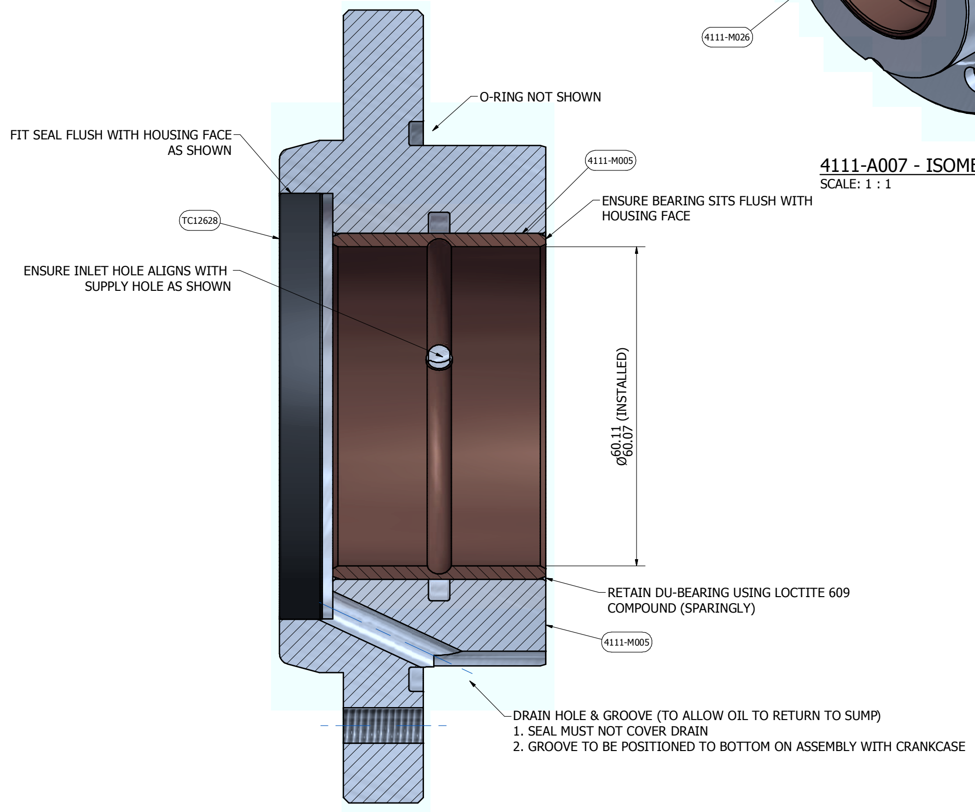


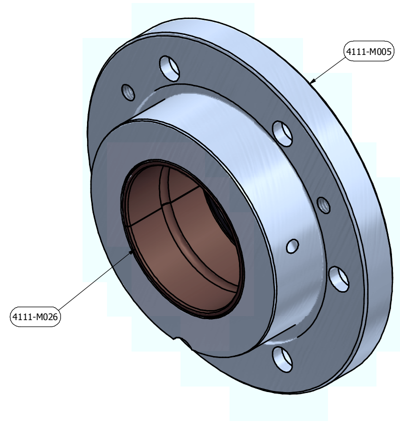
4111-A007 - ASSEMBLY -

SCALE: 2 : 1



SECTION A-A

SCALE: 2 : 1



4111-A007 - ISOMETRIC VIEW -

SCALE: 1 : 1

PART NUMBER	DESCRIPTION	QTY	MASS	KEYWORDS
4111-M005	ENCODER SIDE MAIN BEARING HOUSING	1	0.9 kg	4111-M005
4111-M026	6040DU - LEAD/PTFE BEARING (MODIFIED)	1	0.1 kg	4111-M026
TC12628	80x60x8 Oil seal - Spring loaded, metal cased, with dust-lip DM	1	0 kg	VENDOR SUPPLY

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	10/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN	DATE
KBD	22/07/2014

DESIGNED	DATE
KBD	22/07/2014

CHECKED | DATE

APPROVED | DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: ENCODER SIDE MAIN BEARING ASSEMBLY

JOB No. ENG4111/2 | DRAWING No. 4111-A007 | 1 OF 1 | SIZE A1 | REV. 0

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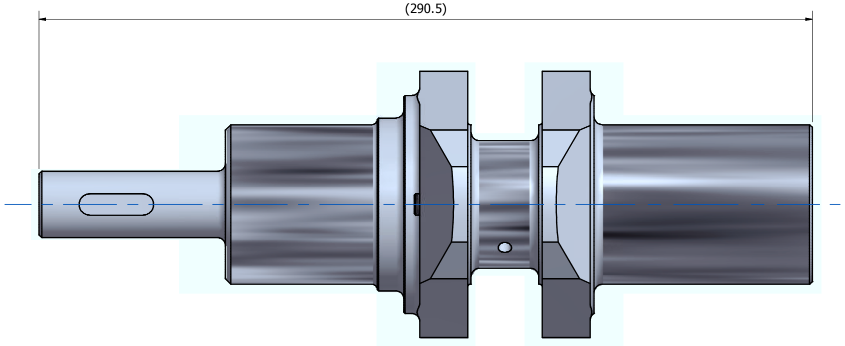
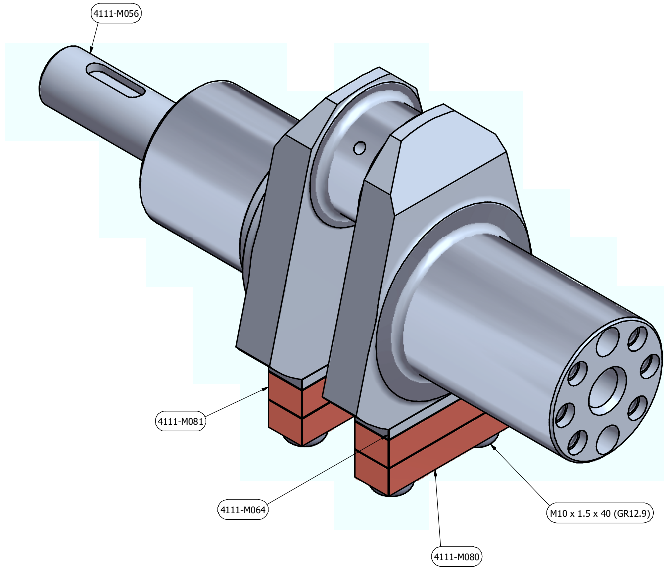
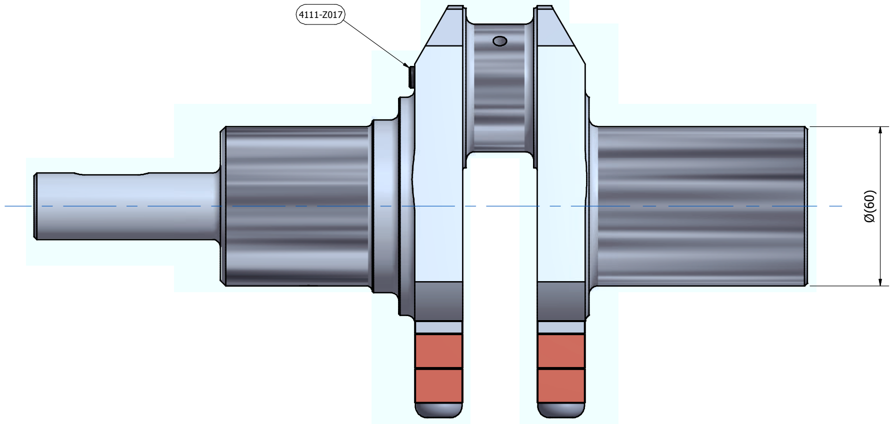


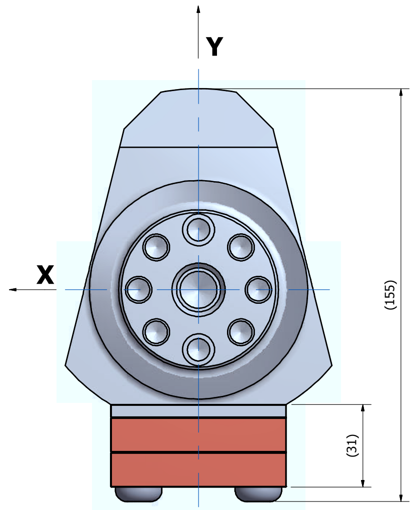
- TOP VIEW -



4111-A010 - ISOMETRIC VIEW -
SCALE: 1 : 1



4111-A010 - SIDE VIEW -
SCALE: 1 : 1



- END VIEW -

ASSEMBLY NOTES:

1. ALL PARTS TO FIT TOGETHER WITHOUT FORCING.
2. BALANCING: CRANKSHAFT ASSEMBLY C.O.G. TO BE POSITIONED Y: -3.9mm ±0.05 X: 0mm ±0.05mm
3. ALL SHARP EDGES TO BE REMOVED BEFORE FITTING.
4. ALL FASTENERS TO BE TIGHTENED TO RECOMMENDED FASTENING TORQUE (REFER 4111-001).
5. LOCTITE BALANCE WEIGHT FASTENERS AND OIL PLUG IN POSITION USING "220" THREAD COMPOUND.
6. CHECK BALANCE BY FITTING COLLAR OF EQUIVALENT CONROD LOWER HALF MASS TO BIG END AND STATICALLY BALANCE CRANK

PARTS LIST

PART NUMBER	MATERIAL	DESCRIPTION	QTY	MASS	KEYWORDS
4111-M056	AS1444 -4340	ENGINE CRANKSHAFT	1	6.448 kg	4111-M056
4111-M064	AS3678 -GR250	CRANK BALANCE WEIGHT	2	0.039 kg	4111-M064
4111-M080	SD170	CRANK BALANCE WEIGHT	2	0.22 kg	4111-M080
4111-M081	SD170	CRANK BALANCE WEIGHT	2	0.219 kg	4111-M081
4111-Z017	MILD STEEL	1/16" BSPT PRESSURE PLUG	1	0.003 kg	VENDOR SUPPLY
4111-Z033	MILD STEEL	4 x 35 MACHINE DOWEL	4	0.003 kg	VENDOR SUPPLY
M10 x 1.5 x 40 (GR12.9)	Steel, Mild	Broached Hexagon Socket Button Head Cap Screw - Metric	4	0.032 kg	VENDOR SUPPLY

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN: KBD	DATE: 3/08/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND	
DESIGNED: KBD	DATE: 3/08/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN	
CHECKED	DATE	DRAWING DESCRIPTION: CRANKSHAFT ASSEMBLY (G.A.)	
APPROVED	DATE	JOB No.: ENG4111/2	DRAWING No.: 4111-A010 1 OF 1 SIZE: A1 REV.: 0

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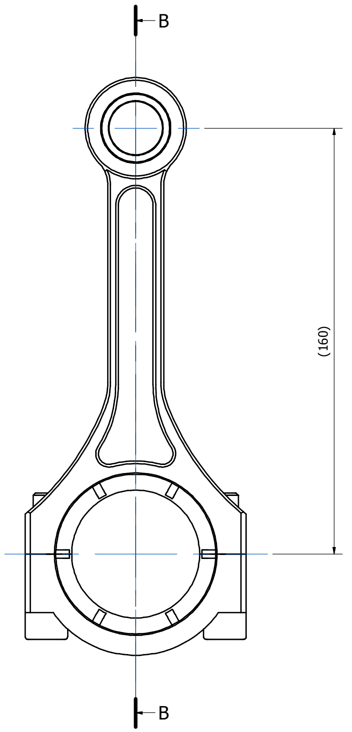
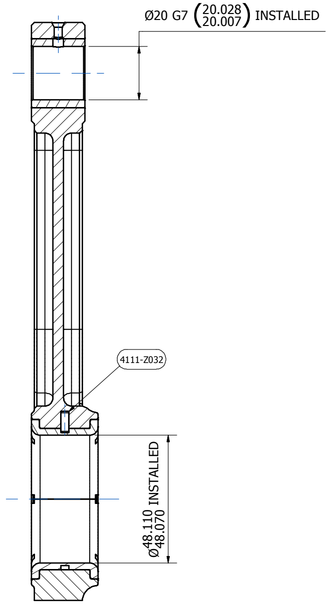
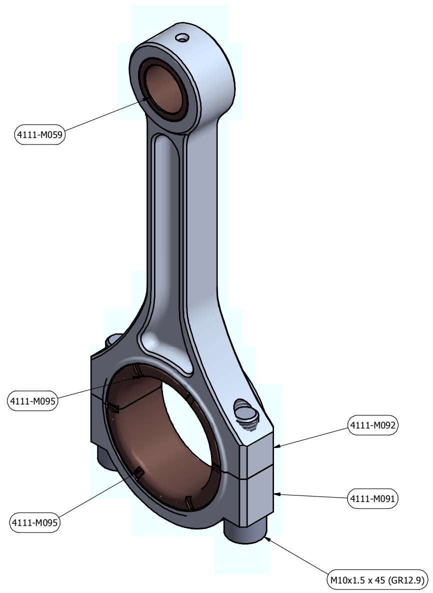


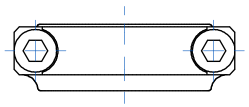
4111-A020 - TOP VIEW -
SCALE: 1 : 1



SECTION B-B
SCALE: 1 : 1



4111-A020 - ISOMETRIC VIEW -
SCALE: 1 : 1



- BOTTOM VIEW -

NOTES:

1. FASTENERS ENGAGED INTO THREAD INSERTS TO BE TORQUED TO THE FOLLOWING SETTINGS:
- M10 =(75 N.m.)
2. ALL FASTENERS MUST BE TIGHTENED WITH 220 LOCTITE THREAD COMPOUND TO PREVENT LOOSENING
3. ALL PARTS TO FIT TOGETHER FREELY WITHOUT FORCING EXCEPT THE LITTLE END BEARING
4. CONROD EXTERNAL SURFACES TO BE FREE OF BURRS & SHARP EDGES.

PARTS LIST

PART NUMBER	DESCRIPTION	QTY	MASS	REFERENCE
4111-M059	LITTLE END BEARING	1	0 kg	4111- M059
4111-M091	CONROD BIG END SADDLE	1	0.2 kg	VENDOR SUPPLY
4111-M092	CONROD	1	0.5 kg	4111-M092
4111-M095	BIG END BEARING SHELL	2	0 kg	4111-M095
4111-Z032	4 x 20 MACHINE DOWEL	1	0 kg	VENDOR SUPPLY
M10x1.5 x 45 (GR12.9)	Broached Socket Head Cap Screw - Metric	2	0 kg	

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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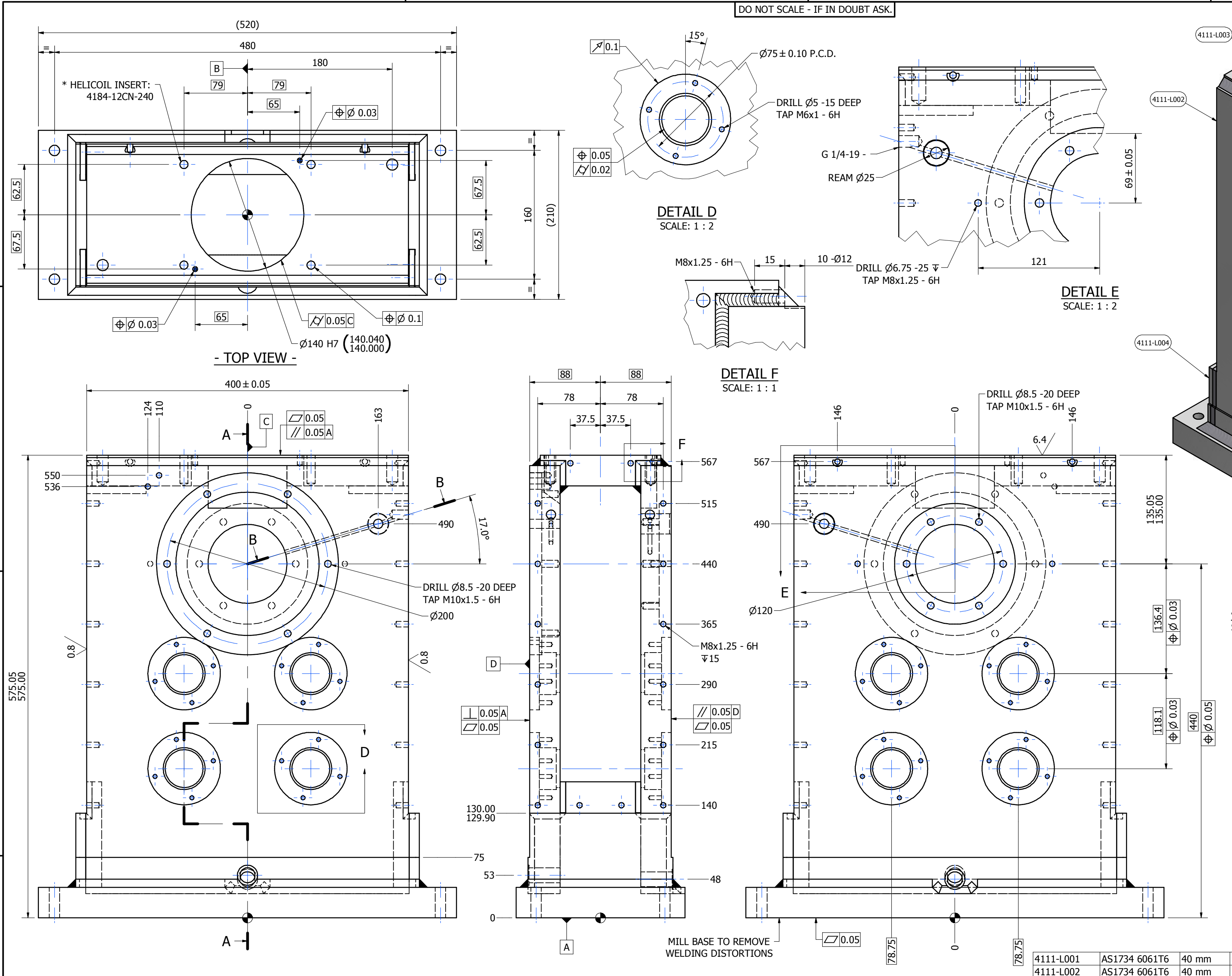
		DATE
DRAWN	KBD	10/07/2014
DESIGNED	KBD	10/07/2014
CHECKED		
APPROVED		

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: CONROD ASSEMBLY

JOB No. ENG4111/2 | DRAWING No. 4111-A021 | 1 OF 1 | SIZE A1 | REV. 0

DO NOT SCALE - IF IN DOUBT ASK.

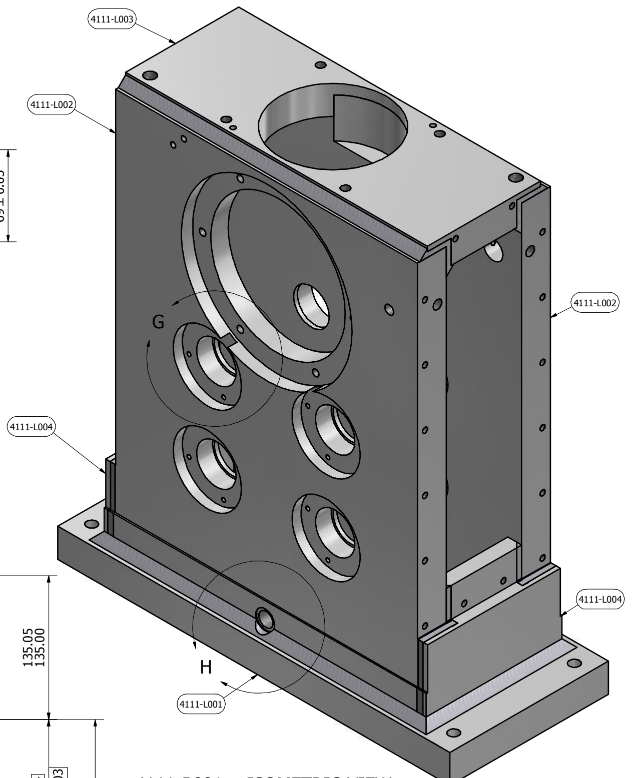


- TOP VIEW -

4111-B001 - FLYWHEEL SIDE VIEW -
SCALE: 1 : 2.5

- R.H.S. VIEW -

- PULLEY SIDE VIEW -



4111-B001 - ISOMETRIC VIEW -
SCALE: 1 : 2.5

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1665:2005

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS	KEYWORDS
4111-L001	AS1734 6061T6	40 mm	BASE PLATE	1	11 kg	4111 - CUT FILE
4111-L002	AS1734 6061T6	40 mm	SIDE PLATE	2	23.1 kg	4111 - CUT FILE
4111-L003	AS1734 6061T6	40 mm	TOP PLATE	1	5.1 kg	4111 - CUT FILE
4111-L004	AS1734 6061T6	32 mm	LOWER END PLATE (SUPERCEEDED BY 4111-B010)	2	1.2 kg	4111 - CUT FILE

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	07/08/14	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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DRAWN: KBD	DATE: 29/04/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 29/04/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	CRANKCASE DETAIL & WELDMENT
APPROVED	DATE	JOB No.: ENG4111/2	DRAWING No.: 4111-B001

1 OF 5 | SIZE: A1 | REV.: 0

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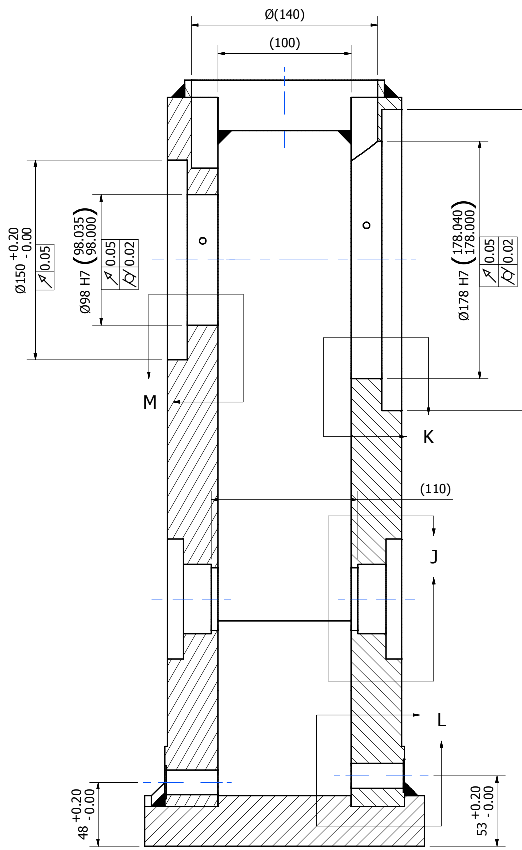
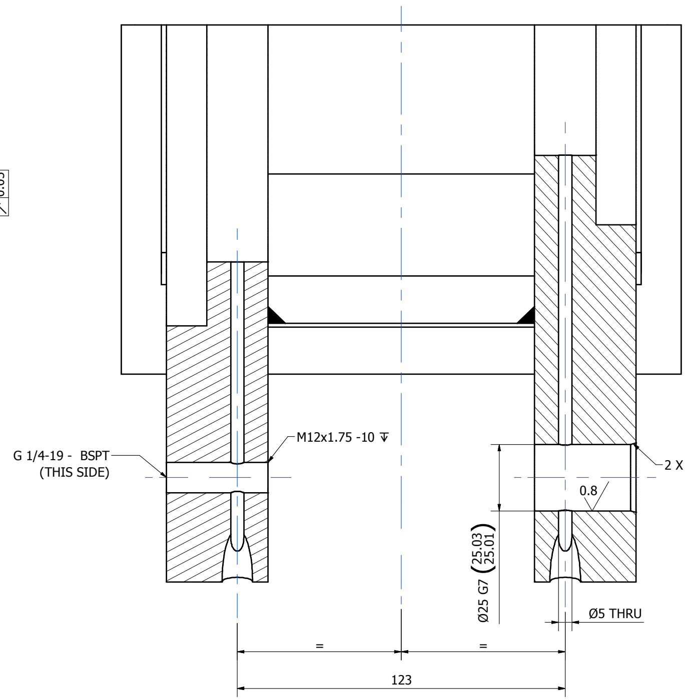
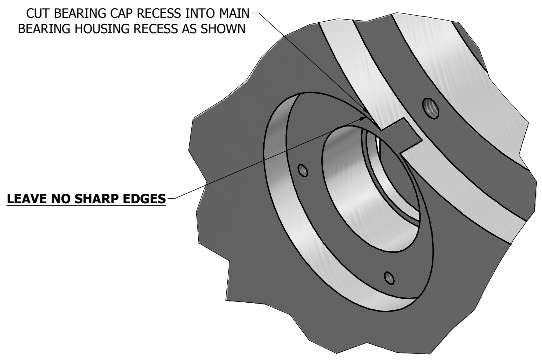


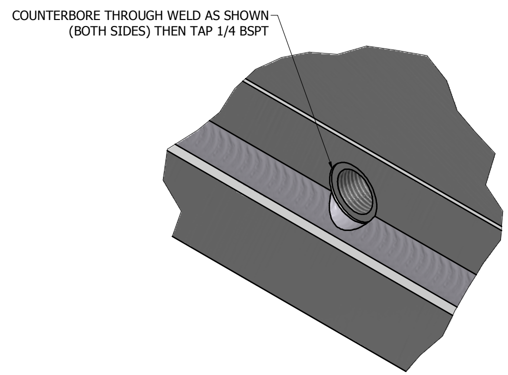
SECTION A-A
SCALE: 1 : 2



SECTION B-B
SCALE: 1 : 1



DETAIL G
SCALE: 1 : 1



DETAIL H
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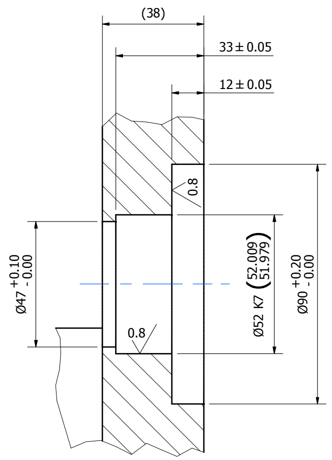

DETAIL J
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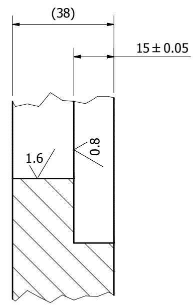

DETAIL K
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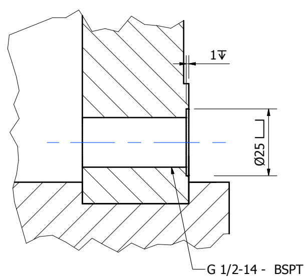
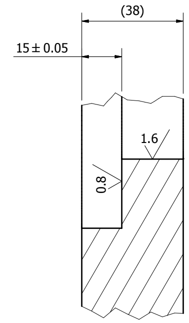


DETAIL L
SCALE: 1 : 1



DETAIL M
SCALE: 1 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1665:2005

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	THE UNIVERSITY OF SOUTHERN QUEENSLAND.
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DESIGNED KBD	DATE 29/04/2014	DESCRIPTION OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION CRANKCASE DETAIL & WELDMENT
APPROVED	DATE	JOB No. ENG4111/2 — DRAWING No. 4111-B001 — 2 OF 5 — SIZE A1 — REV. 0

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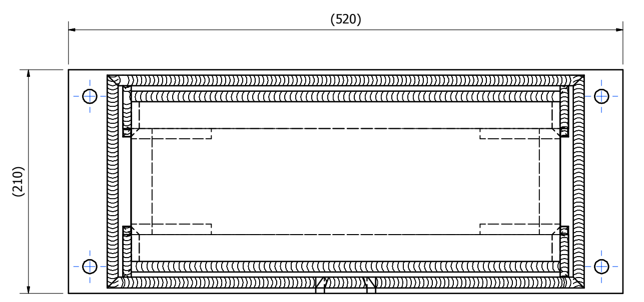
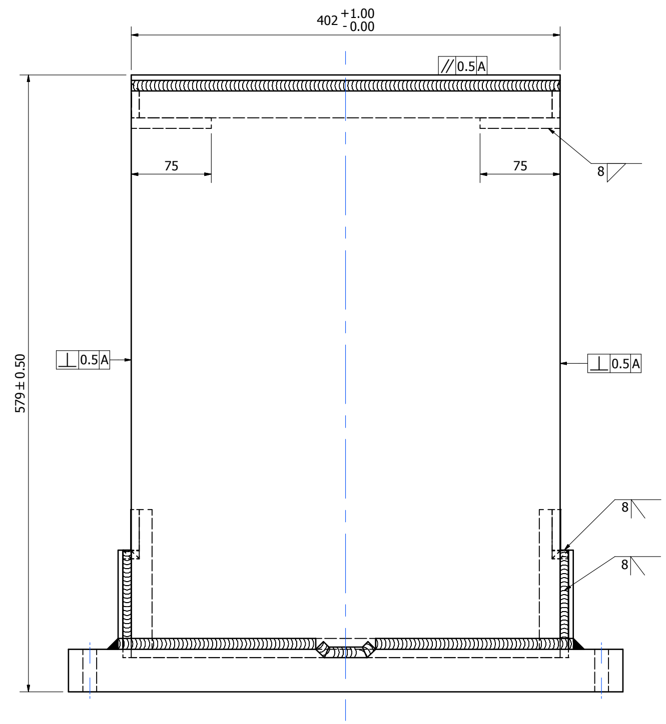
DO NOT SCALE - IF IN DOUBT ASK.

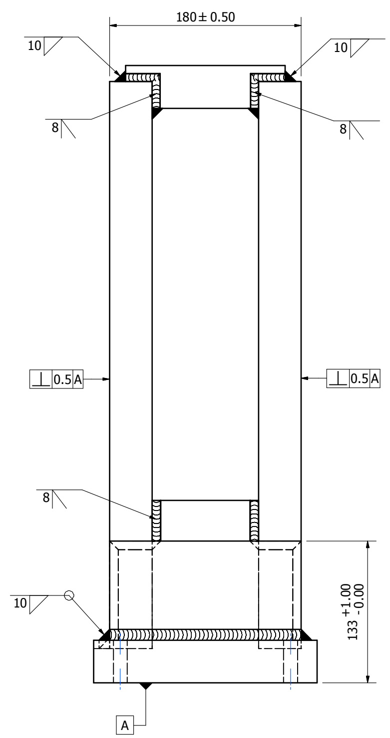


- TOP VIEW -

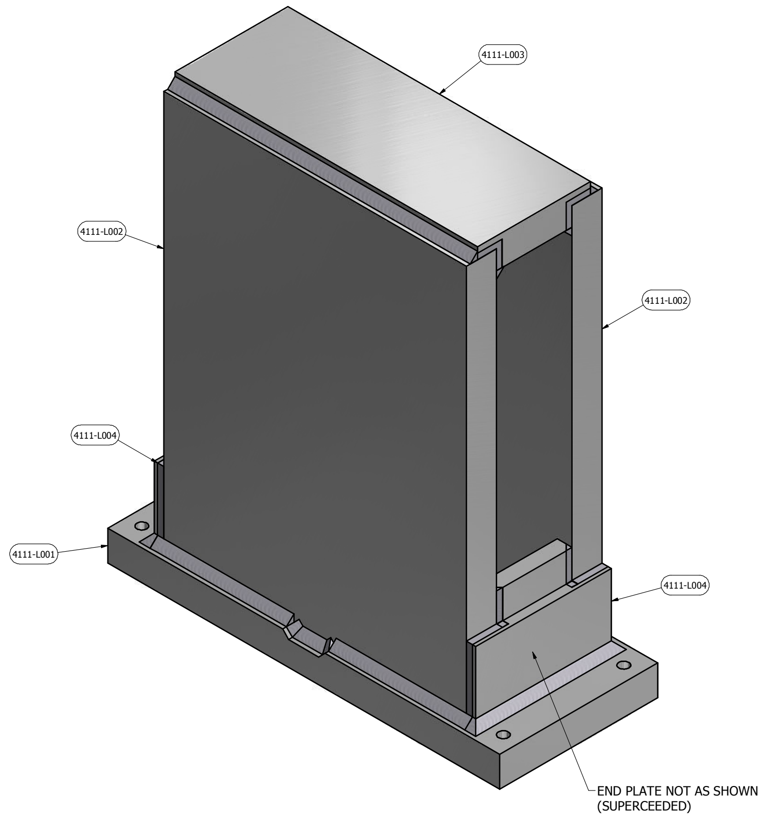


4111-B001 (PRE-MACHINING) / WELDMENT - OIL DRAIN SIDE -

SCALE: 1 : 2.5



- SIDE VIEW -



4111-B001 (PRE-MACHINING) / WELDMENT - ISOMETRIC VIEW -

SCALE: 1 : 2.5

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS
4111-L001	AS1734 6061T6	40 mm	BASE PLATE	1	11 kg
4111-L002	AS1734 6061T6	40 mm	SIDE PLATE	2	23.1 kg
4111-L003	AS1734 6061T6	40 mm	TOP PLATE	1	5.1 kg
4111-L004	AS1734 6061T6	32 mm	LOWER END PLATE (SUPERCEEDED BY 4111-B010)	2	1.2 kg

PARTS LIST

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DESIGNED: KBD	DATE: 29/04/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN	
CHECKED	DATE	DRAWING DESCRIPTION: CRANKCASE DETAIL & WELDMENT	
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-B001 — 3 OF 5 — SIZE A1 — REV. 0

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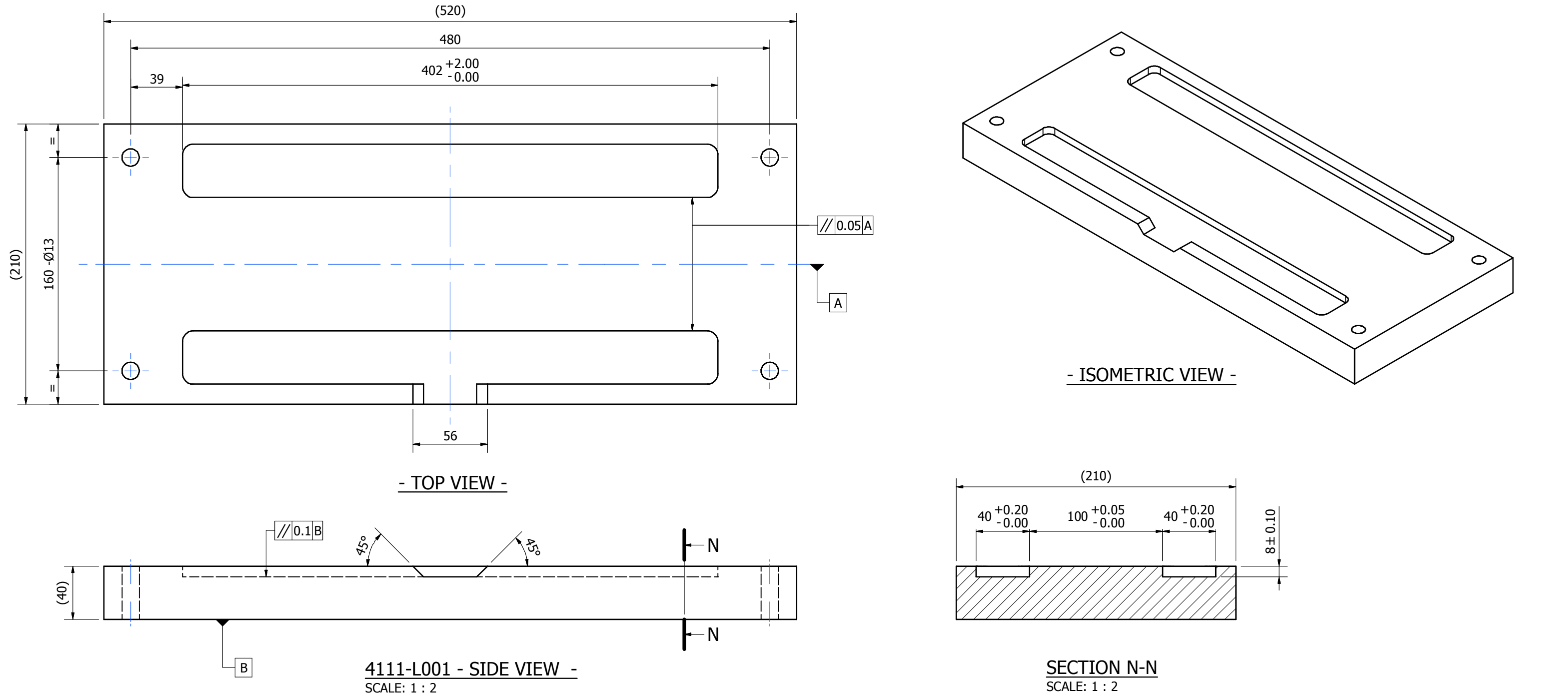

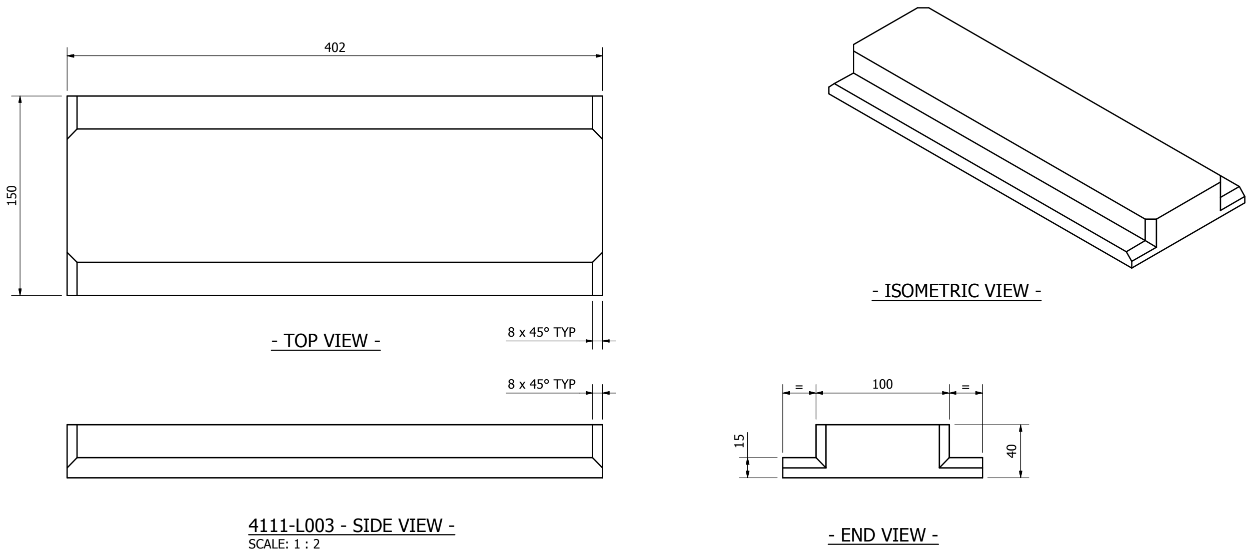

GENERAL TOLERANCES UNLESS INDICATED		ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE
PREPARATION AND ASSEMBLY	MACHINING	COPYRIGHT :- REPRODUCTION OF THIS DRAWING WHOLLY OR IN PART, IS PROHIBITED UNLESS WRITTEN PERMISSION IS OBTAINED FROM THE UNIVERSITY OF SOUTHERN QUEENSLAND (USQ)
< 25mm .... ±1.0mm	0 .... ±0.1mm	
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm	
>600mm .... ±2.0mm	0.00 .... ±0.01mm	
ANGLE .... ±1°	.... 3.2	

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DRAWN: KBD	DATE: 29/04/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND			
DESIGNED: KBD	DATE: 29/04/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN			
CHECKED	DATE	DRAWING DESCRIPTION: CRANKCASE DETAIL & WELDMENT			
APPROVED	DATE	JOB No.: ENG4111/2	DRAWING No.: 4111-B001 4 OF 5	SIZE: A1	REV.: 0

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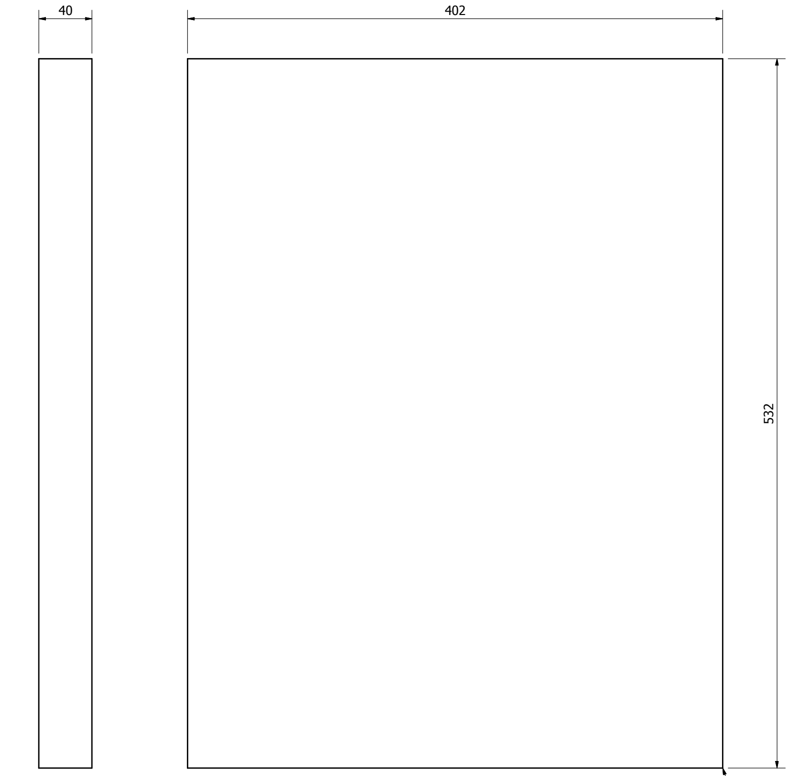
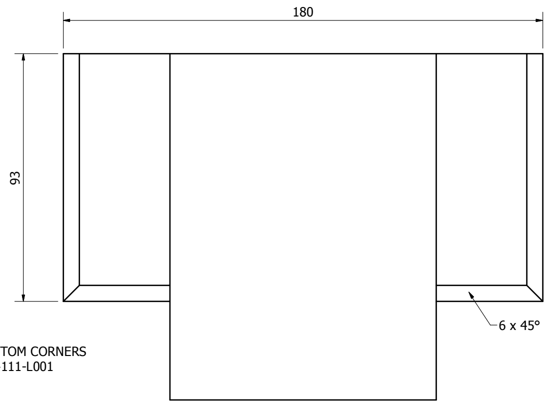
DO NOT SCALE - IF IN DOUBT ASK.



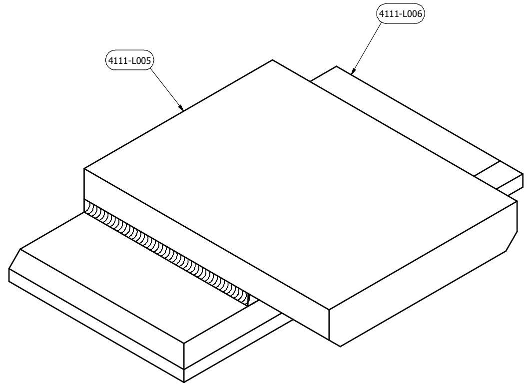
- SIDE VIEW -

4111-L002 - FRONT VIEW -
SCALE: 1 : 2

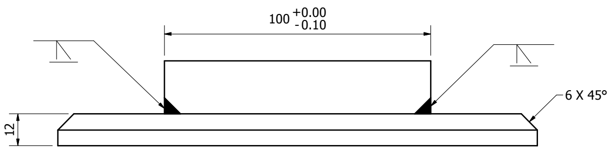
CHAMFER BOTTOM CORNERS
TO FIT INTO 4111-L001



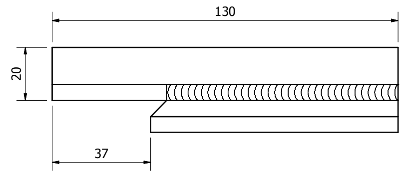
- TOP VIEW -



-ISOMETRIC VIEW -



4111-B010 - DETAIL -
SCALE: 1 : 1
NOTE: ALTERNATIVE END PLATE DESIGN (REPLACE 4111-L004)



- END VIEW -

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS	KEYWORDS
4111-L005	AS1734 6061T6	20 mm	LOWER END PLATE (PART 1)	1	0.7 kg	4111 - CUT FILE
4111-L006	AS1734 6061T6	12 mm	LOWER END PLATE (PART 2)	1	0.5 kg	4111 - CUT FILE

PARTS LIST

GENERAL TOLERANCES UNLESS INDICATED	ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE
PREPARATION AND ASSEMBLY: < 25mm .... ±1.0mm; 25 - 600mm .... ±1.5mm; >600mm .... ±2.0mm; ANGLE .... ±1°	MACHINING: 0 .... ±0.1mm; 0.0 .... ±0.05mm; 0.00 .... ±0.01mm

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
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DATE	29/04/2014
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DATE	29/04/2014
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DATE	
APPROVED	
DATE	

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CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION	CRANKCASE DETAIL & WELDMENT
JOB No.	ENG4111/2
DRAWING No.	4111-B001
	5 OF 5
SIZE	A1
REV.	0

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DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

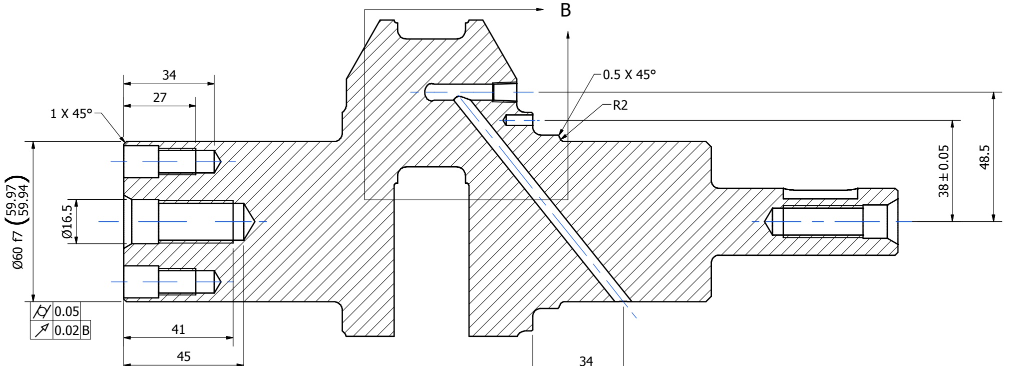
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

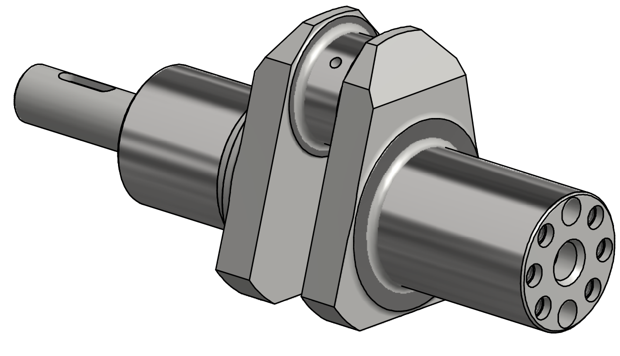
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



SECTION A-A

SCALE: 1 : 1

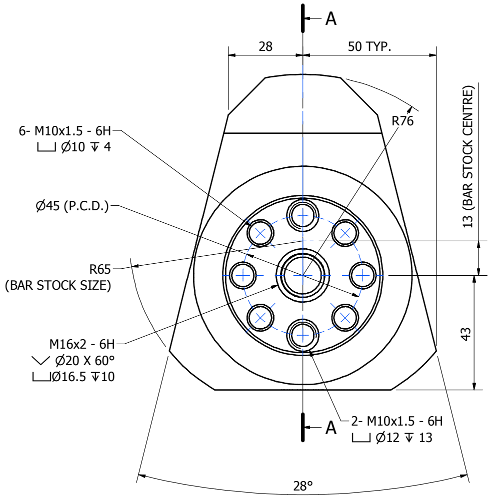


4111-M056 - ISOMETRIC VIEW -

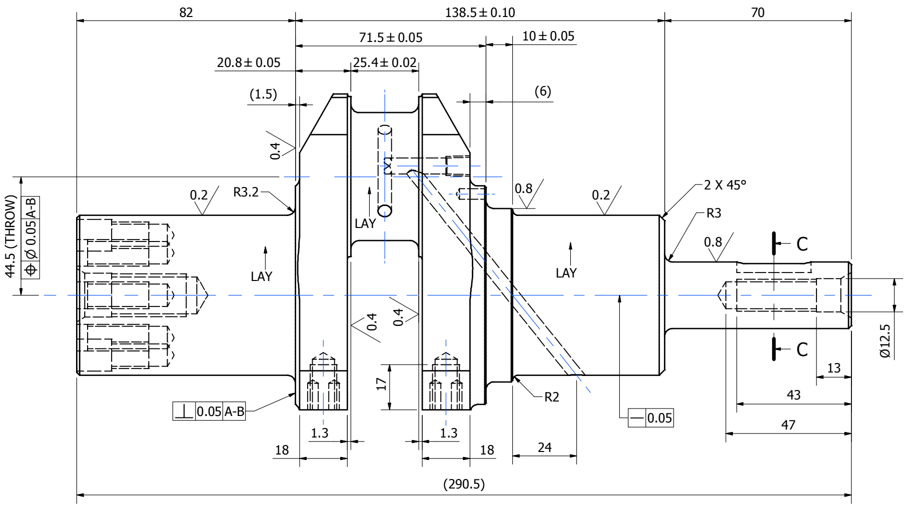
HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- NITRIDE AFTER FINAL MACHINING
- HEAT TO APPROX 150°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

MACHINING:
- ROUGH MACHINE TO 3mm ABOVER FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.
- NOTE THE DIRECTION OF FINISH LAY ↔

IMPORTANT:
- CRANKSHAFT IS OFFSET AXIALLY 2.5mm TOWARDS REAR OF CRANKSHAFT TO ALLOW FOR SAME OFFSET IN CONROD

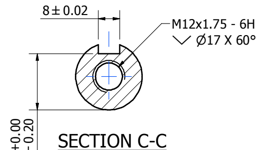


- END VIEW -



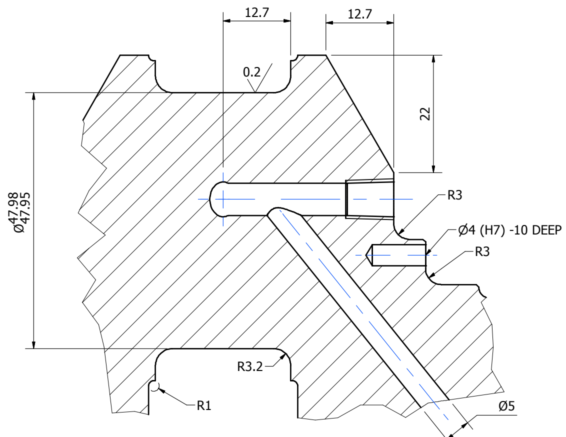
4111-M056 - DETAIL -

SCALE: 1 : 1



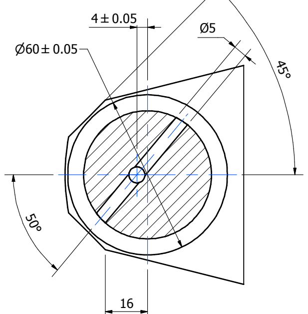
SECTION C-C

SCALE: 1 : 1



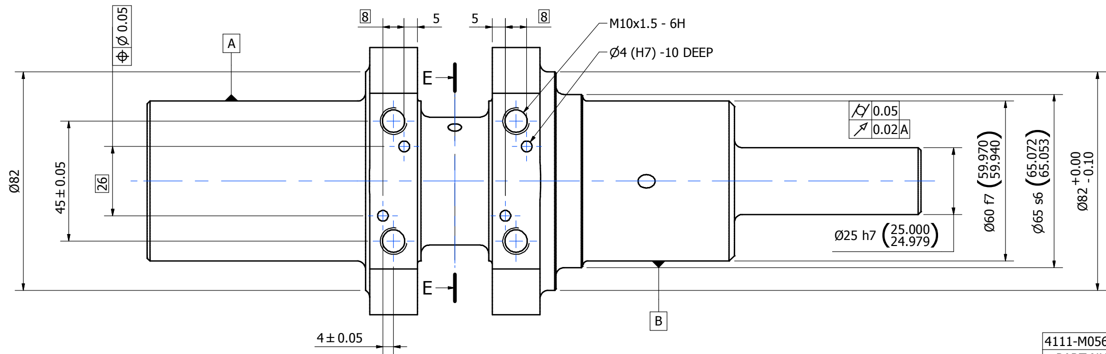
DETAIL B

SCALE: 2 : 1



SECTION E-E

SCALE: 1 : 1



PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M056	1	AS1444 -4340	ENGINE CRANKSHAFT	6.4 kg

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
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UNIVERSITY of SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN
DRAWN: KBD	DATE: 13/07/2014
DESIGNED: KBD	DATE: 13/07/2014
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APPROVED	DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: ENGINE CRANKSHAFT DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M056	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M056.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M056.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

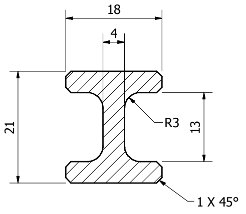
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

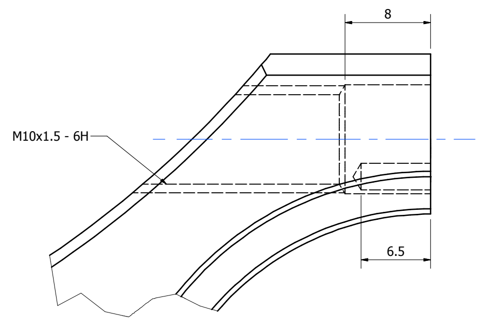
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

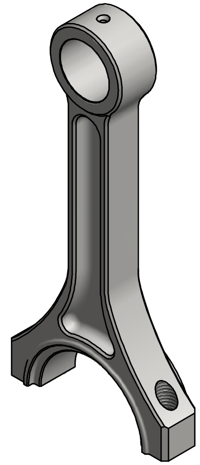
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



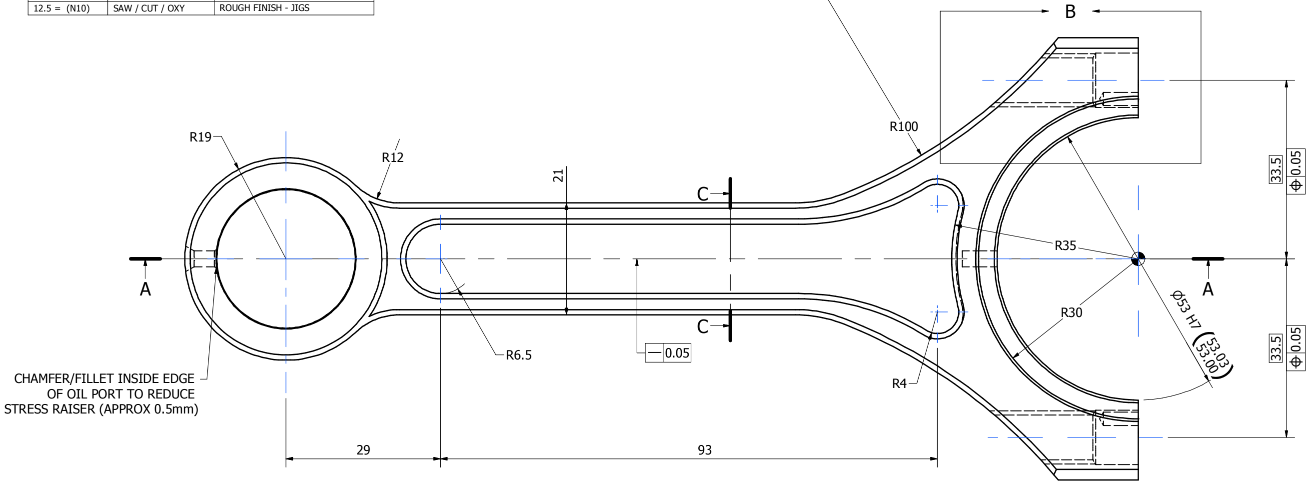
SECTION C-C
SCALE: 2 : 1



DETAIL B
SCALE: 4 : 1



4111-M092 - ISOMETRIC VIEW -
SCALE: 1 : 1



4111-M092 - DETAIL -
SCALE: 2 : 1

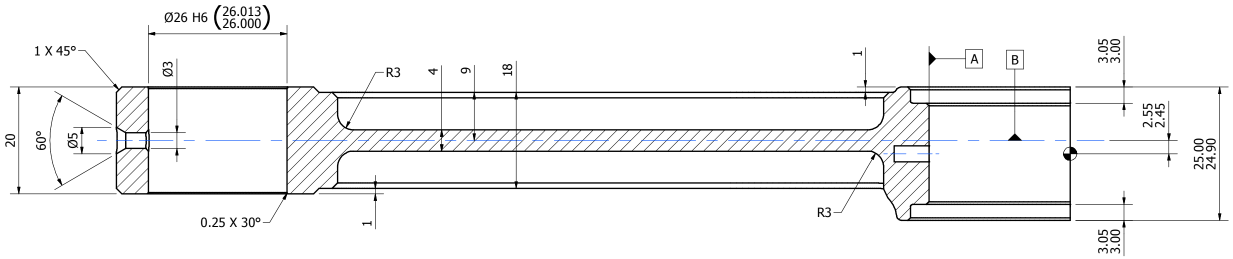
NOTE:

1. DIMENSIONS SHOWN THUS: TO BE FINISH MACHINED WITH CAP ASSEMBLED TO CONNECTING ROD. FIXINGS (BOLTS) TO BE LIGHTLY OILED & TORQUED TO 55Nm FOR MACHINING.
2. CONROD ASSEMBLIES TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A021 +/- 0.03kg, +/- 0.05mm
3. ALL SHARP EDGES TO BE REMOVED
4. ALL TIGHT CORNERS / RADII TO BE IDENTIFIED AND ADDRESSED.
5. ALL PARTS TO FIT TOGETHER WITHOUT FORCING

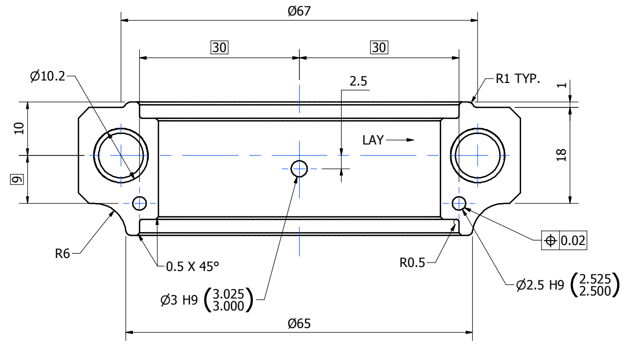
HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- HEAT TO APPROX 150°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

MACHINING:
- ROUGH MACHINE TO 3mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.
- NOTE THE DIRECTION OF FINISH LAY ↔

IMPORTANT:
- CRANKSHAFT IS OFFSET AXIALLY 2.5mm TOWARDS REAR OF CRANKSHAFT TO ALLOW FOR SAME OFFSET IN CONROD



SECTION A-A
SCALE: 2 : 1



- END VIEW -

4111-M092	1	AS1444 -4340	CONROD	0.48 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN KBD	DATE 13/07/2014	CLIENT UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 13/07/2014	DESCRIPTION OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION CONROD DETAIL
APPROVED	DATE	JOB No. ENG4111/2 · DRAWING No. 4111-M092 · 1 OF 1 · SIZE A1 · REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M092.ipt
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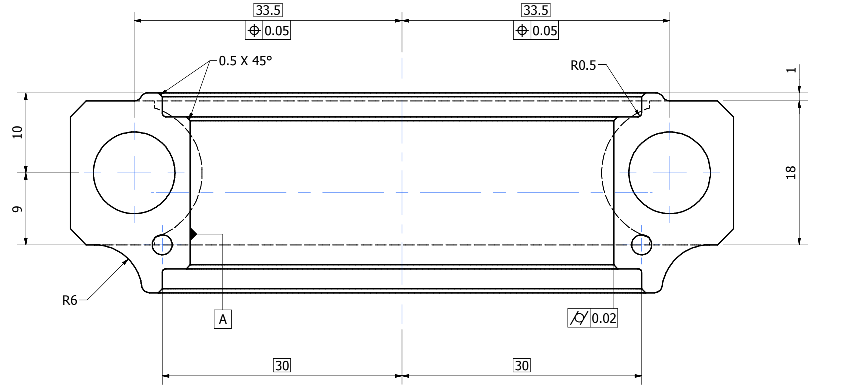
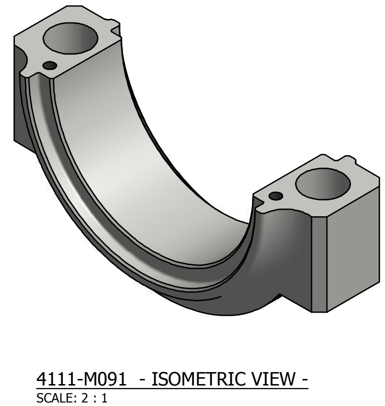
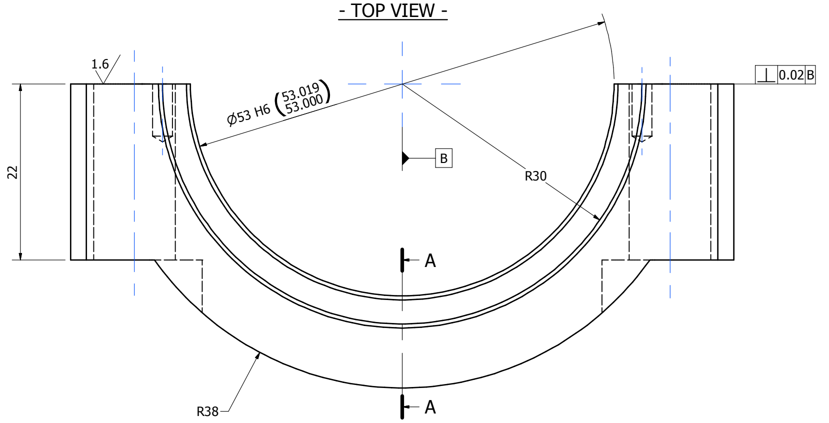
DO NOT SCALE - IF IN DOUBT ASK.

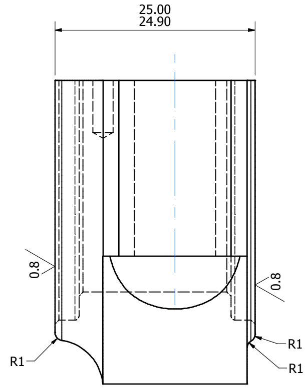


- TOP VIEW -

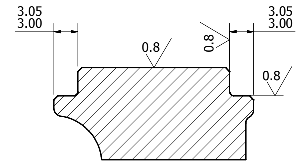




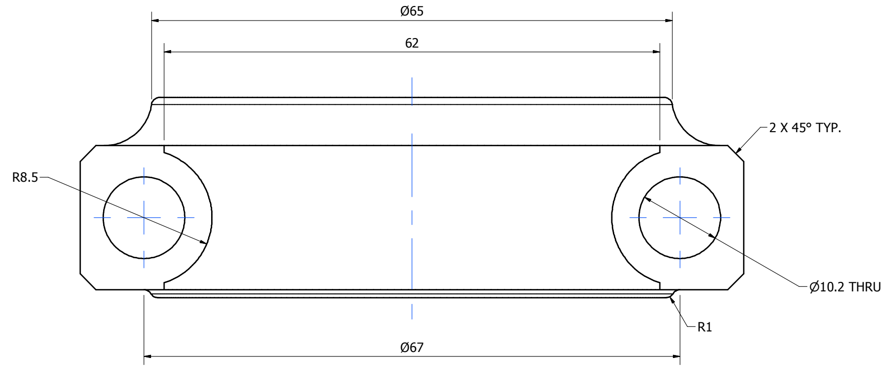
4111-M091 - DETAIL -
SCALE: 3 : 1



- SIDE VIEW -



SECTION A-A
SCALE: 3 : 1



- BOTTOM VIEW -

NOTE:

1. DIMENSIONS SHOWN THUS: TO BE FINISH MACHINED WITH CAP ASSEMBLED TO CONNECTING ROD. FIXINGS (BOLTS) TO BE LIGHTLY OILED & TORQUED TO 55Nm FOR MACHINING.
2. CONROD ASSEMBLIES TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A021 +/- 0.03kg, +/- 0.05mm
3. ALL SHARP EDGES TO BE REMOVED
4. ALL CORNERS / RADII TO BE 0.5mm UNLESS NOTED.
5. ALL PARTS TO FIT TOGETHER WITHOUT FORCING

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- NITRIDE AFTER FINAL MACHINING

MACHINING:
- ROUGH MACHINE TO 3mm ABOVER FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.
- NOTE THE DIRECTION OF FINISH LAY ⟷

SURFACE TREATMENT:
- SURFACE TO BE "BLUED" ON COMPLETION (REFER 4111-A001 FOR DETAILS)

IMPORTANT:
- CRANKSHAFT IS OFFSET AXIALLY 2.5mm TOWARDS REAR OF CRANKSHAFT TO ALLOW FOR SAME OFFSET IN CONROD

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M091	1	AS1444 -4340	CONROD BIG END SADDLE	0.19 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	
DRAWN	KBD	DATE 3/08/2014
DESIGNED	KBD	DATE 3/08/2014
CHECKED		DATE
APPROVED		DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: BIG END CAP DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M091	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M091.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M091.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

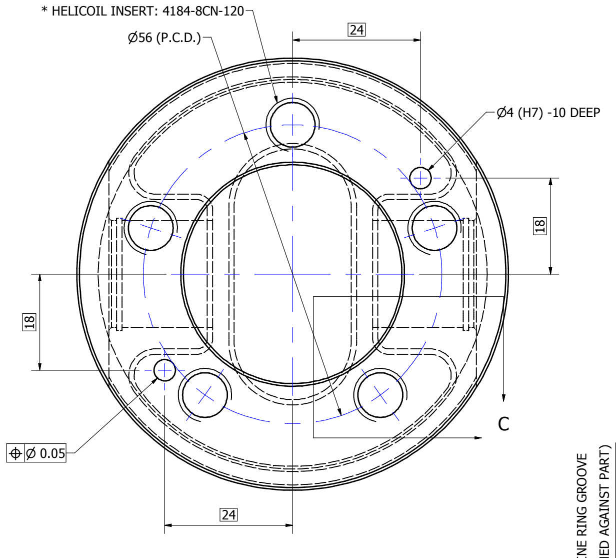
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

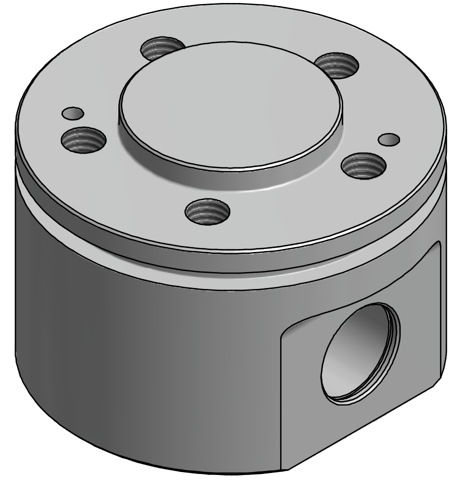
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

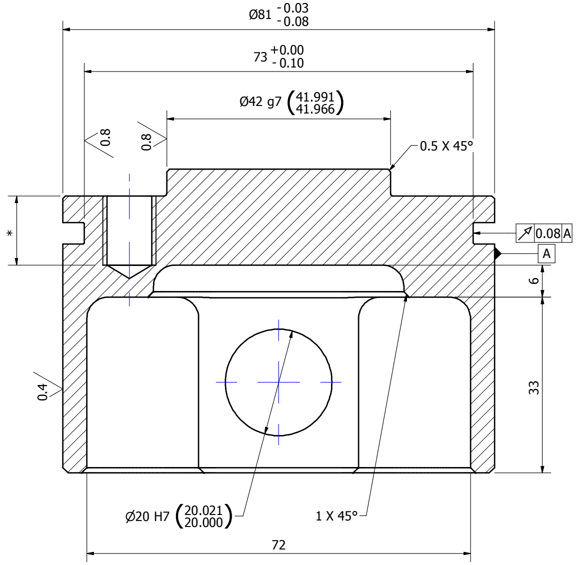
NOTE:

1. FOR DIMENSIONS MARKED "*" REFER HELICOIL MANUAL FOR CORRECT DETAIL ON INSERT INSTALLATION.



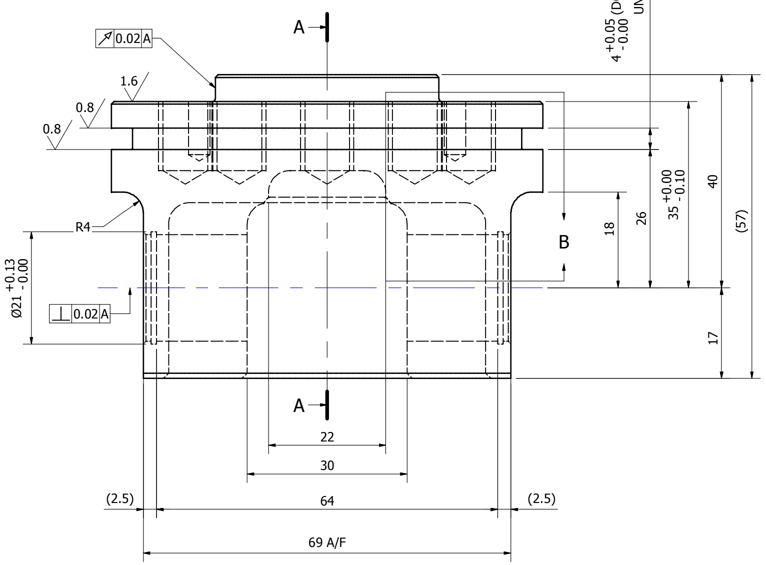
- TOP VIEW -





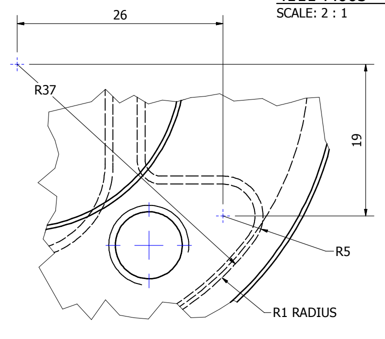
SECTION A-A

SCALE: 2 : 1



4111-M063 - DETAIL -

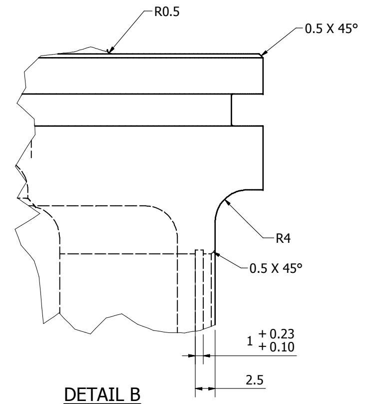
SCALE: 2 : 1



DETAIL C

SCALE: 3 : 1

NOTE: CERTAIN HIDDEN DETAIL OMITTED FOR CLARITY



DETAIL B

SCALE: 3 : 1

NOTE: CERTAIN HIDDEN DETAIL OMITTED FOR CLARITY

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M063	1	AS1866 4032-T6	LOWER PISTON	0.39 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN	KBD	DATE 13/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	DATE 13/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED		DATE	DRAWING DESCRIPTION: LOWER PISTON DETAIL
APPROVED		DATE	JOB No. ENG4111/2 · DRAWING No. 4111-M063 · 1 OF 1 · SIZE A1 · REV. 0

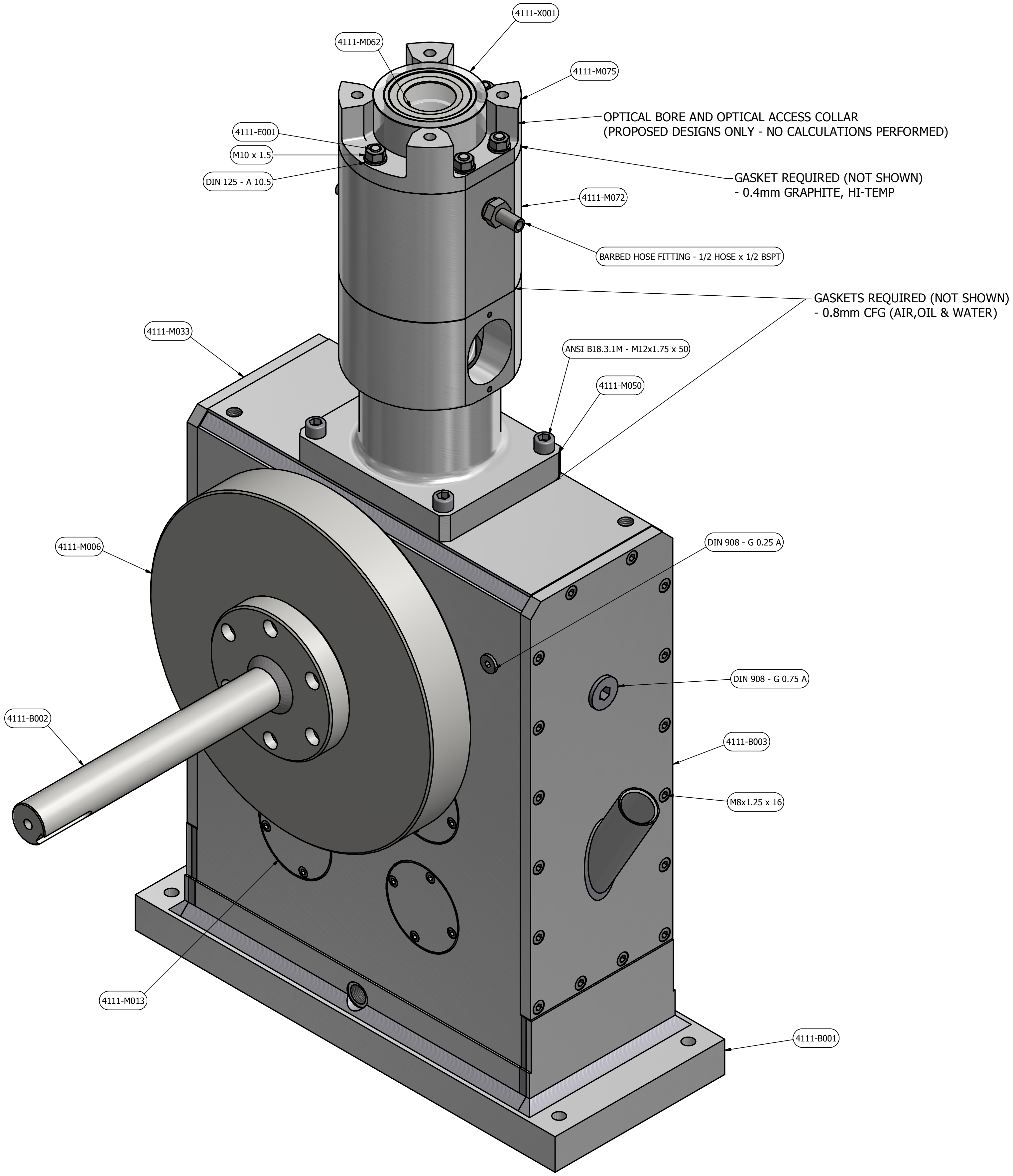
D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M063.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M063.idw

Appendix D: Full Set of Drawings (A3)
(Drawing Transmittal – Revision 0)

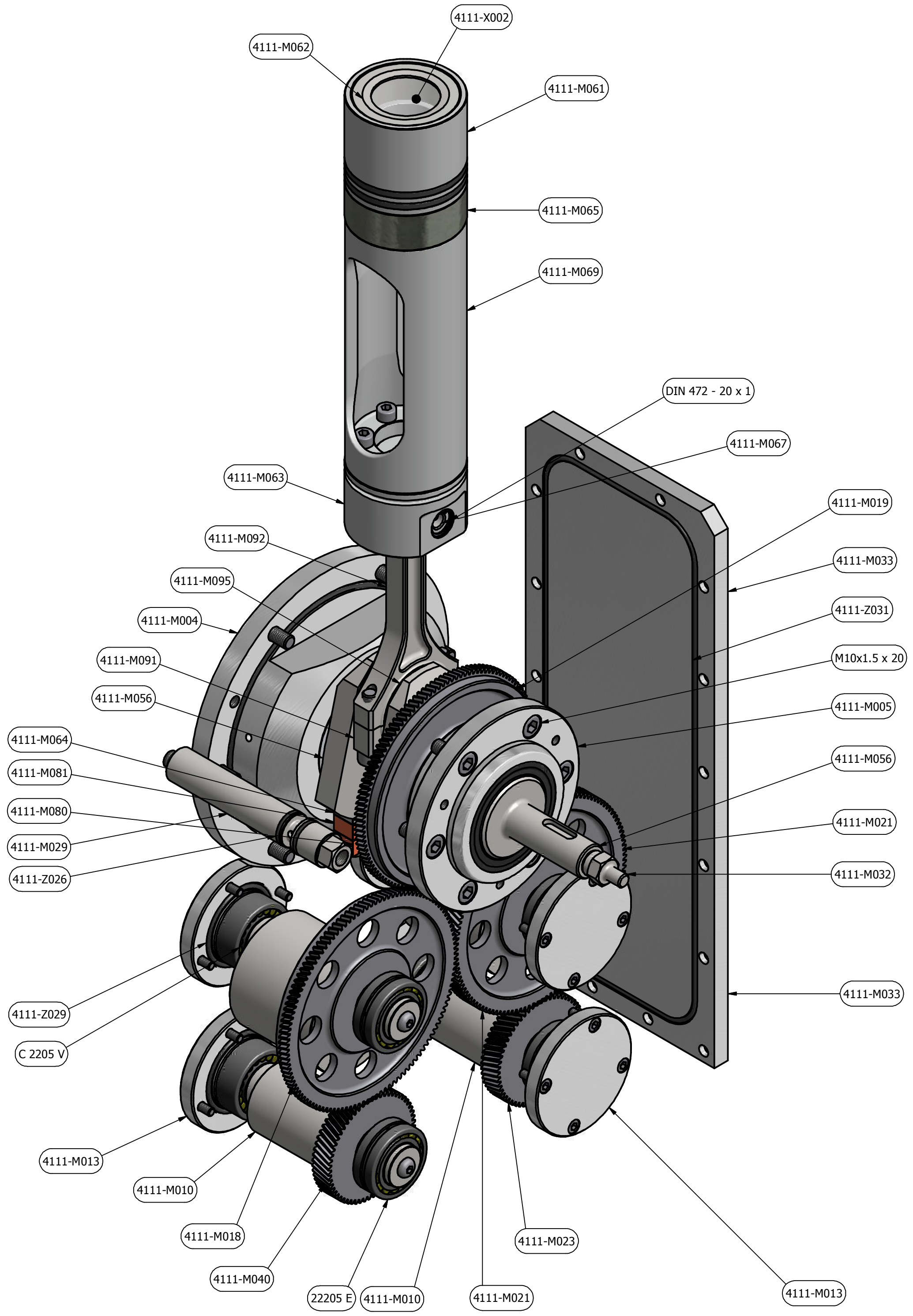
1.	4111 – A001	Main G.A.
2.	4111 – A003	Piston Assembly
3.	4111 – A006	Flywheel Side Main Bearing
4.	4111 – A007	Encoder Side Main Bearing
5.	4111 – A010	Crankshaft G.A.
6.	4111 – A021	Conrod Assembly
7.	4111 – B001	Crankcase Detail
8.	4111 – B002	Drive Attachment
9.	4111 – B003	Side Cover
10.	4111 – E001	Tie Rod
11.	4111 – M004	Bearing Housing Detail
12.	4111 – M005	Bearing Housing Detail
13.	4111 – M006	Flywheel Detail
14.	4111 – M007	Balancer Shaft Detail
15.	4111 – M008	Retaining Washer Detail
16.	4111 – M009	Primary Balance Weight Detail
17.	4111 – M010	Secondary Balance Weight
18.	4111 – M011	Spacer Detail
19.	4111 – M012	Spacer Detail
20.	4111 – M013	Bearing Cap Detail
21.	4111 – M018	Gear Detail
22.	4111 – M019	Gear Detail
23.	4111 – M021	Gear Detail
24.	4111 – M023	Gear Detail
25.	4111 – M025	Bearing Cap Detail
26.	4111 – M026	Main Bearing Detail
27.	4111 – M027	Thrust Bearing Detail
28.	4111 – M028	Thrust Collar Detail
29.	4111 – M029	Oil Supply Tube Detail
30.	4111 – M032	Encoder Shaft Detail
31.	4111 – M033	Side Cover Detail
32.	4111 – M034	Side Cover Detail
33.	4111 – M040	Gear Detail
34.	4111 – M050	Lower Barrel Detail
35.	4111 – M051	Lower Cylinder Sleeve Detail
36.	4111 – M056	Crankshaft Detail
37.	4111 – M059	Little End Bearing Detail
38.	4111 – M061	Piston Head Detail
39.	4111 – M062	Optical Window Collar Detail
40.	4111 – M063	Lower piston Detail
41.	4111 – M064	Balance Weight Detail
42.	4111 – M065	Sliding Bush Detail

43.	4111 – M066	Low Friction Rings Detail
44.	4111 – M067	Wrist Pin Detail
45.	4111 – M069	Piston Extension Detail
46.	4111 – M072	Upper Barrel Detail
47.	4111 – M075	Optical Access Collar
48.	4111 – M078	Upper Cylinder Sleeve Detail
49.	4111 – M080	Balance Weight Detail
50.	4111 – M081	Balance Weight Detail
51.	4111 – M091	Conrod Bearing Cap Detail
52.	4111 – M092	Conrod Detail
53.	4111 – M095	Big End Bearing Detail
54.	4111 – S001	Oil Cover Detail
55.	4111 – X001	Optical Bore Detail
56.	4111 – X002	Optical Access Window Detail

DO NOT SCALE - IF IN DOUBT ASK.



4111-A001 - L.H.S. ISOMETRIC VIEW -



4111-A001 - R.H.S. ISOMETRIC VIEW -

NOTE: CRANKCASE,SIDECOVER & MISC REMOVED FOR INTERNAL VIEWING

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	29/04/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

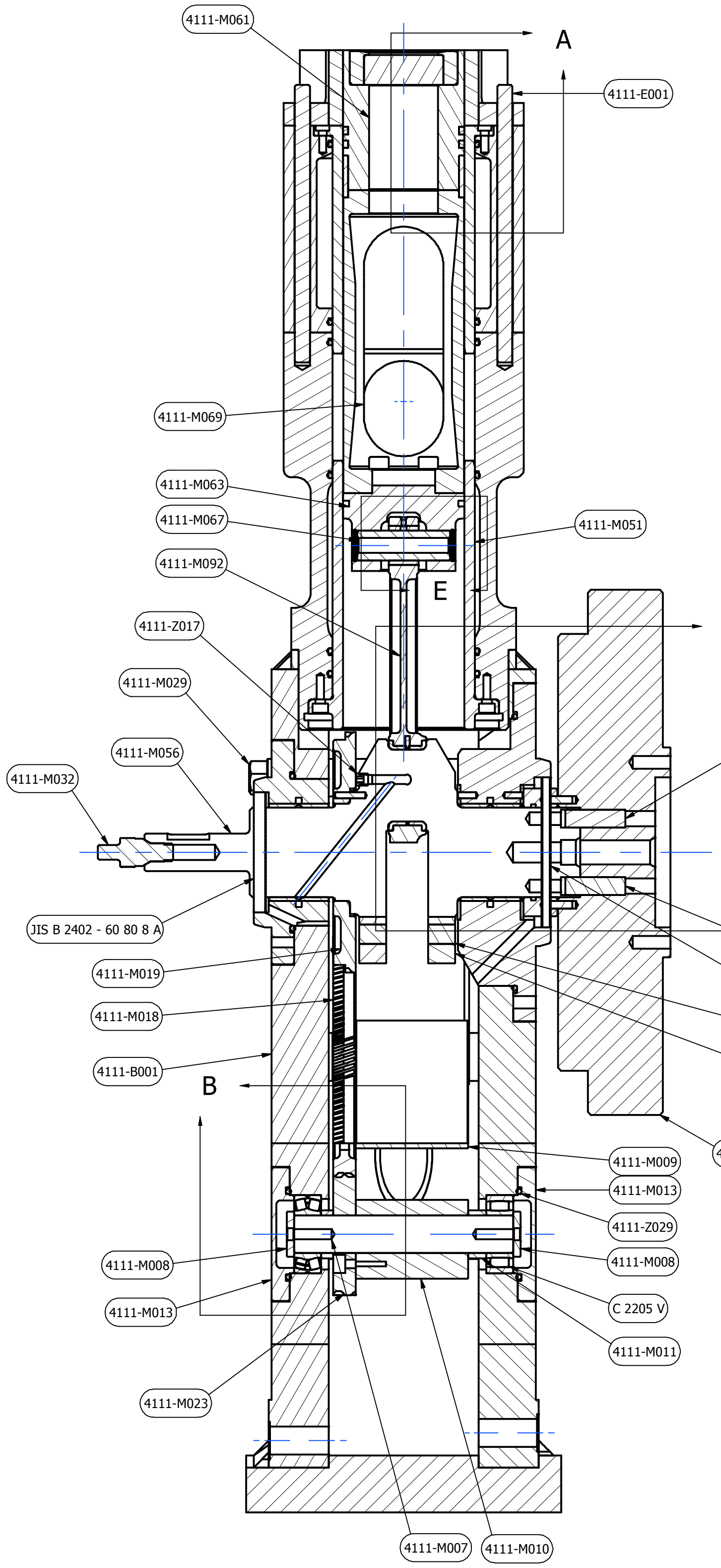
SCALE	AS SHOWN
DRAWN	KBD — DATE 29/04/2014
DESIGNED	KBD — DATE 29/04/2014
CHECKED	DATE
APPROVED	DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

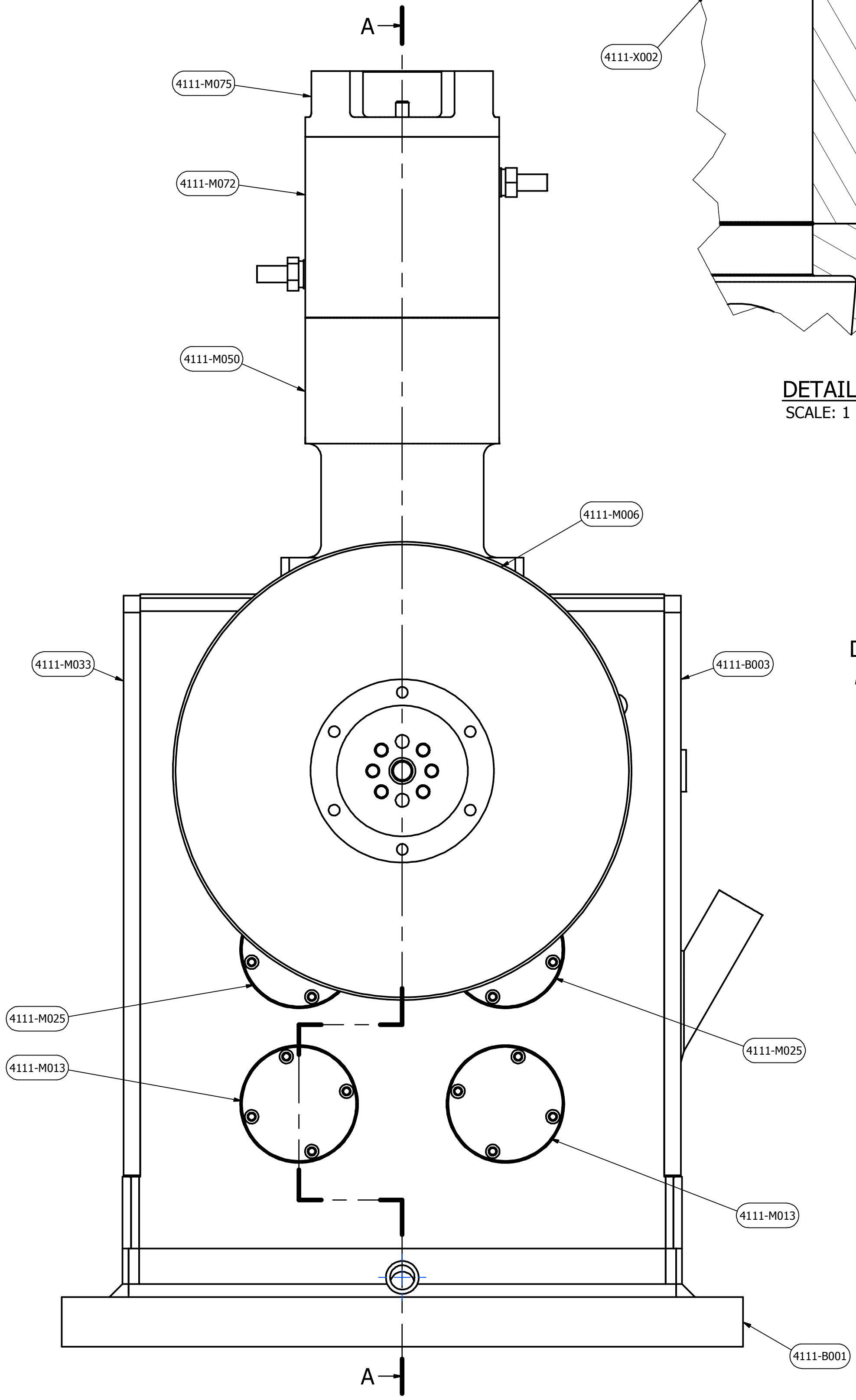
CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION	GENERAL ASSEMBLY & PRESENTATION
JOB No.	ENG4111/2
DRAWING No.	4111-A001
SHEET	1 OF 3
SIZE	A1
REV.	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Assemblies\4111-A001.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-A001.idw

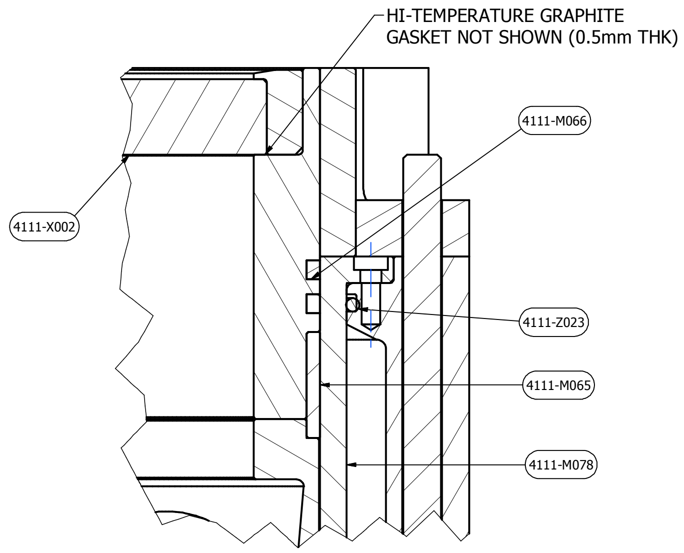
DO NOT SCALE - IF IN DOUBT ASK.



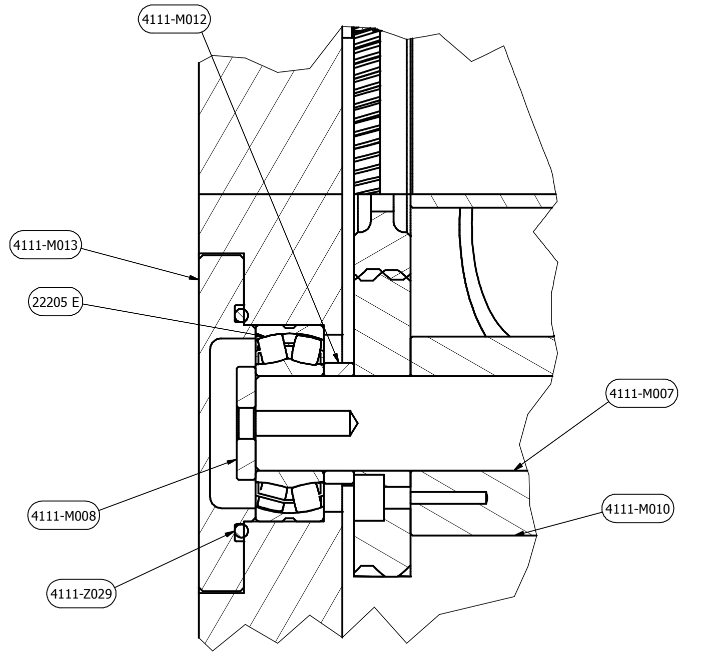
SECTION A-A
SCALE: 1 : 2.5



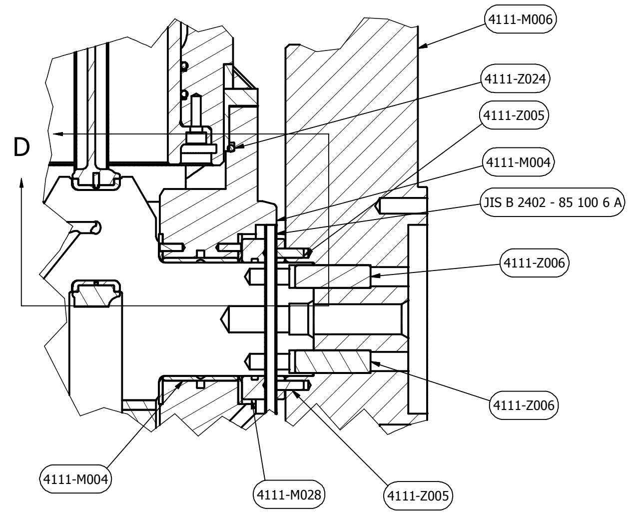
4111-A001 - FLYWHEEL SIDE VIEW -
SCALE: 1 : 2.5
NOTE: ALL FASTENERS SUPPRESSED



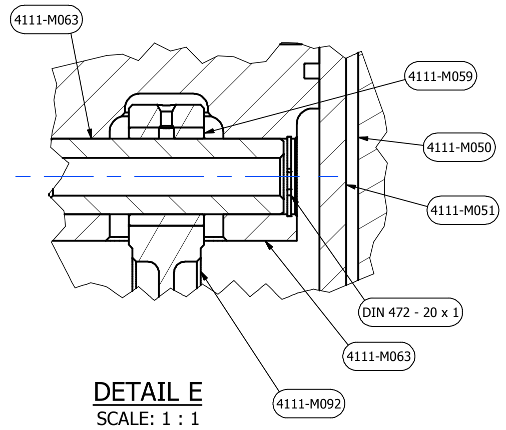
DETAIL A
SCALE: 1 : 1



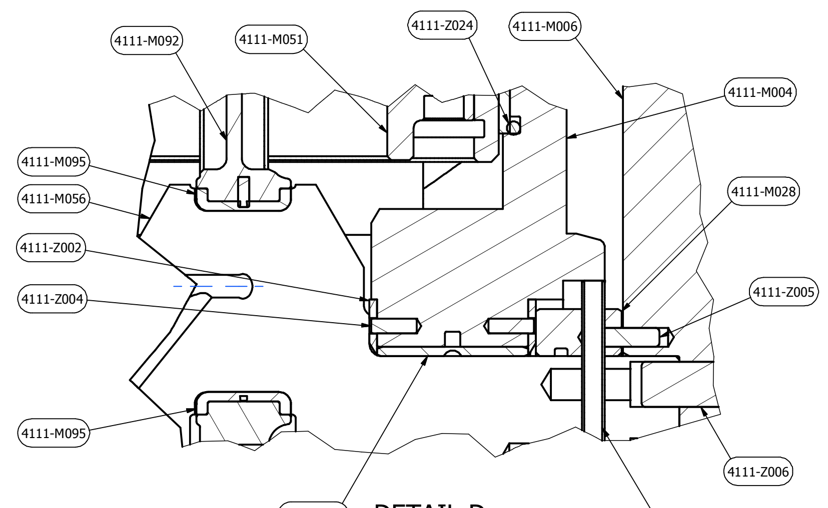
DETAIL B
SCALE: 1 : 1



DETAIL C
SCALE: 1 : 2



DETAIL E
SCALE: 1 : 1



DETAIL D
SCALE: 1 : 1

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN
DRAWN	KBD
DATE	29/04/2014
DESIGNED	KBD
DATE	29/04/2014
CHECKED	
DATE	
APPROVED	
DATE	

THE UNIVERSITY OF SOUTHERN QUEENSLAND

CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION	GENERAL ASSEMBLY & PRESENTATION
JOB No.	ENG4111/2
DRAWING No.	4111-A001
SHEET	2 OF 3
SIZE	A1
REV.	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Assemblies\4111-A001.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-A001.idw

DO NOT SCALE - IF IN DOUBT ASK.

PARTS LIST (PARTS PER ONE MAIN ASSEMBLY ONLY)

PART NUMBER	MATERIAL	DESCRIPTION	MASS	QTY	REFERENCE
22205 E	N/A	SRB, series CC/W33-Standard spherical roller bearings	0.05 kg	4	VENDOR SUPPLY
4111-B001	Aluminum 6061, Welded	MAIN CRANKCASE	52.72 kg	1	4111-B001
4111-B002	Steel, Mild, Welded	DRIVE ATTACHMENT	5.63 kg	1	4111-B002
4111-B003	Aluminum 6061, Welded	SIDE COVER (OIL FILL)	2.7 kg	1	4111-B003
4111-E001	AS1444 -4140	TIE ROD	0.04 kg	6	4111-E001
4111-M004	AS1865 -2011T6	FLYWHEEL SIDE MAIN BEARING HOUSING	3.41 kg	1	4111-M004
4111-M005	AS1865 -2011T6	ENCODER SIDE MAIN BEARING HOUSING	0.91 kg	1	4111-M005
4111-M006	AS3678 -GR250	FLYWHEEL	47.45 kg	1	4111-M006
4111-M007	AS1444 -4140	COUNTER BALANCE SHAFT	0.55 kg	4	4111-M007
4111-M008	AS1444 -1020	COUNTER BALANCE SHAFT WASHER	0.03 kg	8	4111-M008
4111-M009	AS1444 -1020	PRIMARY BALANCE WEIGHT	3.08 kg	2	4111-M009
4111-M010	AS1444 -1020	SECONDARY BALANCE WEIGHT	1 kg	2	4111-M010
4111-M011	AS1444 -1020	COUNTER BALANCE SHAFT SPACER LONG	0.03 kg	4	4111-M011
4111-M012	AS1444 -1020	COUNTER BALANCE SHAFT SPACER SHORT	0.02 kg	4	4111-M012
4111-M013	AS1865 -2011T6	BEARING CAP	0.16 kg	6	4111-M013
4111-M018	AS1444 EN36A	PRIMARY COUNTER BALANCE SHAFT GEAR	1.3 kg	1	4111-M018
4111-M019	AS1444 EN36A	CRANKSHAFT DRIVE PINION	1.38 kg	1	4111-M019
4111-M021	AS1444 EN36A	PRIMARY COUNTER BALANCE SHAFT GEAR	1.42 kg	1	4111-M021
4111-M023	AS1444 EN36A	SECONDARY COUNTER BALANCE SHAFT GEAR	0.5 kg	1	4111-M023
4111-M025	6061-T6	BEARING CAP (PRIMARY, FLYWHEEL SIDE)	0.15 kg	2	4111-M025
4111-M026	DU BEARING	6040DU - LEAD/PTFE BEARING (MODIFIED)	0.05 kg	2	4111-M026
4111-M028	AS1444 -1020	THRUST COLLAR	0.49 kg	1	4111-M028
4111-M029	AS1444 -1020	OIL SUPPLY FITTING	0.48 kg	1	4111-M029
4111-M032	AS1444 -4140	ENCODER ATTACHMENT	0.06 kg	1	4111-M032
4111-M033	AS1734 6061T6	CRANKCASE SIDE COVER	2.57 kg	1	4111-M033
4111-M040	AS1444 EN36A	SECONDARY COUNTER BALANCE SHAFT GEAR	0.5 kg	1	4111-M040
4111-M050	AS1866 6061T6	LOWER BARREL	6.47 kg	1	4111-M050
4111-M051	2P Cast Iron	LOWER CYLINDER SLEEVE	2.81 kg	1	4111-M051
4111-M056	AS1444 -4340	ENGINE CRANKSHAFT	6.45 kg	1	4111-M056
4111-M059	954 - AL. BRONZE	LITTLE END BEARING	0.03 kg	1	4111- M059
4111-M061	AS1866 4032-T6	PISTON HEAD	0.6 kg	1	4111-M061
4111-M062	SS-304 ROUND	OPTICAL WINDOW COLLAR	0.33 kg	1	4111-M062
4111-M063	AS1866 4032-T6	LOWER PISTON	0.39 kg	1	4111-M063
4111-M064	AS3678 -GR250	CRANK BALANCE WEIGHT	0.04 kg	2	4111-M064
4111-M065	Tetron B	SLIDING BUSH	0.09 kg	1	4111-M065
4111-M066	Tetron C	LOW FRICTION PISTON RINGS	0.01 kg	2	4111-M066
4111-M067	AS1444 -4340	WRIST PIN	0.11 kg	1	4111-M067
4111-M069	AS1866 4032-T6	PISTON EXTENSION	0.67 kg	1	4111-M069
4111-M072	AS1865 -2011T6	UPPER BARREL	3.61 kg	1	4111-M072
4111-M075	AS1865 -2011T6	OPTICAL ACCESS COLLAR	0.77 kg	1	4111-M075
4111-M076	AS1444 -4140	HEAD BOLT FERRULE	0.01 kg	4	4111-M076
4111-M078	2P Cast Iron	UPPER CYLINDER SLEEVE	2.29 kg	1	4111-M078
4111-M080	SD170	CRANK BALANCE WEIGHT	0.22 kg	2	4111-M080
4111-M081	SD170	CRANK BALANCE WEIGHT	0.22 kg	2	4111-M081
4111-M091	AS1444 -4340	CONROD BIG END SADDLE	0.19 kg	1	4111-M091
4111-M092	AS1444 -4340	CONROD	0.48 kg	1	4111-M092
4111-M095	954 - AL. BRONZE	BIG END BEARING SHELL	0.05 kg	2	4111-M095
4111-X001	FUSED SILICA (QUARTZ)	OPTICAL BORE	0.3 kg	1	4111-X001
4111-X002	FUSED SILICA (QUARTZ)	OPTICAL ACCESS WINDOW	0.1 kg	1	4111-X002
4111-Z002	DU BEARING	WC60DU - LEAD/PTFE THRUST BEARING (MODIFIED)	0.04 kg	2	4111-Z002
4111-Z004	MILD STEEL	4x12 MACHINE DOWEL	0 kg	4	VENDOR SUPPLY
4111-Z005	MILD STEEL	5 x 20 MACHINE DOWEL	0 kg	4	VENDOR SUPPLY
4111-Z006	MILD STEEL	12 x 20 MACHINE DOWEL	0.04 kg	2	VENDOR SUPPLY
4111-Z010	MILD STEEL	1/4" BSPT PRESSURE PLUG	0.01 kg	2	VENDOR SUPPLY
4111-Z015	MILD STEEL	2.5 x 12 MACHINE DOWEL	0 kg	2	VENDOR SUPPLY
4111-Z017	MILD STEEL	1/16" BSPT PRESSURE PLUG	0 kg	1	VENDOR SUPPLY
4111-Z022	MILD STEEL	4 x 20 MACHINE DOWEL	0 kg	5	VENDOR SUPPLY
4111-Z023	Rubber	#-240 (3-3/4" x 1/8") -N7 NITRILE O-RING	0 kg	6	VENDOR SUPPLY

PARTS LIST (PARTS PER ONE MAIN ASSEMBLY ONLY)

PART NUMBER	MATERIAL	DESCRIPTION	MASS	QT	REFERENCE
4111-Z024	Rubber	#-241 (3-7/8" x 1/8") -N7 NITRILE O-RING	0 kg	1	VENDOR SUPPLY
4111-Z025	Rubber	#-017 (11/16" x 1/16) -N7 NITRILE O-RING	0 kg	1	VENDOR SUPPLY
4111-Z026	Rubber	#-117 (13/16" x 3/32) -N7 NITRILE O-RING	0 kg	2	VENDOR SUPPLY
4111-Z028	Rubber	#-241 (3-7/8" x 1/8") -N7 NITRILE O-RING	0 kg	1	VENDOR SUPPLY
4111-Z029	Rubber	#-227 (2-1/8 x 1/8") -N7 NITRILE O-RING	0 kg	8	VENDOR SUPPLY
4111-Z031	Rubber	(1/8") -N7 NITRILE O-RING (CUSTOM)	0 kg	2	VENDOR SUPPLY
4111-Z032	MILD STEEL	4 x 20 MACHINE DOWEL	0 kg	1	VENDOR SUPPLY
4111-Z033	MILD STEEL	4 x 35 MACHINE DOWEL	0 kg	4	VENDOR SUPPLY
ANSI B18.3.1M - M6x1 x 25	Steel, Mild	Socket Head Cap Screw - Metric	0 kg	5	VENDOR SUPPLY
ANSI B18.3.1M - M6x1 x 70	Steel, Mild	Socket Head Cap Screw - Metric	0 kg	5	VENDOR SUPPLY
ANSI B18.3.4M - M10 x 1.5 x 40	Steel, Mild	Socket Button Head Cap Screw - Metric	0 kg	4	VENDOR SUPPLY
ANSI B18.3.4M - M8 x 1.25 x 16	Steel, Mild	Socket Button Head Cap Screw - Metric	0 kg	8	VENDOR SUPPLY
BARBED HOSE FITTING - 1/2 HOSE x 1/2 BSPT	Stainless Steel, Austenitic	HOSE FITTING (COOLANT)	0.1 kg	2	VENDOR SUPPLY
C 2205 V	N/A	CARB, cylindrical bore-Toroidal roller bearings	0 kg	4	VENDOR SUPPLY
DIN 125 - A 10.5	Steel, Mild	Flat Washer	0 kg	6	VENDOR SUPPLY
DIN 472 - 20 x 1	Steel, Mild	Spring Retaining Ring	0 kg	2	VENDOR SUPPLY
DIN 6912 - M5 x 12	Steel, Mild	METRIC LOW HEAD CAP SCREW	0 kg	10	VENDOR SUPPLY
DIN 908 - G 0.25 A	Steel, Mild	METRIC LOW HEAD CAP SCREW	0 kg	1	VENDOR SUPPLY
DIN 908 - G 0.5 A	Steel, Mild	METRIC LOW HEAD CAP SCREW	0 kg	1	VENDOR SUPPLY
DIN 908 - G 0.75 A	Steel, Mild	METRIC LOW HEAD CAP SCREW	0.1 kg	1	VENDOR SUPPLY
JIS B 2402 - 60 80 8 A	Rubber	Oil seals - Spring loaded, metal cased, with dust-lp DM	0 kg	1	VENDOR SUPPLY
JIS B 2402 - 85 100 6 A	Rubber	Oil seals - Spring loaded, non-spring, rubber covered G	0 kg	1	VENDOR SUPPLY
M10 x 1.5	Steel, Mild	Hex Nut	0 kg	6	VENDOR SUPPLY
M10x1.5 x 20	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	12	VENDOR SUPPLY
M10x1.5 x 30	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	6	VENDOR SUPPLY
M10x1.5 x 45	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	2	VENDOR SUPPLY
M10x1.5 x 70	Steel, Mild	Broached Socket Head Cap Screw - Metric	0.1 kg	6	VENDOR SUPPLY
M6x1 x 16	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	32	VENDOR SUPPLY
M6x1 x 20	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	6	VENDOR SUPPLY
M8x1.25 x 16	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	36	VENDOR SUPPLY
ANSI B18.3.1M - M12x1.75 x 50	Steel, Mild	Broached Socket Head Cap Screw - Metric	0.1 kg	4	
ANSI B18.3.1M - M8x1.25 x 30	Steel, Mild	Broached Socket Head Cap Screw - Metric	0 kg	5	

NOTES:

GENERAL

1. ALL SHARP EDGES TO BE REMOVED
2. ALL TIGHT CORNERS / RADII TO BE IDENTIFIED AND ADDRESSED.
3. ALL PARTS TO FIT TOGETHER WITHOUT FORCING EXCEPT WHERE TOLERANCED SO.
4. ALL FASTENERS TO BE TORQUED TO THE SETTINGS PROVIDED IN THE APPROPRIATE TABLES
5. ALL FASTENERS TO BE LUBRICATED USING ANTI-SEIZE LUBRICANT BEFORE ASSEMBLY EXCEPT WHERE NOTED TO USE A THREAD LOCKING COMPOUND
6. ALL MATING PARTS TO BE LIGHTLY OILED BEFORE ASSEMBLY
7. ALL GASKETS TO BE ASSEMBLED DRY
8. ALL O-RINGS TO BE LUBRICATED USING SILCONE GREASE BEFORE ASSEMBLY
9. ENGINE TO BE FILED WITH SAE 30 ENGINE OIL AS MINIMUM, SAE 40 ALSO APPROPRIATE.
10. ALL PARTS TO BE ASSEMBLED FREE OF DIRT AND UNWANTED CONTAMINANTS

SURFACE TREAT

- HEAT SURFACE (DO NOT SOAK!) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION. HERE-IN KNOWN AS "BLUED"
- ALL EXPOSED FERROUS PARTS TO BE LIGHTLY OILED TO PREVENT SURFACE CORROSION

HEAT TREATMENT

4340:
NORMALIZE: 850-880°C, FOR 1hr MIN, AIR COOL.
HARDEN: 830-860°C , OIL QUENCH
TEMPER: 540-680°C HOLD FOR 1 hr MIN, AT TEMPERATURE , AIR COOL. (REFER BOHLER TEMPERATURE CHART) HEREIN KNOWN AS "T" CONDITION
NITIRIDE: NITRIDE DEPTH TO 0.5mm UNLESS NOTED OTHERWISE

EN36A:
NORMALIZE: 850-880°C, FOR 1hr MIN, AIR COOL.
ANNEAL: 650°-700°C , COOL SLOWLY IN CONTROLLED FURNACE
CASE HARDEN: HERE-IN KNOWN AS "CH" CONDITION, MINIMUM DEPTH 0.5mm, PROCESS TO BE DETERMINED BY HEAT TREATMENT PROVIDER.

CONROD

1. NORMALIZE AS SPECIFIED IN HEAT TREATMENT BEFORE MACHINING.
2. ROUGH MACHINE TO 3mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
3. DIMENSIONS SHOWN THUS: TO BE FINISH MACHINED WITH CAP ASSEMBLED TO CONNECTING ROD.
4. FIXINGS (BOLTS) TO BE LIGHTLY OILED & TORQUED TO 55Nm FOR MACHINING THE BIG END BEARING.
5. CONROD ASSEMBLIES TO BE BALANCED TO MASS & C.O.G SHOWN IN 4111-A021 +/- 0.03kg, +/- 0.05mm
6. CONROD AND SHELL TO BE "BLUED" AS NOTED IN SURFACE TREATMENT

CRANKSHAFT

1. NORMALIZE AS SPECIFIED IN HEAT TREATMENT BEFORE MACHINING.
2. ROUGH MACHINE TO 3mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
3. FINAL CRANKSHAFT ASSSEMBLY TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A010 +/- 0.03kg, +/- 0.05mm
4. CRANKSHAFT TO BE NITRIDED AFTER FINAL MACHINING.
5. CRANKSHAFT TO BE "BLUED" AS NOTED IN SURFACE TREATMENT

TIGHTENING TORQUE

FASTENERS ENGAGED INTO THREAD INSERTS ARE TO BE TORQUED TO THE FOLLOWING SETTINGS:
- M6 =(3.5 N.m.) - M8 =(8.5 N.m.)
- M10 =(17 N.m.) - M12=(40 N.m.)

ALL REMAINING FASTENERS TO BE TORQUED TO THE FOLLOWING SETTINGS UNLESS NOTED OTHERWISE:
(DO NOT TORQUE FASTENERS SCREWED DIRECTLY INTO ALLOY)
- M6 =(9.0 N.m.) - M10=(44 N.m.)
- M8 =(22 N.m.) - M12=(77 N.m.)

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE N/A		THE UNIVERSITY OF SOUTHERN QUEENSLAND.	
DRAWN KBD	DATE 29/04/2014	CLIENT	
DESIGNED KBD	DATE 29/04/2014	DESCRIPTION	
CHECKED	DATE	DRAWING DESCRIPTION GENERAL ASSEMBLY & PRESENTATION	
APPROVED	DATE	JOB No.	DRAWING No. 4111-A001 3 OF 3 SIZE A1 REV. 0

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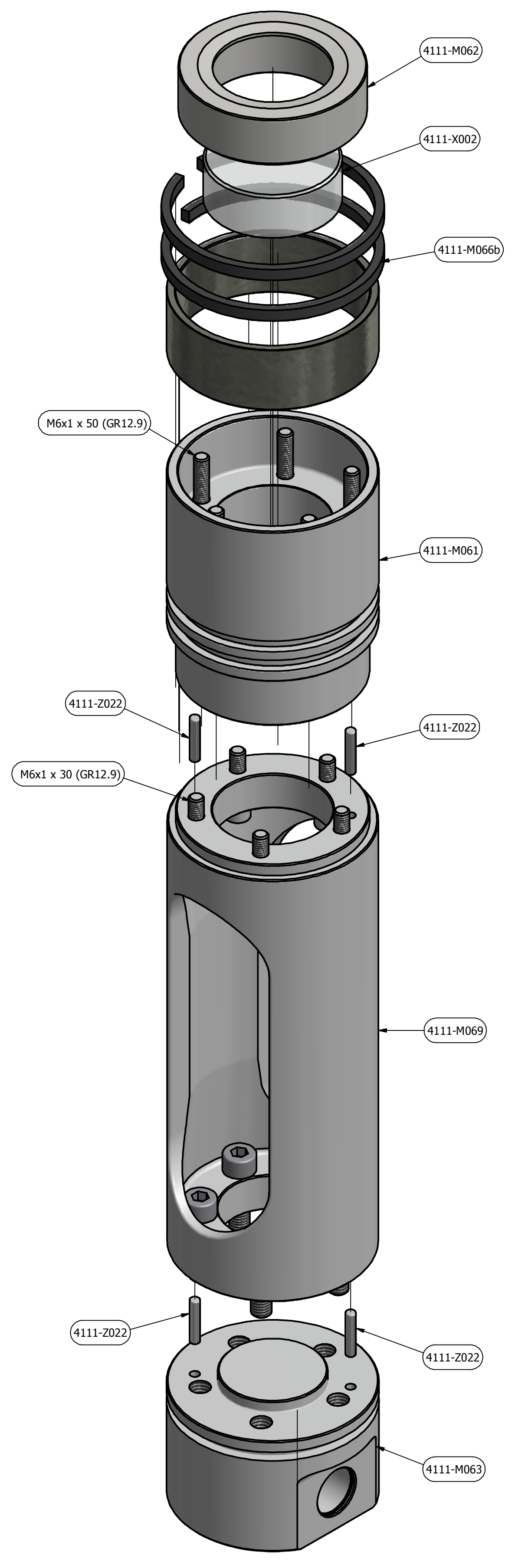
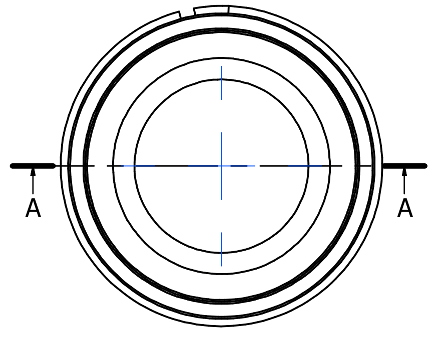
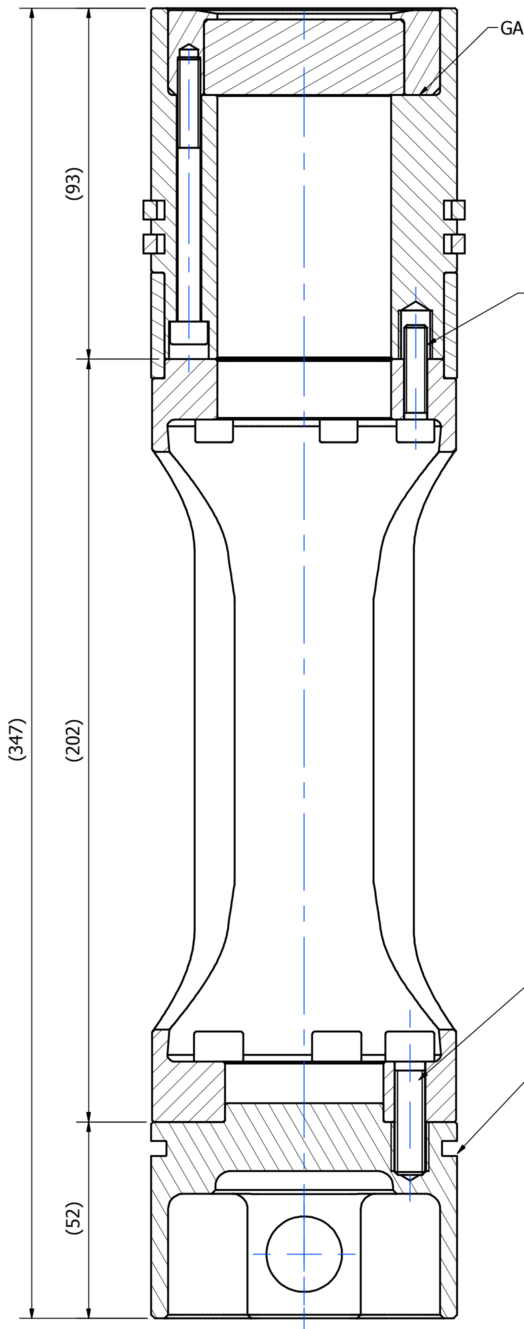
DO NOT SCALE - IF IN DOUBT ASK.



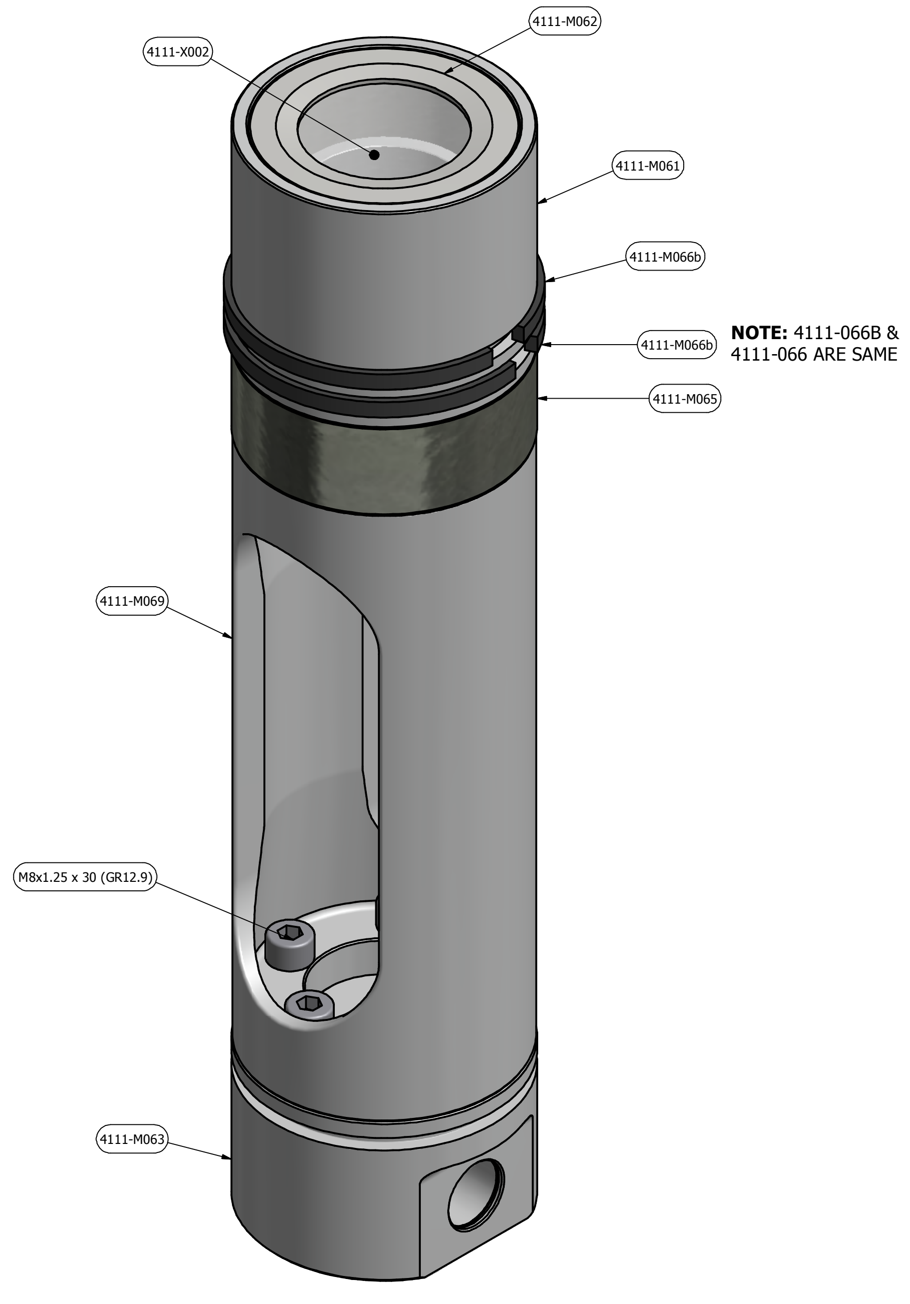
4111-A003 - EXPLODED VIEW -
SCALE: 1 : 1.25



4111-A003 - TOP VIEW -
SCALE: 1 : 1



SECTION A-A
SCALE: 1 : 1



4111-A003 - ISOMETRIC VIEW -
SCALE: 1 : 1

NOTES:

1. FASTENERS ENGAGED INTO THREAD INSERTS TO BE TORQUED TO THE FOLLOWING SETTINGS:
   - M6 =(3.0 N.m.)
   - M8 =(8.0 N.m.)
   - M10 =(10.5 N.m.)
2. ALL FASTENERS MUST BE LUBRICATED BEFORE ASSEMBLY.
3. ALL INSERTS TO BE INSTALLED IN ACCORDANCE WITH 'HELICOIL" PROCEDURES & SPECIFICATIONS.
3. ALL PARTS TO FIT TOGETHER FREELY WITHOUT FORCING
4. MAXIMUM ALLOWABLE RUN-OUT OF ANY PISTON SEGMENT WITH RESPECT TO ANOTHER IS 0.03mm
5. PISTON EXTERNAL SURFACES AND RING GROOVES TO BE FREE OF BURRS & SHARP EDGES.

PART NUMBER	DESCRIPTION	QTY	MASS	REFERENCE
M6x1 x 30 (GR12.9)	Broached Socket Head Cap Screw - Metric	5	0 kg	VENDOR SUPPLY
M6x1 x 50 (GR12.9)	Broached Socket Head Cap Screw - Metric	5	0 kg	VENDOR SUPPLY
4111-M061	PISTON HEAD	1	0.6 kg	4111-M061
4111-M062	OPTICAL WINDOW COLLAR	1	0.3 kg	4111-M062
4111-M063	LOWER PISTON	1	0.4 kg	4111-M063
4111-M065	SLIDING BUSH	1	0.1 kg	4111-M065
4111-M066b	LOW FRICTION PISTON RINGS	2	0 kg	4111-M066
4111-M069	PISTON EXTENSION	1	0.7 kg	4111-M069
4111-X002	OPTICAL ACCESS WINDOW	1	0.1 kg	4111-X002
4111-Z022	4 x 20 MACHINE DOWEL	4	0 kg	VENDOR SUPPLY
M8x1.25 x 30 (GR12.9)	Broached Socket Head Cap Screw - Metric	5	0 kg	VENDOR SUPPLY

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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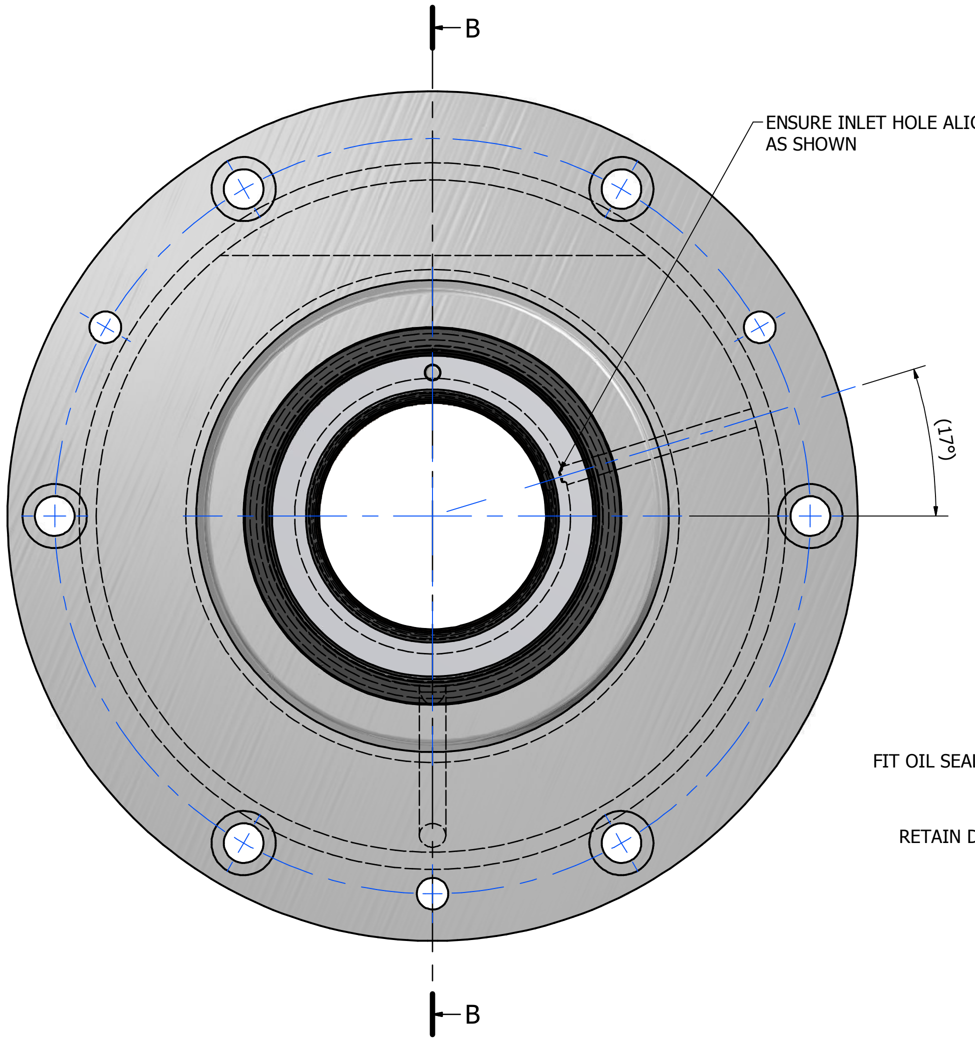
DRAWN	DATE
KBD	10/07/2014
DESIGNED: KBD	DATE: 10/07/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: BOWDITCH PISTON ASSEMBLY

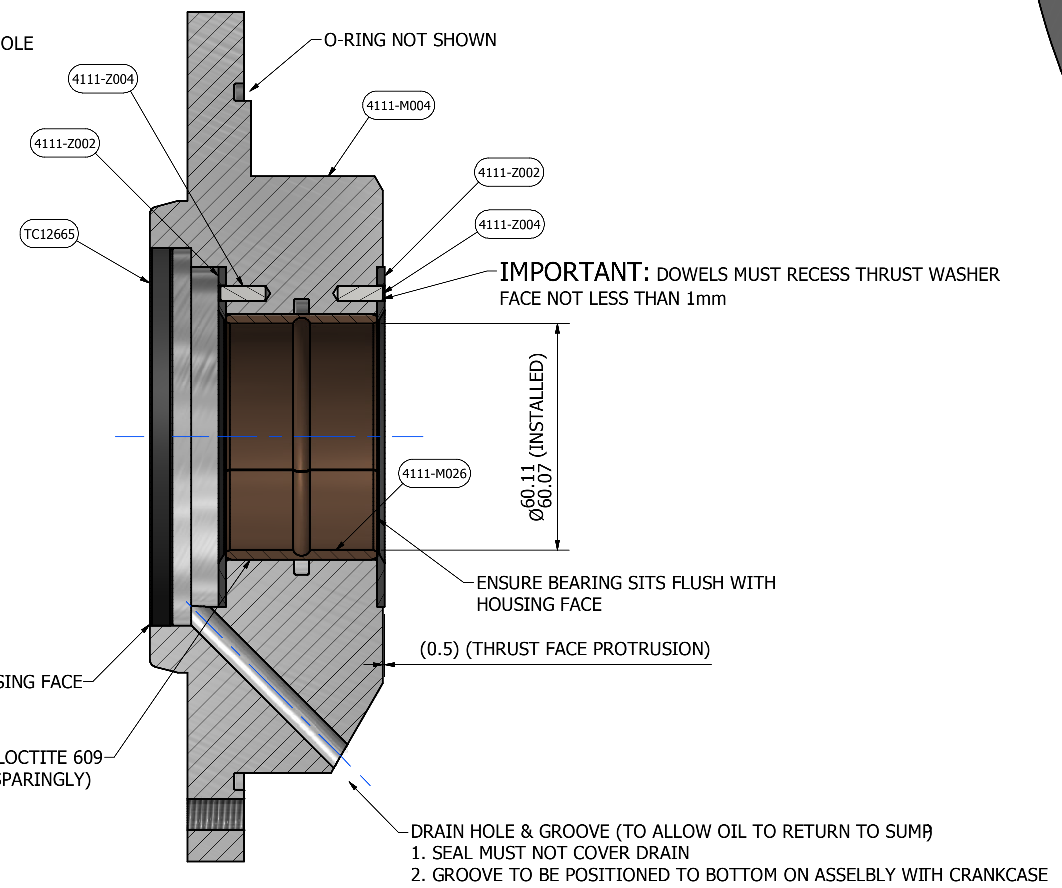
JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-A003	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-A003.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-A003.idw

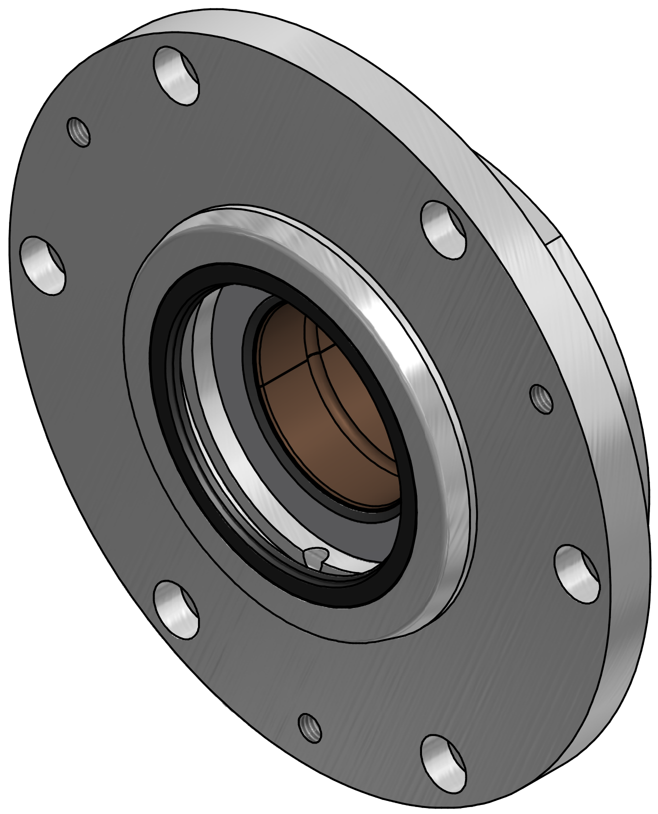
DO NOT SCALE - IF IN DOUBT ASK.



4111-A006 - ASSEMBLY -
SCALE: 1 : 1



SECTION B-B
SCALE: 1 : 1



4111-A006 - ISOMETRIC VIEW -
SCALE: 1 : 1

PARTS LIST

PART NUMBER	DESCRIPTION	QTY	MASS	KEYWORDS
4111-M004	FLYWHEEL SIDE MAIN BEARING HOUSING	1	3.4 kg	4111-M004
4111-M026	6040DU - LEAD/PTFE BEARING (MODIFIED)	1	0.1 kg	4111-M026
4111-Z002	WC60DU - LEAD/PTFE THRUST BEARING (MODIFIED)	2	0 kg	4111-Z002
4111-Z004	4x12 MACHINE DOWEL	2	0 kg	VENDOR SUPPLY
TC12665	100x80x10 Oil seal - Spring loaded, metal cased, with dust-lip DM	1	0 kg	VENDOR SUPPLY

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	10/08/2014	
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West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
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EMAIL: study@usq.edu.au

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DRAWN	KBD	DATE 22/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	DATE 22/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED		DATE	DRAWING DESCRIPTION: FLYWHEEL SIDE MAIN BEARING
APPROVED		DATE	JOB No. ENG4111/2 · DRAWING No. 4111-A006 · 1 OF 1 · SIZE A1 · REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-A006.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-A006.idw

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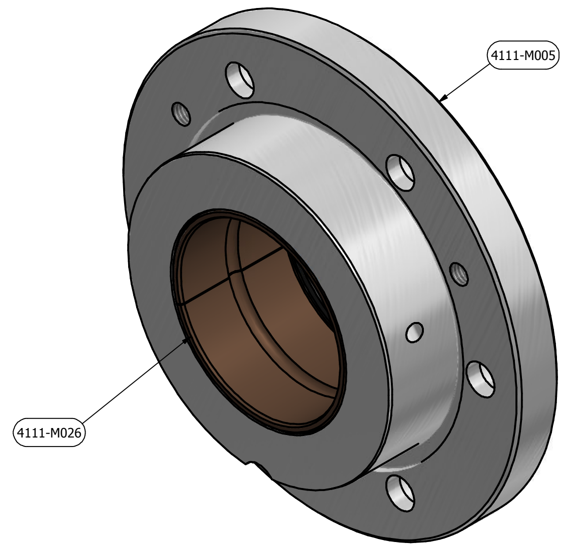
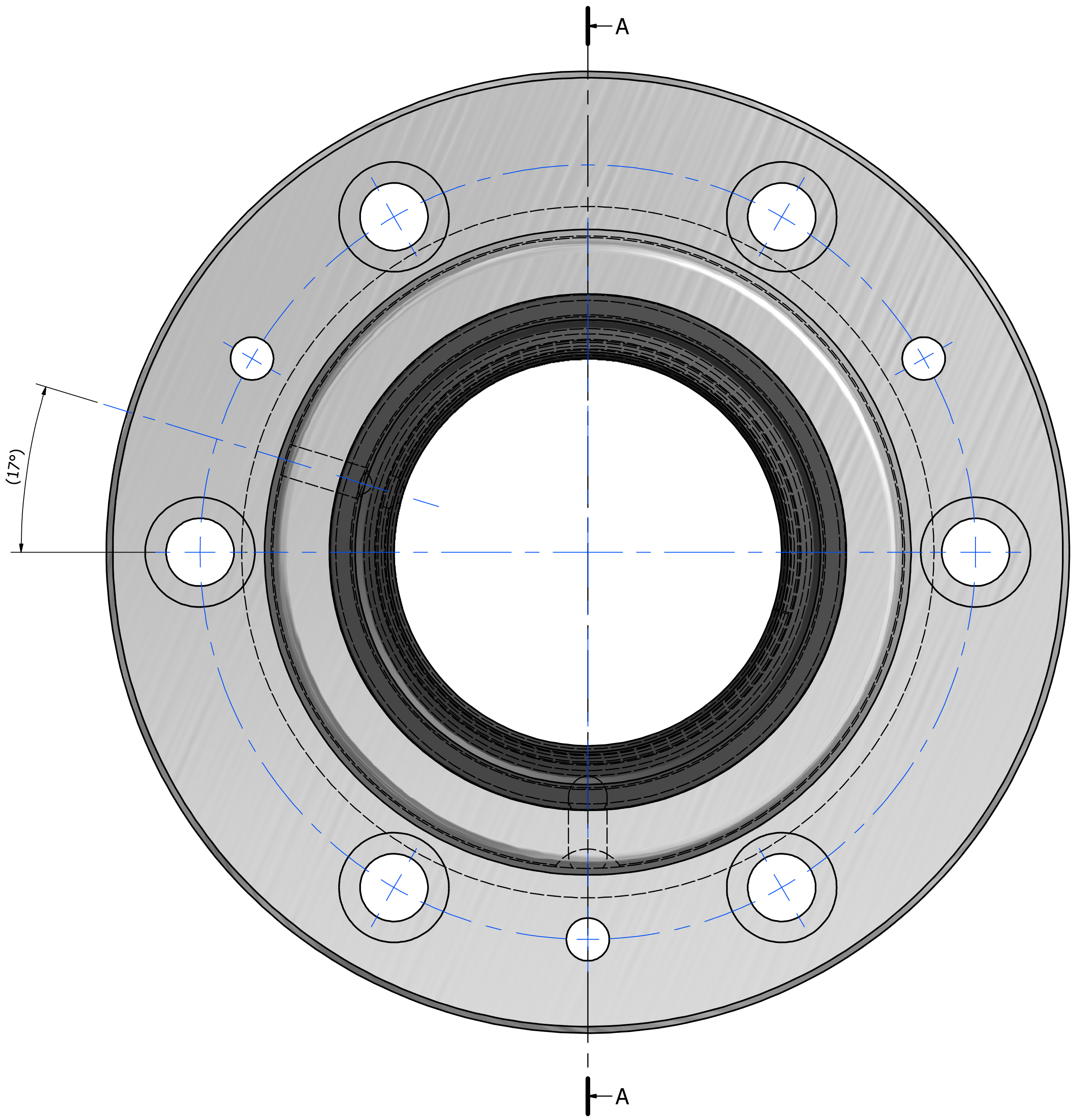
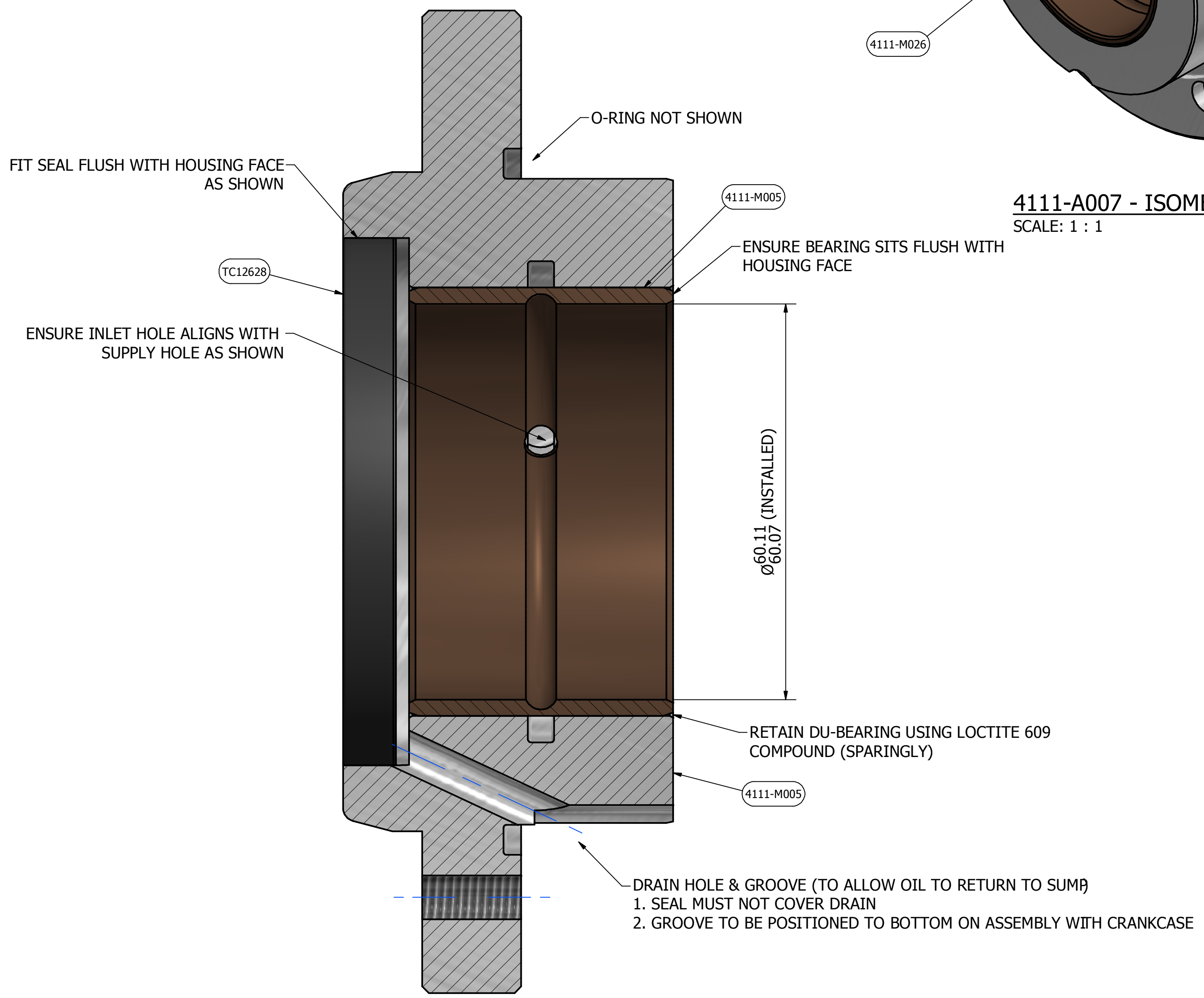


4111-A007 - ISOMETRIC VIEW -
SCALE: 1 : 1



4111-A007 - ASSEMBLY -
SCALE: 2 : 1



SECTION A-A
SCALE: 2 : 1

PARTS LIST

PART NUMBER	DESCRIPTION	QTY	MASS	KEYWORDS
4111-M005	ENCODER SIDE MAIN BEARING HOUSING	1	0.9 kg	4111-M005
4111-M026	6040DU - LEAD/PTFE BEARING (MODIFIED)	1	0.1 kg	4111-M026
TC12628	80x60x8 Oil seal - Spring loaded, metal cased, with dust-lip DM	1	0 kg	VENDOR SUPPLY

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	10/08/2014	
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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DESIGNED	KBD	DATE	22/07/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: ENCODER SIDE MAIN BEARING ASSEMBLY

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-A007	1 OF 1	A1	0

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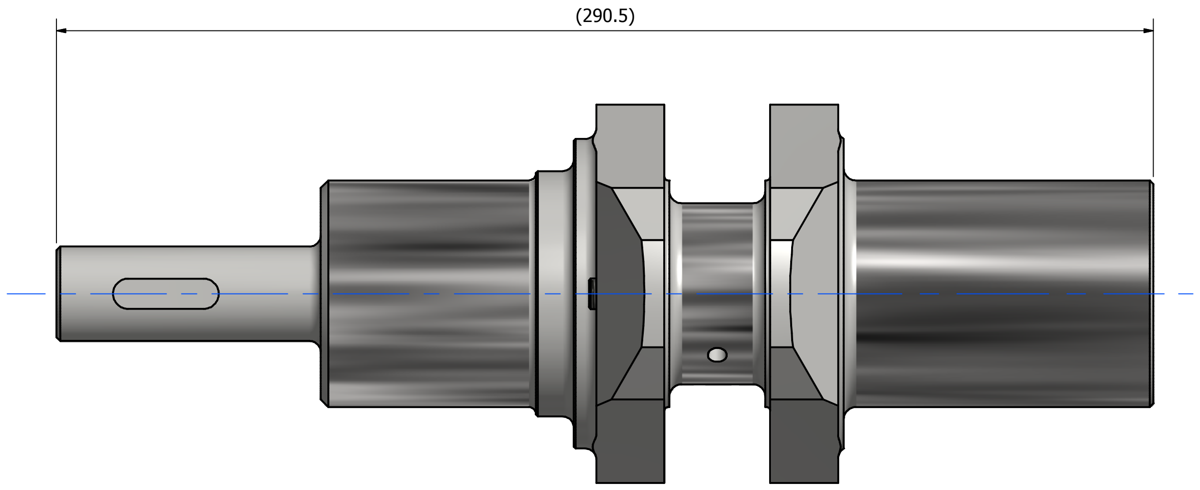
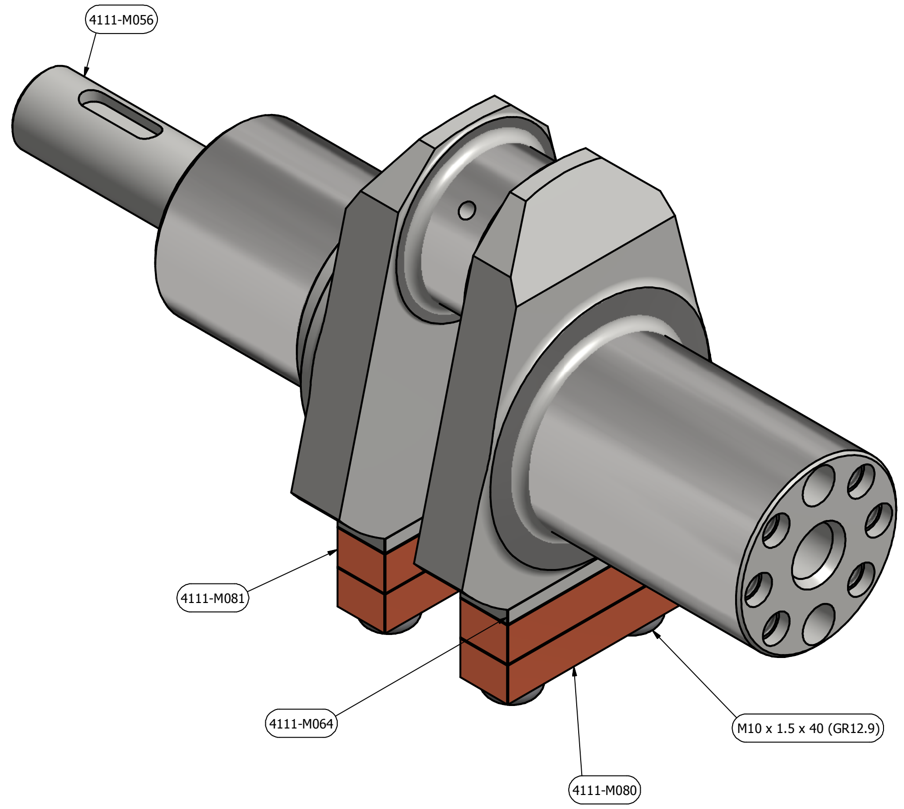
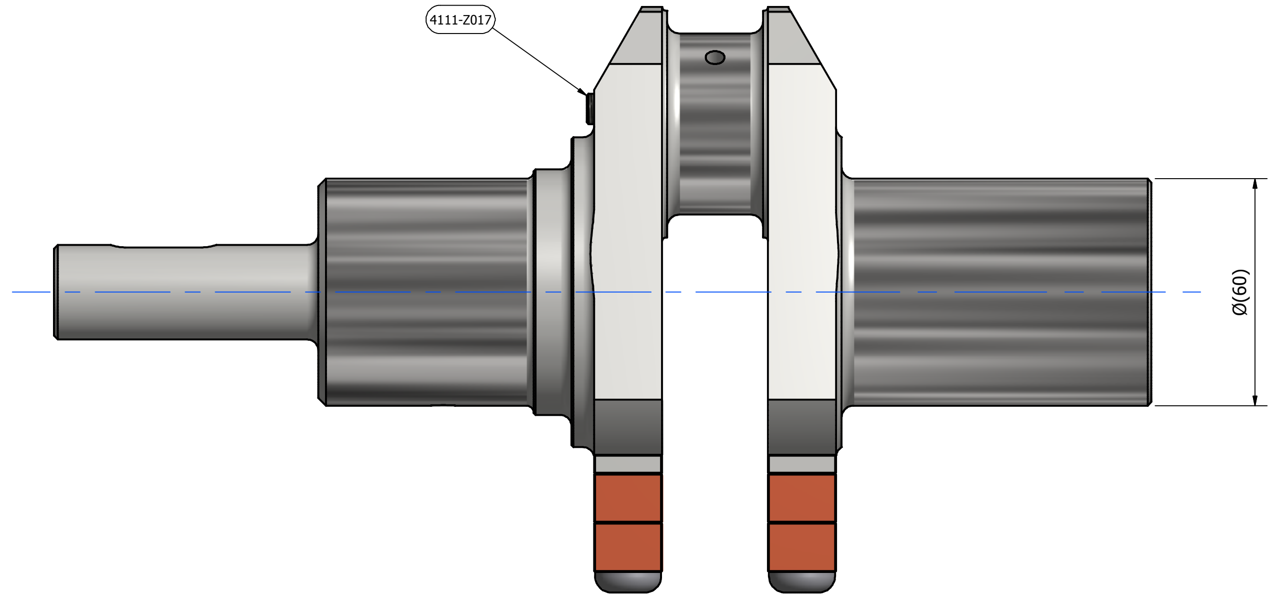


- TOP VIEW -



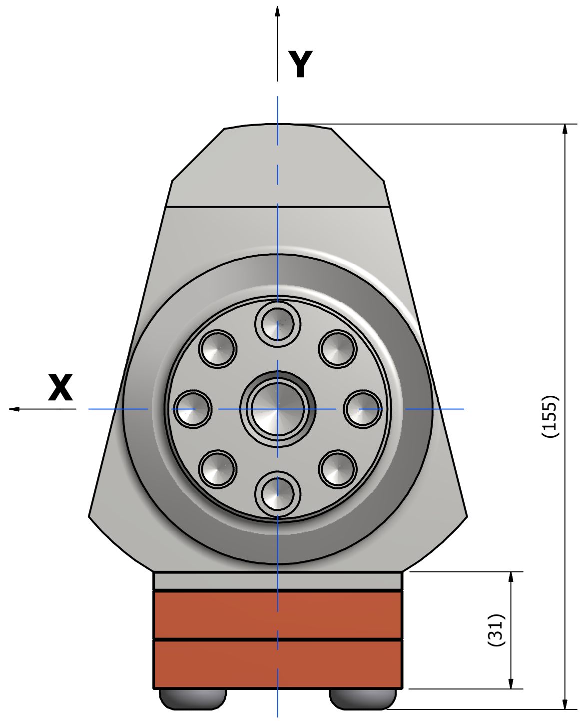
4111-A010 - ISOMETRIC VIEW -

SCALE: 1 : 1



4111-A010 - SIDE VIEW -

SCALE: 1 : 1



- END VIEW -

ASSEMBLY NOTES:

1. ALL PARTS TO FIT TOGETHER WITHOUT FORCING.
2. BALANCING: CRANKSHAFT ASSEMBLY C.O.G. TO BE POSITIONED Y: -3.9mm ±0.05 X: 0mm ±0.05mm
3. ALL SHARP EDGES TO BE REMOVED BEFORE FITTING.
4. ALL FASTENERS TO BE TIGHTENED TO RECOMMENDED FASTENING TORQUE (REFER 4111-001).
5. LOCTITE BALANCE WEIGHT FASTENERS AND OIL PLUG IN POSITION USING "220" THREAD COMPOUND.
6. CHECK BALANCE BY FITTING COLLAR OF EQUIVALENT CONROD LOWER HALF MASS TO BIG END AND STATICALLY BALANCE CRANK

PART NUMBER	MATERIAL	DESCRIPTION	QTY	MASS	KEYWORDS
4111-M056	AS1444 -4340	ENGINE CRANKSHAFT	1	6.448 kg	4111-M056
4111-M064	AS3678 -GR250	CRANK BALANCE WEIGHT	2	0.039 kg	4111-M064
4111-M080	SD170	CRANK BALANCE WEIGHT	2	0.22 kg	4111-M080
4111-M081	SD170	CRANK BALANCE WEIGHT	2	0.219 kg	4111-M081
4111-Z017	MILD STEEL	1/16" BSPT PRESSURE PLUG	1	0.003 kg	VENDOR SUPPLY
4111-Z033	MILD STEEL	4 x 35 MACHINE DOWEL	4	0.003 kg	VENDOR SUPPLY
M10 x 1.5 x 40 (GR12.9)	Steel, Mild	Broached Hexagon Socket Button Head Cap Screw - Metric	4	0.032 kg	VENDOR SUPPLY

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
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West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN: KBD DATE: 3/08/2014
DESIGNED: KBD DATE: 3/08/2014
CHECKED: DATE:
APPROVED: DATE:

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: CRANKSHAFT ASSEMBLY (G.A.)

JOB No. ENG4111/2 | DRAWING No. 4111-A010 | 1 OF 1 | SIZE A1 | REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-A010.iam
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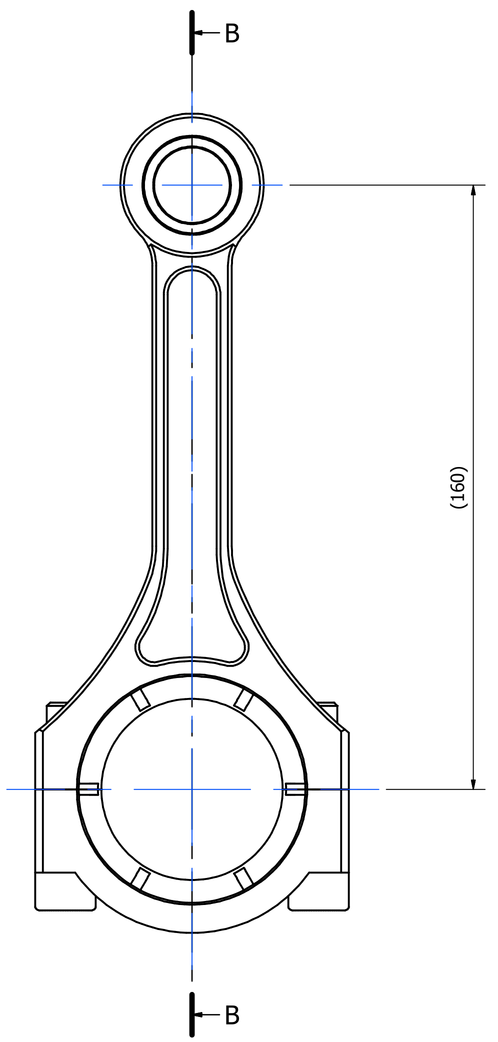
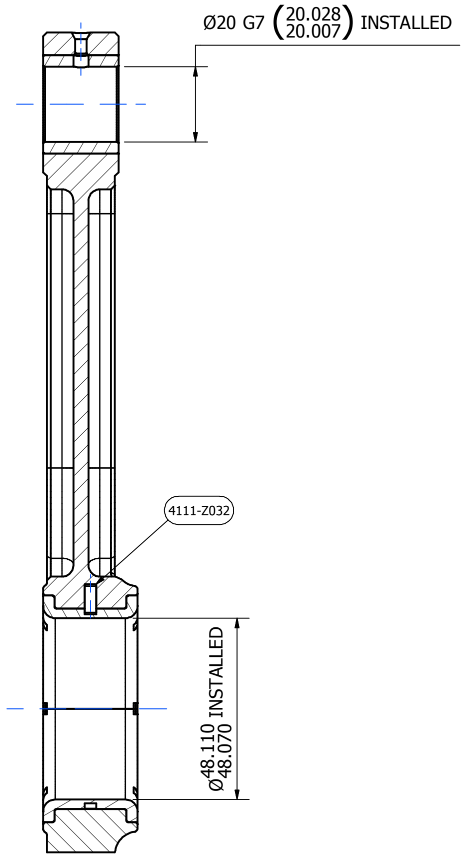
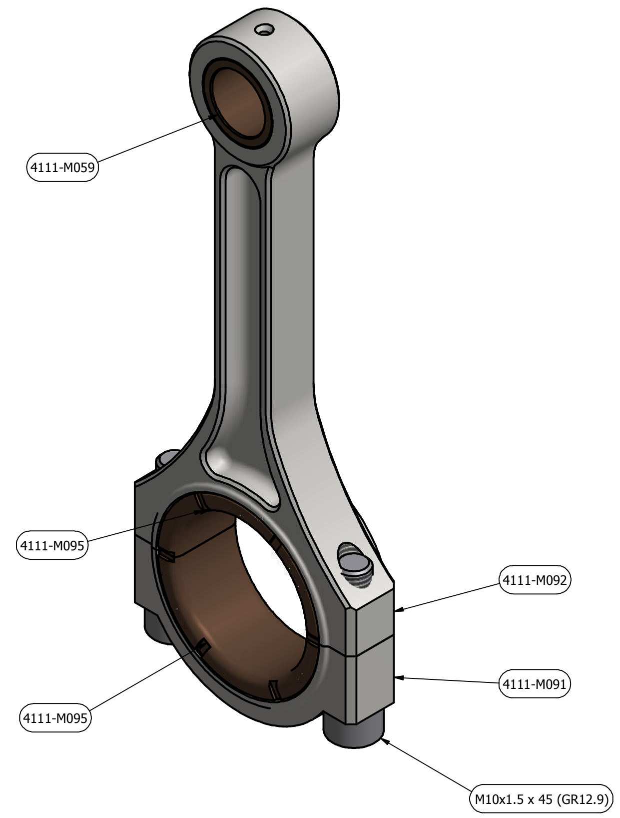


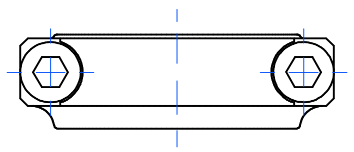
4111-A020 - TOP VIEW -
SCALE: 1 : 1



SECTION B-B
SCALE: 1 : 1



4111-A020 - ISOMETRIC VIEW -
SCALE: 1 : 1



- BOTTOM VIEW -

NOTES:

1. FASTENERS ENGAGED INTO THREAD INSERTS TO BE TORQUED TO THE FOLLOWING SETTINGS:
   - M10 =(75 N.m.)
2. ALL FASTENERS MUST BE TIGHTENED WITH 220 LOCTITE THREAD COMPOUND TO PREVENT LOOSENING
3. ALL PARTS TO FIT TOGETHER FREELY WITHOUT FORCING EXCEPT THE LITTLE END BEARING
4. CONROD EXTERNAL SURFACES TO BE FREE OF BURRS & SHARP EDGES.

PARTS LIST

PART NUMBER	DESCRIPTION	QTY	MASS	REFERENCE
4111-M059	LITTLE END BEARING	1	0 kg	4111- M059
4111-M091	CONROD BIG END SADDLE	1	0.2 kg	VENDOR SUPPLY
4111-M092	CONROD	1	0.5 kg	4111-M092
4111-M095	BIG END BEARING SHELL	2	0 kg	4111-M095
4111-Z032	4 x 20 MACHINE DOWEL	1	0 kg	VENDOR SUPPLY
M10x1.5 x 45 (GR12.9)	Broached Socket Head Cap Screw - Metric	2	0 kg	

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
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DRAWN	DATE	DESIGNED	DATE	CHECKED	DATE	APPROVED	DATE
KBD	10/07/2014	KBD	10/07/2014				

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: CONROD ASSEMBLY

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-A021	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-A021.iam
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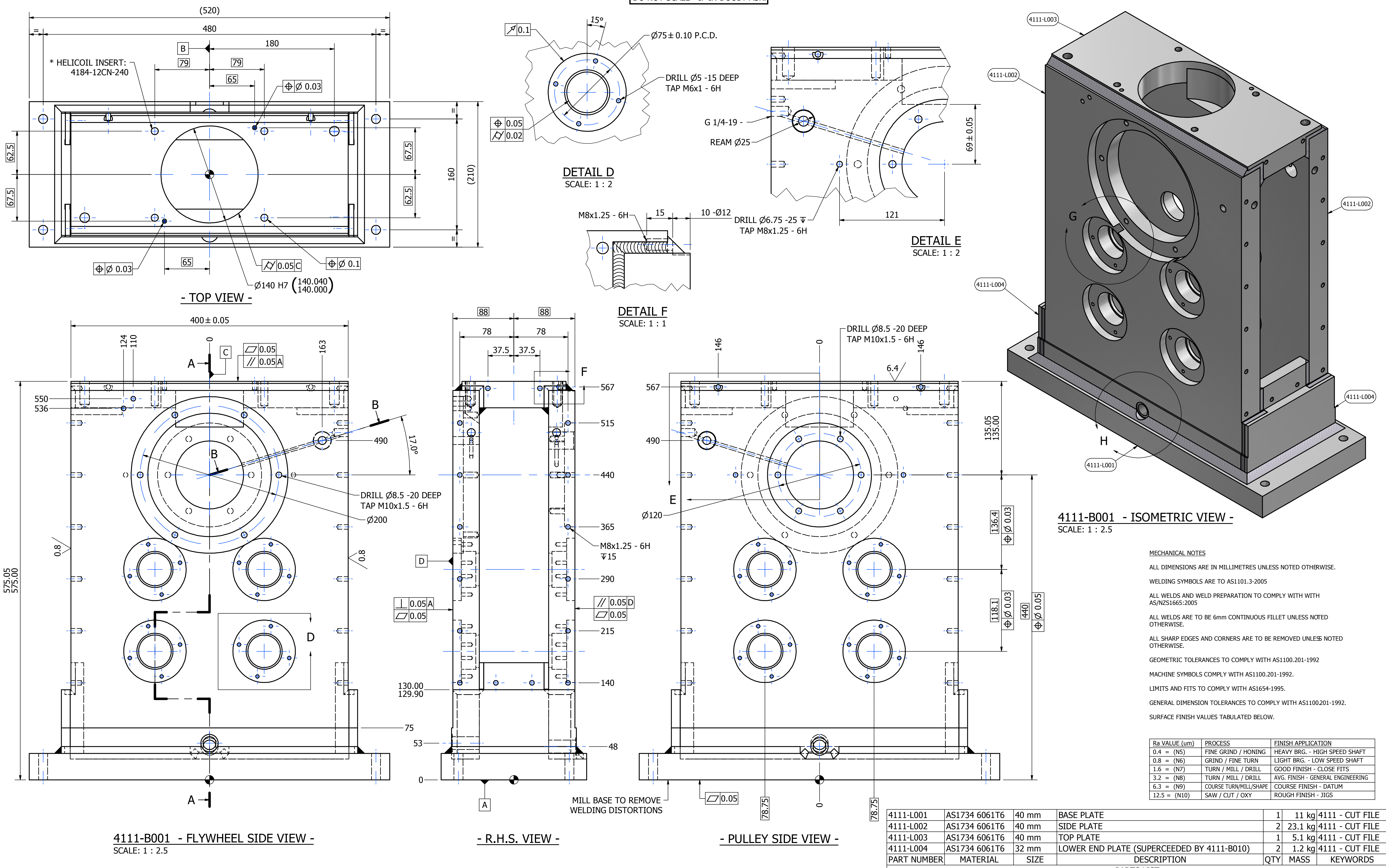

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1665:2005

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS	KEYWORDS
4111-L001	AS1734 6061T6	40 mm	BASE PLATE	1	11 kg	4111 - CUT FILE
4111-L002	AS1734 6061T6	40 mm	SIDE PLATE	2	23.1 kg	4111 - CUT FILE
4111-L003	AS1734 6061T6	40 mm	TOP PLATE	1	5.1 kg	4111 - CUT FILE
4111-L004	AS1734 6061T6	32 mm	LOWER END PLATE (SUPERCEEDED BY 4111-B010)	2	1.2 kg	4111 - CUT FILE

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm ...... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	07/08/14	
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DRAWN	KBD	DATE 29/04/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	DATE 29/04/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED		DATE	DRAWING DESCRIPTION: CRANKCASE DETAIL & WELDMENT
APPROVED		DATE	JOB No. ENG4111/2 · DRAWING No. 4111-B001 · 1 OF 5 · SIZE A1 · REV. 0

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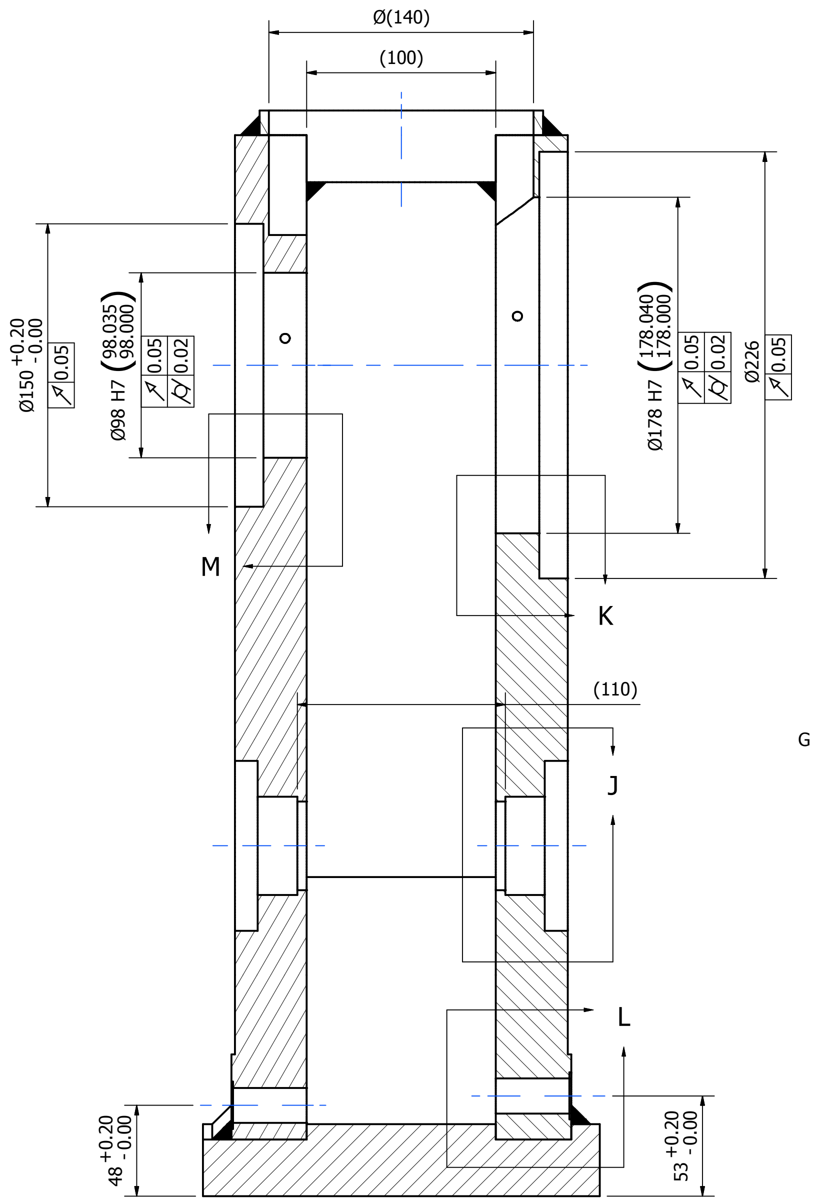
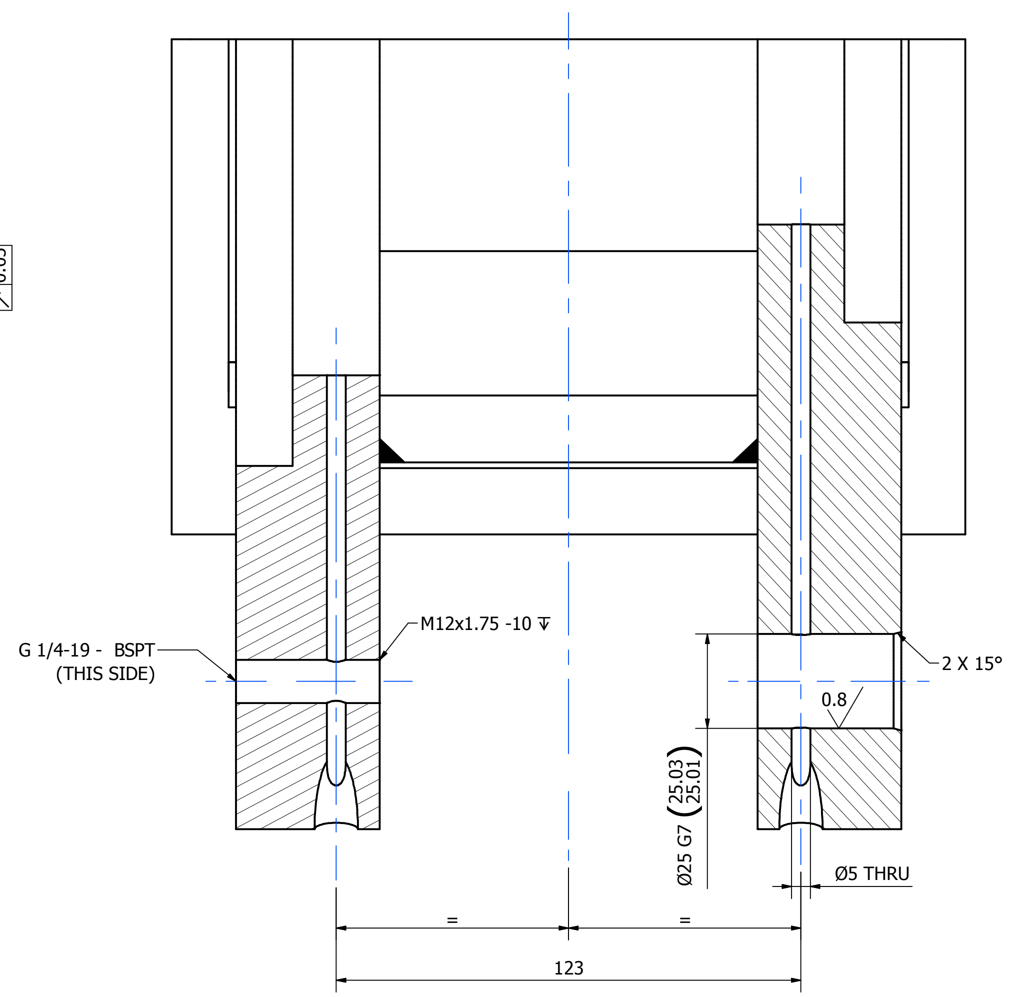
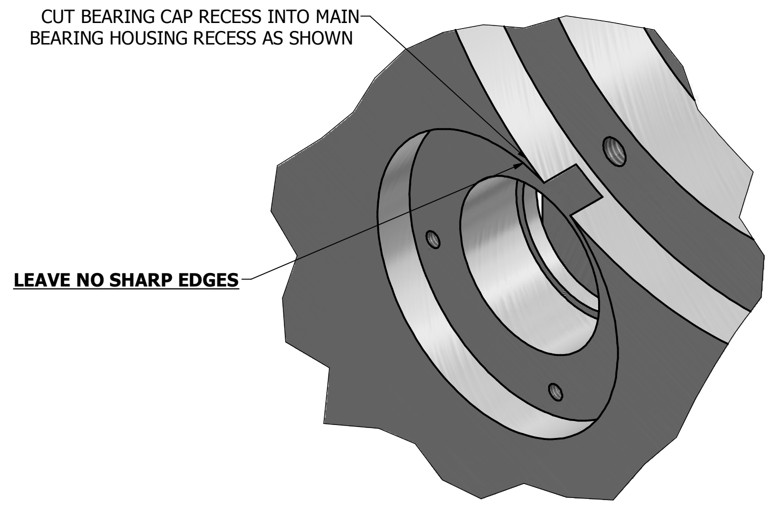


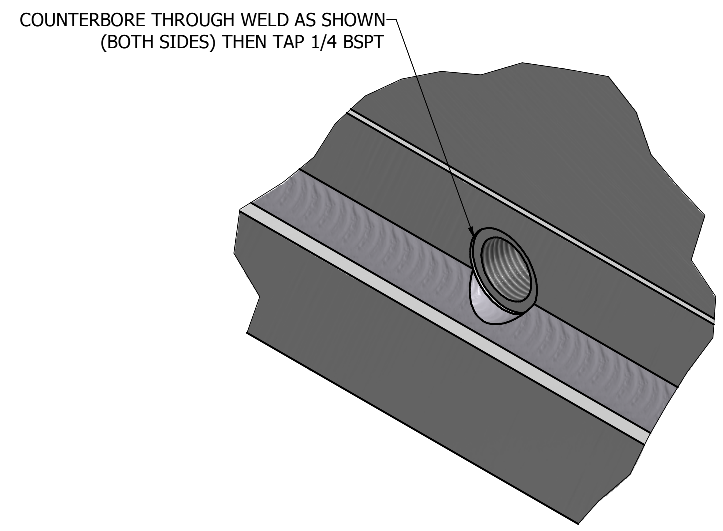
SECTION A-A
SCALE: 1 : 2



SECTION B-B
SCALE: 1 : 1



DETAIL G
SCALE: 1 : 1



DETAIL H
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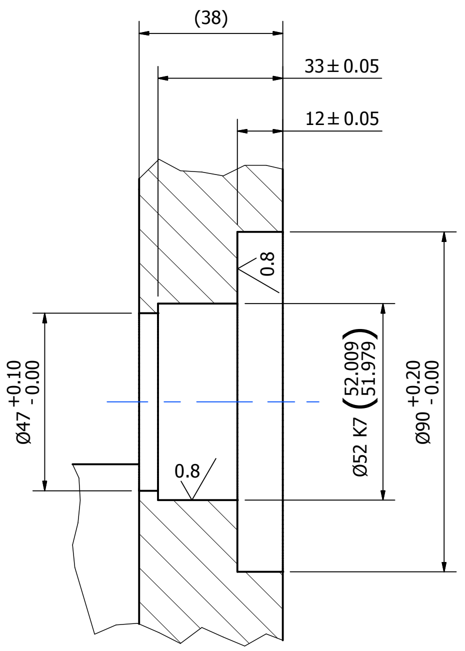

DETAIL J
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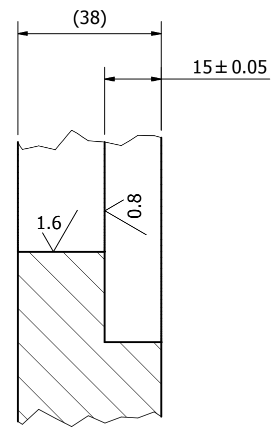
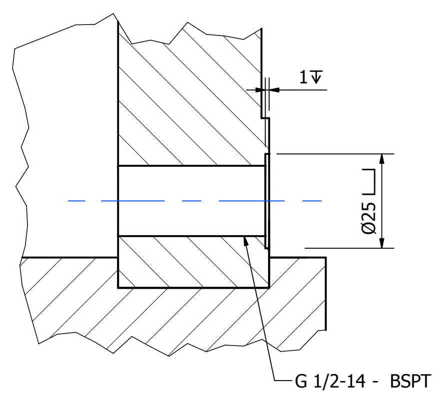
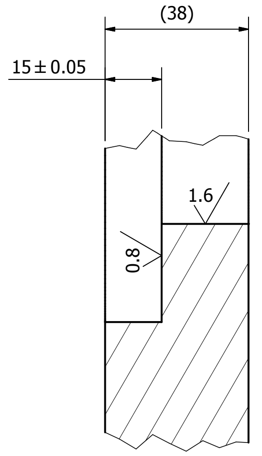


DETAIL K
SCALE: 1 : 1



DETAIL L
SCALE: 1 : 1



DETAIL M
SCALE: 1 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1665:2005

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN		
DRAWN	KBD	DATE	29/04/2014
DESIGNED	KBD	DATE	29/04/2014
CHECKED		DATE	
APPROVED		DATE	

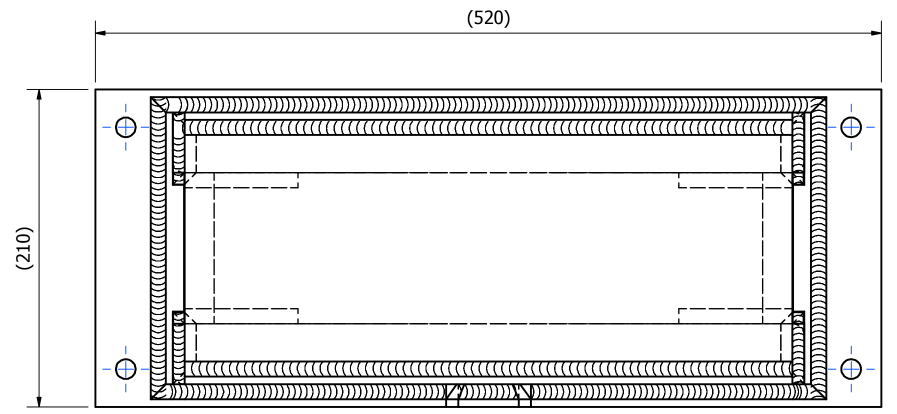
THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: CRANKCASE DETAIL & WELDMENT

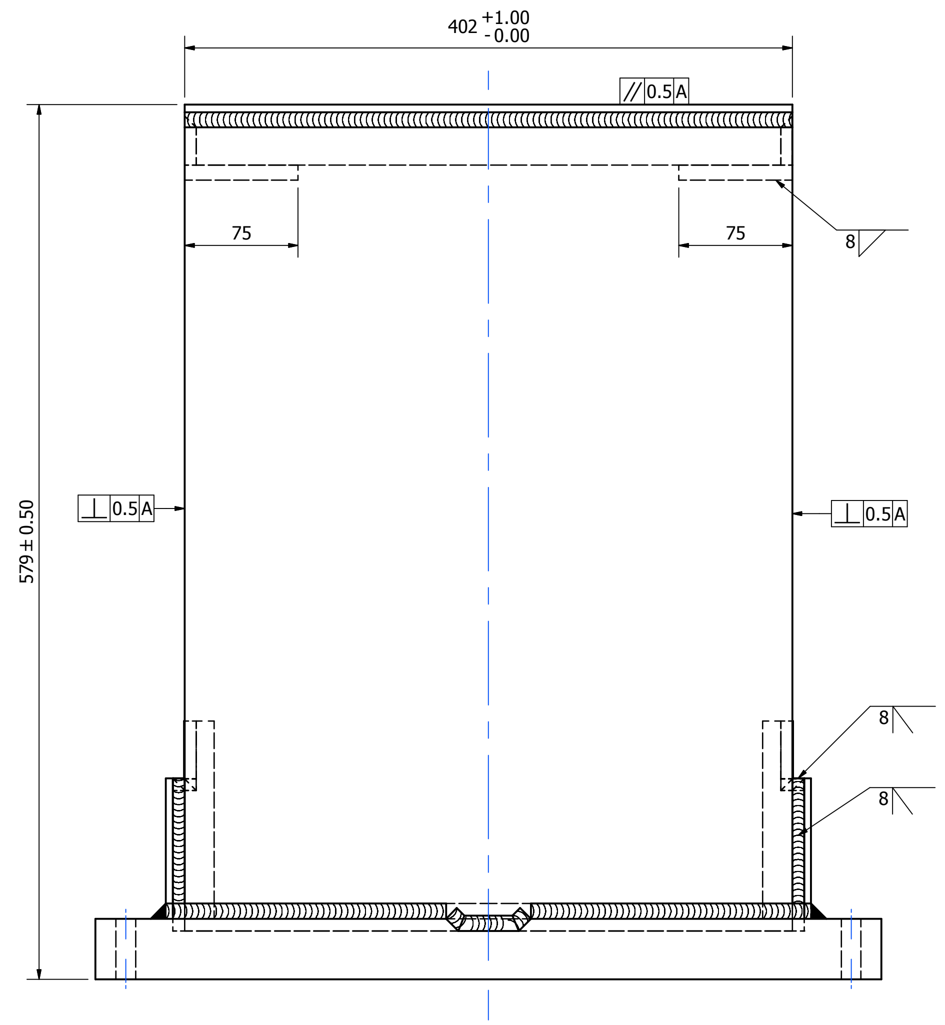
JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-B001	2 OF 5	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-B001.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-B001.idw

DO NOT SCALE - IF IN DOUBT ASK.

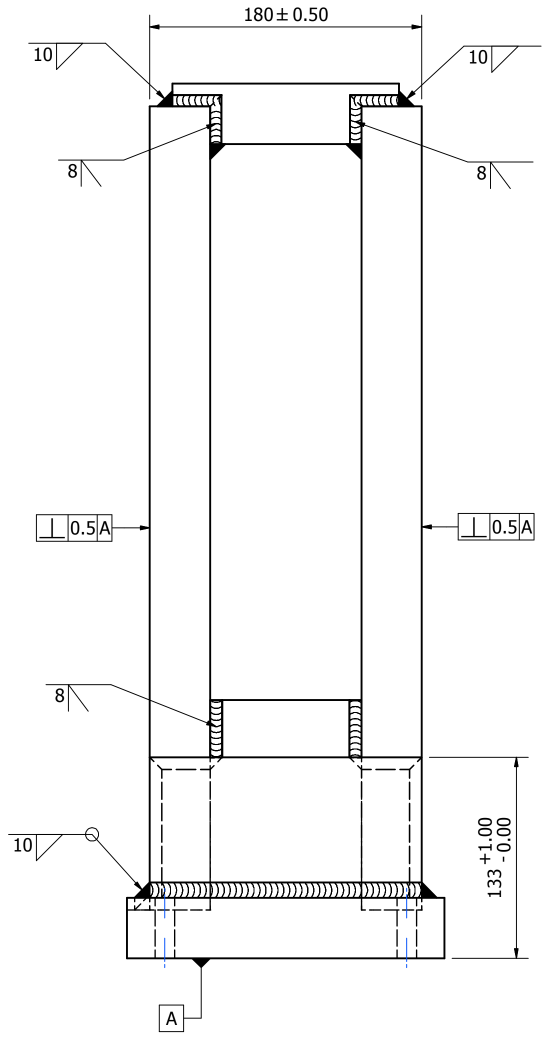


- TOP VIEW -

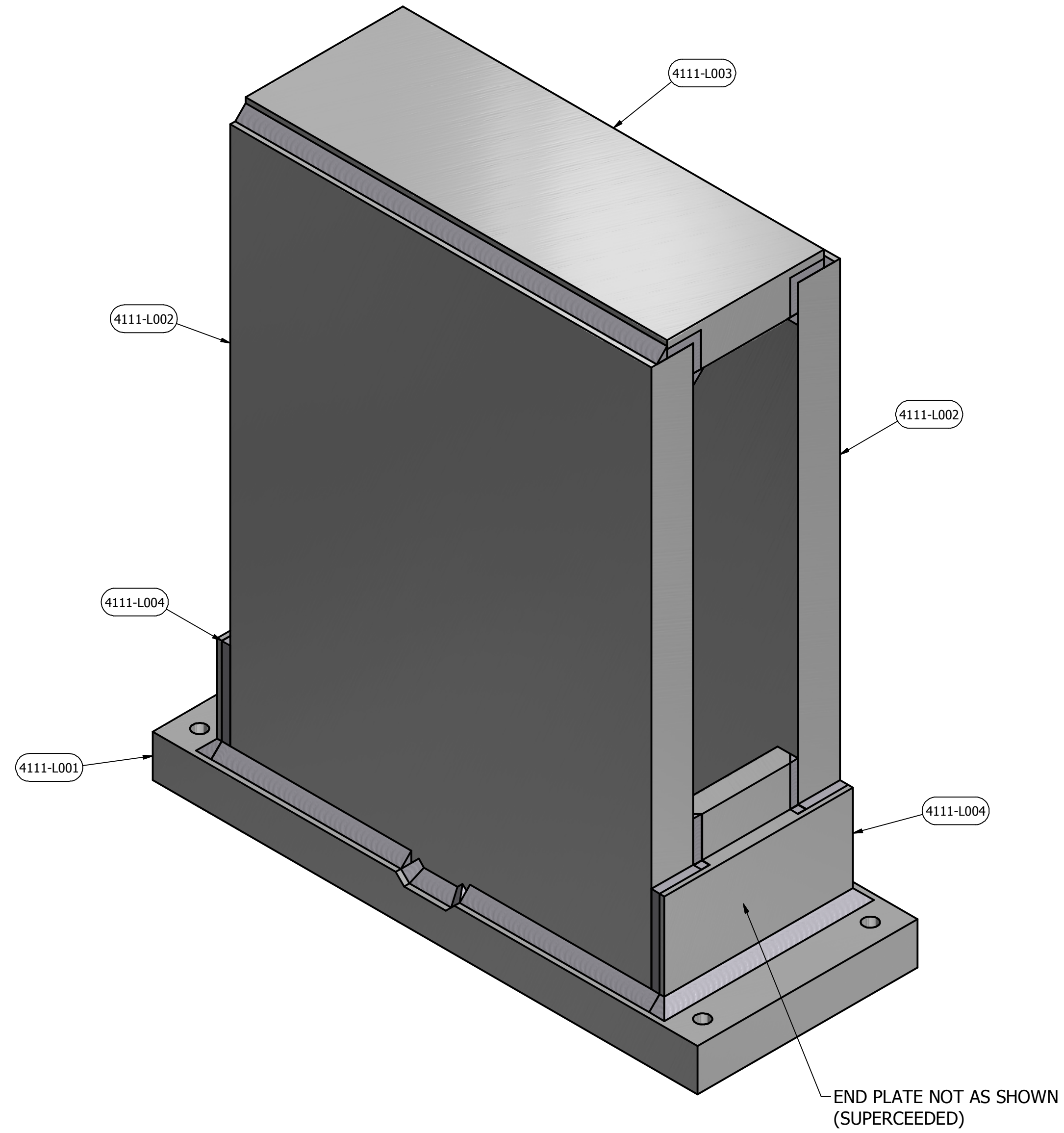


4111-B001 (PRE-MACHINING) / WELDMENT - OIL DRAIN SIDE -

SCALE: 1 : 2.5



- SIDE VIEW -



4111-B001 (PRE-MACHINING) / WELDMENT - ISOMETRIC VIEW -

SCALE: 1 : 2.5

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS
4111-L001	AS1734 6061T6	40 mm	BASE PLATE	1	11 kg
4111-L002	AS1734 6061T6	40 mm	SIDE PLATE	2	23.1 kg
4111-L003	AS1734 6061T6	40 mm	TOP PLATE	1	5.1 kg
4111-L004	AS1734 6061T6	32 mm	LOWER END PLATE (SUPERCEEDED BY 4111-B010)	2	1.2 kg

PARTS LIST

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
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EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN	KBD	DATE	29/04/2014
DESIGNED	KBD	DATE	29/04/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: CRANKCASE DETAIL & WELDMENT

JOB No. ENG4111/2 | DRAWING No. 4111-B001 | 3 OF 5 | SIZE A1 | REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-B001 (PRE-MACHINING).iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-B001.idw

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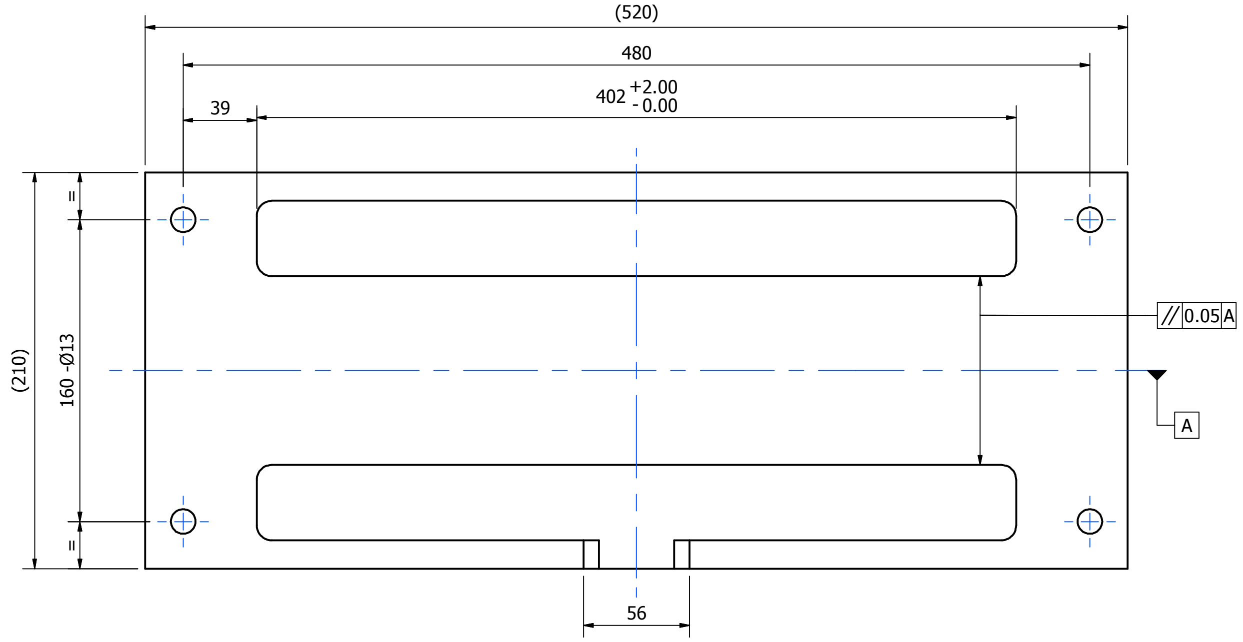
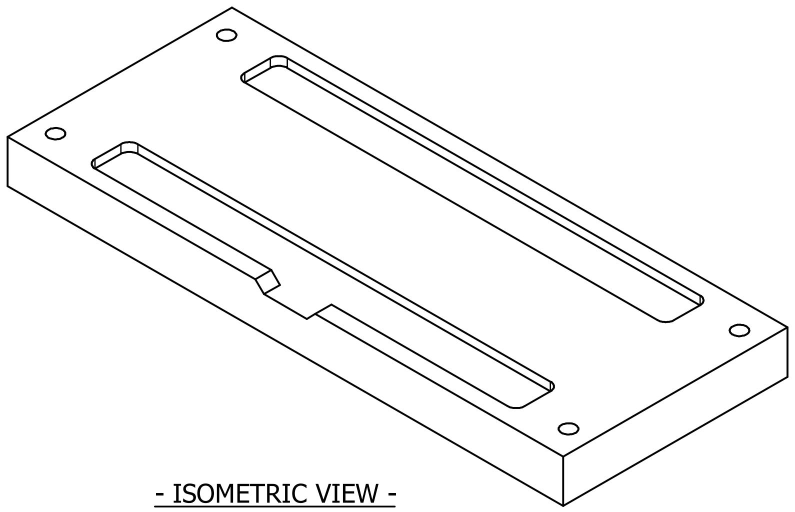
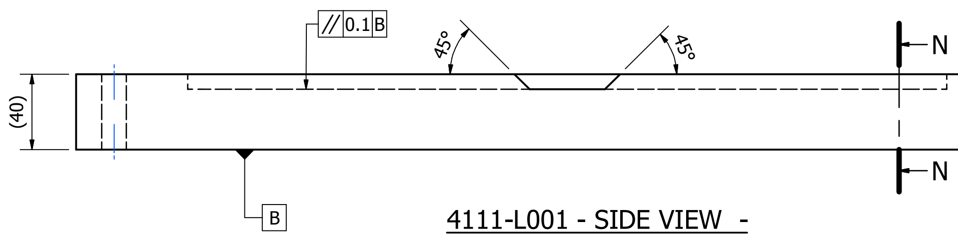


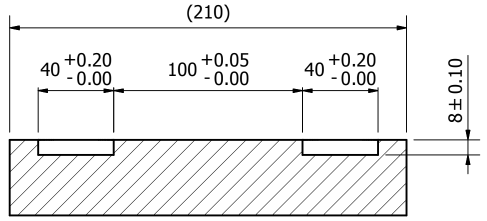
- TOP VIEW -



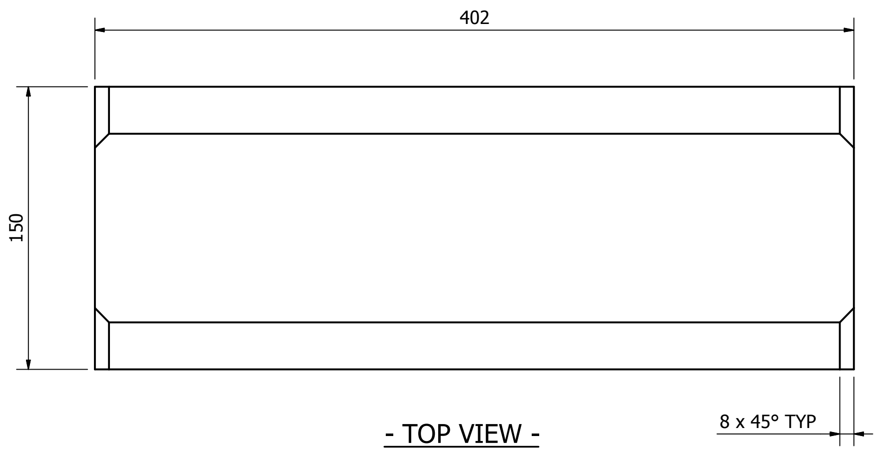
- ISOMETRIC VIEW -



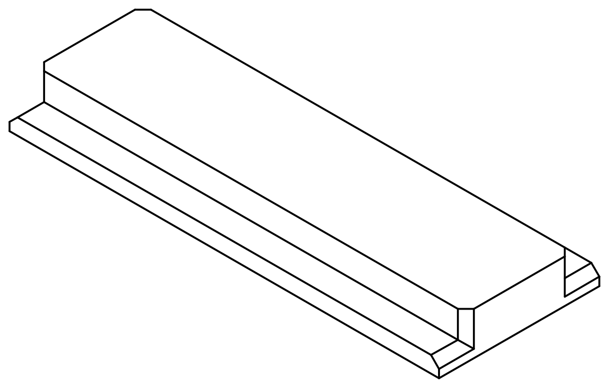
4111-L001 - SIDE VIEW -
SCALE: 1 : 2



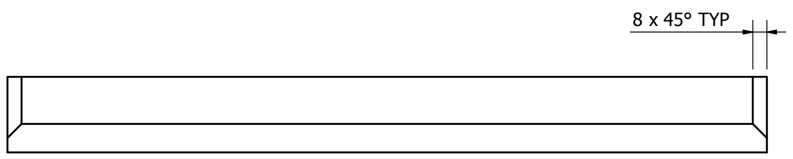
SECTION N-N
SCALE: 1 : 2



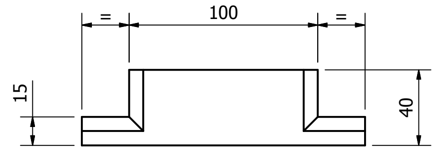
- TOP VIEW -



- ISOMETRIC VIEW -



4111-L003 - SIDE VIEW -
SCALE: 1 : 2



- END VIEW -

GENERAL TOLERANCES UNLESS INDICATED		ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE
PREPARATION AND ASSEMBLY	MACHINING	
< 25mm .... ±1.0mm	0 .... ±0.1mm	
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm	
>600mm .... ±2.0mm	0.00 .... ±0.01mm	
ANGLE .... ±1°	3.2	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
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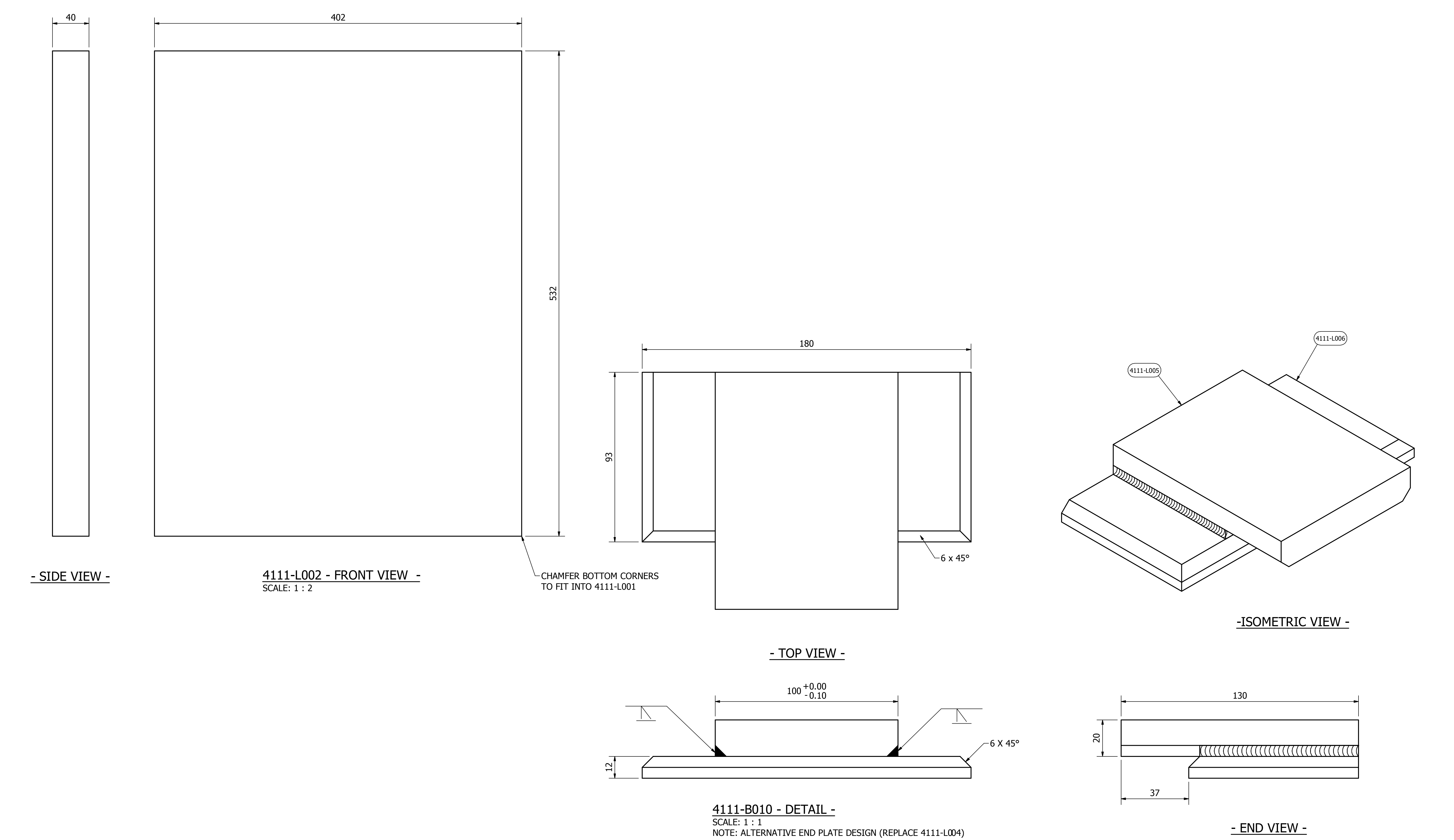
SCALE	AS SHOWN	THE UNIVERSITY OF SOUTHERN QUEENSLAND.	
DRAWN KBD	DATE 29/04/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 29/04/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	CRANKCASE DETAIL & WELDMENT
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-B001 4 OF 5 · SIZE A1 · REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-L001.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-B001.idw

DO NOT SCALE - IF IN DOUBT ASK.



PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS	KEYWORDS
4111-L005	AS1734 6061T6	20 mm	LOWER END PLATE (PART 1)	1	0.7 kg	4111 - CUT FILE
4111-L006	AS1734 6061T6	12 mm	LOWER END PLATE (PART 2)	1	0.5 kg	4111 - CUT FILE

PARTS LIST

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

University of Southern Queensland
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PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN	KBD	DATE	29/04/2014
DESIGNED	KBD	DATE	29/04/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: CRANKCASE DETAIL & WELDMENT

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-B001	5 OF 5	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-L002.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-B001.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITHAS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

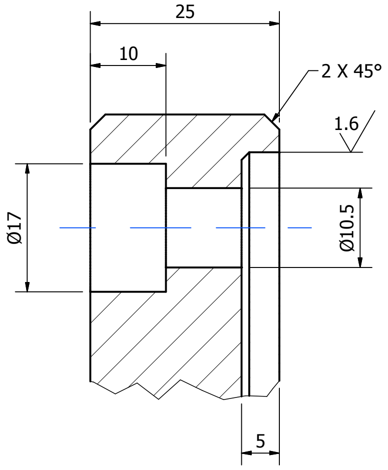
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

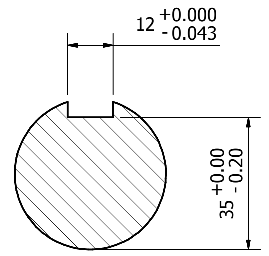
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

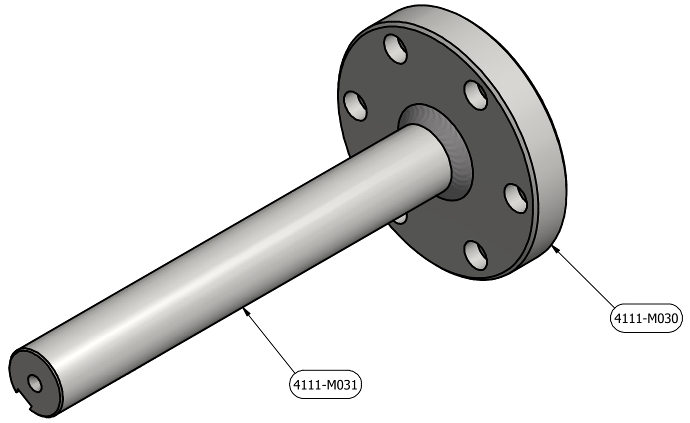
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



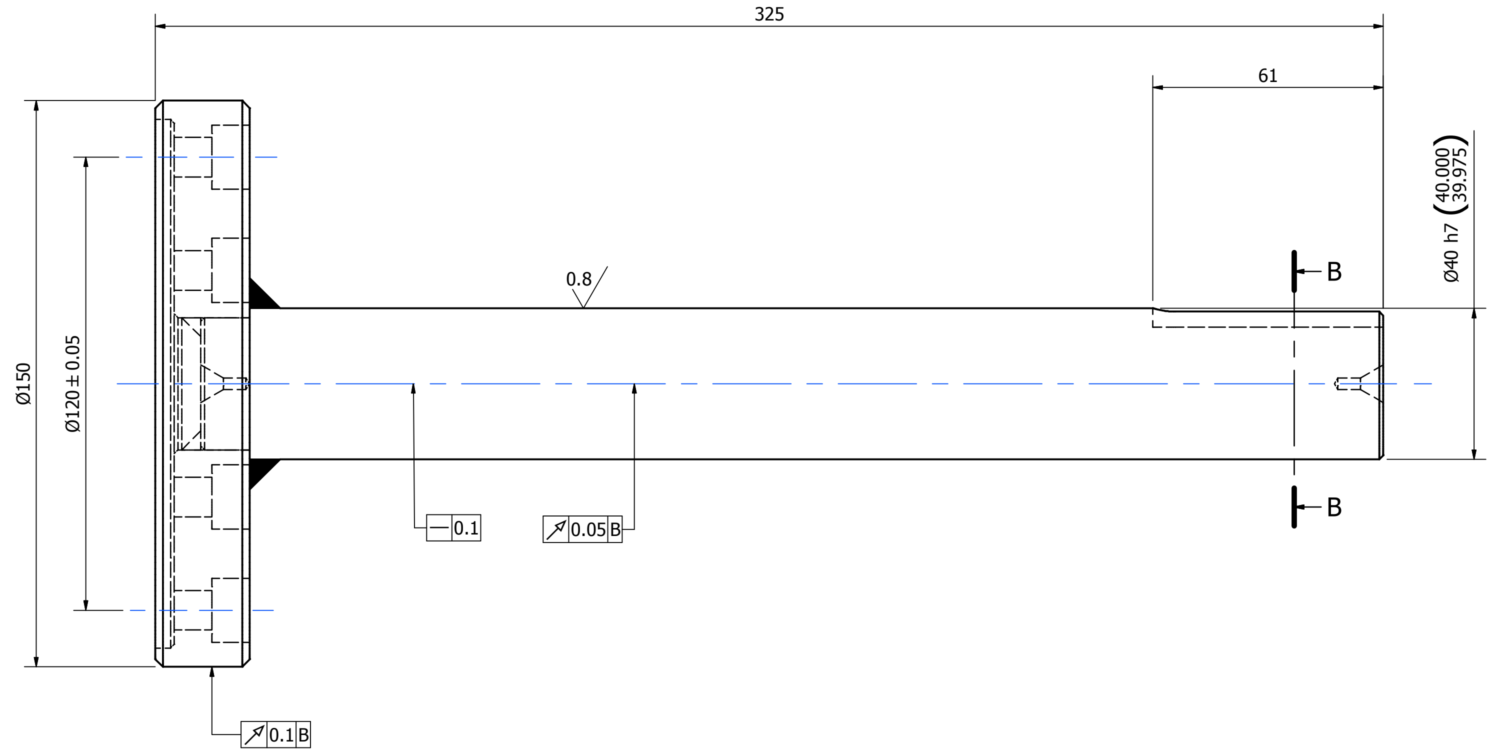
DETAIL A
SCALE: 2 : 1



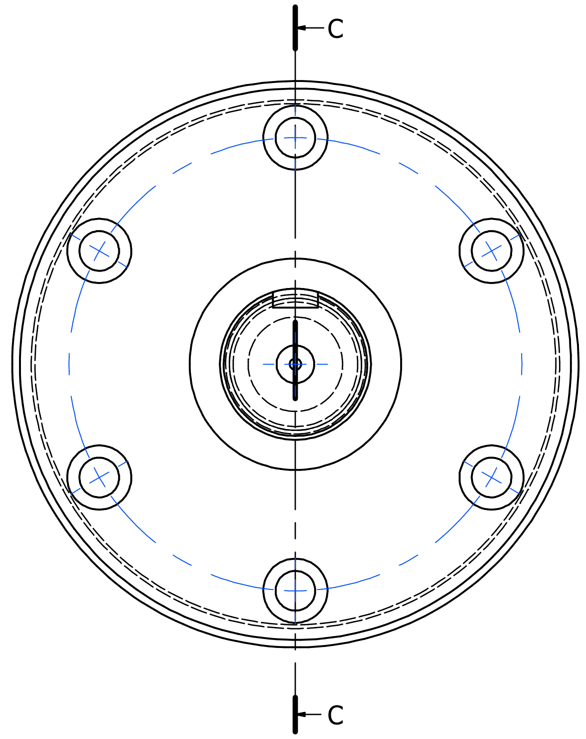
SECTION B-B
SCALE: 1 : 1



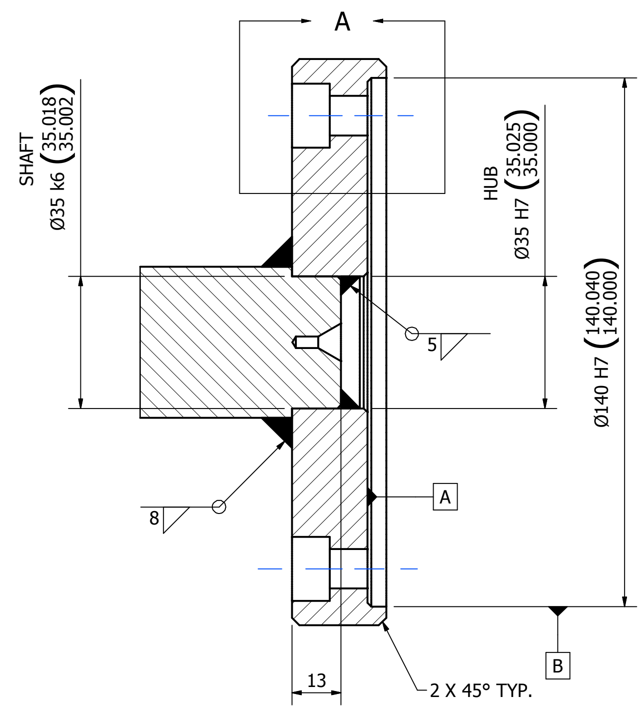
4111-B002 - ISOMETRIC VIEW -
SCALE: 1 : 2



4111-B002 - DETAIL -
SCALE: 1 : 1



- END VIEW -



SECTION C-C
SCALE: 1 : 1

PART NUMBER	QTY	MATERIAL	SIZE	LENGTH	MASS	REFERENCE
4111-M030	1	AS3678 -GR250	25 mm	N/A	2.6 kg	SEE DETAIL THIS SHEET
4111-M031	1	AS1444 -1020	40 mm	313.00000 mm	3 kg	SEE DETAIL THIS SHEET

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD	DATE: 12/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 12/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED:	DATE:	DRAWING DESCRIPTION: DRIVE ATTACHMENT
APPROVED:	DATE:	JOB No.: ENG4111/2

DRAWING No.: 4111-B002 | 1 OF 1 | SIZE: A1 | REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-B002.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-B002.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1665:2004

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

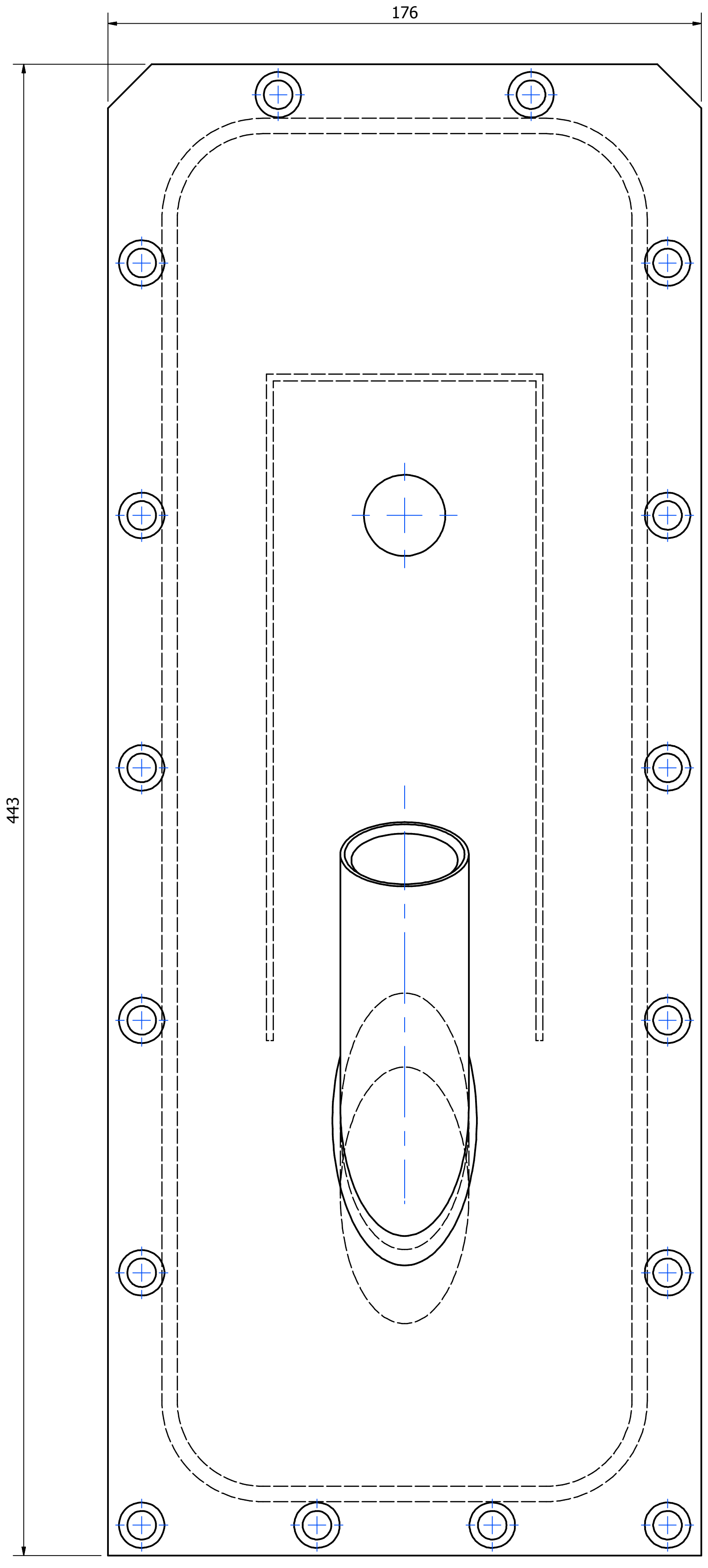
MACHINE SYMBOLS COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

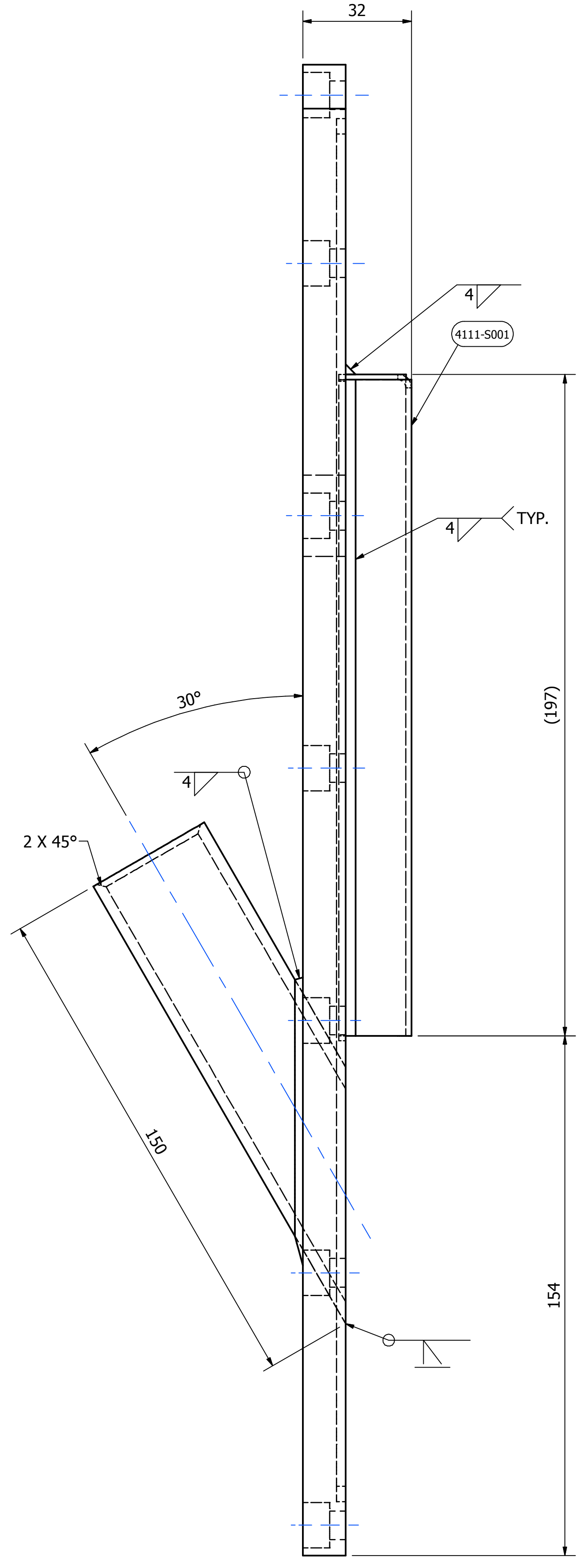
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

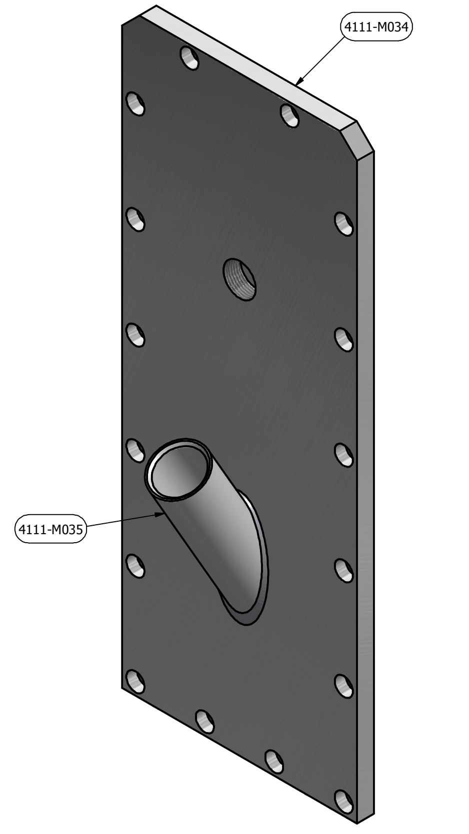
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



4111-B003 - DETAIL -
SCALE: 1 : 1



- SIDE VIEW -



4111-B003 - ISOMETRIC VIEW -
SCALE: 1 : 2

4111-M034	1	AS1734 6061T6	AL. PLATE	12.7 mm	N/A	2.5 kg	4111-M034 ; 4111-CUT FILE
4111-M035	1	AS1867 6061T6	AL. TUBE	38.1 mm	150.00000 mm	0.1 kg	SEE DETAIL THIS SHEET
4111-S001	1	AS1734 6061T6	AL. SHEET	1.6 mm	N/A	0.1 kg	411-S004; 4111-CUT FILE
PART NUMBER	QTY	MATERIAL	CATEGORY	SIZE	LENGTH	MASS	REFERENCE

PARTS LIST

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD	DATE: 10/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 10/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	SIDE COVER (OIL FILL)
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-B003 1 OF 1 SIZE A1 REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-B003.iam
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-B003.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

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GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

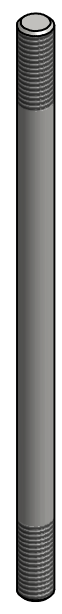
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

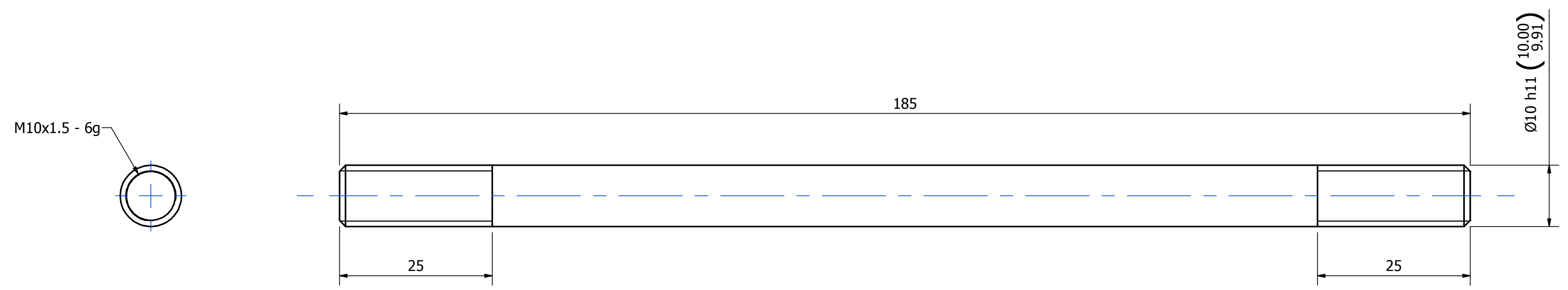
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



4111-E001 - ISOMETRIC VIEW -

SCALE: 1 : 1



4111-E001 - DETAIL -

SCALE: 2 : 1

NOTE:

1. THREAD TO CONFORM WITH AS1275 AND NOTED CLASS OF FIT.
2. TIE RODS AND THREADS TO BE FREE FROM DAMAGE OR EXCESSIVE MACHINING THAT MAY INDUCE STRESS CONCENTRATIONS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-E001	6	AS1444 -4140	TIE ROD	0 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

		DATE	
DRAWN	KBD		10/08/2014
DESIGNED	KBD		10/08/2014
CHECKED			
APPROVED			

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

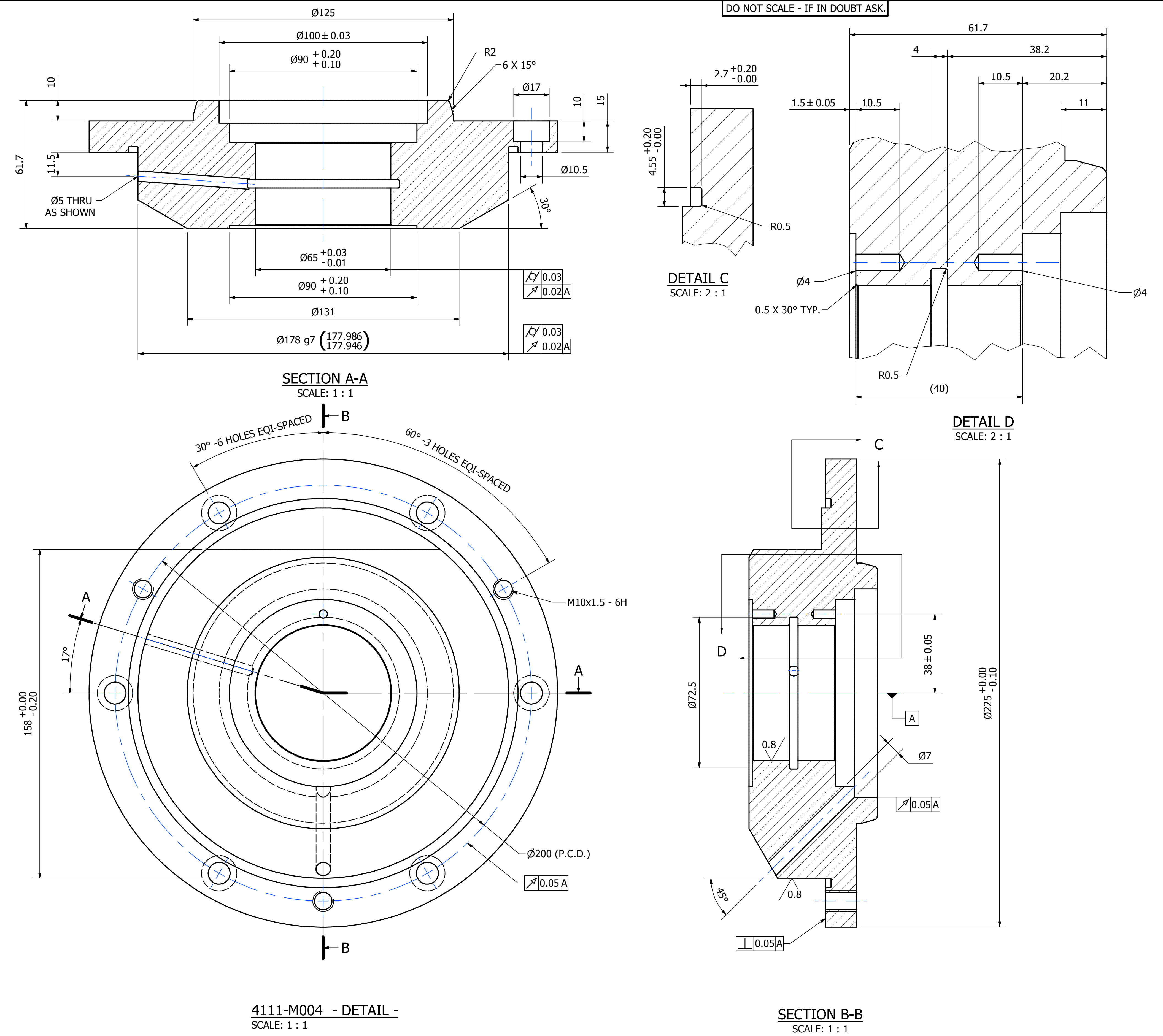
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

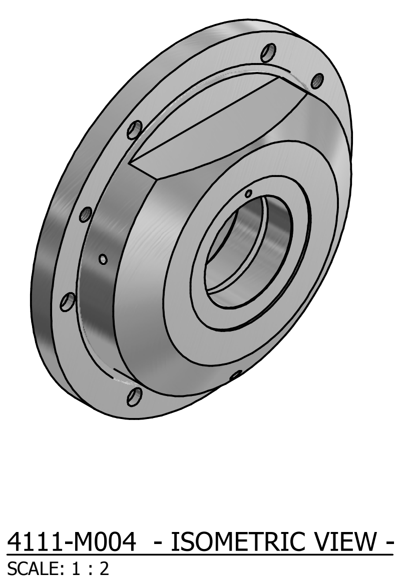
DRAWING DESCRIPTION: CYLINDER TIE ROD DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-E001	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-E001.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-E001.idw

DO NOT SCALE - IF IN DOUBT ASK.





MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M004	1	AS1865 -2011T6	FLYWHEEL SIDE MAIN BEARING HOUSING	3.4 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

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A	PRELIMINARY ISSUE	09/08/2014	
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PH: +61 7 4631 2100
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DRAWN	KBD
DATE	10/07/2014
DESIGNED	KBD
DATE	10/07/2014
CHECKED	
APPROVED	
CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION	MAIN BEARING HOUSING (FLYWHEEL SIDE)
JOB No.	ENG4111/2
DRAWING No.	4111-M004 1 OF 1
SIZE	A1
REV.	0

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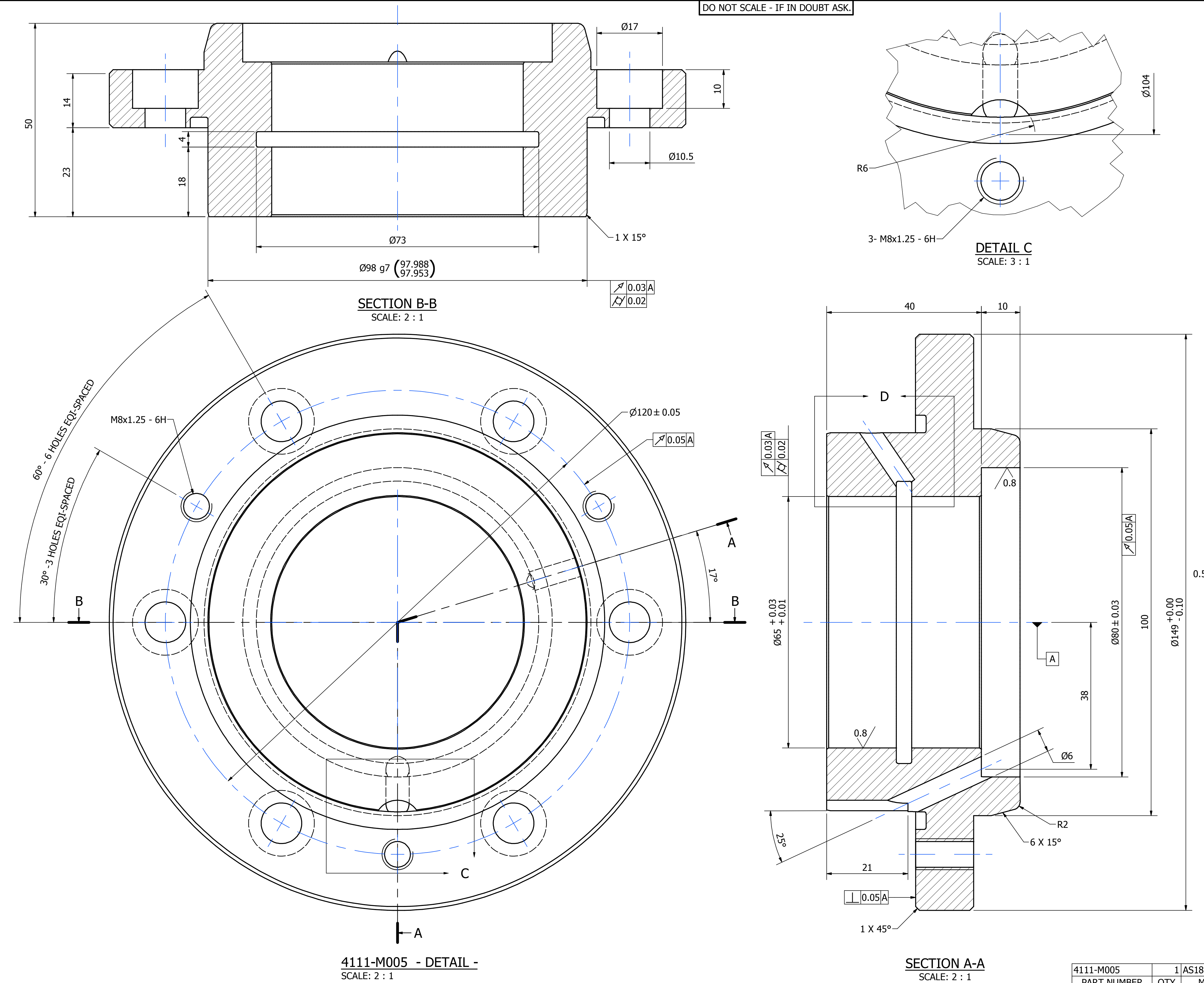

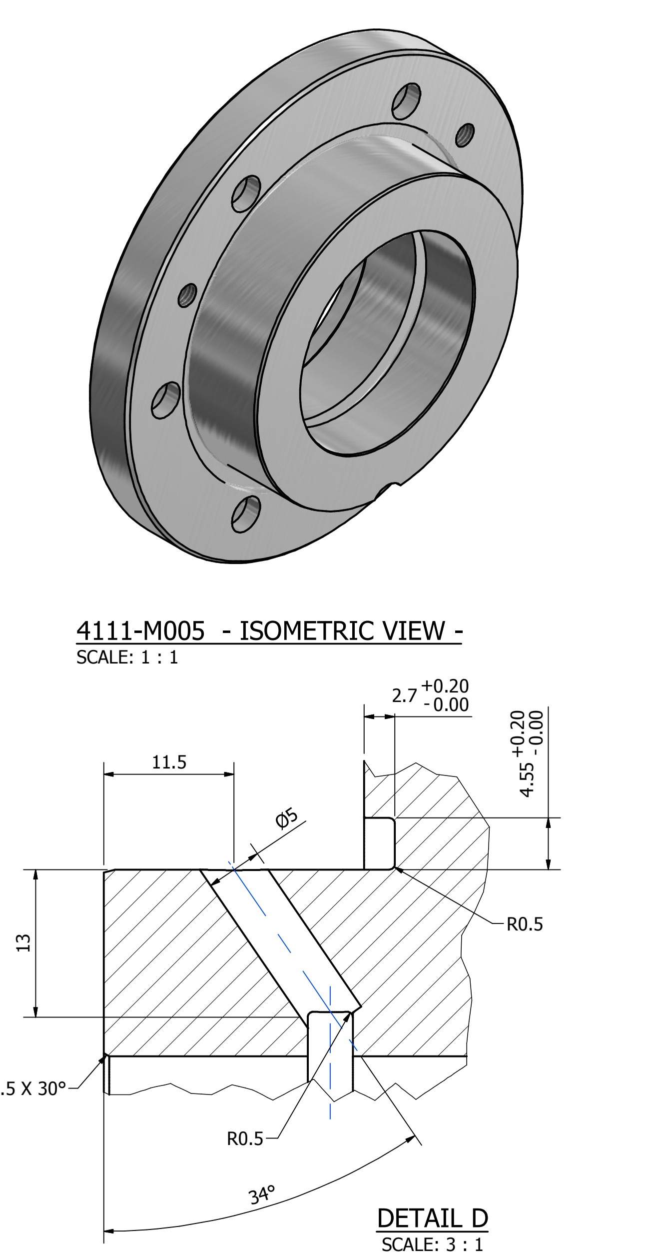

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M005	1	AS1865 -2011T6	ENCODER SIDE MAIN BEARING HOUSING	0.9 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
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West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
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DESIGNED	KBD	DATE	10/07/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: MAIN BEARING HOUSING (DRIVE SIDE)

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M005	1 OF 1	A1	0

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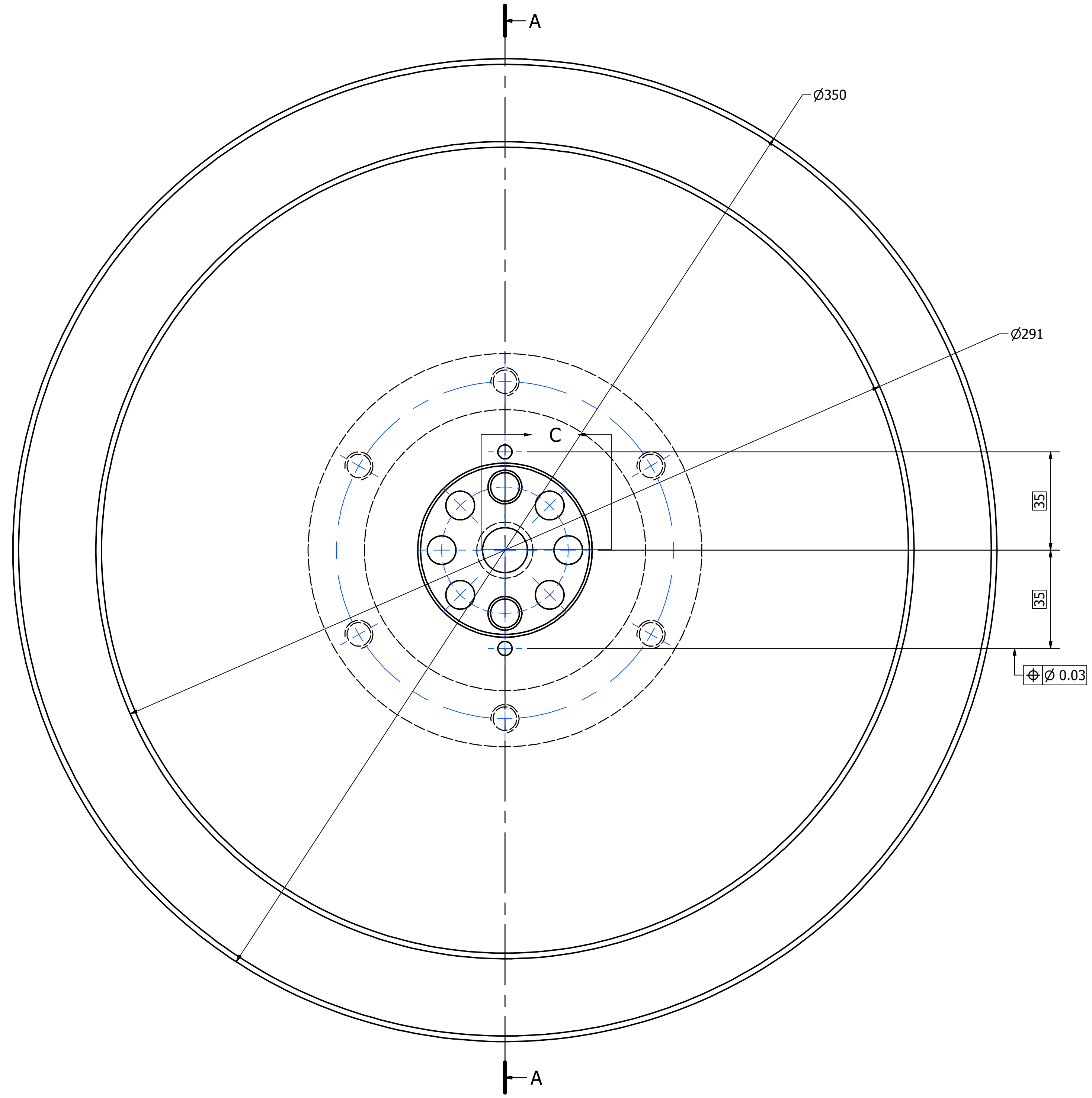
DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

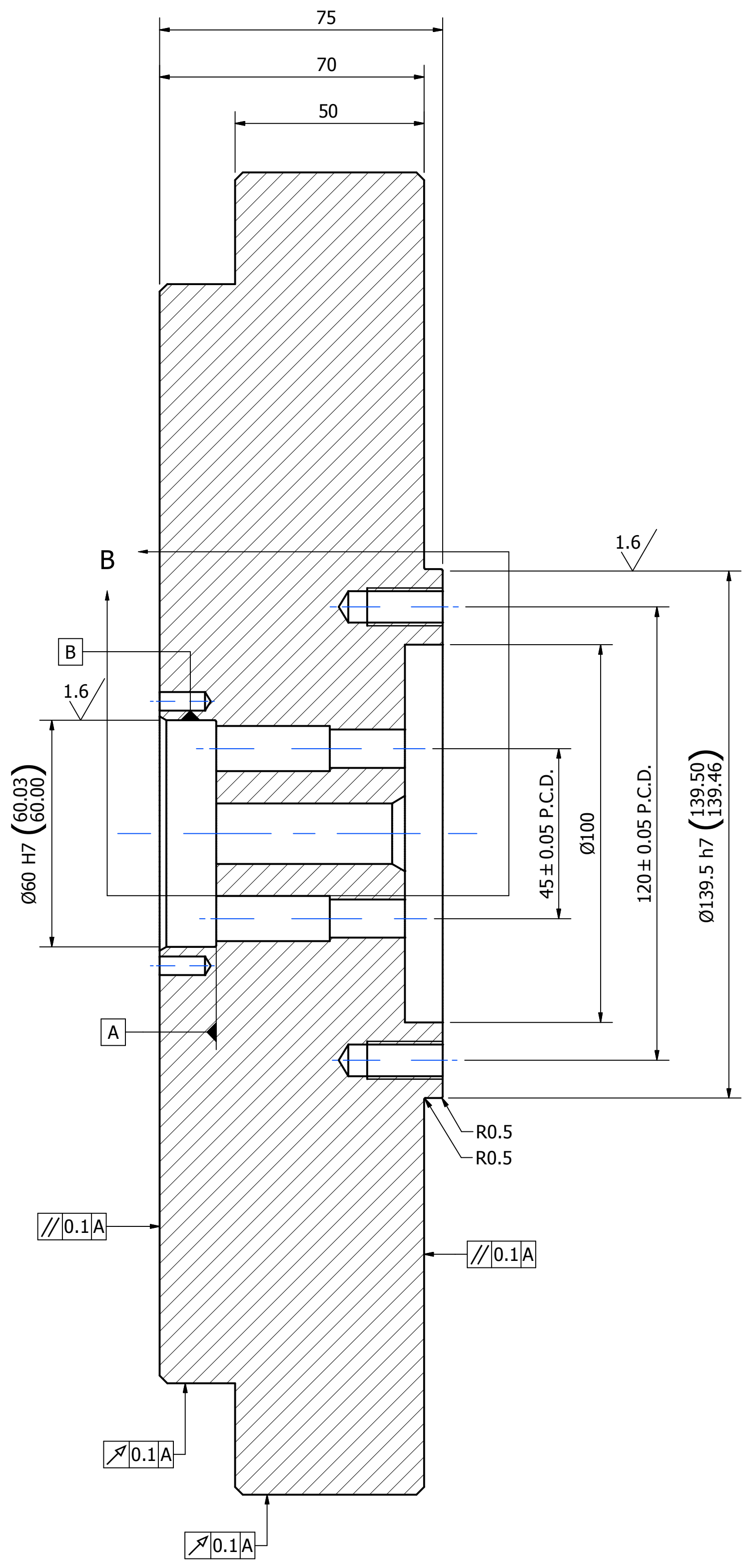
- ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.
- WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005
- ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1
- ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.
- ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.
- HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.
- GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992
- MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.
- LIMITS AND FITS TO COMPLY WITH AS1654-1995.
- GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.
- SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



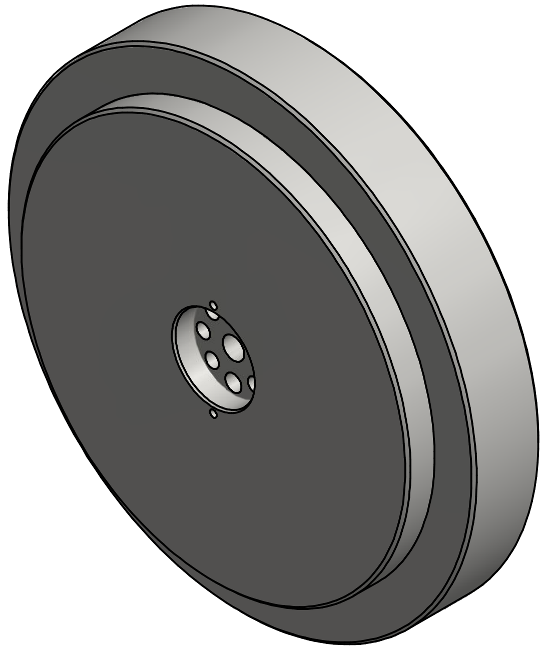
4111-M006 - DETAIL -

SCALE: 1 : 1



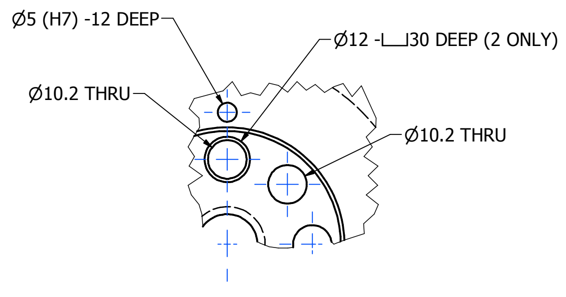
SECTION A-A

SCALE: 1 : 1



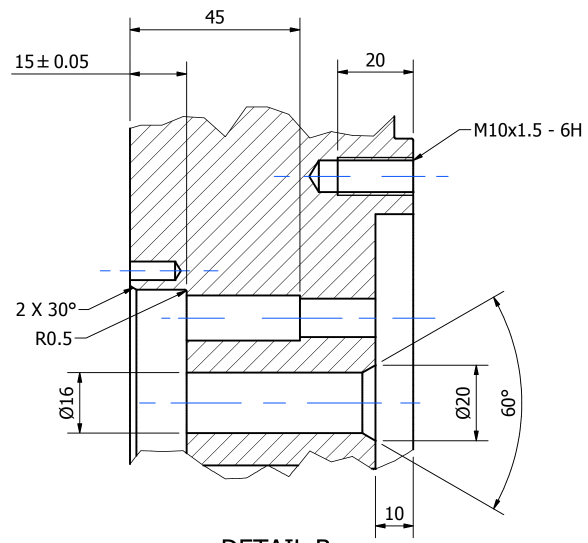
4111-M006 - ISOMETRIC VIEW -

SCALE: 1 : 2



DETAIL C

SCALE: 1 : 1



DETAIL B

SCALE: 1 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M006	1	AS3678 -GR250	FLYWHEEL	47.4 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN	KBD	DATE 12/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	DATE 12/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED		DATE	DRAWING DESCRIPTION: FLYWHEEL DETAIL
APPROVED		DATE	JOB No. ENG4111/2 · DRAWING No. 4111-M006 · 1 OF 1 · SIZE A1 · REV. 0

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

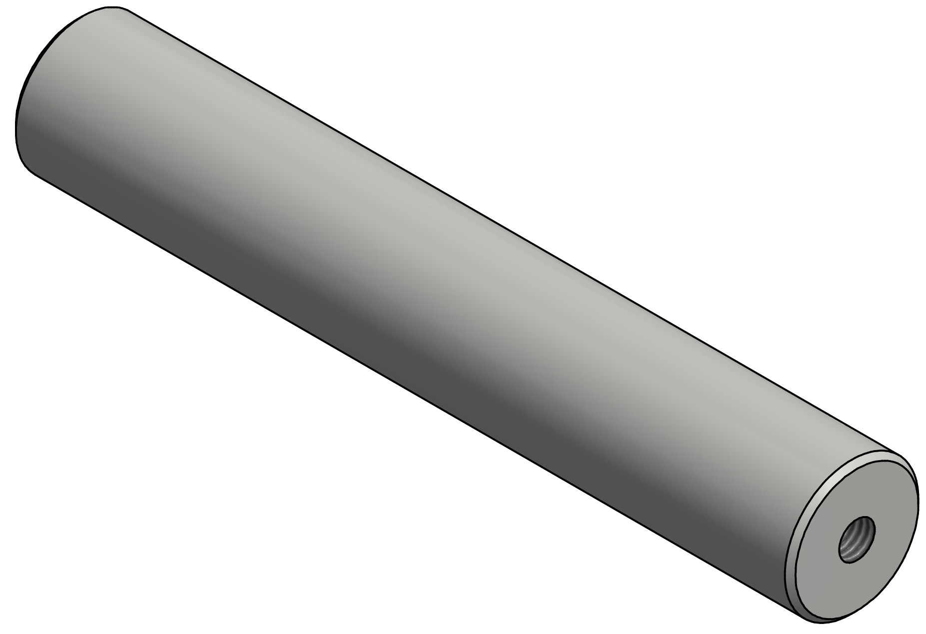
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

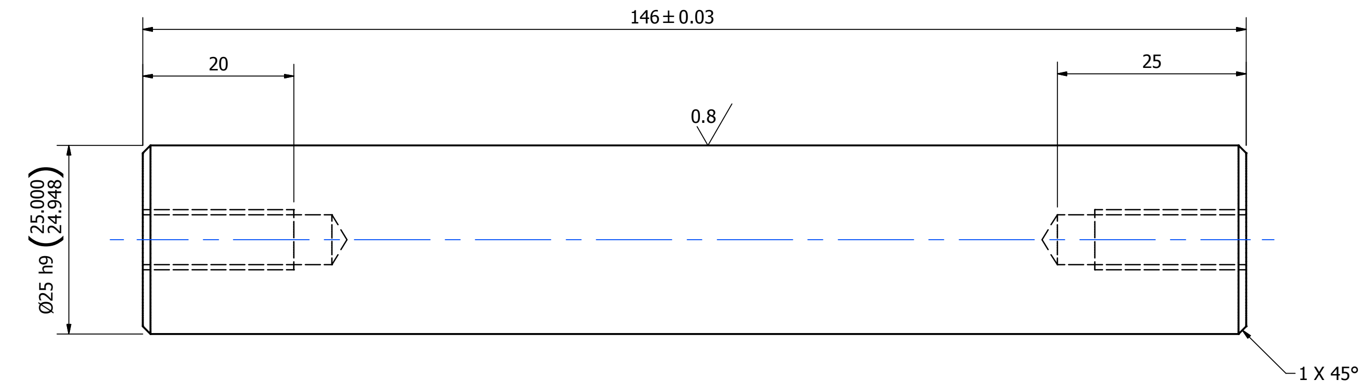
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



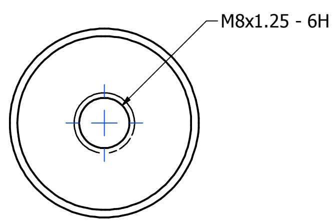
4111-M007 - ISOMETRIC VIEW -

SCALE: 2 : 1



4111-M007 - DETAIL -

SCALE: 2 : 1



- END VIEW -

4111-M007	4	AS1444 -4140	COUNTER BALANCE SHAFT	0.5 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
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DRAWN	DATE
KBD	10/07/2014
DESIGNED: KBD	DATE: 10/07/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: COUNTER BALANCE SHAFT DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M007	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M007.ipt
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DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

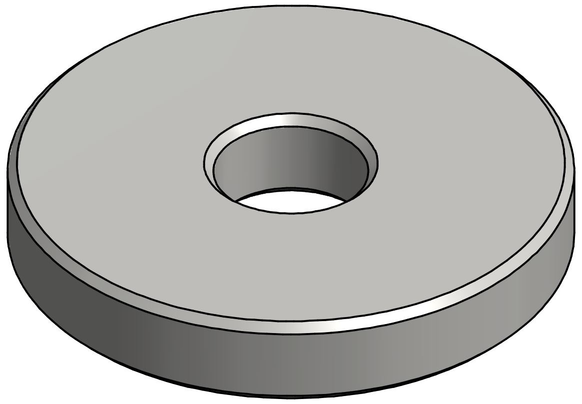
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

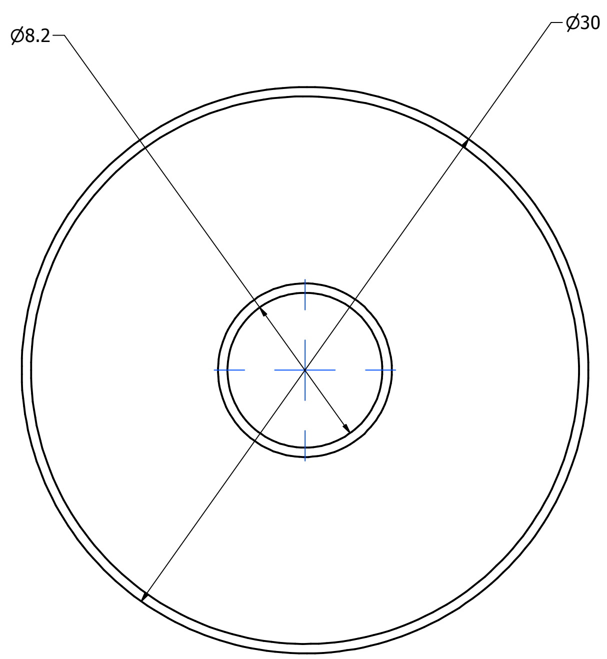
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



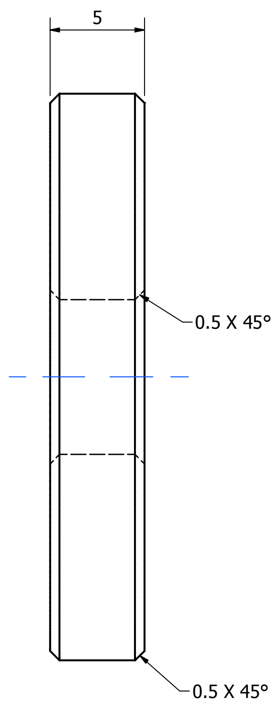
4111-M008 - ISOMETRIC VIEW -

SCALE: 5 : 1



4111-M008 - DETAIL -

SCALE: 5 : 1



- SIDE VIEW -

4111-M008	8	AS1444 -1020	COUNTER BALANCE SHAFT WASHER	0 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

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University of Southern Queensland
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DRAWN KBD	DATE 12/07/2014	CLIENT UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 12/07/2014	DESCRIPTION OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION BALANCE SHAFT RETAINING WASHERS
APPROVED	DATE	JOB No. ENG4111/2 · DRAWING No. 4111-M008 · 1 OF 1 · SIZE A1 · REV. 0

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DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

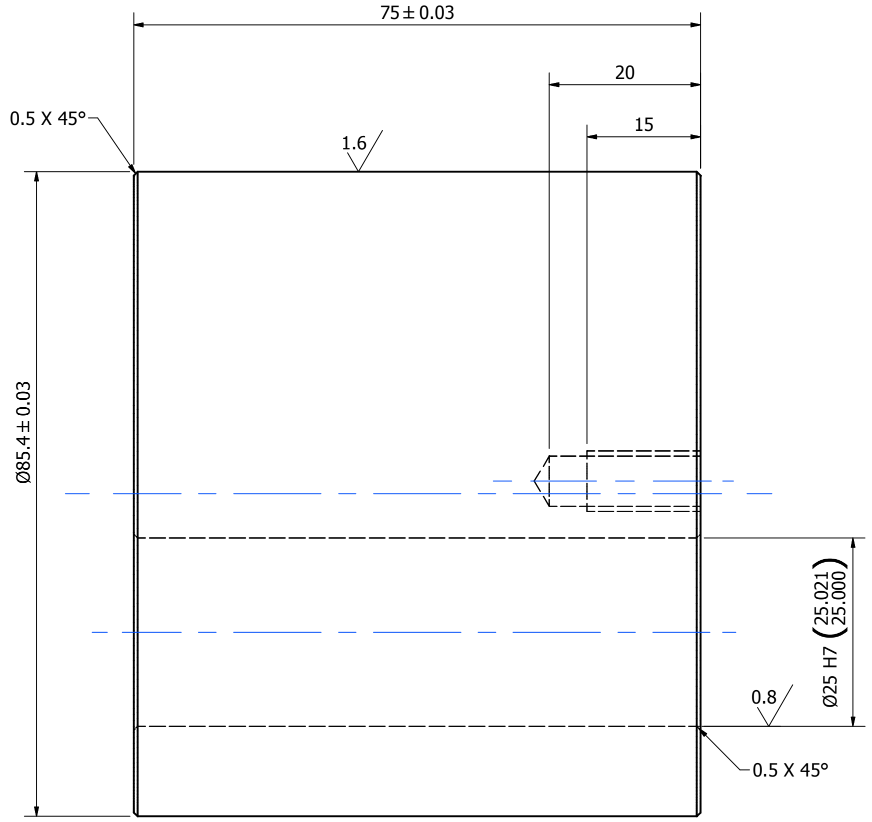
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

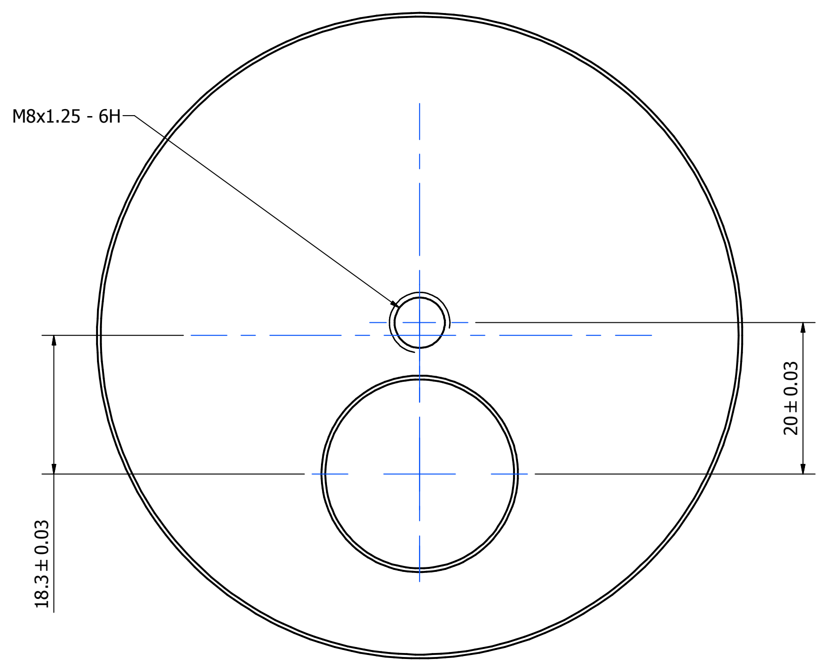
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

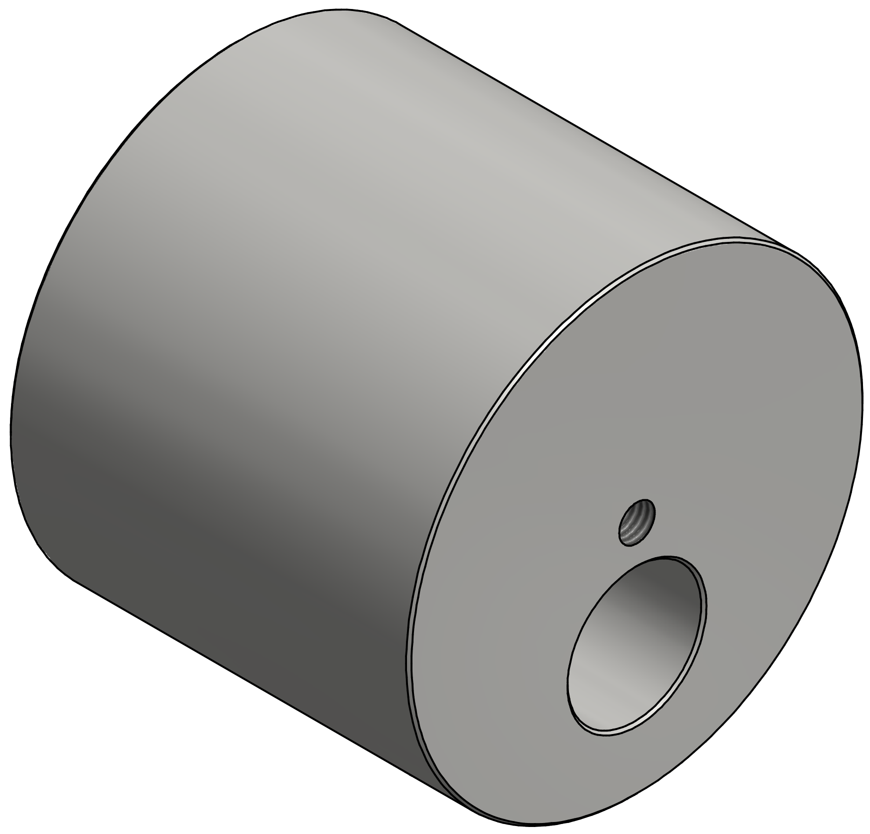


- SIDE VIEW -



4111-M009 - DETAIL -

SCALE: 2 : 1



4111-M009 - ISOMETRIC VIEW -

SCALE: 2 : 1

4111-M009	2	AS1444 -1020	PRIMARY BALANCE WEIGHT	3.08 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD	DATE: 10/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 10/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	PRIMARY BALANCE WEIGHT
APPROVED	DATE		

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M009	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M009.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M009.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

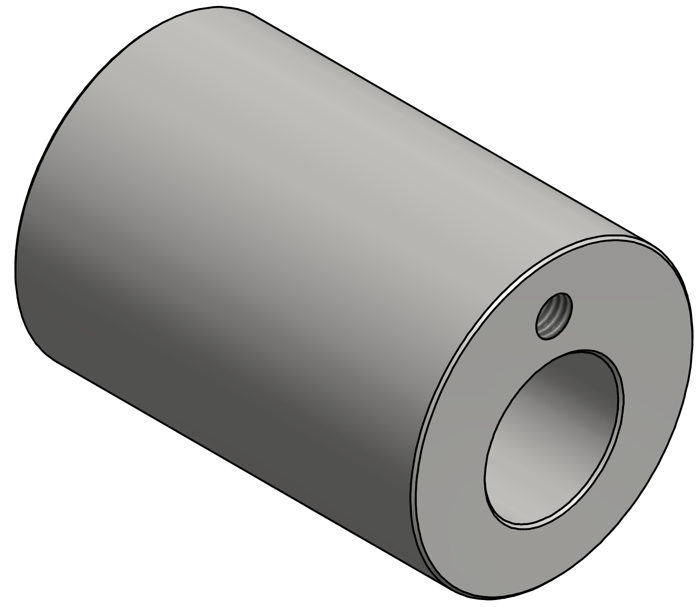
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

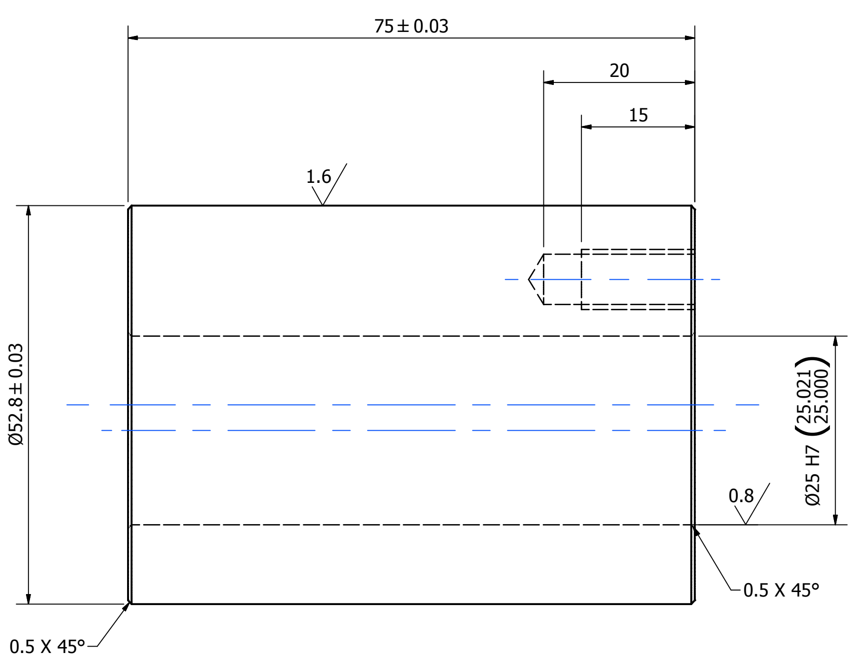
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

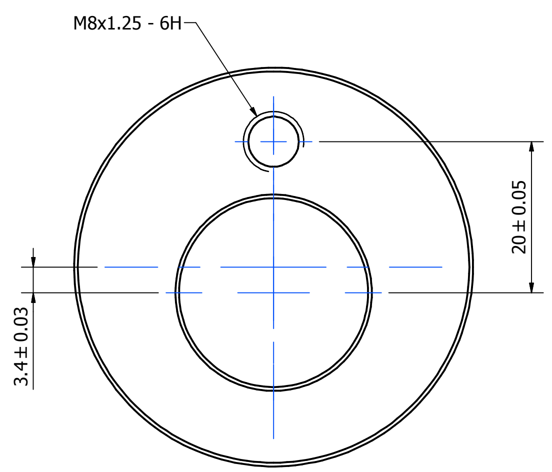
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



4111-M010 - ISOMETRIC VIEW -
SCALE: 2 : 1



- SIDE VIEW -



4111-M010 - DETAIL -
SCALE: 2 : 1

4111-M010	2	AS1444 -1020	SECONDARY BALANCE WEIGHT	1 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	08/09/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN	DATE
KBD	10/07/2014
DESIGNED: KBD	10/07/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: SECONDARY COUNTER BALANCE SHAFT DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M010	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M010.ipt
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DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

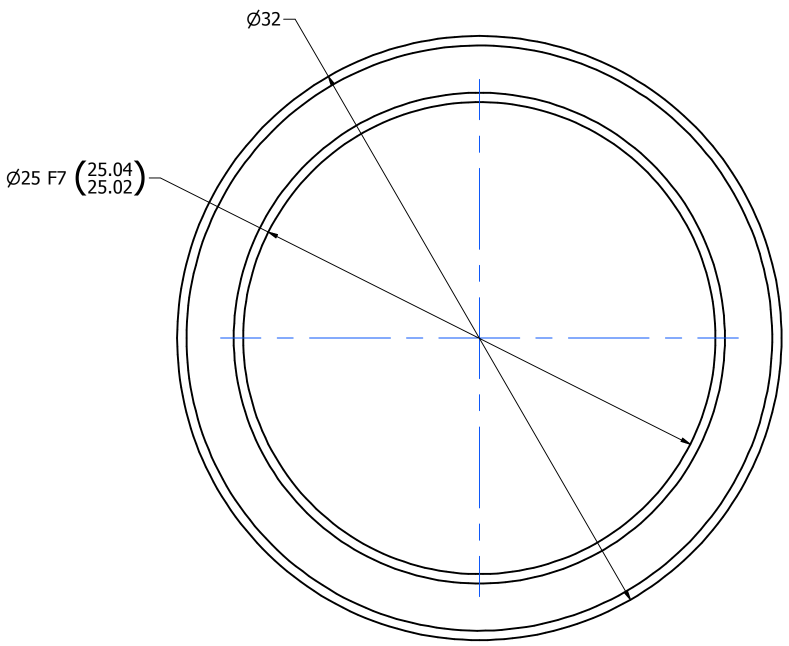
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

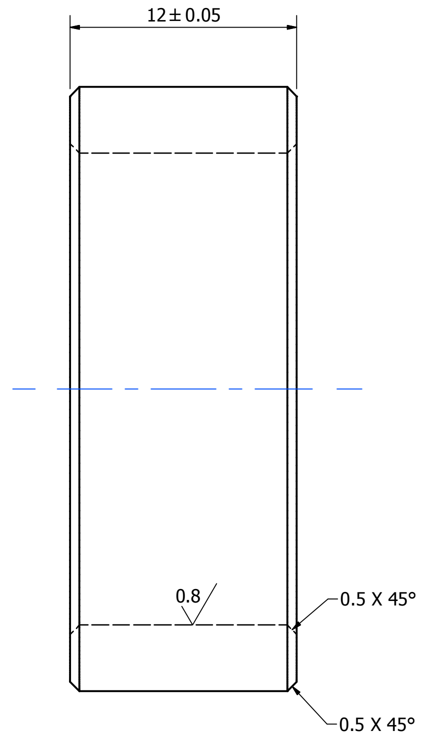
4111-M011 - ISOMETRIC VIEW -

SCALE: 5 : 1



4111-M011 - DETAIL -

SCALE: 5 : 1



- SIDE VIEW -

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M011	4	AS1444 -1020	COUNTER BALANCE SHAFT SPACER LONG	0 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN	DATE
KBD	12/07/2014

DESIGNED	DATE
KBD	12/07/2014

CHECKED | DATE

APPROVED | DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: COUNTER BALANCE SHAFT SPACER

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M011	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M011.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M011.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

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ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

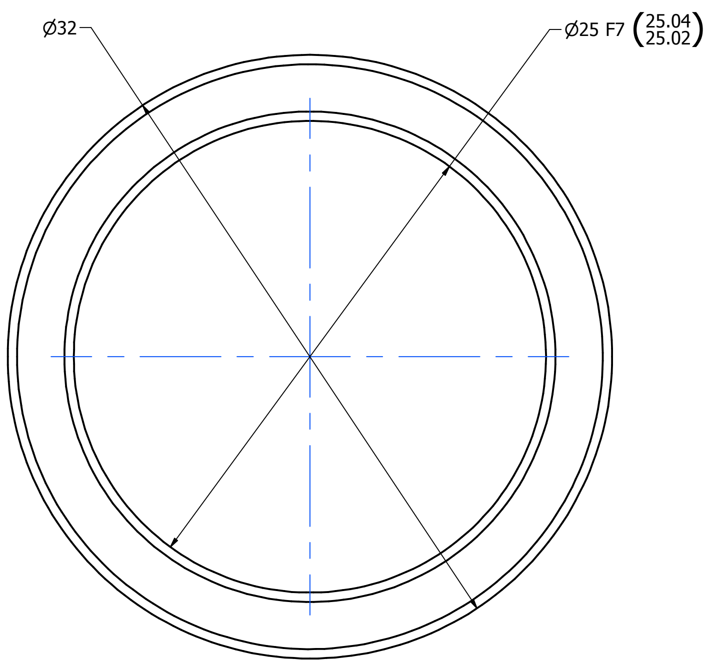
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

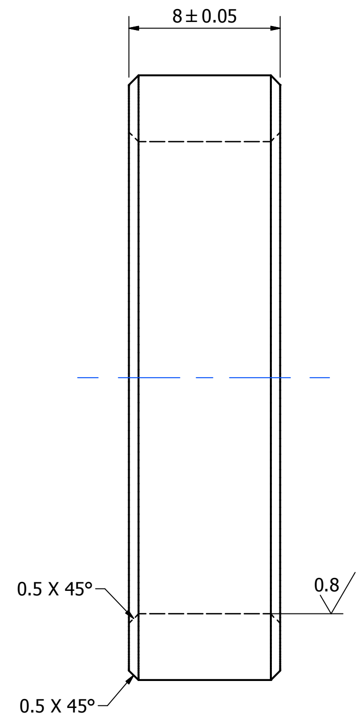
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



4111-M012 - DETAIL -

SCALE: 5 : 1



- SIDE VIEW -

4111-M012 - ISOMETRIC VIEW -

SCALE: 5 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M012	4	AS1444 -1020	COUNTER BALANCE SHAFT SPACER SHORT	0.02 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN KBD	DATE 12/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 12/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	COUNTER BALANCE SHAFT SPACER
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-M012

1 OF 1 SIZE A1 REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M012.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M012.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

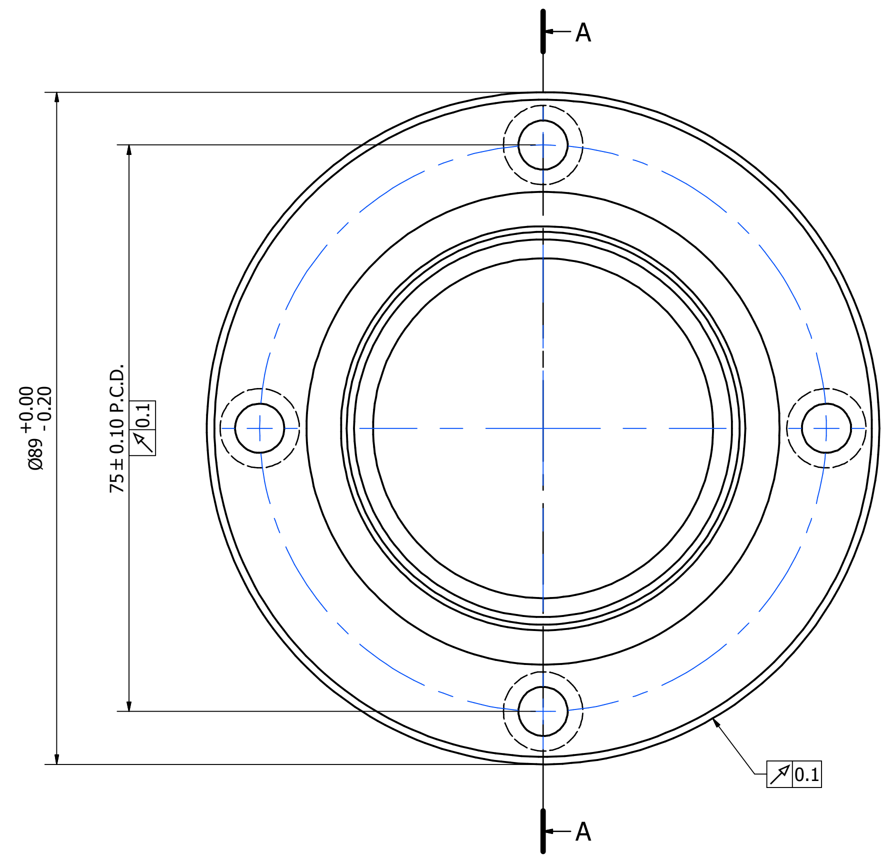
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

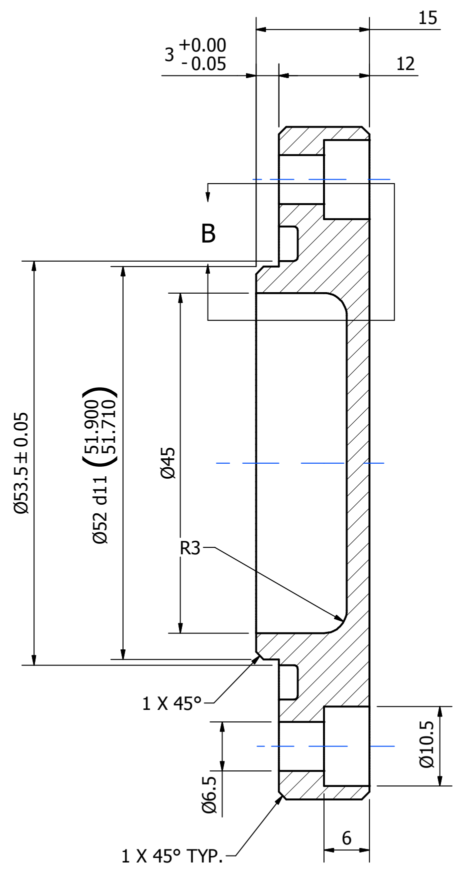
4111-M013 - ISOMETRIC VIEW -

SCALE: 2 : 1



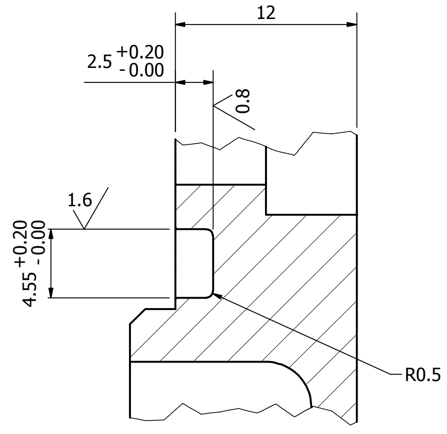
4111-M013 - DETAIL -

SCALE: 2 : 1



SECTION A-A

SCALE: 2 : 1



DETAIL B

SCALE: 4 : 1

4111-M013	1	AS1865 -2011T6	BEARING CAP	0.2 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/14	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN: KBD	DATE: 10/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 10/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION: BEARING CAP DETAIL
APPROVED	DATE	JOB No.: ENG4111/2; DRAWING No.: 4111-M013; 1 OF 1; SIZE: A1; REV.: 0

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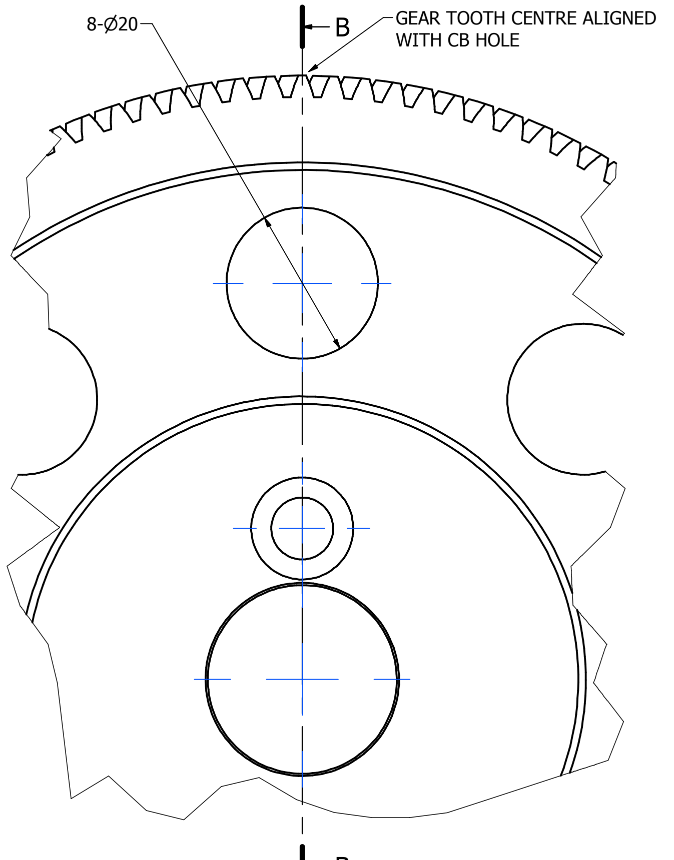
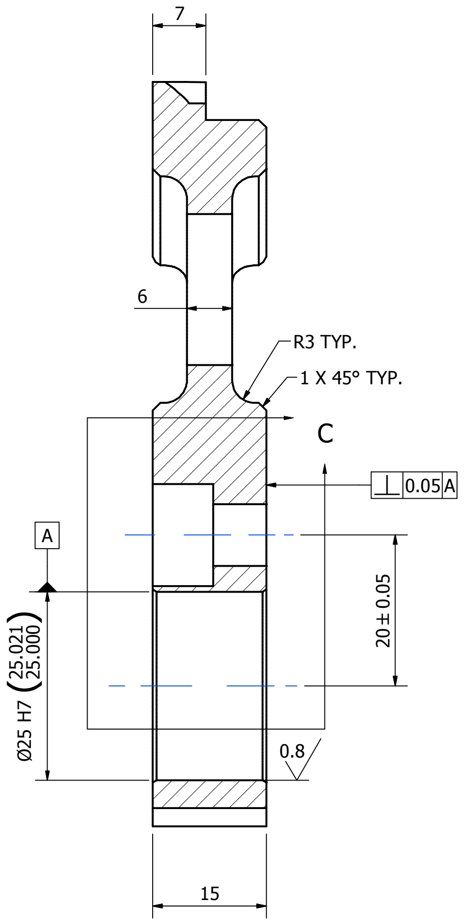
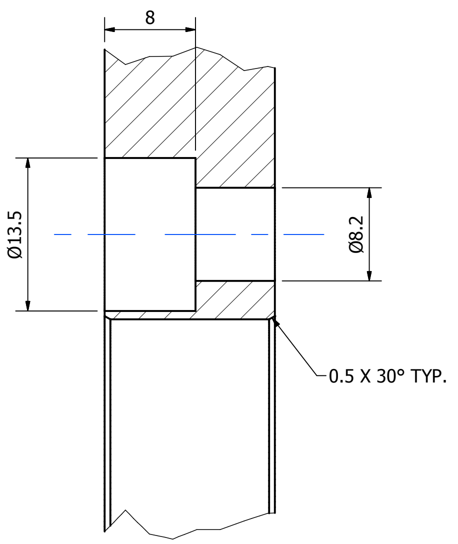
DO NOT SCALE - IF IN DOUBT ASK.



DETAIL A
SCALE: 2 : 1



SECTION B-B
SCALE: 2 : 1

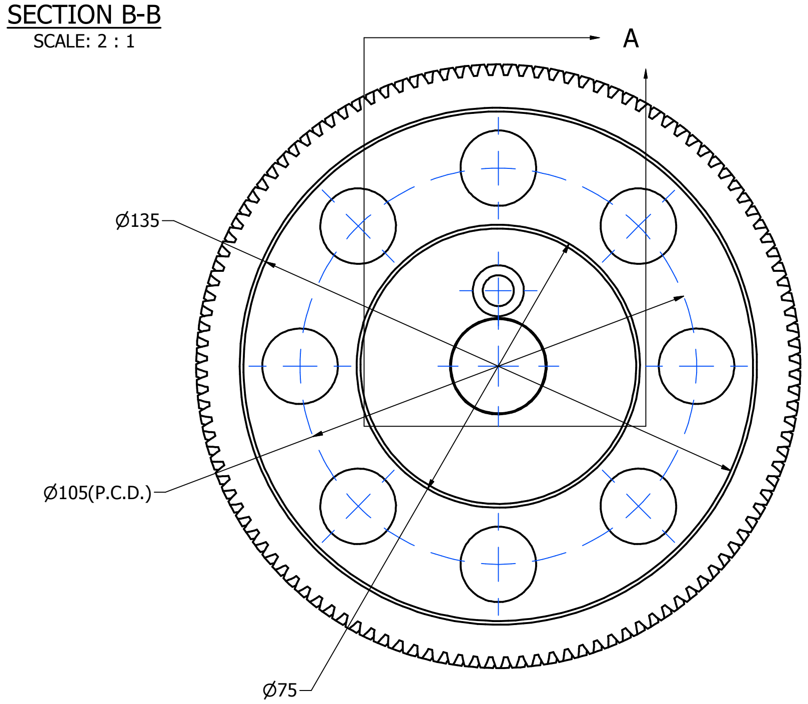


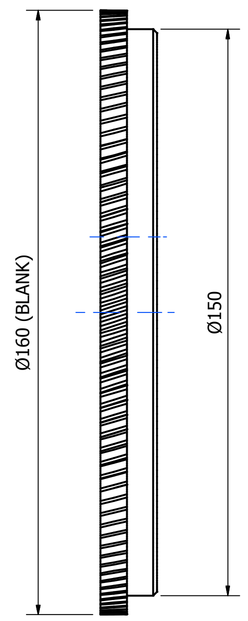
DETAIL C
SCALE: 3 : 1

4111-M018 - ISOMETRIC VIEW -
SCALE: 1 : 1

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- CASE HARDEN AS SPECIFIED IN 4111-A001 ("CH" CONDITION)
- HEAT SURFACE (**DO NOT SOAK**) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

IMPORTANT:
-ASSEMBLE GEAR CLUSTER AND TEST FOR EXCESSIVE BACKLASH OR POOR FIT BEFORE FINAL HEAT TREATMENT.





4111-M018 - DETAIL -
SCALE: 1 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GEAR DATA

No.	DETAIL	VALUE
1	TOOTH FORM	INVOLUTE
2	No. OF TEETH	120
3	MODULE	1.25
4	HELIX ANGLE	17.75 (L.H.)
5	PRESSURE ANGLE	20
6	MATING GEAR CENTRES	157.5mm
7	FINAL HARDNESS	55 HRC
8	TOOTH SURFACE FINISH	0.4 Ra
9	MATING GEAR	120T

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M018	1	AS1444 EN36A	PRIMARY COUNTER BALANCE SHAFT GEAR	1.3 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETRES UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

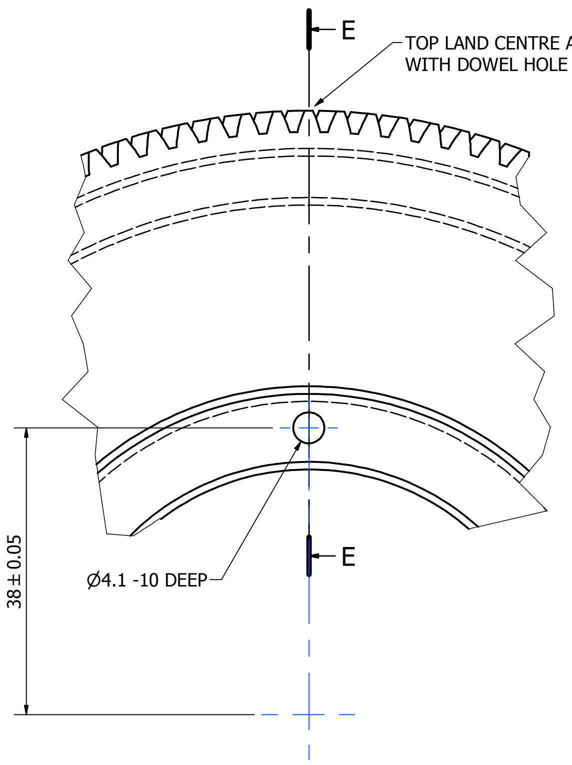
THE UNIVERSITY OF SOUTHERN QUEENSLAND.

		DATE	
DRAWN	KBD	4/08/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	4/08/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED			DRAWING DESCRIPTION: PRIMARY COUNTER BALANCE SHAFT GEAR DETAIL
APPROVED			

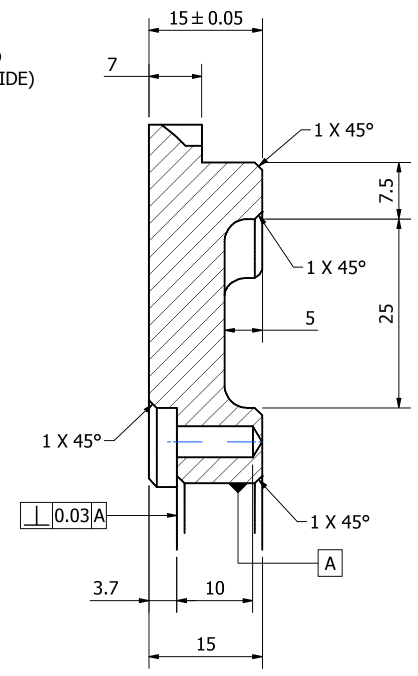
JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M018	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M018.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M018.idw

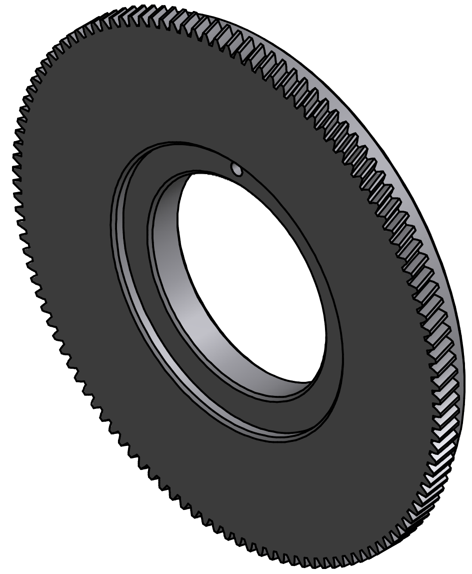
DO NOT SCALE - IF IN DOUBT ASK.



DETAIL D
SCALE: 2 : 1



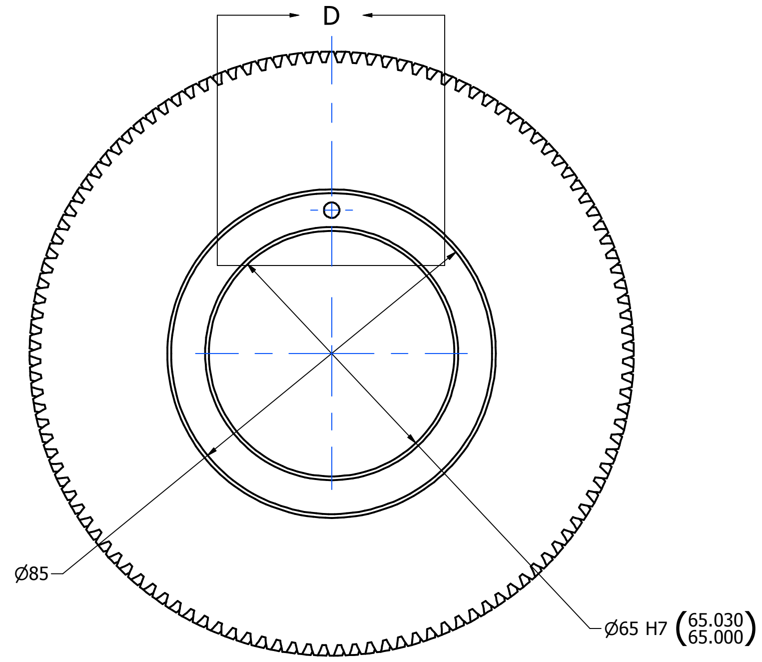
SECTION E-E
SCALE: 2 : 1

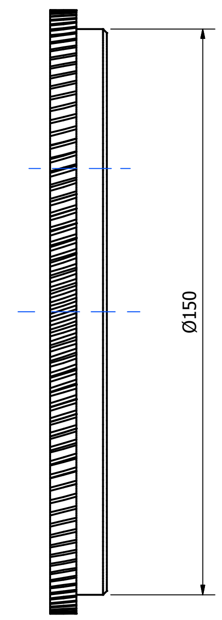


4111-M019 - ISOMETRIC VIEW -
SCALE: 1 : 1

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- CASE HARDEN AS SPECIFIED IN 4111-A001 ("CH" CONDITION)
- HEAT SURFACE (**DO NOT SOAK**) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

IMPORTANT:
-ASSEMBLE GEAR CLUSTER AND TEST FOR EXCESSIVE BACKLASH OR POOR FIT BEFORE FINAL HEAT TREATMENT.





4111-M019 - DETAIL -
SCALE: 1 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GEAR DATA		
No.	DETAIL	VALUE
1	TOOTH FORM	INVOLUTE
2	No. OF TEETH	120
3	MODULE	1.25
4	HELIX ANGLE	17.75 (L.H.)
5	PRESSURE ANGLE	20
6	MATING GEAR CENTRES	157.51 mm
7	FINAL HARDNESS	55 HRC
8	TOOTH SURFACE FINISH	0.4 Ra
9	MATING GEAR	120 T

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M019	1	AS1444 EN36A	CRANKSHAFT DRIVE PINION	1.4 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY			
REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

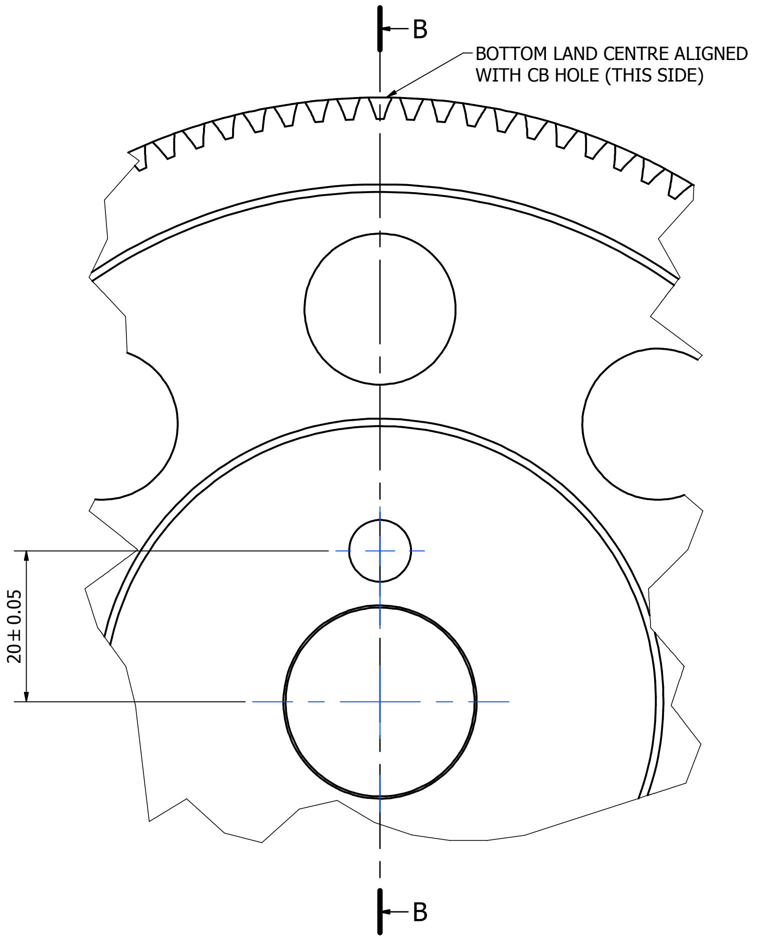
THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD DATE: 23/07/2014
DESIGNED: KBD DATE: 23/07/2014
CHECKED: DATE:
APPROVED: DATE:

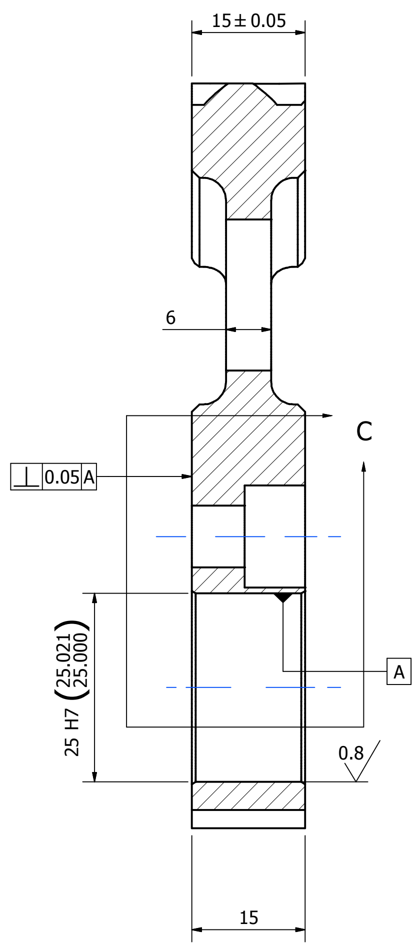
CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: CRANKSHAFT PINION DETAIL
JOB No.: ENG4111/2
DRAWING No.: 4111-M019 1 OF 1
SIZE: A1
REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M019.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M019.idw

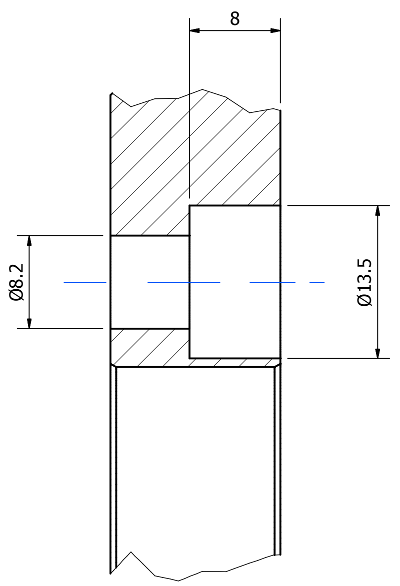
DO NOT SCALE - IF IN DOUBT ASK.



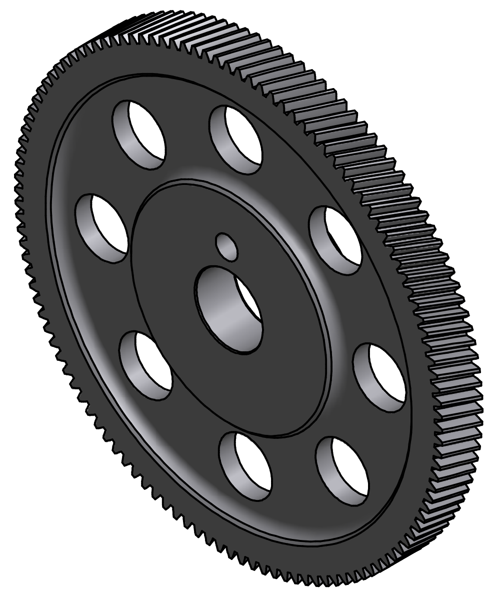
DETAIL A
SCALE: 2 : 1



SECTION B-B
SCALE: 2 : 1



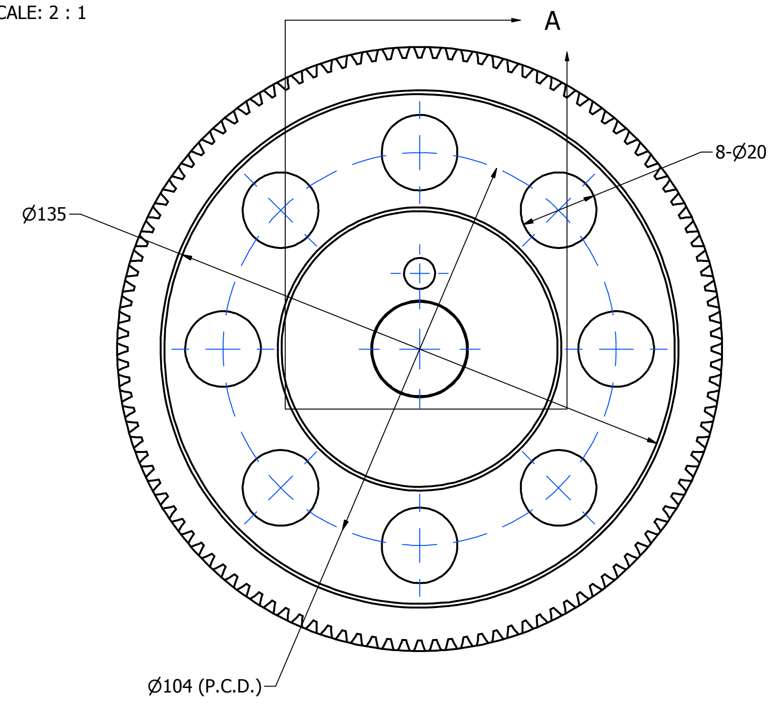
DETAIL C
SCALE: 3 : 1



4111-M021 - ISOMETRIC VIEW -
SCALE: 1 : 1

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- CASE HARDEN AS SPECIFIED IN 4111-A001 ("CH" CONDITION)
- HEAT SURFACE (**DO NOT SOAK**) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

IMPORTANT:
-ASSEMBLE GEAR CLUSTER AND TEST FOR EXCESSIVE BACKLASH OR POOR FIT BEFORE FINAL HEAT TREATMENT.



4111-M021 - DETAIL -
SCALE: 1 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GEAR DATA

No.	DETAIL	VALUE
1	TOOTH FORM	INVOLUTE
2	No. OF TEETH	120
3	MODULE	1.25
4	HELIX ANGLE	17.75 (R.H.)
5	PRESSURE ANGLE	20
6	MATING GEAR CENTRES	157.51mm
7	FINAL HARDNESS	55 HRC
8	TOOTH SURFACE FINISH	0.4 Ra
9	MATING GEAR	120 T

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M021	1	AS1444 EN36A	PRIMARY COUNTER BALANCE SHAFT GEAR	1.4 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

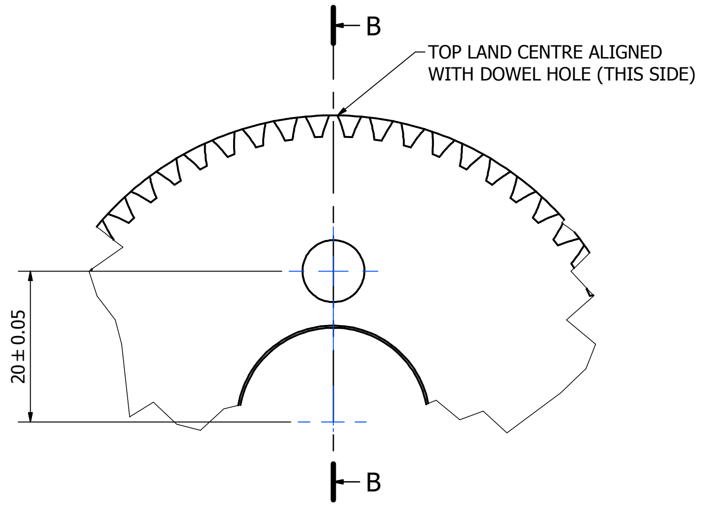
SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

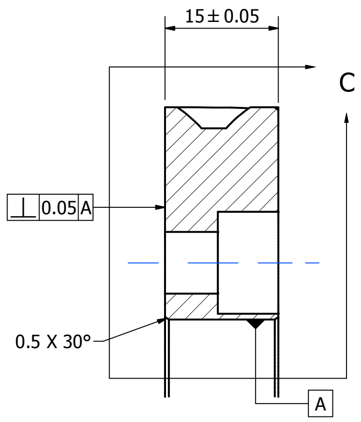
DRAWN: KBD	DATE: 23/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 23/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	PRIMARY COUNTER BALANCE SHAFT GEAR DETAIL
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-M021 1 OF 1 SIZE A1 REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M021.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M021.idw

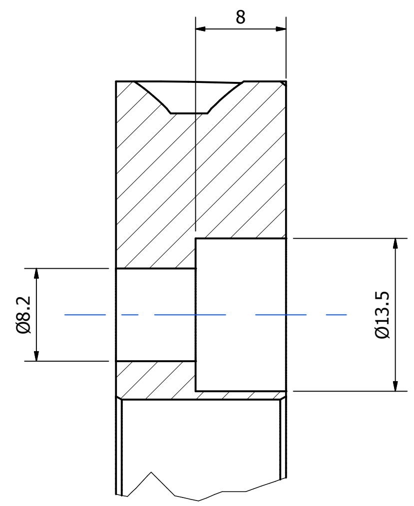
DO NOT SCALE - IF IN DOUBT ASK.



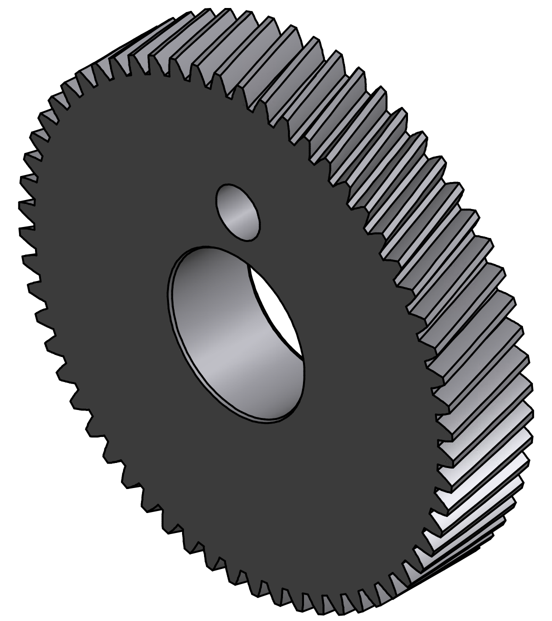
DETAIL A
SCALE: 2 : 1



SECTION B-B
SCALE: 2 : 1



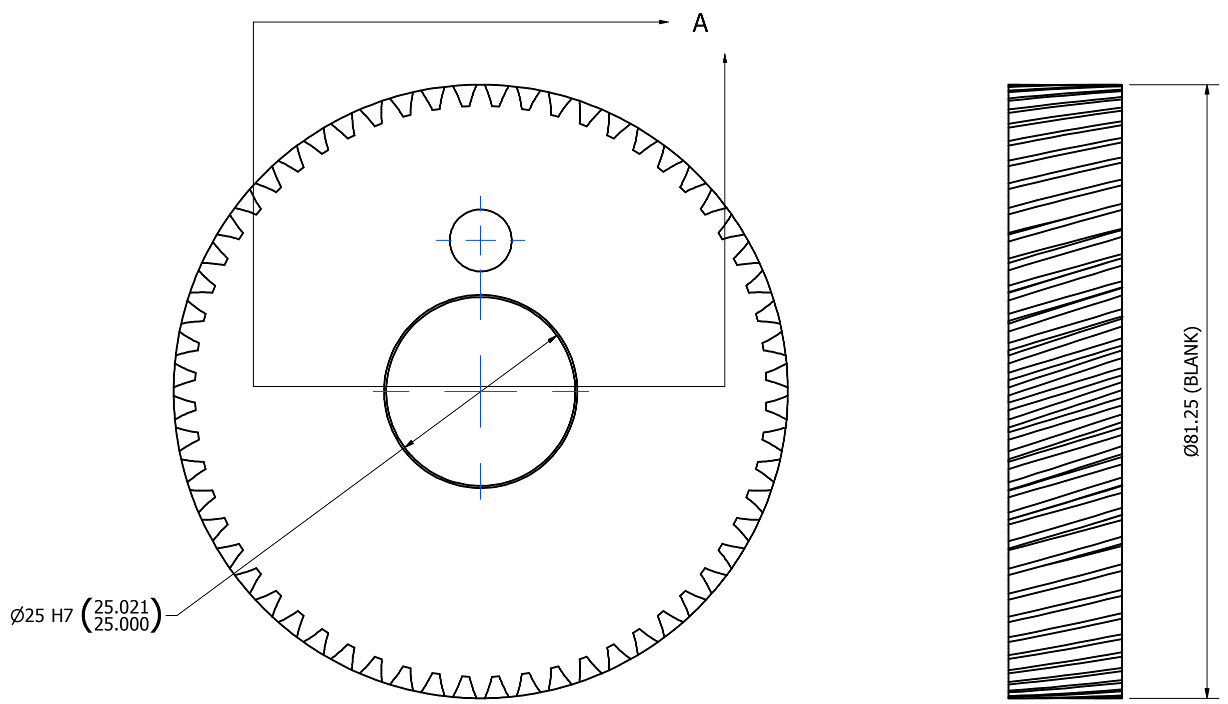
DETAIL C
SCALE: 3 : 1



4111-M023 - ISOMETRIC VIEW -
SCALE: 2 : 1

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- CASE HARDEN AS SPECIFIED IN 4111-A001 ("CH" CONDITION)
- HEAT SURFACE (**DO NOT SOAK**) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

IMPORTANT:
-ASSEMBLE GEAR CLUSTER AND TEST FOR EXCESSIVE BACKLASH OR POOR FIT BEFORE FINAL HEAT TREATMENT.



4111-M023 - DETAIL -
SCALE: 2 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GEAR DATA

No.	DETAIL	VALUE
1	TOOTH FORM	INVOLUTE
2	No. OF TEETH	120
3	MODULE	1.25
4	HELIX ANGLE	17.75 (L.H.)
5	PRESSURE ANGLE	20
6	MATING GEAR CENTRES	118.12
7	FINAL HARDNESS	55 HRC
8	TOOTH SURFACE FINISH	0.4 Ra
9	MATING GEAR	120 T

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M023	1	AS1444 EN36A	SECONDARY COUNTER BALANCE SHAFT GEAR	0.5 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm ...... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

DRAWN: KBD DATE: 23/07/2014
DESIGNED: KBD DATE: 23/07/2014
CHECKED: DATE:
APPROVED: DATE:

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: SECONDARY COUNTER BALANCE SHAFT GEAR

JOB No.: ENG4111/2 DRAWING No.: 4111-M023 1 OF 1 SIZE: A1 REV.: 0

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

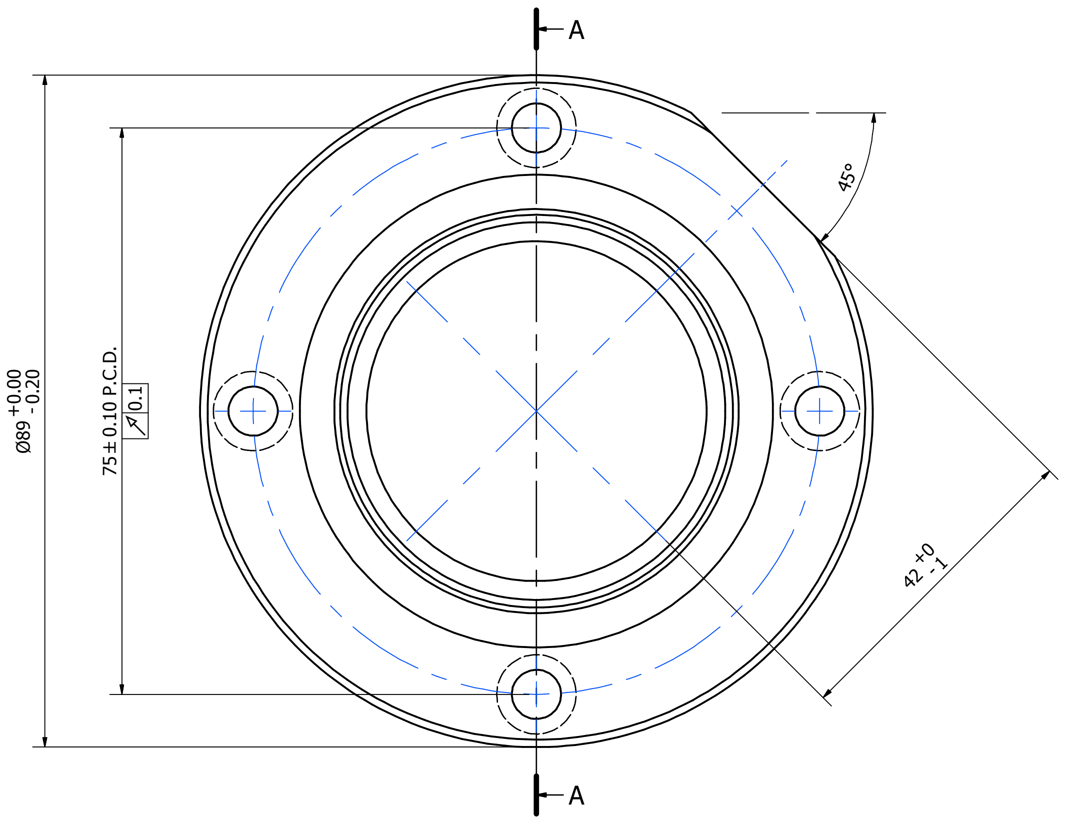
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

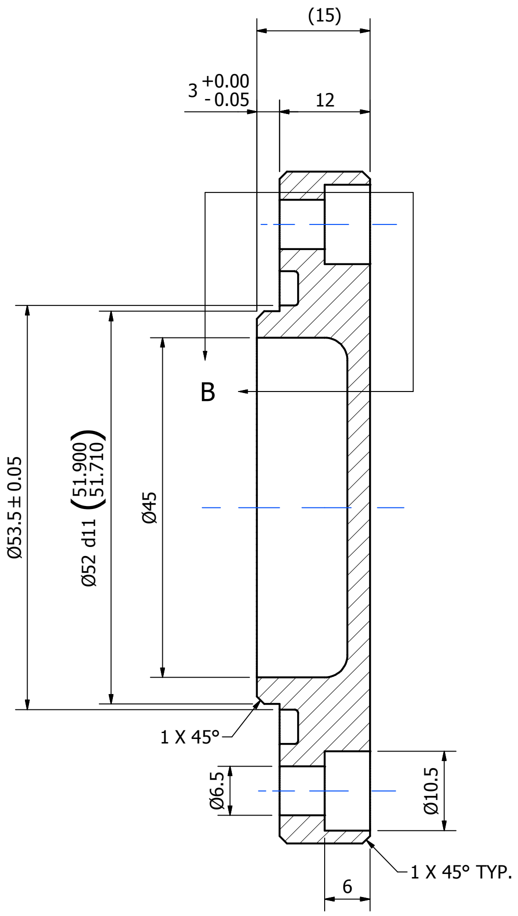
4111-M025 - ISOMETRIC VIEW -

SCALE: 2 : 1



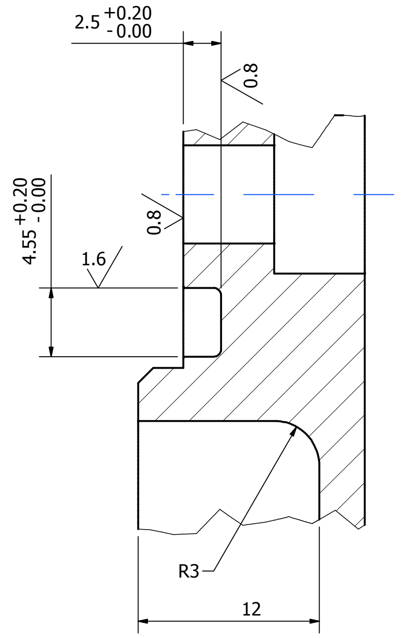
4111-M025 - DETAIL -

SCALE: 2 : 1



SECTION A-A

SCALE: 2 : 1



DETAIL B

SCALE: 4 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M025	1	6061-T6	BEARING CAP (PRIMARY, FLYWHEEL SIDE)	0.2 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/14	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	
DRAWN	KBD	DATE 9/08/2014
DESIGNED	kevin	DATE 9/08/2014
CHECKED		DATE
APPROVED		DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: BEARING CAP (EDGED)

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M025	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M025.ipt
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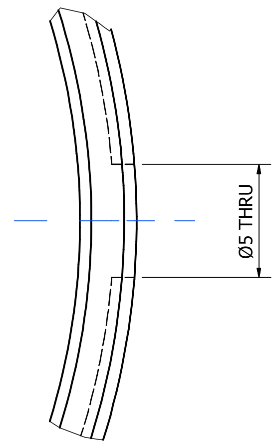
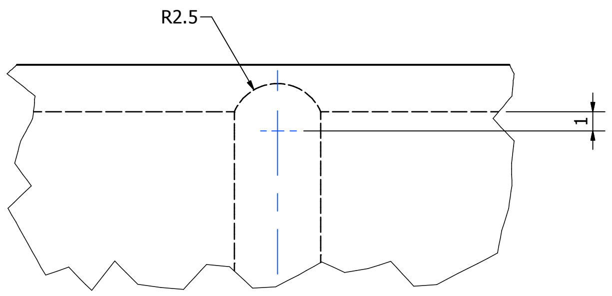
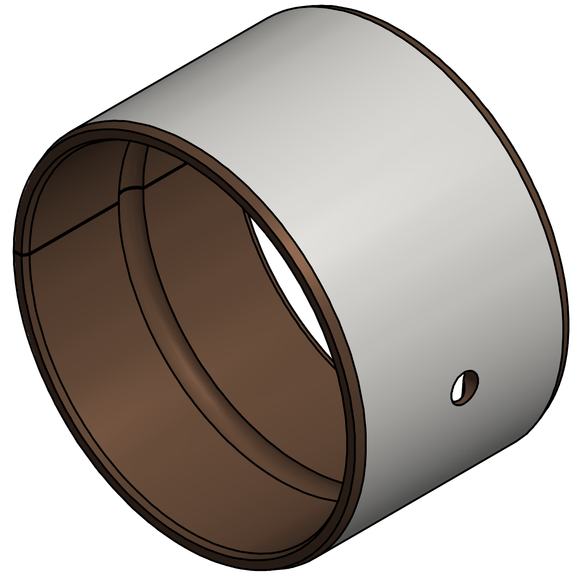
DO NOT SCALE - IF IN DOUBT ASK.



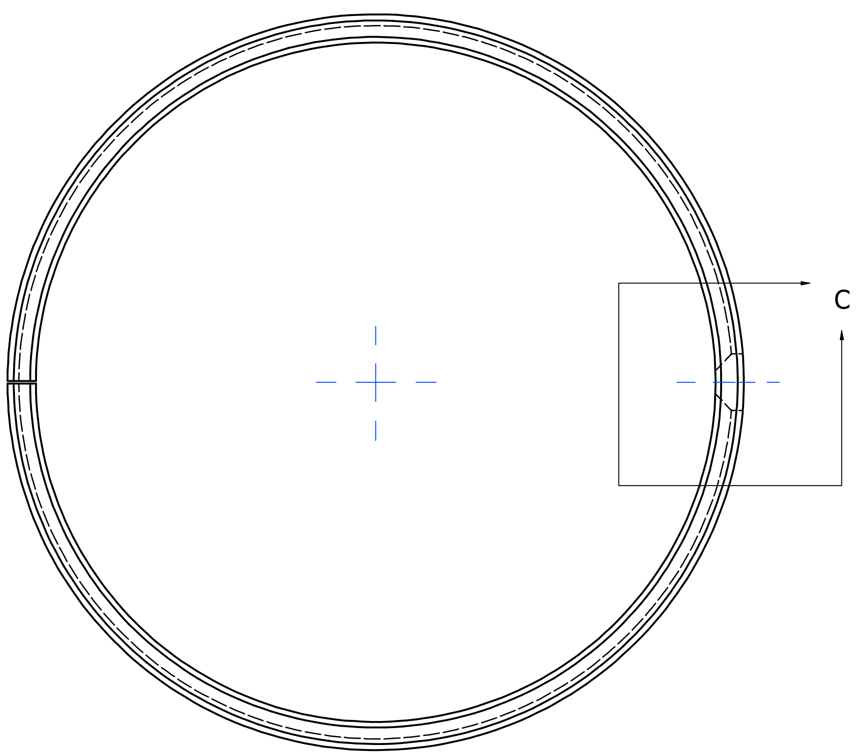
DETAIL C
SCALE: 6 : 1



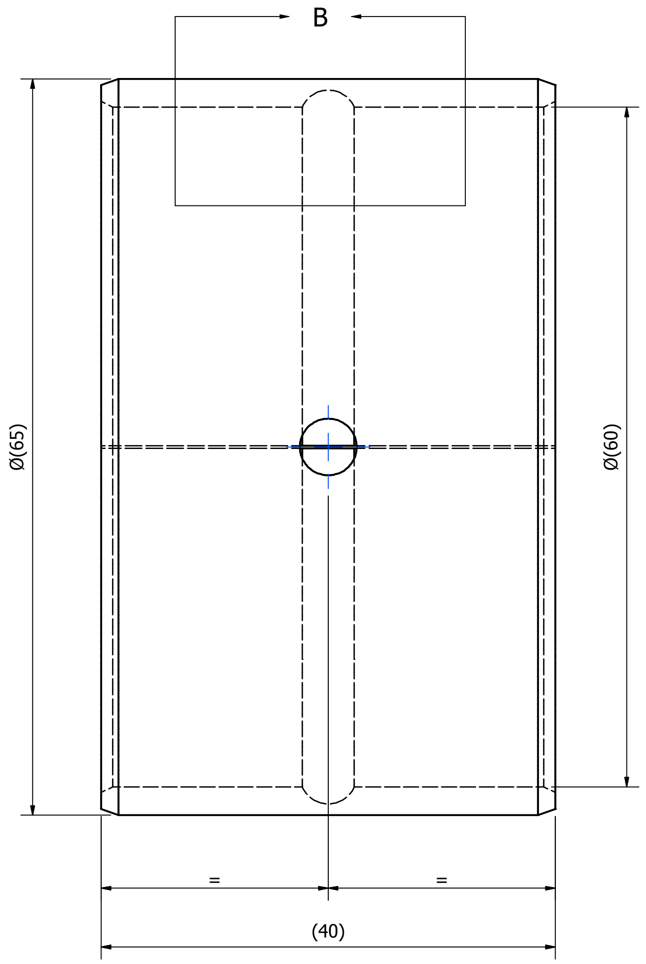
DETAIL B
SCALE: 5 : 1



4111-M026 - ISOMETRIC VIEW -
SCALE: 2 : 1



4111-M026 - DETAIL -
SCALE: 3 : 1



- SIDE VIEW -

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M026	2	DU BEARING	6040DU - LEAD/PTFE BEARING (MODIFIED)	0.05 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

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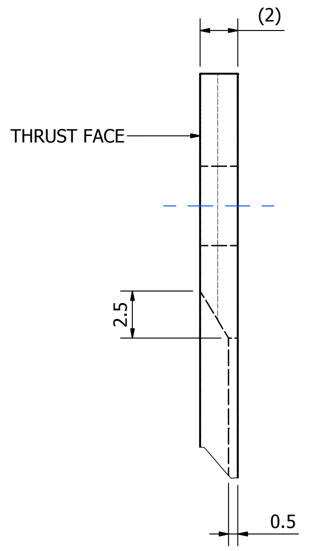
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

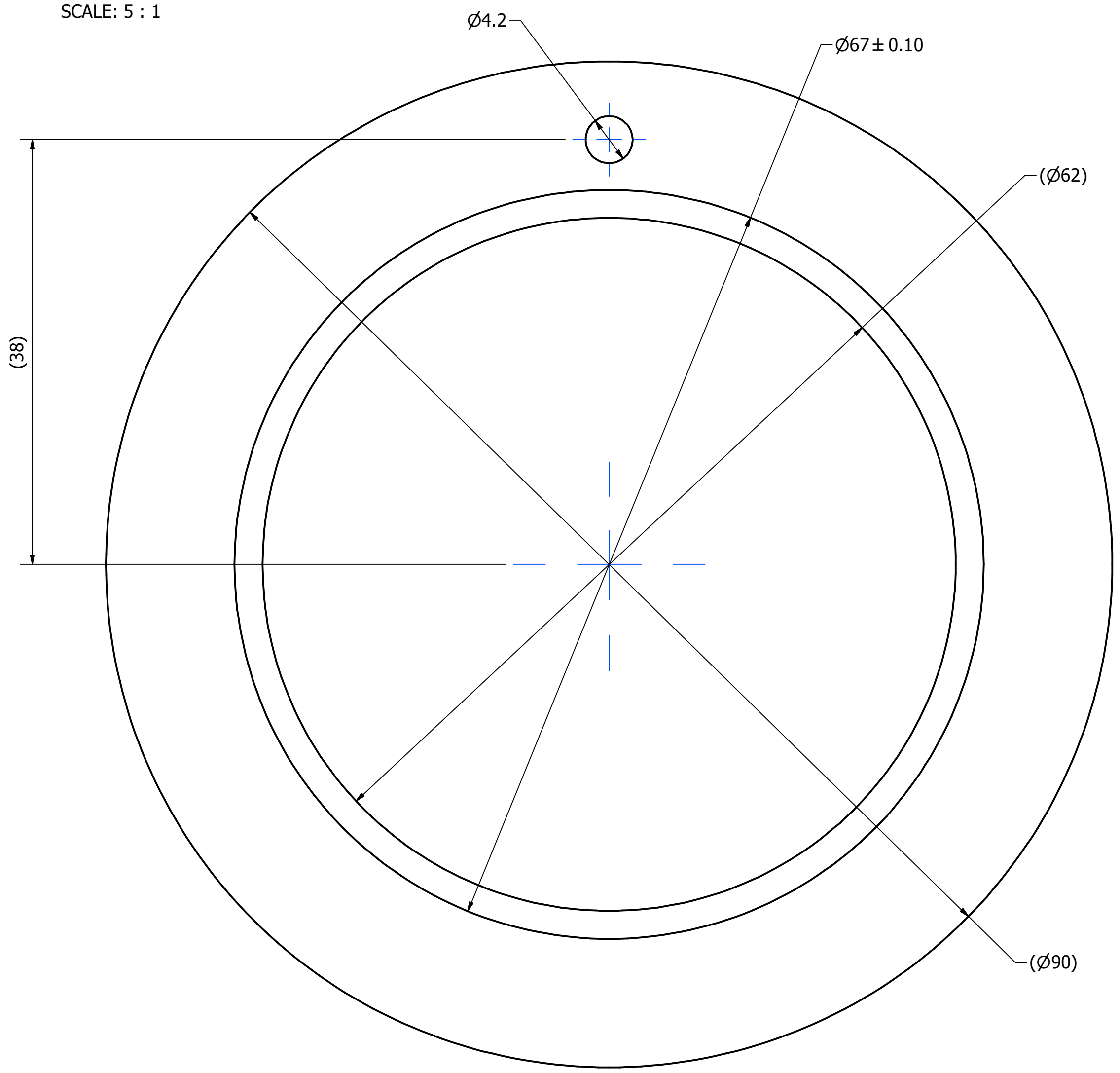
SCALE	AS SHOWN	THE UNIVERSITY OF SOUTHERN QUEENSLAND.	
DRAWN KBD	DATE 12/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 12/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	MAIN BEARING DETAIL
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-M026 1 OF 1 SIZE A1 REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M026.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M026.idw

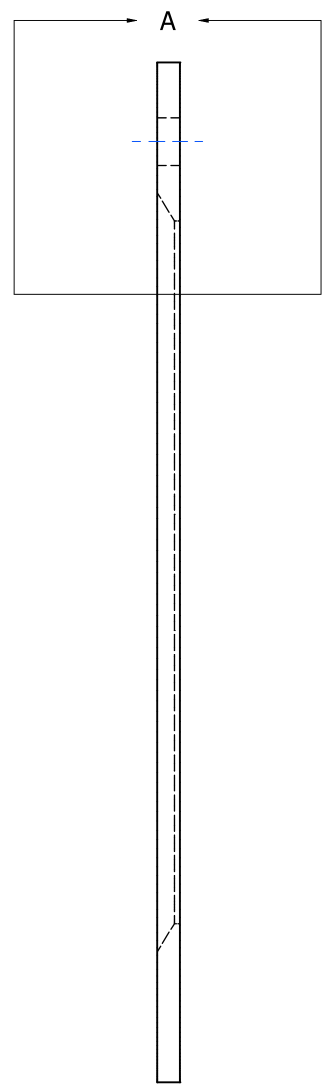
DO NOT SCALE - IF IN DOUBT ASK.



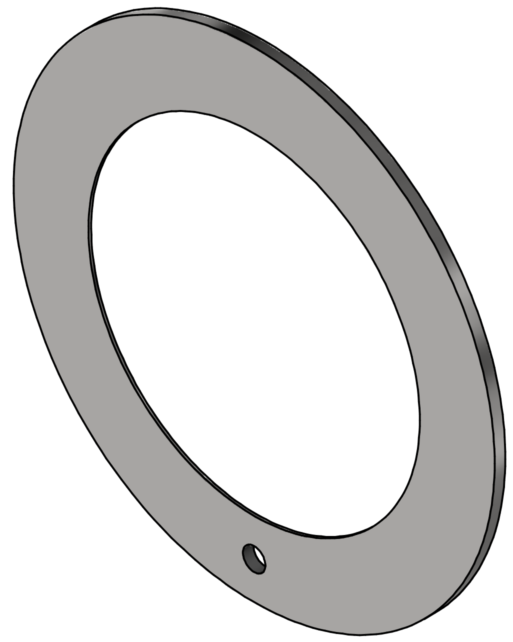
DETAIL A
SCALE: 5 : 1



4111-Z002 - DETAIL -
SCALE: 3 : 1



- END VIEW -



4111-Z002 - ISOMETRIC VIEW -
SCALE: 2 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-Z002	2	DU BEARING	WC60DU - LEAD/PTFE THRUST BEARING (MODIFIED)	0 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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DRAWN	DATE	DESIGNED	DATE	CHECKED	DATE	APPROVED	DATE
KBD	12/07/2014	KBD	12/07/2014				

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: THRUST BEARING DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M027	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-Z002.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M027.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
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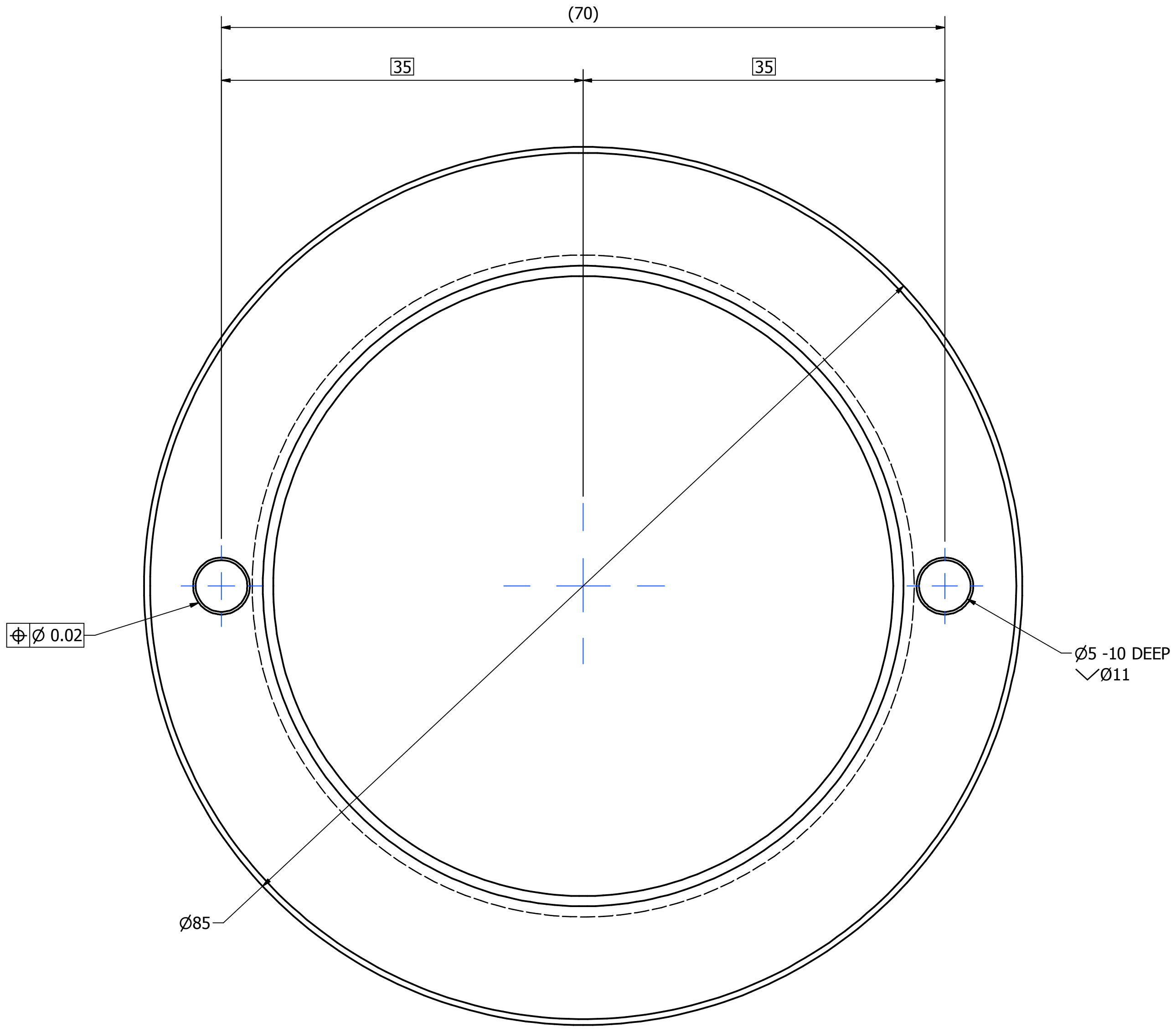
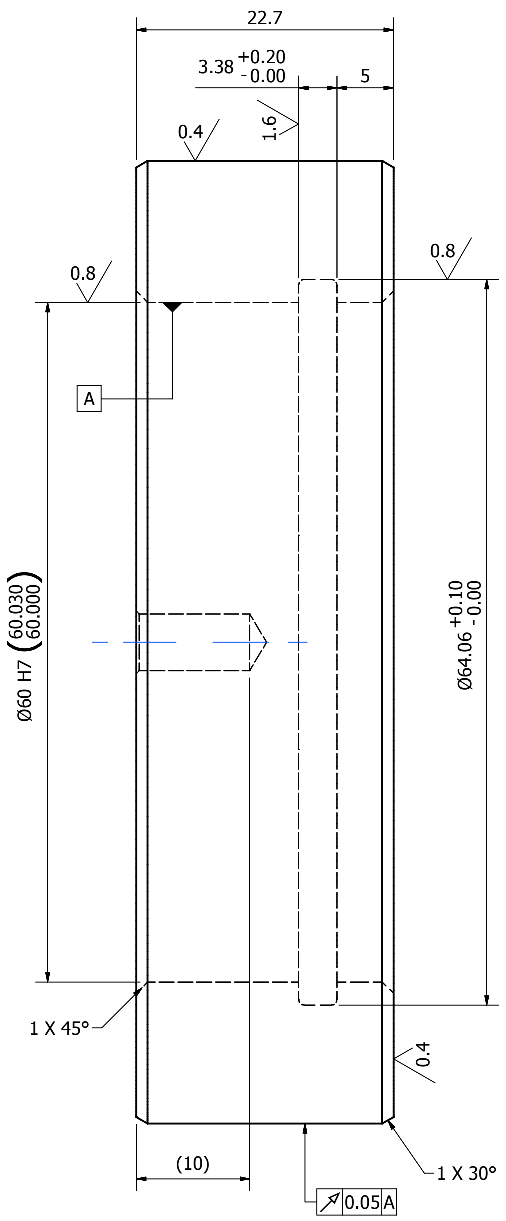
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

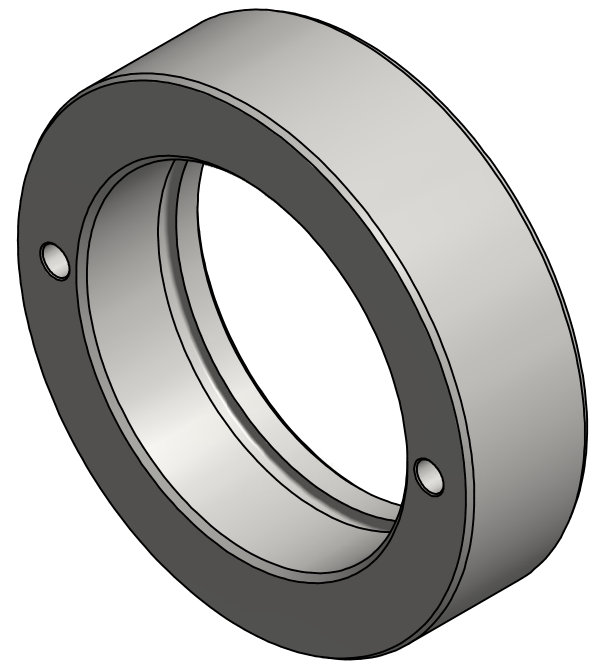


4111-M028 - DETAIL -

SCALE: 3 : 1



- SIDE VIEW -



4111-M028 - ISOMETRIC VIEW -

SCALE: 2 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M028	1	AS1444 -1020	THRUST COLLAR	0.492 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
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PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

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DRAWN	DATE
KBD	12/07/2014
DESIGNED	DATE
KBD	12/07/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: THRUST COLLAR DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M028	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M028.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M028.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

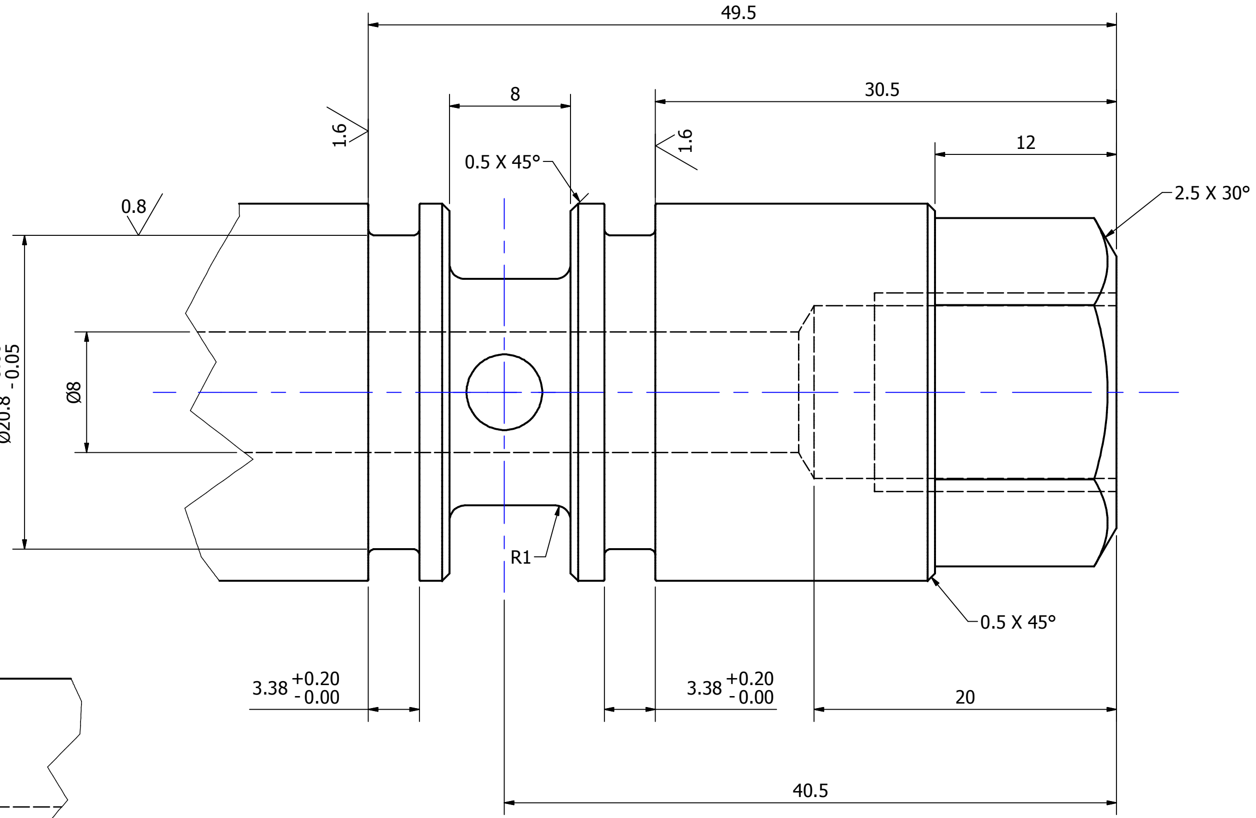
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

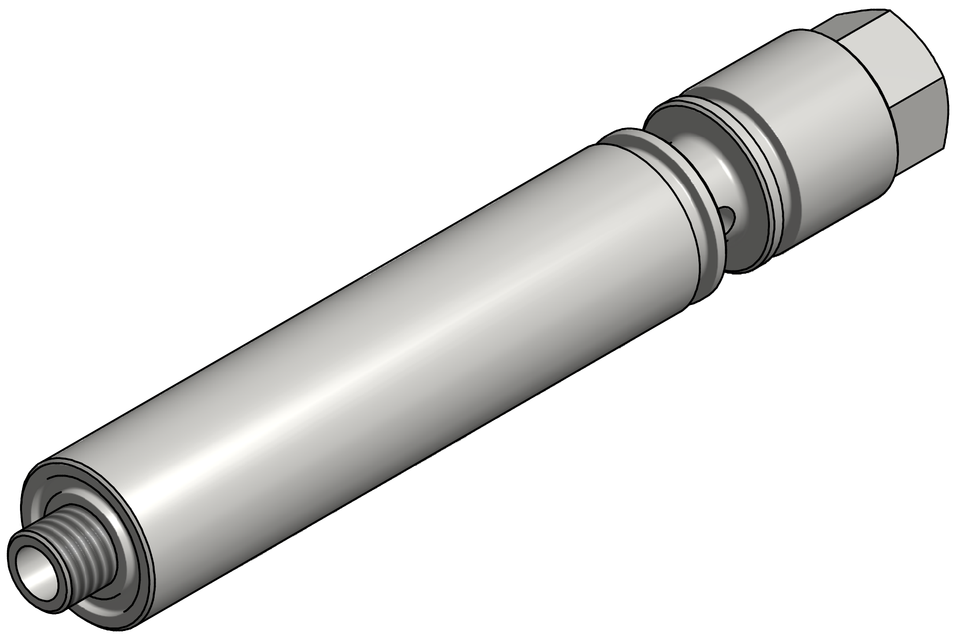
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

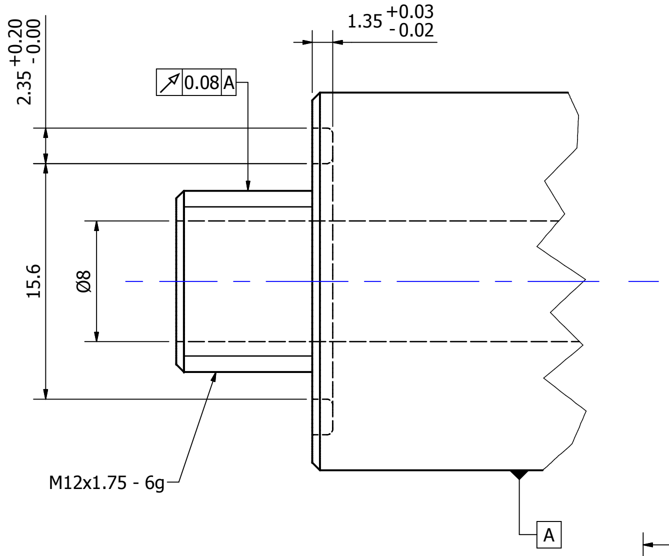
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



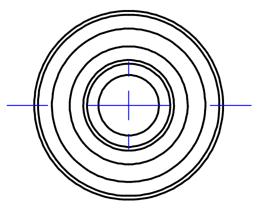
DETAIL A
SCALE: 4 : 1



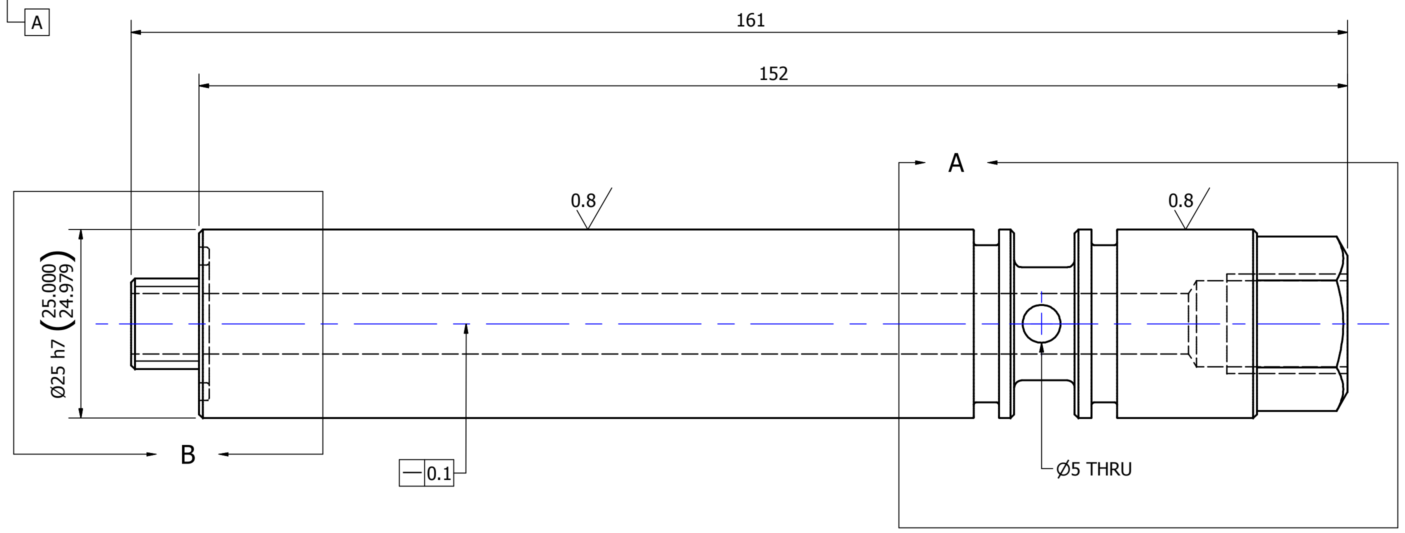
4111-M029 - ISOMETRIC VIEW -
SCALE: 2 : 1



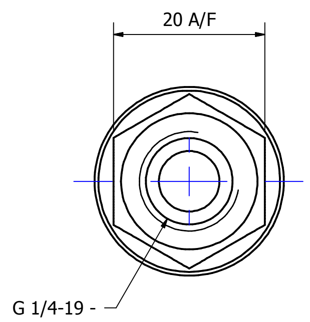
DETAIL B
SCALE: 4 : 1



- END VIEW -



4111-M029 - DETAIL -
SCALE: 2 : 1



- END VIEW -

4111-M029	1	AS1444 -1020	OIL SUPPLY FITTING		0.5 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION		MASS

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
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THE UNIVERSITY OF SOUTHERN QUEENSLAND.

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DESIGNED	KBD	DATE	9/08/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: OIL SUPPLY TUBE

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M029	1 OF 1	A1	0

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D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M029.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITHAS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

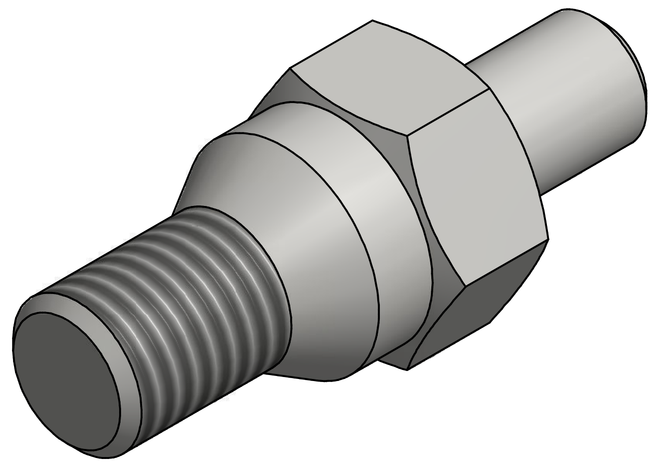
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

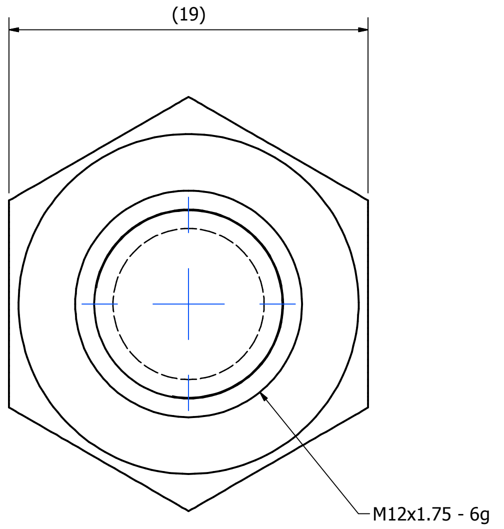


4111-M032 - ISOMETRIC VIEW -

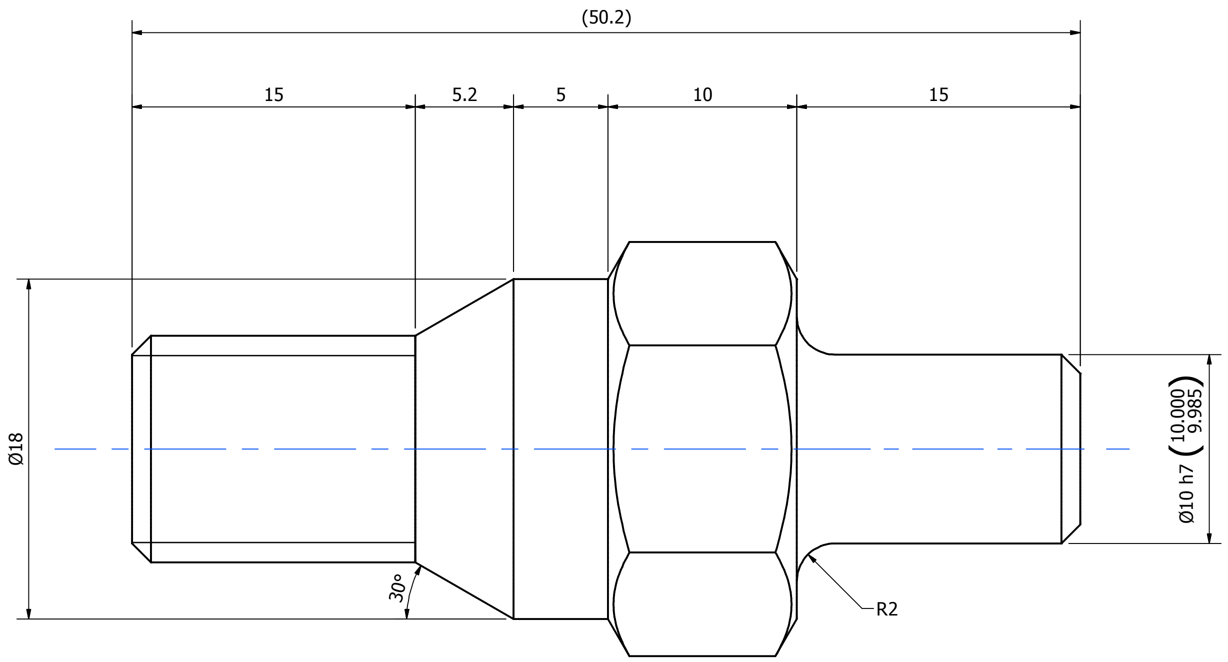
SCALE: 4 : 1

NOTE:

CONFIRM ENCODER SELECTION BEFORE FABRICATING THIS PART

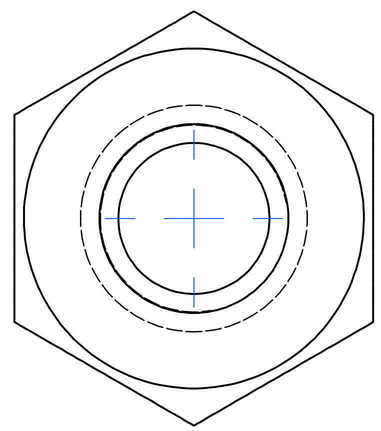


- END VIEW -



4111-M032 - DETAIL -

SCALE: 5 : 1



- END VIEW -

4111-M032	1	AS1444 -4140	ENCODER ATTACHMENT	0.1 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

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PH: +61 7 4631 2100
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EMAIL: study@usq.edu.au

SCALE: AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN: KBD	DATE: 9/08/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 9/08/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED:	DATE:	DRAWING DESCRIPTION: ENCODER SHAFT
APPROVED:	DATE:	JOB No.: ENG4111/2 · DRAWING No.: 4111-M032 · 1 OF 1 · SIZE: A1 · REV.: 0

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D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M032.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

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HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
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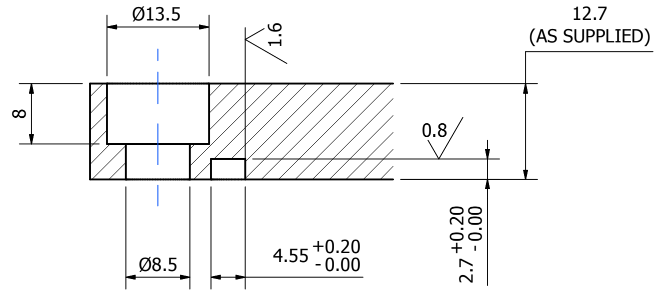
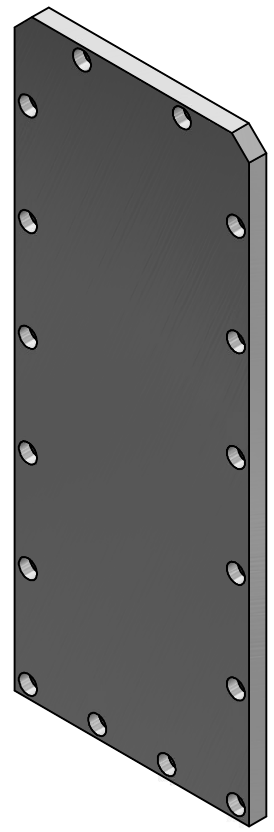
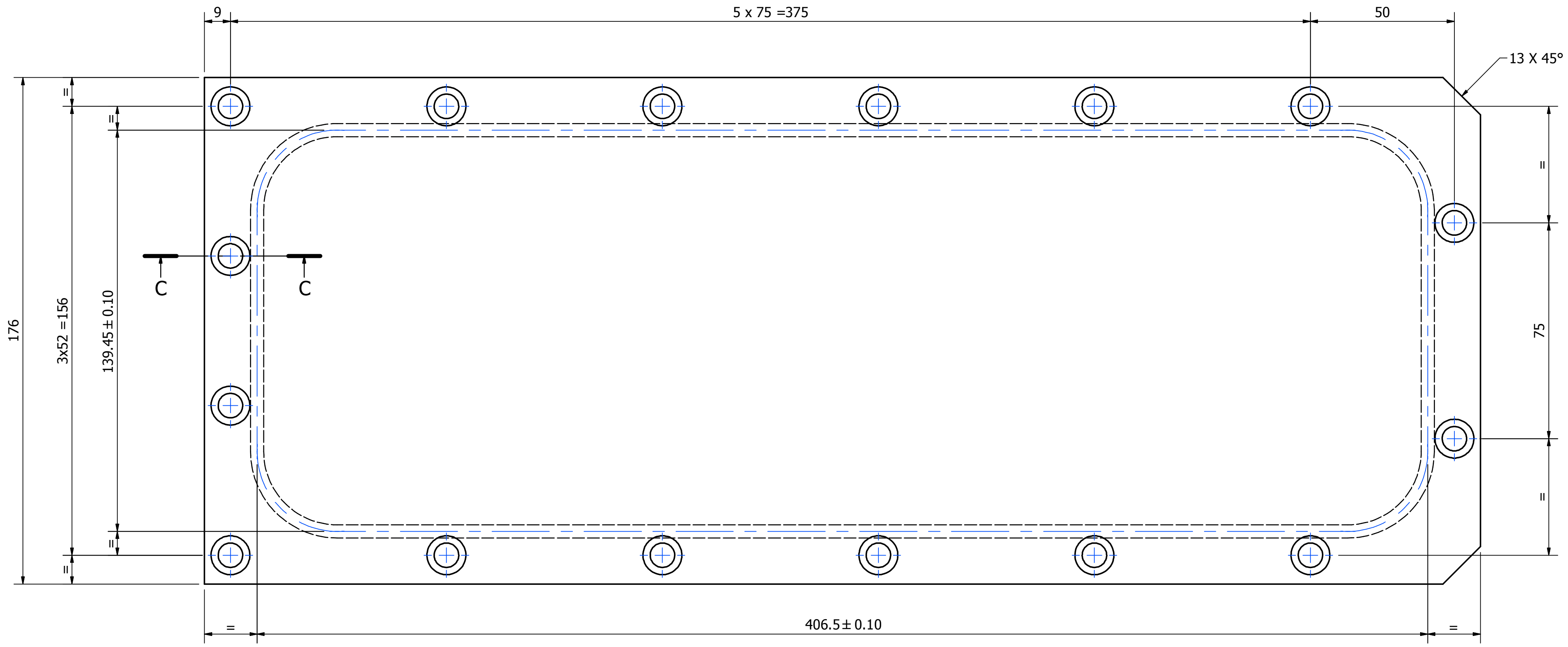
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



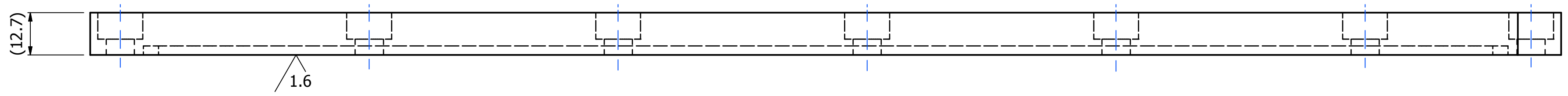
SECTION C-C
SCALE: 2 : 1



4111-M033 - ISOMETRIC VIEW -
SCALE: 1 : 2



4111-M033 - DETAIL -
SCALE: 1 : 1



- SIDE VIEW -

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS	KEYWORDS
4111-M033	AS1734 6061T6	12.7 mm	CRANKCASE SIDE COVER	1	2.6 kg	4111-M033

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm ..... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	08/08/14	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	
DRAWN	KBD	DATE 10/07/2014
DESIGNED	KBD	DATE 10/07/2014
CHECKED		DATE
APPROVED		DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: SIDE COVER DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M033	1 OF 1	A1	0

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

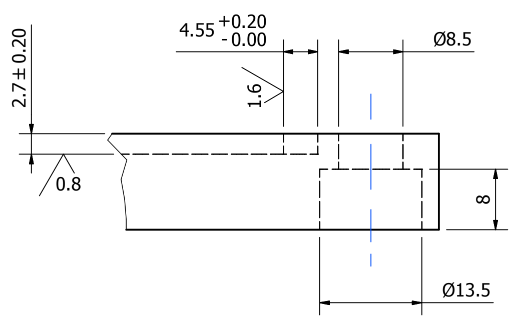
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

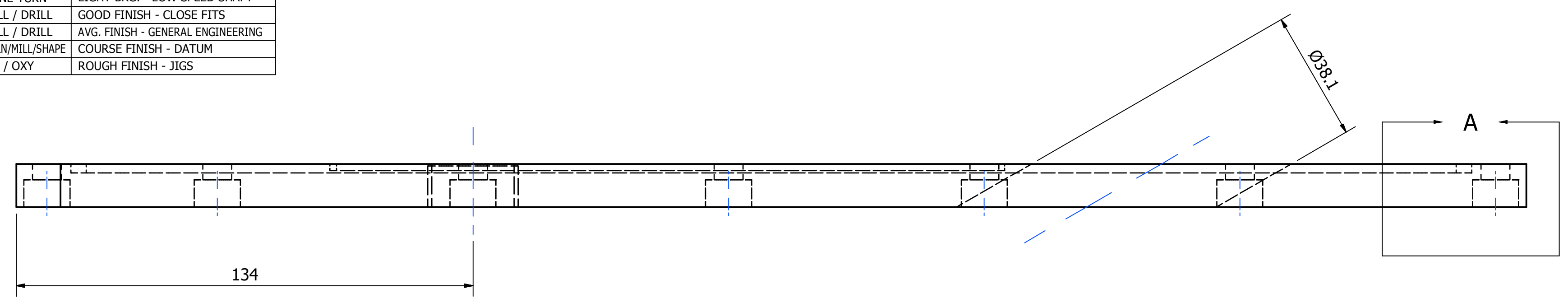
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

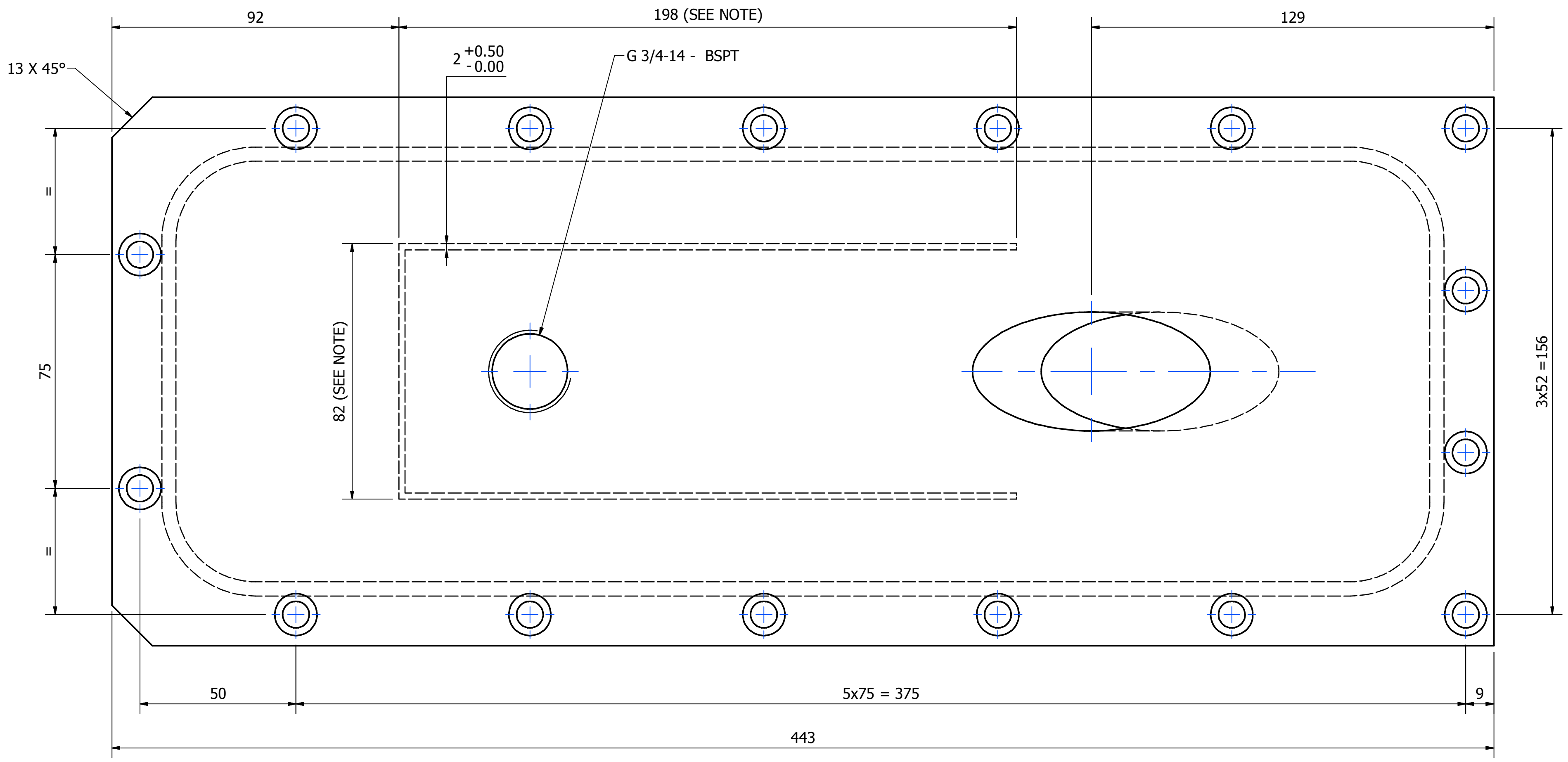
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

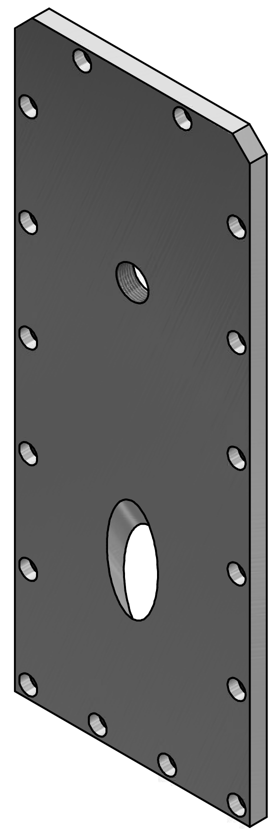


DETAIL A
SCALE: 2 : 1





4111-M034 - DETAIL -
SCALE: 1 : 1



4111-M034 - ISOMETRIC VIEW -
SCALE: 1 : 2

NOTE:
1. MODIFY GROOVE DIMENSIONS TO SUIT FOLDED PART 4111-S001
2. REMOVE ALL SHARP EDGES & BURRS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M034	1	AS1734 6061T6	SIDECOVER PLATE	2.5 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm ..... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

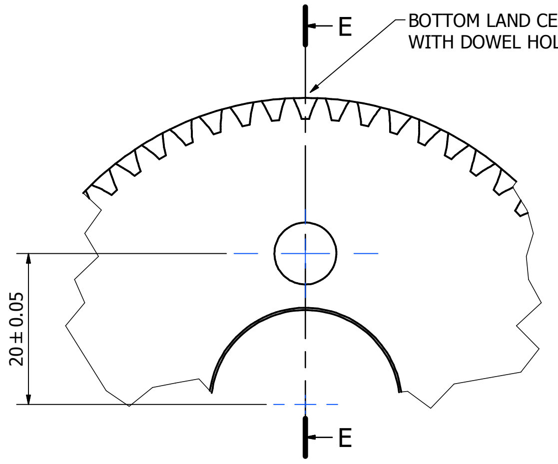
THE UNIVERSITY OF SOUTHERN QUEENSLAND.

SCALE	AS SHOWN	
DRAWN	KBD	DATE 9/08/2014
DESIGNED	KBD	DATE 9/08/2014
CHECKED		DATE
APPROVED		DATE

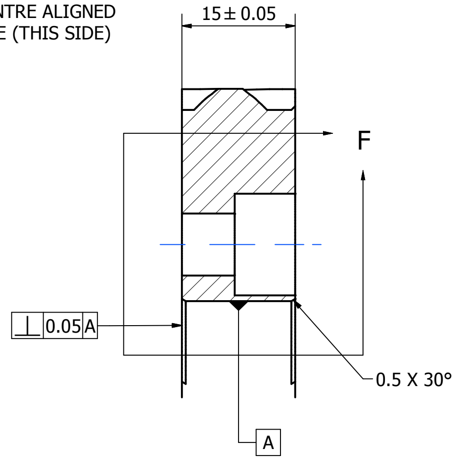
CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: SIDE COVER PLATE (OIL FILL)
JOB No.: ENG4111/2
DRAWING No.: 4111-M034
1 OF 1
SIZE: A1
REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M034.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M034.idw

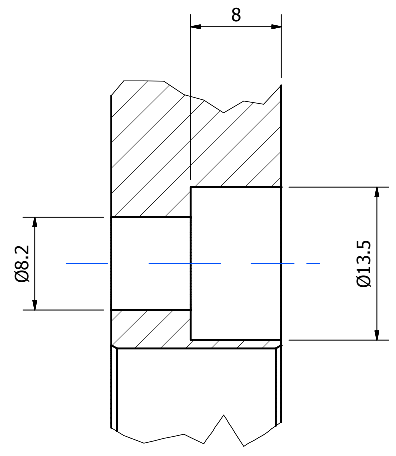
DO NOT SCALE - IF IN DOUBT ASK.



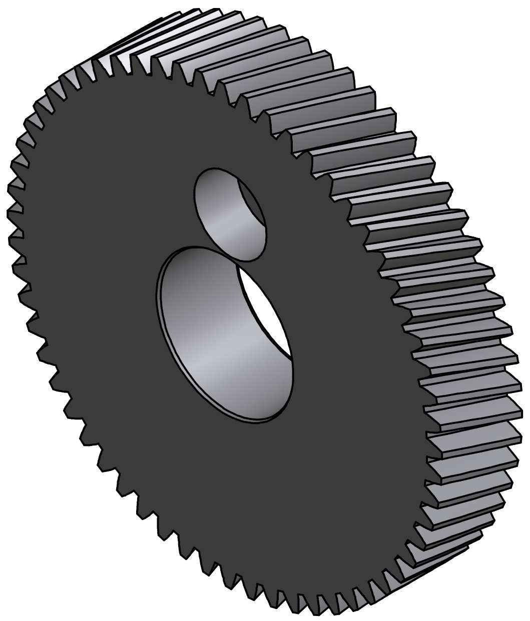
DETAIL D
SCALE: 2 : 1



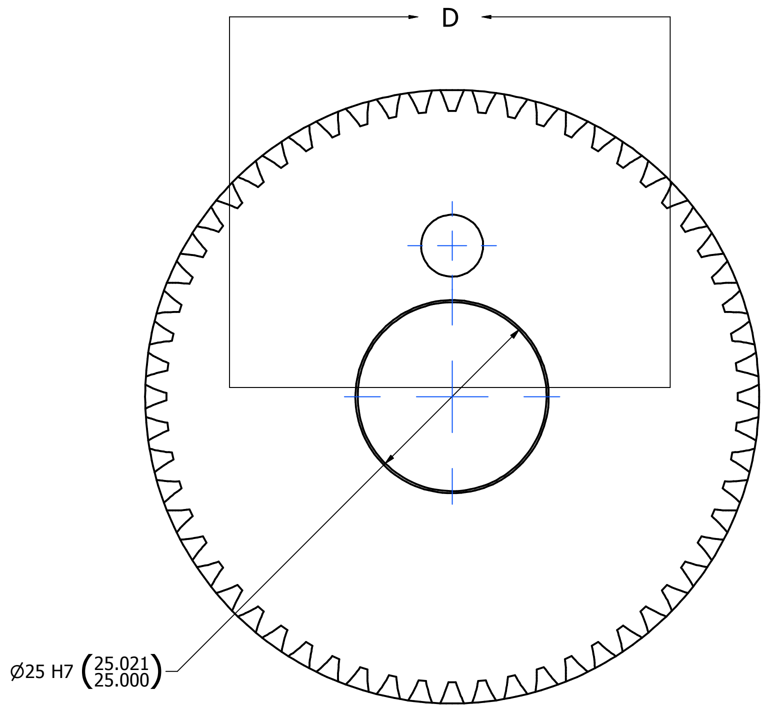
SECTION E-E
SCALE: 2 : 1

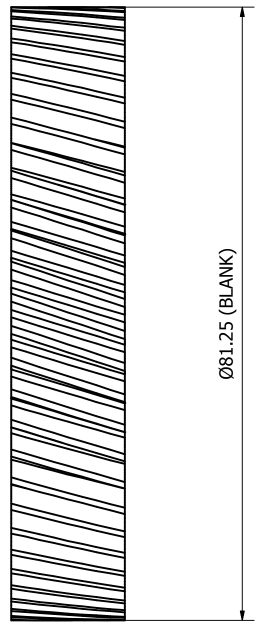


DETAIL F
SCALE: 3 : 1



4111-M040 - ISOMETRIC VIEW -
SCALE: 2 : 1





4111-M040 - DETAIL -
SCALE: 2 : 1

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- CASE HARDEN AS SPECIFIED IN 4111-A001 ("CH" CONDITION)
- HEAT SURFACE (**DO NOT SOAK**) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

IMPORTANT:
-ASSEMBLE GEAR CLUSTER AND TEST FOR EXCESSIVE BACKLASH OR POOR FIT BEFORE FINAL HEAT TREATMENT.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

GEAR DATA		
No.	DETAIL	VALUE
1	TOOTH FORM	INVOLUTE
2	No. OF TEETH	120
3	MODULE	1.25
4	HELIX ANGLE	17.75 (R.H.)
5	PRESSURE ANGLE	20
6	MATING GEAR CENTRES	118.12
7	FINAL HARDNESS	55 HRC
8	TOOTH SURFACE FINISH	0.4 Ra
9	MATING GEAR	120 T

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M040	1	AS1444 EN36A	SECONDARY COUNTER BALANCE SHAFT GEAR	0.5 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	
DRAWN	KBD	DATE 23/07/2014
DESIGNED	KBD	DATE 23/07/2014
CHECKED		DATE
APPROVED		DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: SECONDARY COUNTER BALANCE SHAFT GEAR

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M040	1 OF 1	A1	0

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

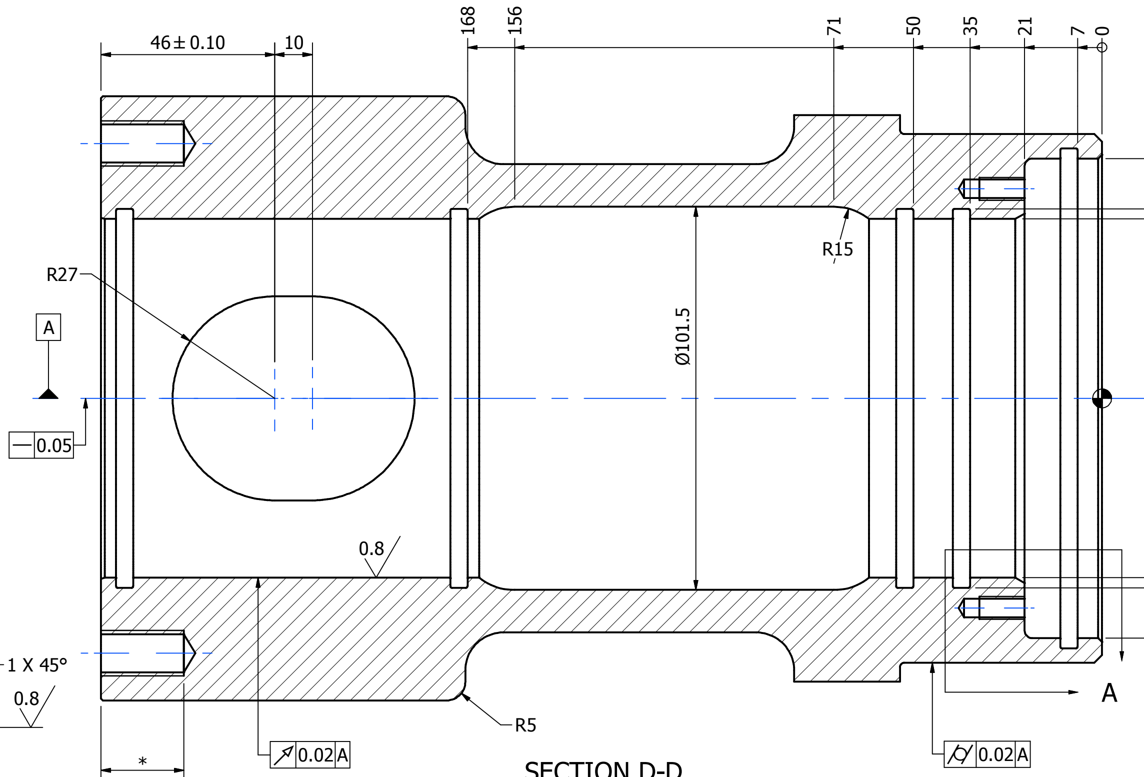
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

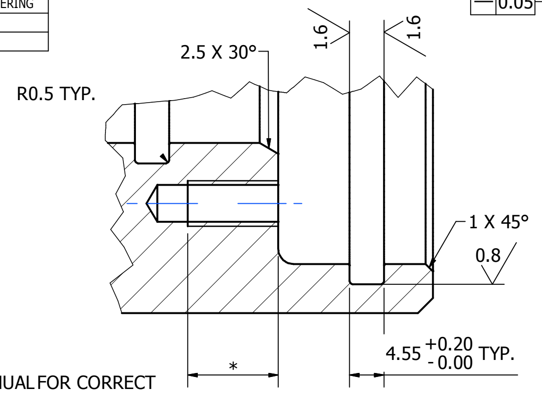
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



SECTION D-D

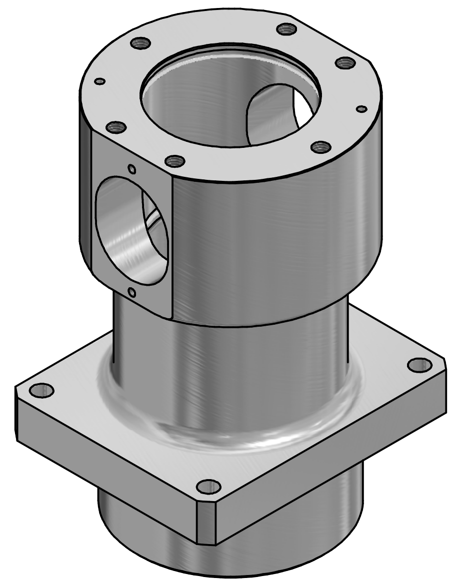
SCALE: 1 : 1



DETAIL A

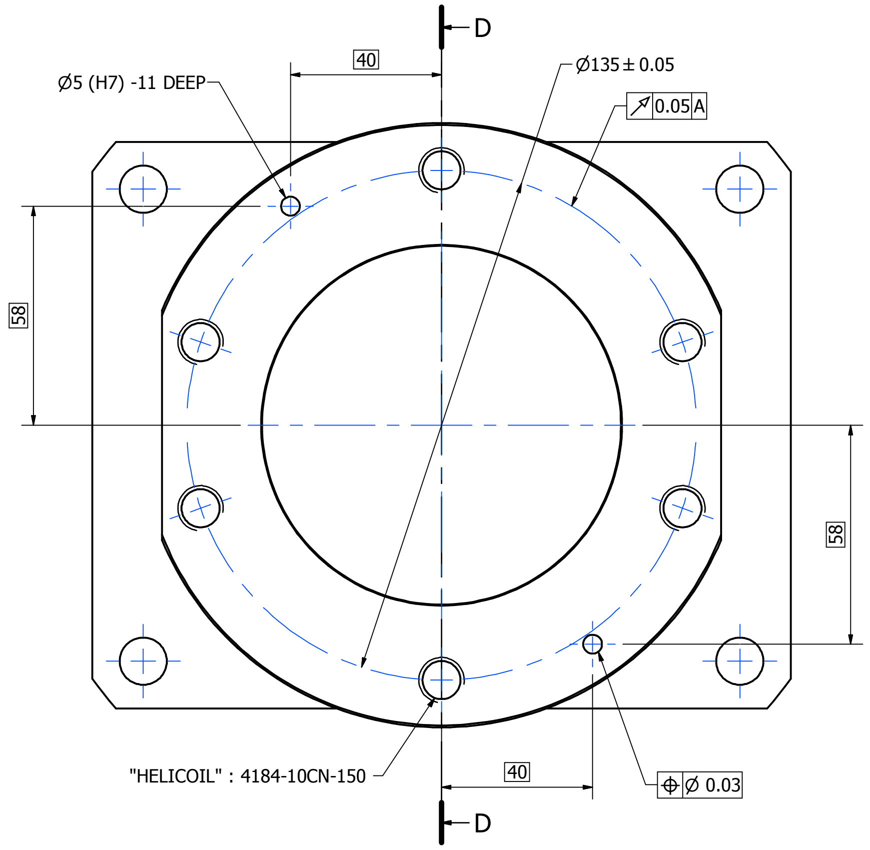
SCALE: 2 : 1

NOTE:
1. FOR DIMENSIONS MARKED "*" REFER HELICOIL MANUAL FOR CORRECT DETAIL ON INSERT INSTALLATION.
2. BARREL TO BE FREE OF BURRS & SHARP EDGES BEFORE ASSEMBLY

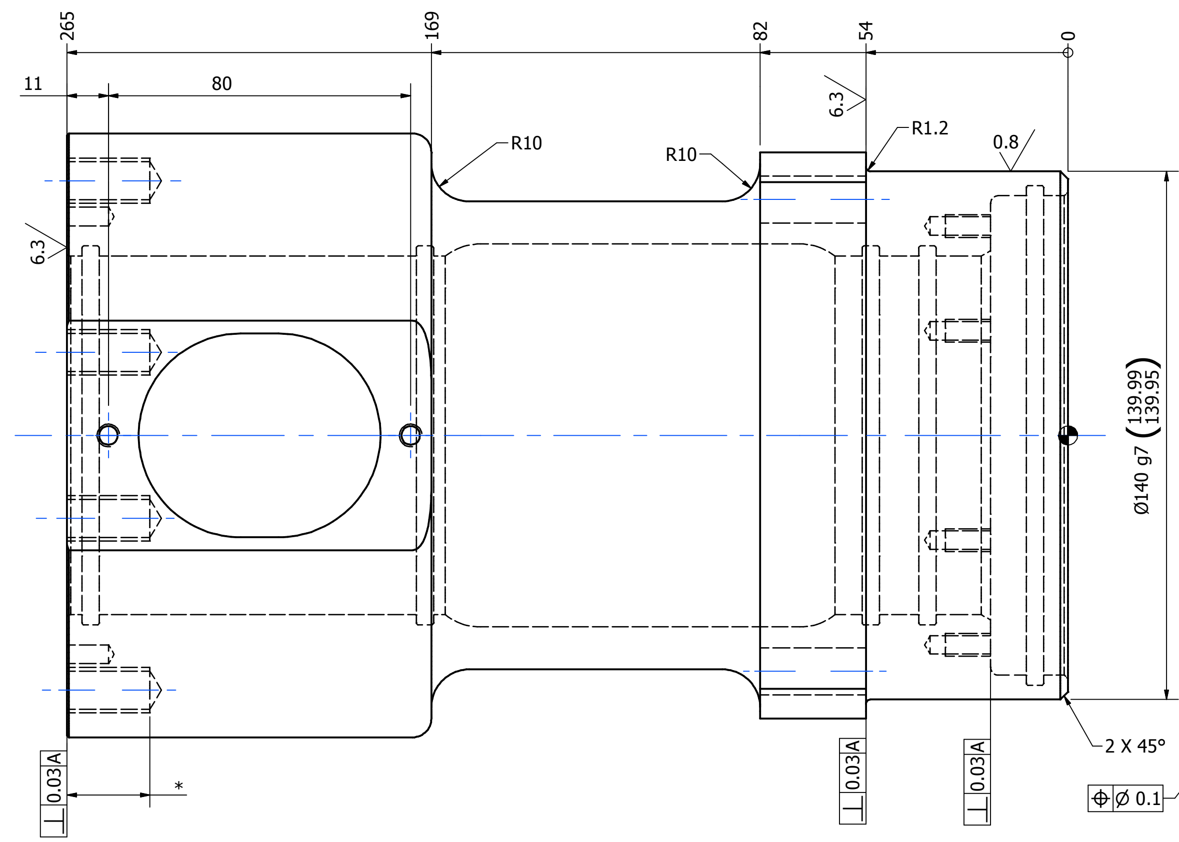


4111-M050 - ISOMETRIC VIEW -

SCALE: 1 : 2

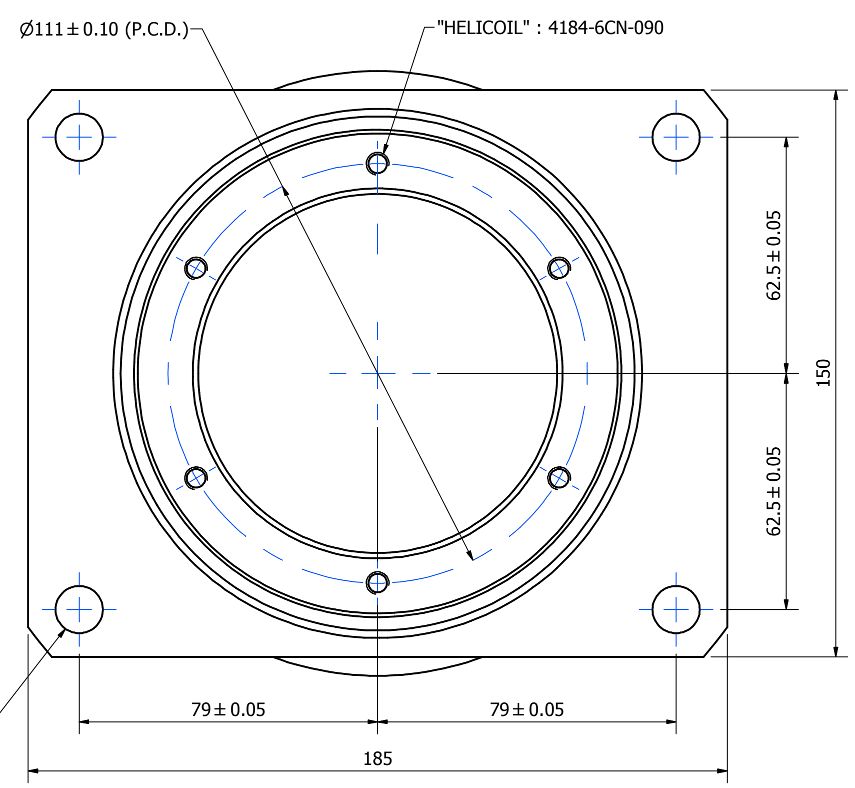


- TOP VIEW -



4111-M050 - DETAIL -

SCALE: 1 : 1



PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M050	1	AS1866 6061T6	LOWER BARREL	6.5 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	06/08/14	
0	ISSUED TO USQ	18/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE		
AS SHOWN		
DRAWN: KBD	DATE: 13/07/2014	
DESIGNED: KBD	DATE: 13/07/2014	
CHECKED	DATE	
APPROVED	DATE	

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: LOWER BARREL DESIGN

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M050	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M050.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M050.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

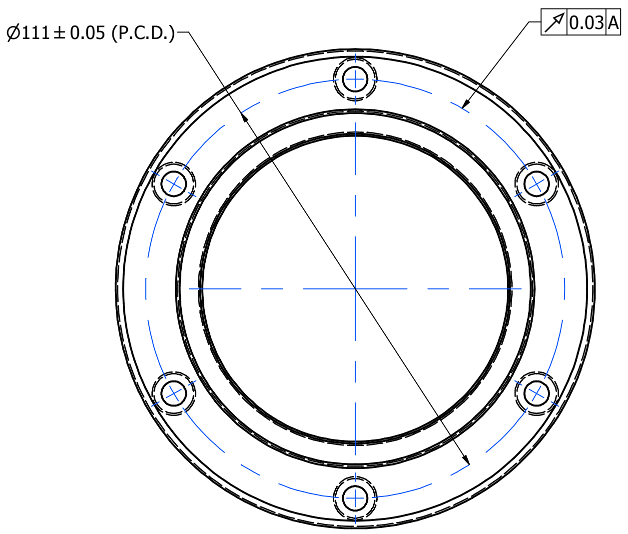
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

NOTES:

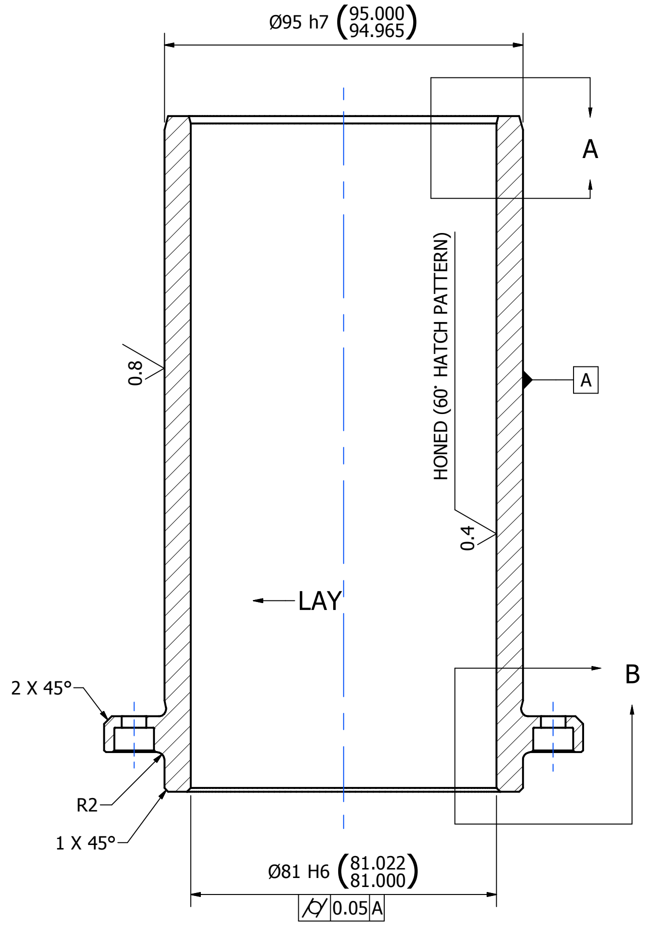
1. CASTING TO BE FREE OF CRACKS OR POROSITY
2. STRESS RELIEVE FINISHED COMPONENT
   - HEAT FROM A COLD OVEN TO 550°C - 580°C
   - SOAK FOR 2+ HRS AT TEMPERATURE
   - SWITCH OFF OVEN AND ALLOW TO COOL TO 100°C BEFORE REMOVAL
3. HONE CROSS HATCH PATTERN USING A SILICON CARBIDE HONE (45-65° PATTERN) NOTE "LAY" DIRECTION
4. WIPE BORE CLEAN AFTER HONING USING A CLEAN LIGHTLY OILED CLOTH (MORE THAN ONCE IF REQ'D).



- TOP VIEW -

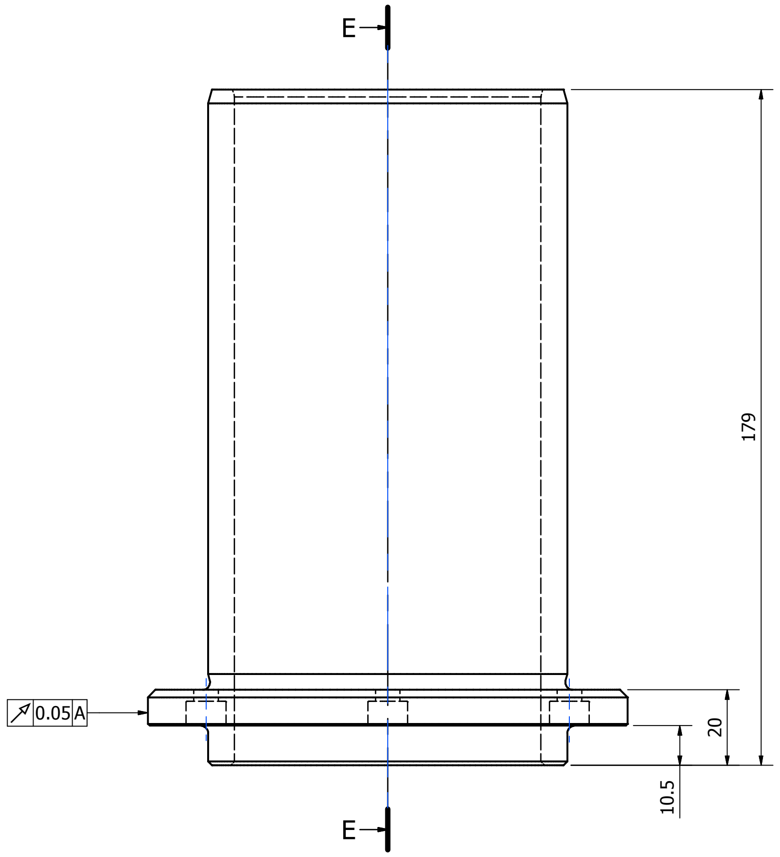
4111-M051 - ISOMETRIC VIEW -

SCALE: 1 : 1



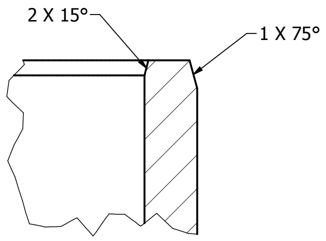
SECTION E-E

SCALE: 1 : 1



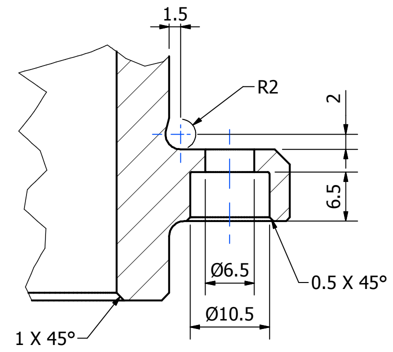
4111-M051 - DETAIL -

SCALE: 1 : 1



DETAIL A

SCALE: 2 : 1



DETAIL B

SCALE: 2 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M051	1	2P Cast Iron	LOWER CYLINDER SLEEVE	2.8 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	06/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD	DATE: 13/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 13/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	LOWER CYLINDER SLEEVE DETAIL
APPROVED	DATE		

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M051	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M051.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M051.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

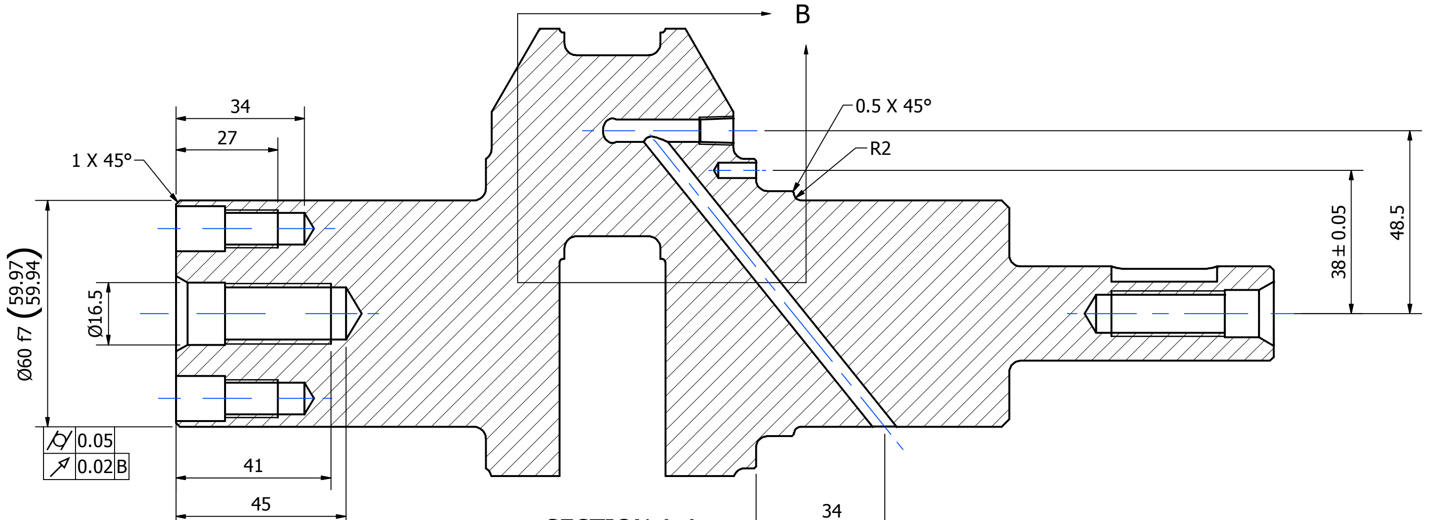
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

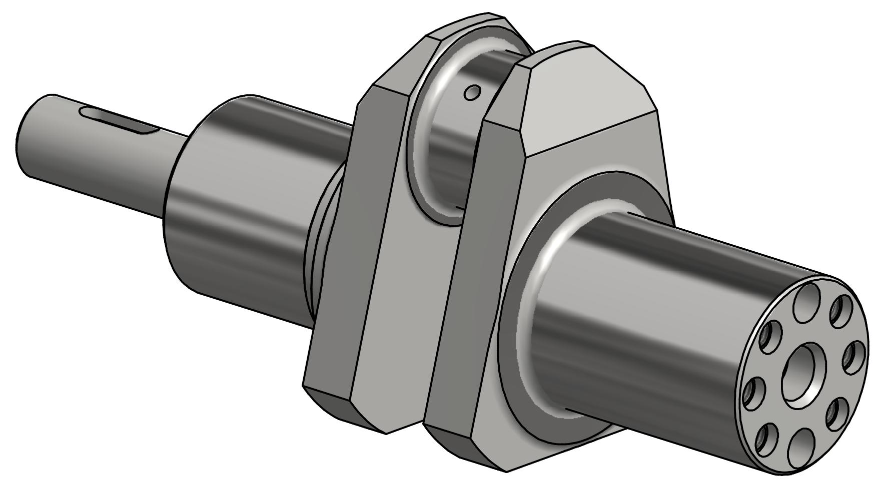
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



SECTION A-A
SCALE: 1 : 1

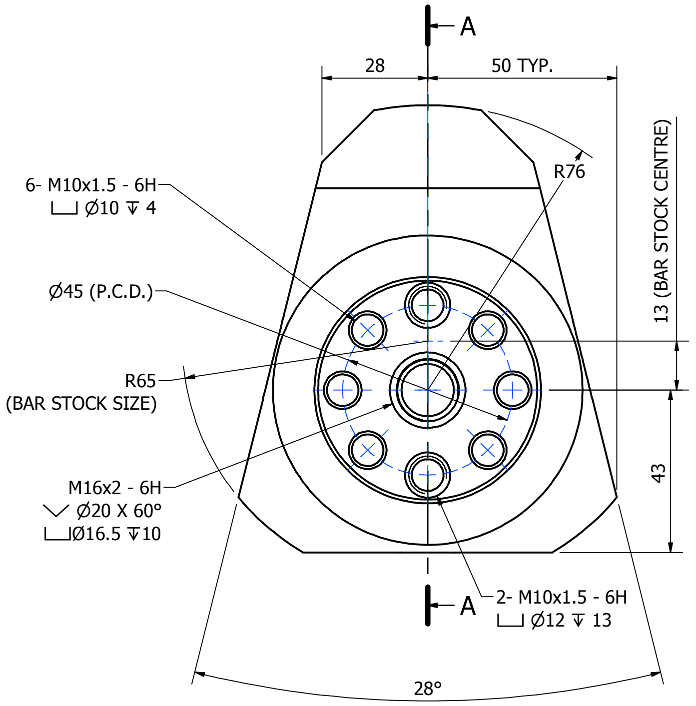


4111-M056 - ISOMETRIC VIEW -

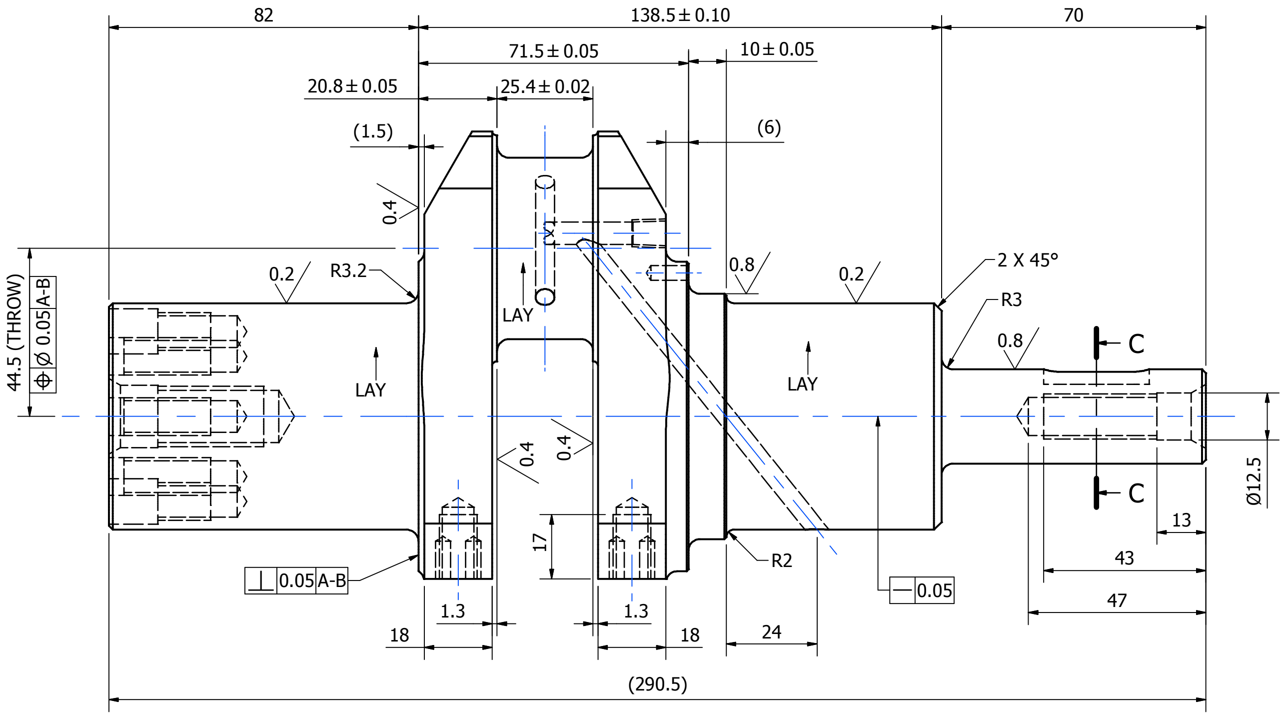
HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- NITRIDE AFTER FINAL MACHINING
- HEAT TO APPROX 150°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

MACHINING:
- ROUGH MACHINE TO 3mm ABOVER FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.
- NOTE THE DIRECTION OF FINISH LAY

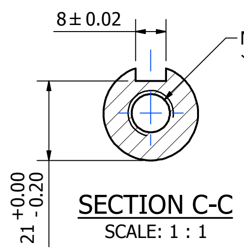
IMPORTANT:
- CRANKSHAFT IS OFFSET AXIALLY 2.5mm TOWARDS REAR OF CRANKSHAFT TO ALLOW FOR SAME OFFSET IN CONROD



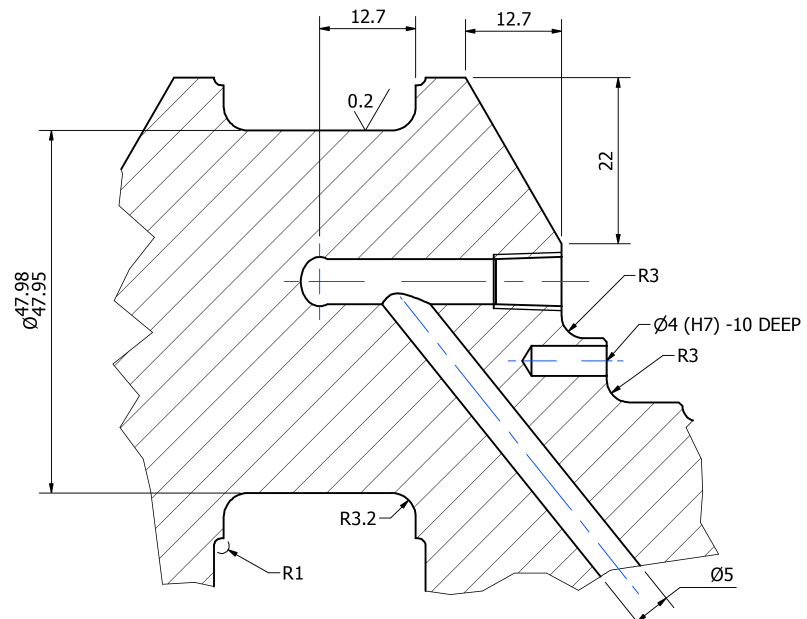
- END VIEW -



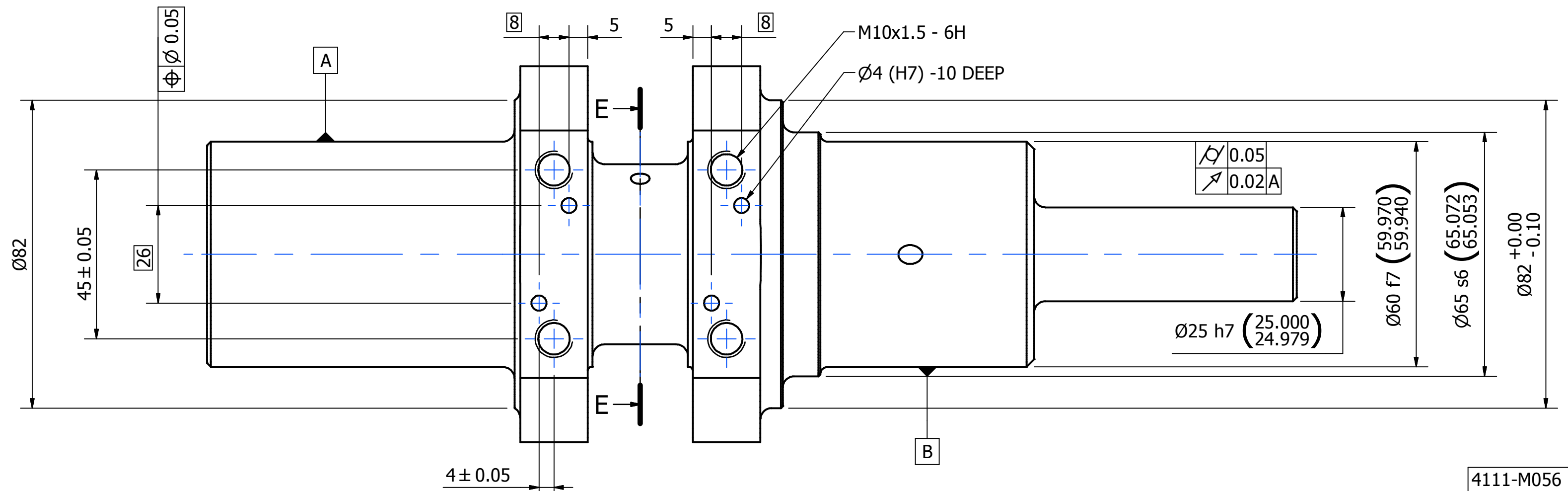
4111-M056 - DETAIL -
SCALE: 1 : 1

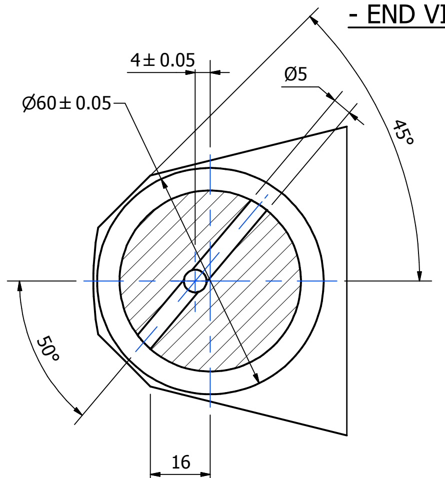


SECTION C-C
SCALE: 1 : 1



DETAIL B
SCALE: 2 : 1





SECTION E-E
SCALE: 1 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M056	1	AS1444 -4340	ENGINE CRANKSHAFT	6.4 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND
DRAWN	KBD	DATE 13/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	DATE 13/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED		DATE	DRAWING DESCRIPTION: ENGINE CRANKSHAFT DETAIL
APPROVED		DATE	JOB No. ENG4111/2 · DRAWING No. 4111-M056 · 1 OF 1 · SIZE A1 · REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M056.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M056.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

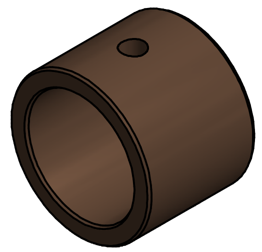
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

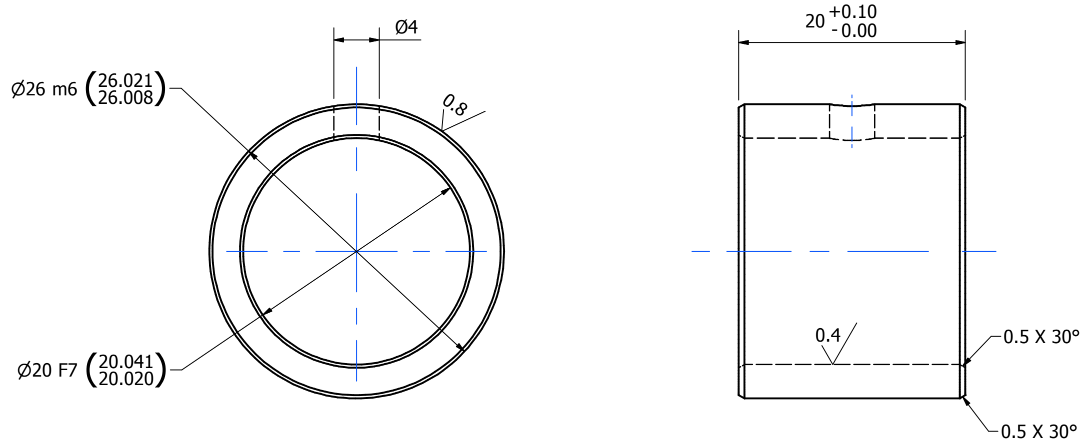
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



4111-M059 - ISOMETRIC VIEW -
SCALE: 2 : 1



4111-M059 - DETAIL -
SCALE: 3 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M059	1	954 - AL. BRONZE	LITTLE END BEARING	0 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	10/08/2014	
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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

DRAWN: KBD DATE: 10/08/2014

DESIGNED: KBD DATE: 10/08/2014

CHECKED: DATE:

APPROVED: DATE:

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

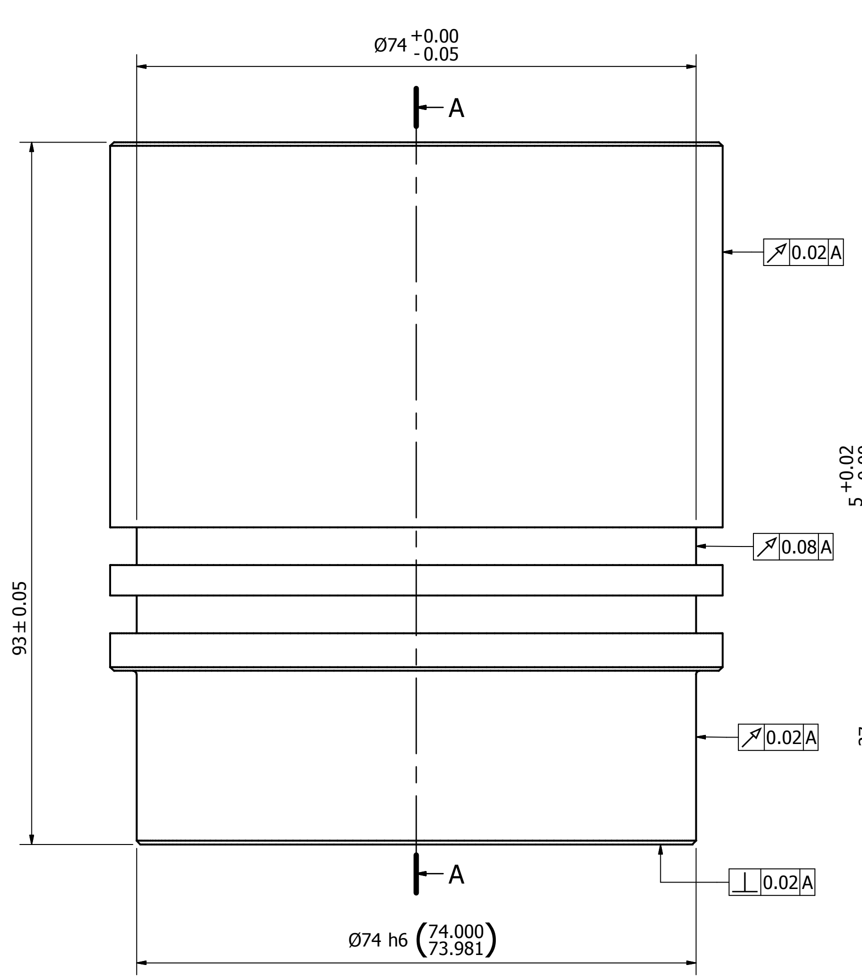
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: LITTLE END BEARING

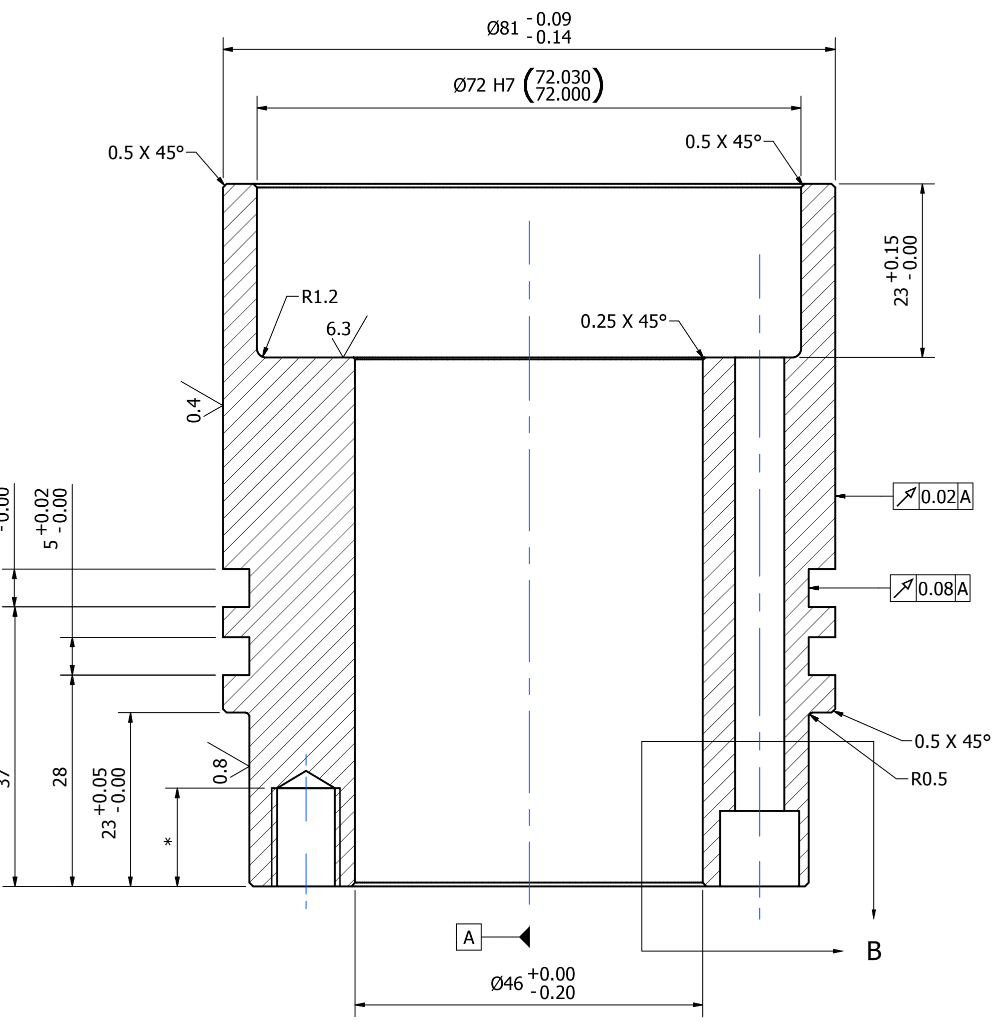
JOB No. ENG4111/2 | DRAWING No. 4111-M059 | 1 OF 1 | SIZE A1 | REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M059.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M059.idw

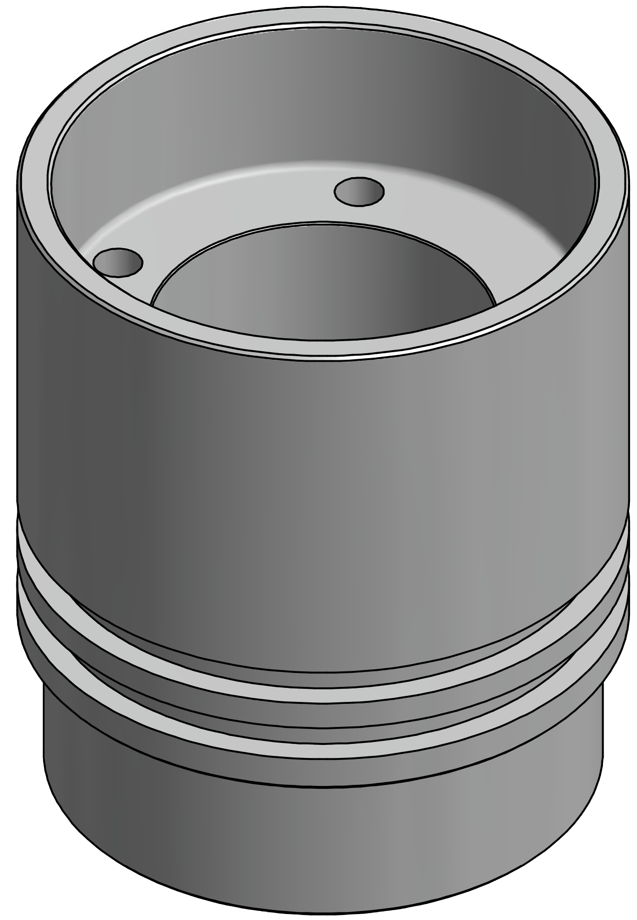
DO NOT SCALE - IF IN DOUBT ASK.



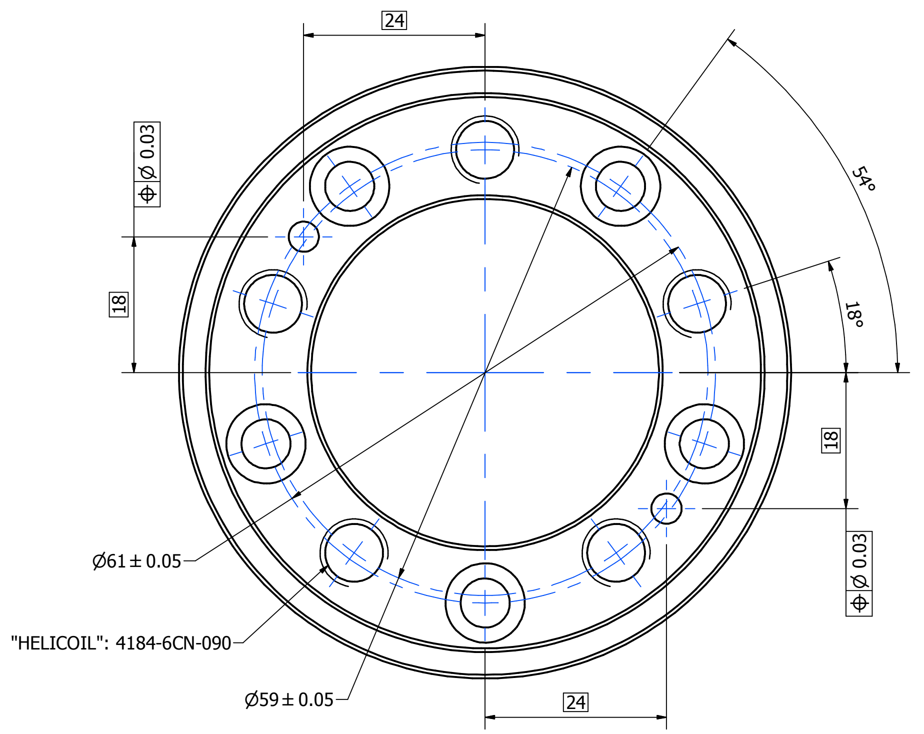
4111-M061 - DETAIL -
SCALE: 2 : 1



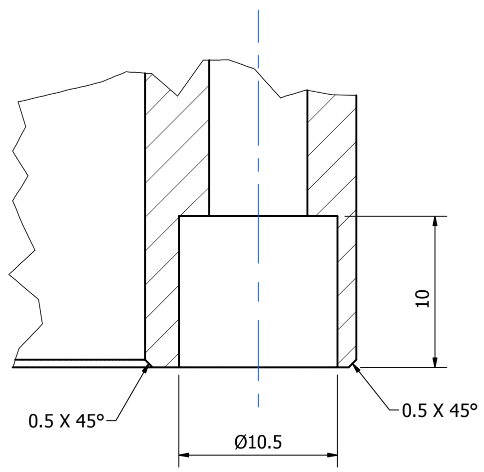
SECTION A-A
SCALE: 2 : 1



4111-M061 - ISOMETRIC VIEW -
SCALE: 2 : 1



- BOTTOM VIEW -



DETAIL B
SCALE: 4 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

NOTE:
1. FOR DIMENSIONS MARKED "*" REFER HELICOIL MANUAL FOR CORRECT DETAIL ON INSERT INSTALLATION.
2. PISTON TO BE FREE OF BURRS & SHARP EDGES BEFORE ASSEMBLY

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M061	1	AS1866 4032-T6	PISTON HEAD	0.6 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

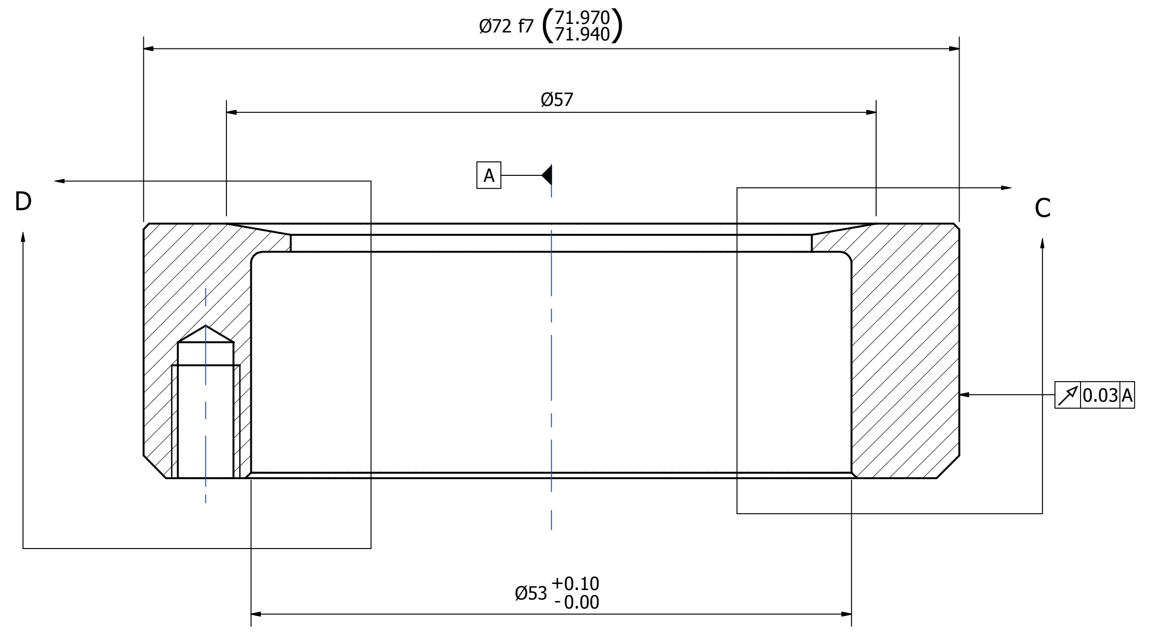
SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

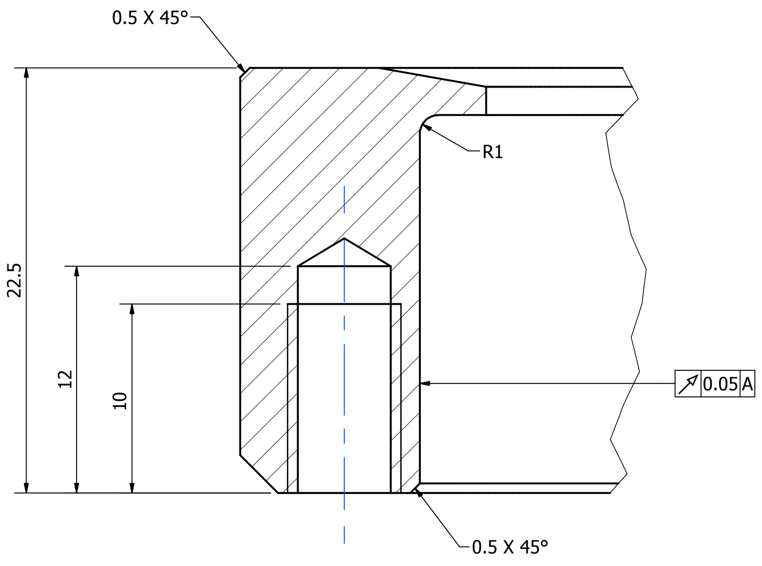
DRAWN: KBD	DATE: 13/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 13/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION: PISTON HEAD DETAIL
APPROVED	DATE	JOB No.: ENG4111/2 — DRAWING No.: 4111-M061 — 1 OF 1 — SIZE: A1 — REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M061.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M061.idw

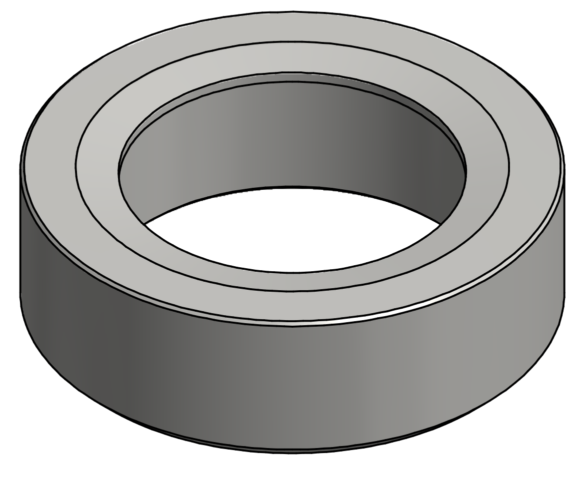
DO NOT SCALE - IF IN DOUBT ASK.



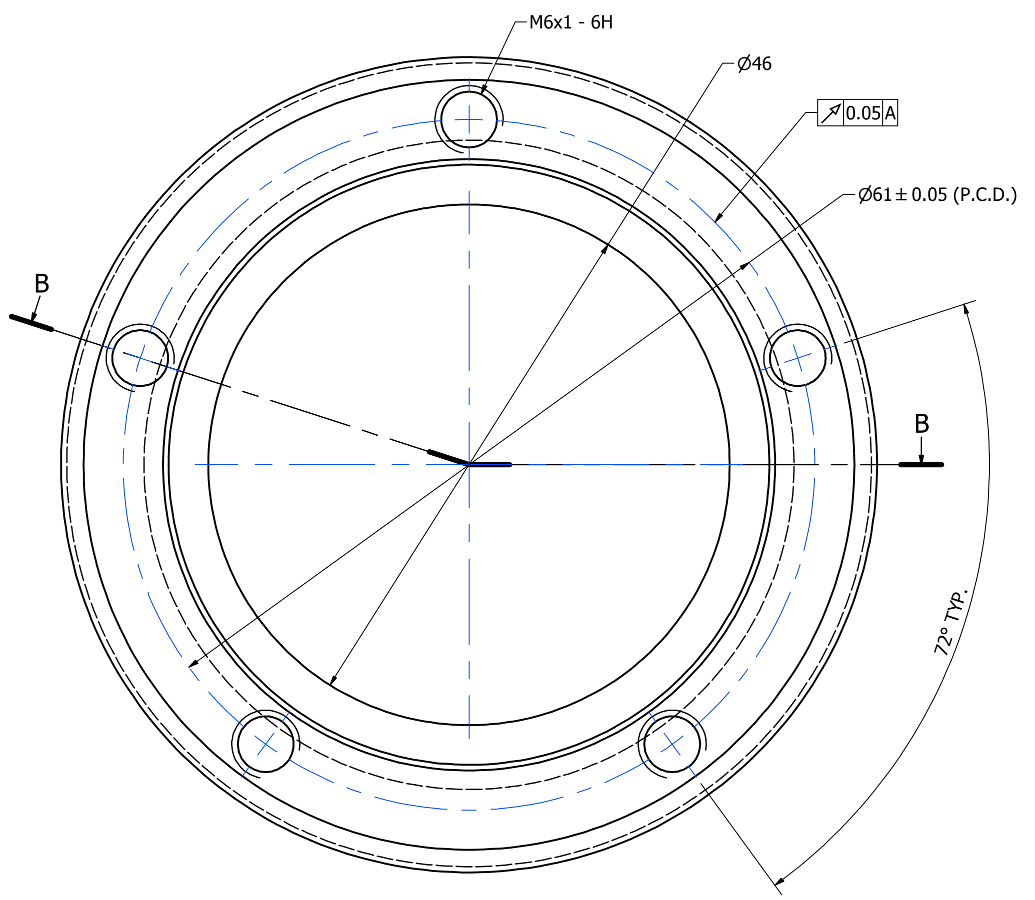
SECTION B-B
SCALE: 3 : 1



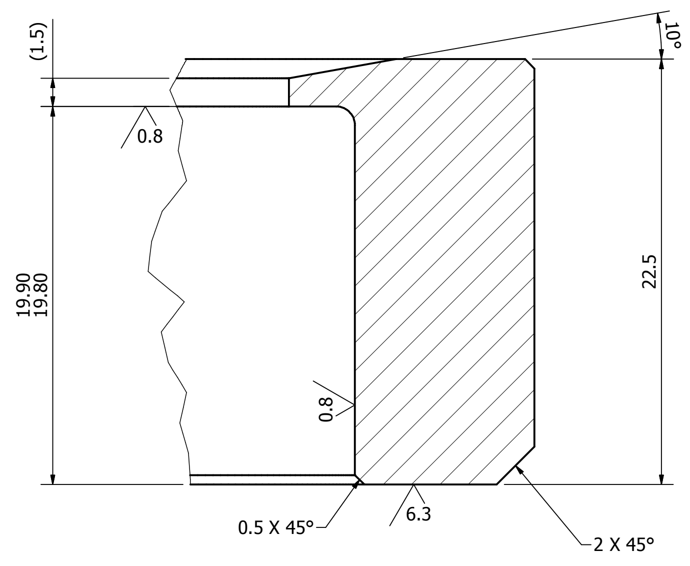
DETAIL D
SCALE: 5 : 1



4111-M062 - ISOMETRIC VIEW -
SCALE: 2 : 1



4111-M062 - BOTTOM VIEW -
SCALE: 3 : 1



DETAIL C
SCALE: 5 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M062	1	SS-304 ROUND	OPTICAL WINDOW COLLAR	0.33 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ...... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN	KBD	DATE	13/07/2014
DESIGNED	KBD	DATE	13/07/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: OPTICAL WINDOW COLLAR DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M062	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M062.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M062.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

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HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

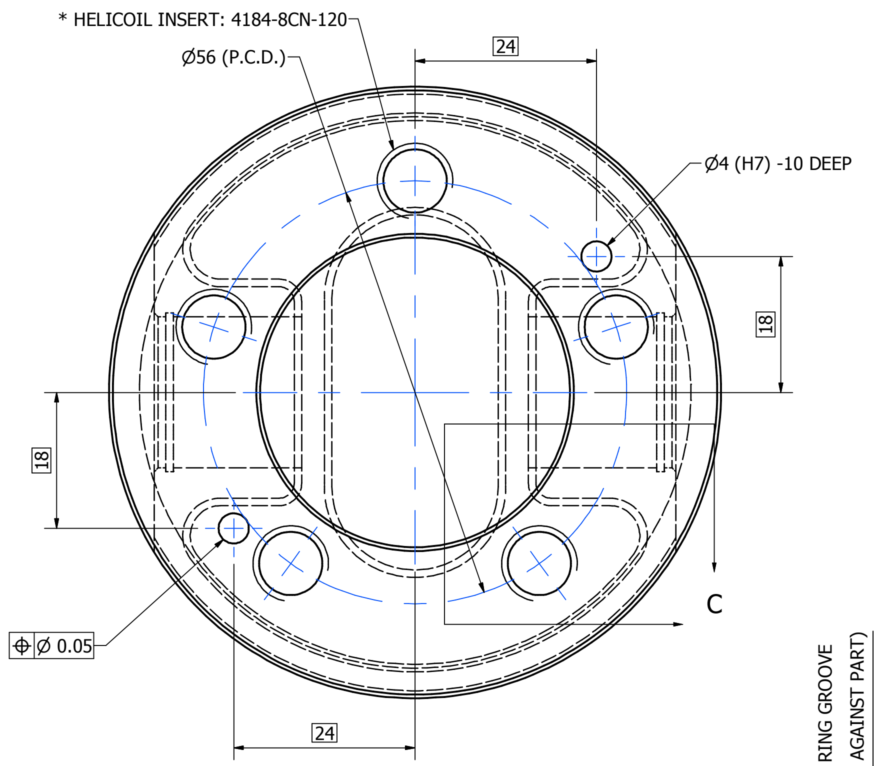
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

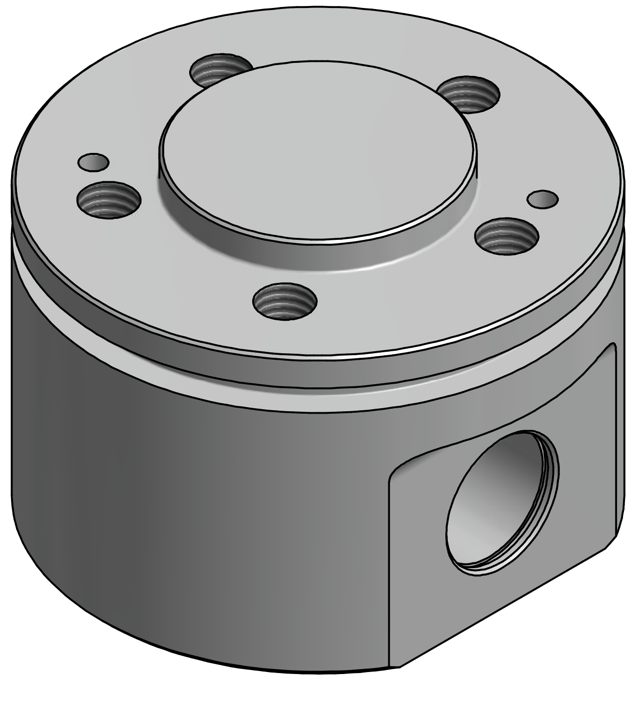
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

NOTE:

1. FOR DIMENSIONS MARKED "*" REFER HELICOIL MANUAL FOR CORRECT DETAIL ON INSERT INSTALLATION.

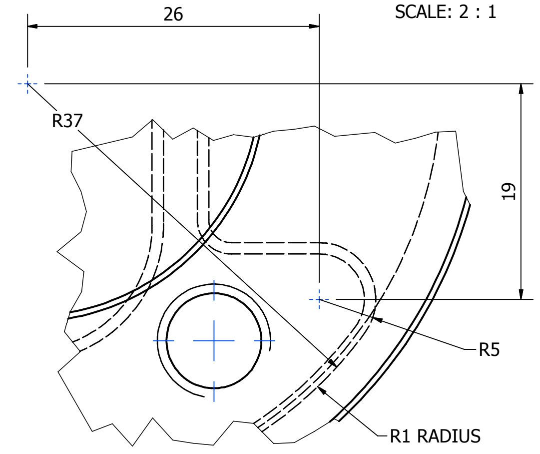


- TOP VIEW -



4111-M063 - ISOMETRIC VIEW -

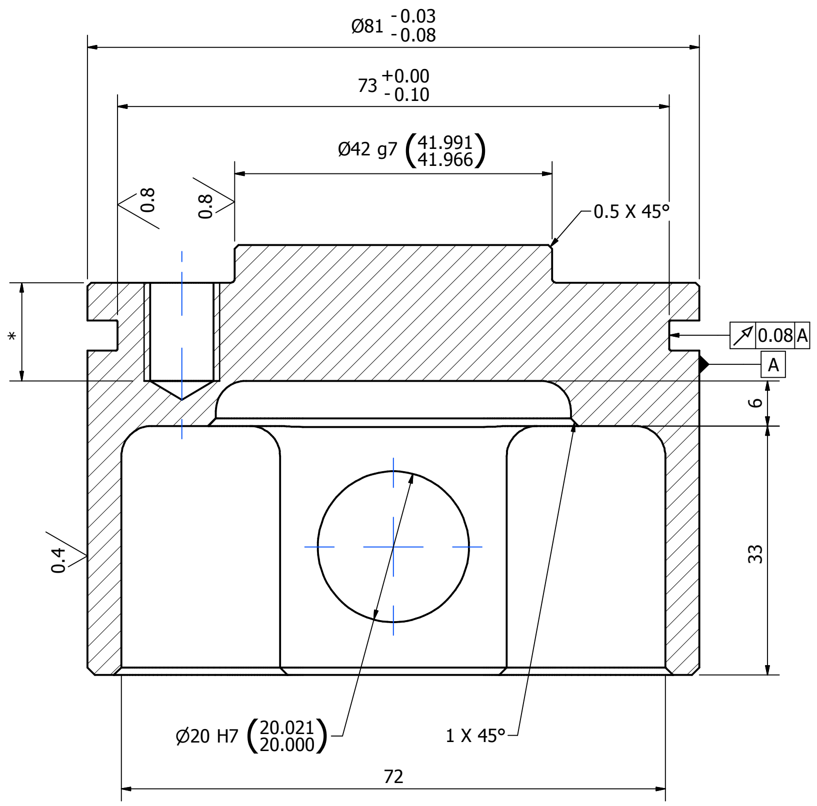
SCALE: 2 : 1



DETAIL C

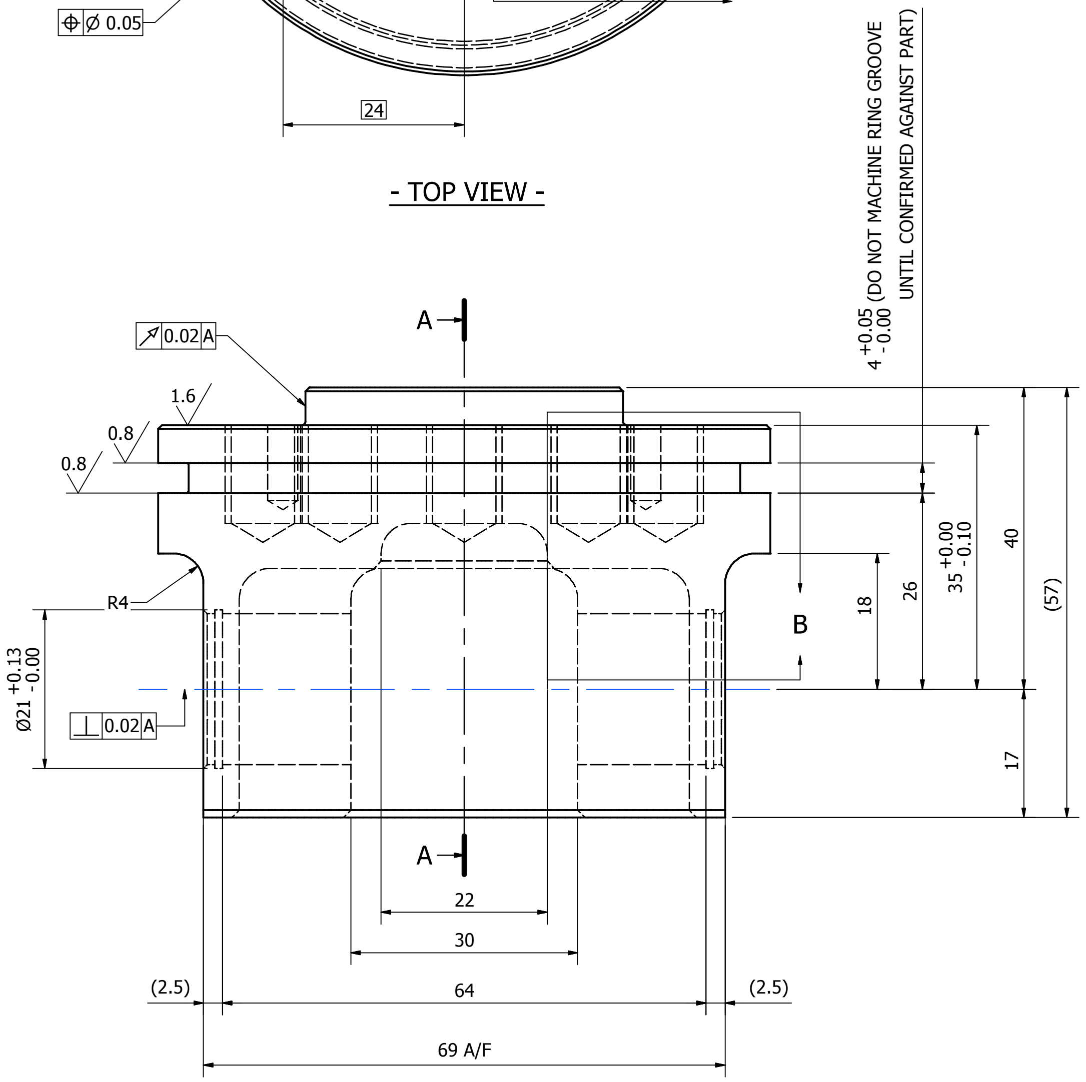
SCALE: 3 : 1

NOTE: CERTAIN HIDDEN DETAIL OMITTED FOR CLARITY



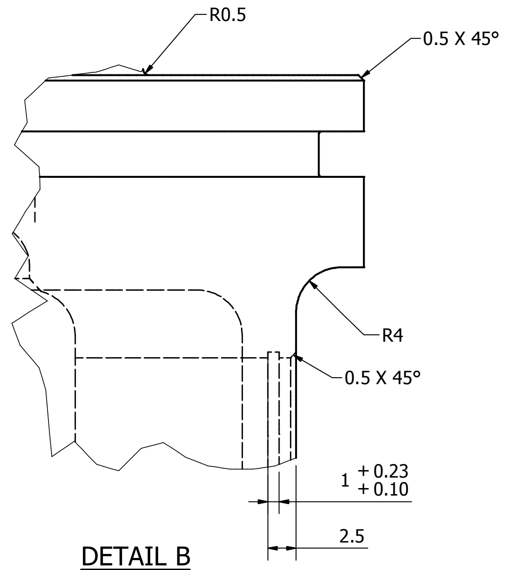
SECTION A-A

SCALE: 2 : 1



4111-M063 - DETAIL -

SCALE: 2 : 1



DETAIL B

SCALE: 3 : 1

NOTE: CERTAIN HIDDEN DETAIL OMITTED FOR CLARITY

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M063	1	AS1866 4032-T6	LOWER PISTON	0.39 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETRES UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.	
DRAWN: KBD	DATE: 13/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND	
DESIGNED: KBD	DATE: 13/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN	
CHECKED	DATE	DRAWING DESCRIPTION: LOWER PISTON DETAIL	
APPROVED	DATE	JOB No.: ENG4111/2	DRAWING No.: 4111-M063 — 1 OF 1 — SIZE: A1 — REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M063.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M063.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

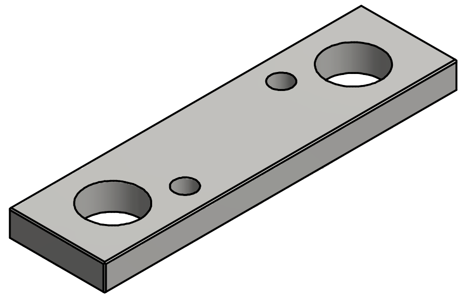
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

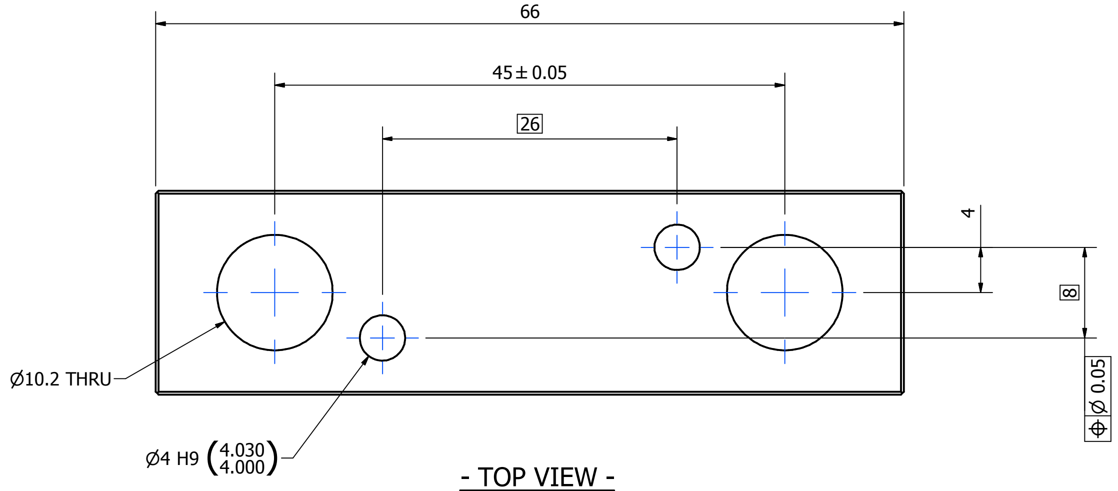
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

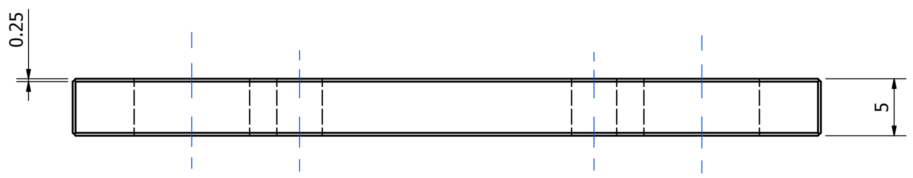


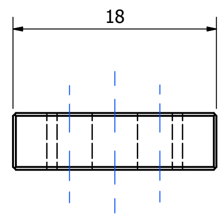
4111-M064 - ISOMETRIC VIEW -

SCALE: 2 : 1



- TOP VIEW -





- END VIEW -

4111-M064 - DETAIL -

SCALE: 3 : 1

4111-M064	1	AS3678 -GR250	CRANK BALANCE WEIGHT	0.039 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN	KBD	DATE	13/07/2014
DESIGNED	KBD	DATE	13/07/2014
CHECKED		DATE	
APPROVED		DATE	

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: CRANK BALANCE WEIGHT DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M064	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M064.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M064.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

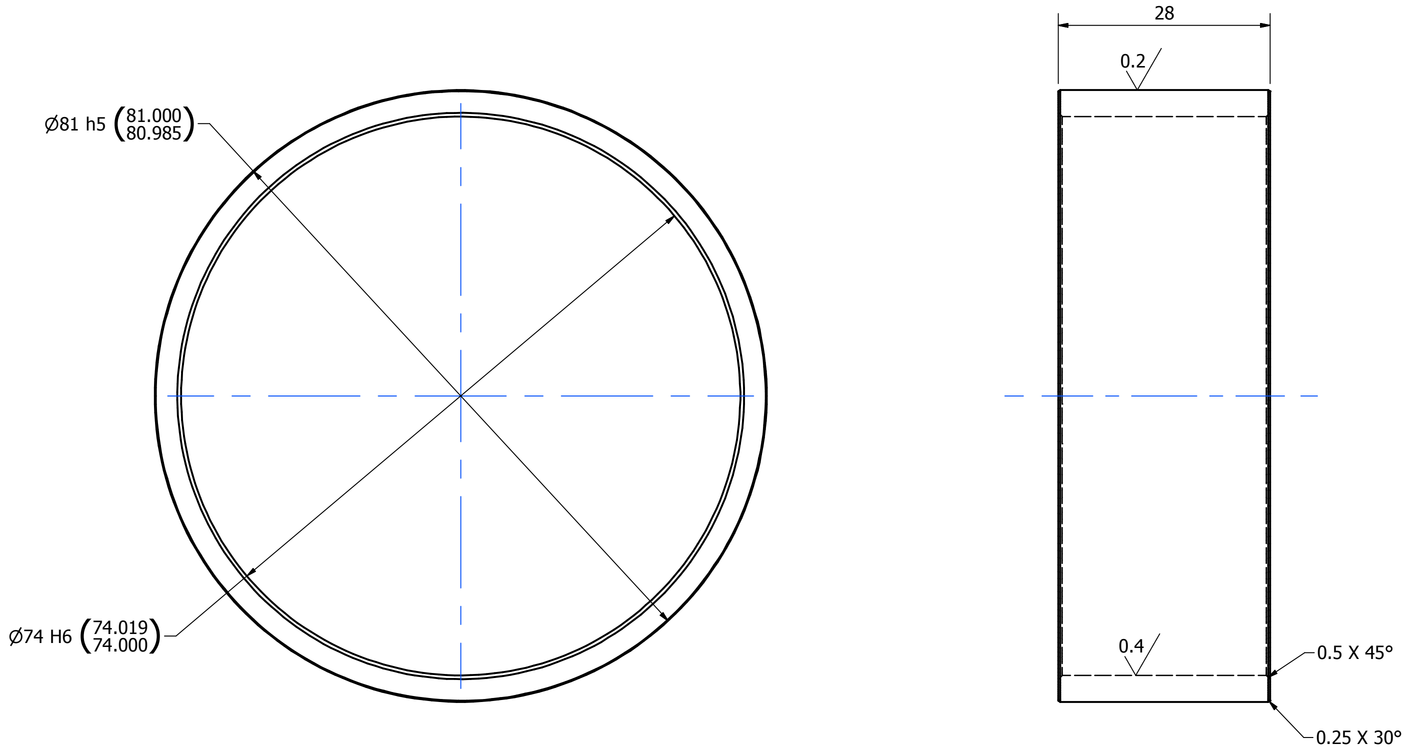
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



4111-M065 - DETAIL -
SCALE: 2 : 1

4111-M065 - ISOMETRIC VIEW -
SCALE: 2 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M065	1	Tetron B	SLIDING BUSH	92.792 g

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
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UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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DRAWN: KBD	DATE: 9/08/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 9/08/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION: SLIDING BUSH DETAIL
APPROVED	DATE	JOB No.: ENG4111/2 · DRAWING No.: 4111-M065 · 1 OF 1 · SIZE: A1 · REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M065.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M065.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

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ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

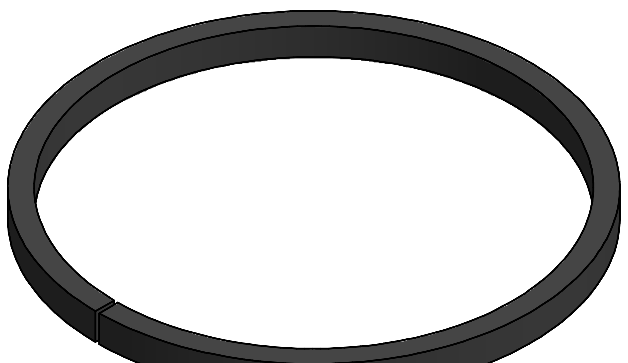
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

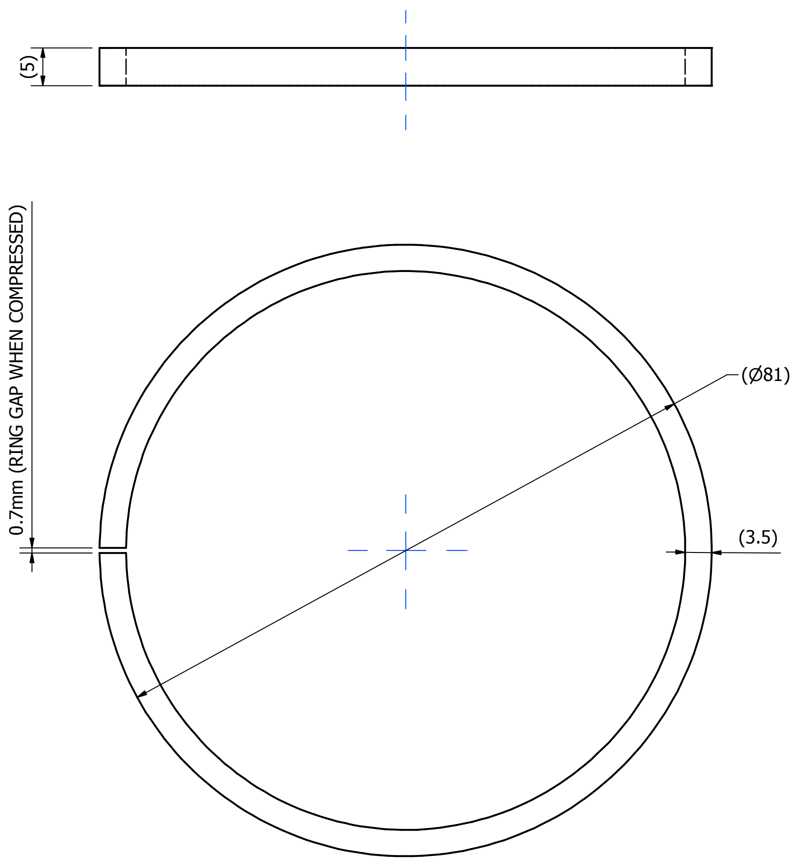
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



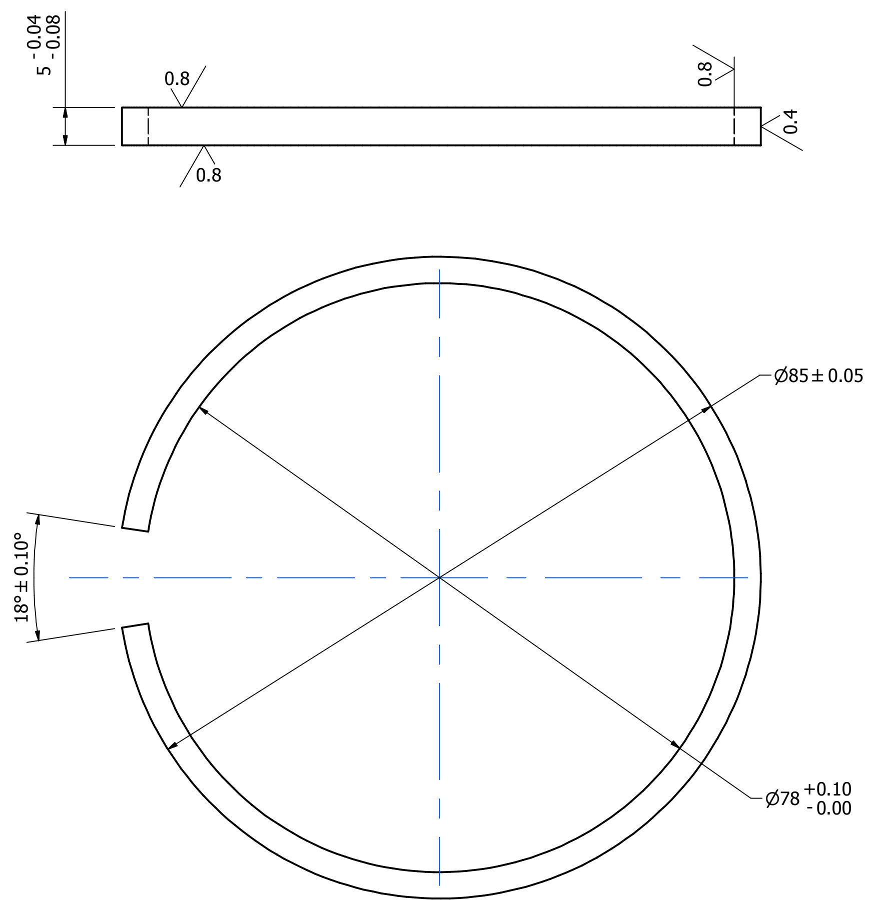
4111-M066 - ISOMETRIC VIEW -

SCALE: 2 : 1



4111-M066 - DETAIL -

SCALE: 2 : 1



OPEN RING - DETAIL-

NOTE: MACHINE RINGS FROM BILLET TO THESE DIMENSIONS; LABELLED AS 4111-M066b

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M066	2	Tetron C	LOW FRICTION PISTON RINGS	0 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09-08-2014	
0	ISSUED TO USQ	30/10/2014	

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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

		DATE	
DRAWN	KBD	DATE	9/08/2014
DESIGNED	KBD	DATE	9/08/2014
CHECKED		DATE	
APPROVED		DATE	

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: LOW FRICTION PISTON RINGS

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M066	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M066.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M066.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

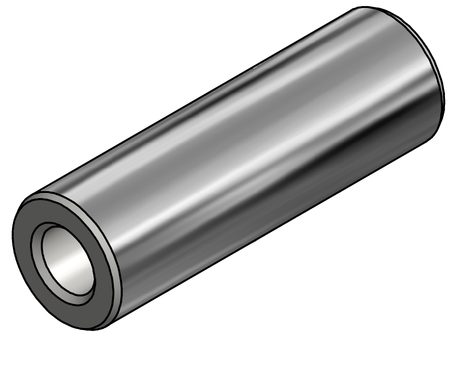
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

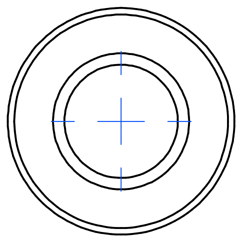
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

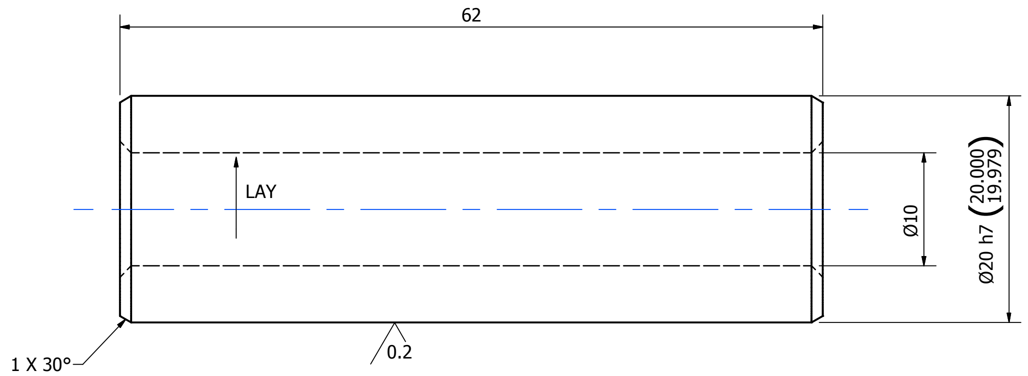


4111-M067 - ISOMETRIC VIEW -

SCALE: 2 : 1



- END VIEW -



4111-M067 - DETAIL -

SCALE: 3 : 1

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- WRIST PIN TO BE NITRIDED AFTER FINAL MACHINING.
- HEAT SURFACE (DO NOT SOAK) TO APPROX 250°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

MACHINING:
- ROUGH MACHINE TO 1mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.

- NOTE THE DIRECTION OF FINISH LAY ◄►

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M067	1	AS1444 -4340	WRIST PIN	0.1 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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DRAWN	DATE
KBD	9/08/2014
DESIGNED	DATE
KBD	9/08/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: WRIST PIN DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M067	1 OF 1	A1	0

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DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

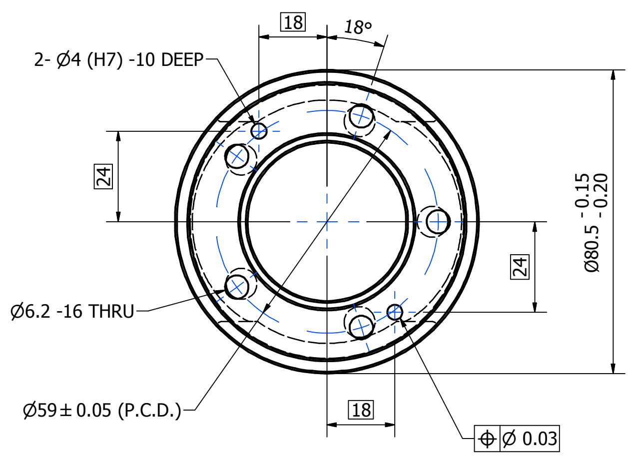
LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

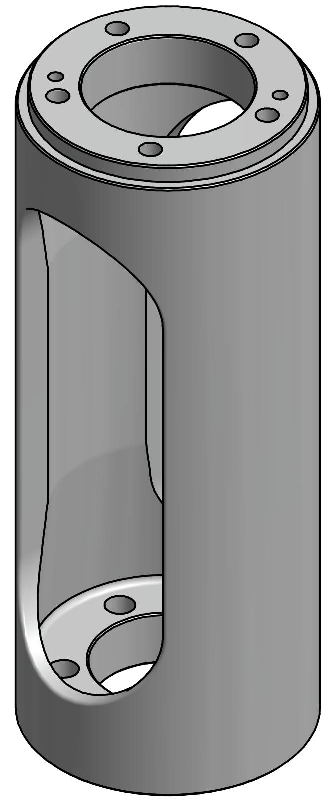
SURFACE FINISH VALUES TABULATED BELOW.

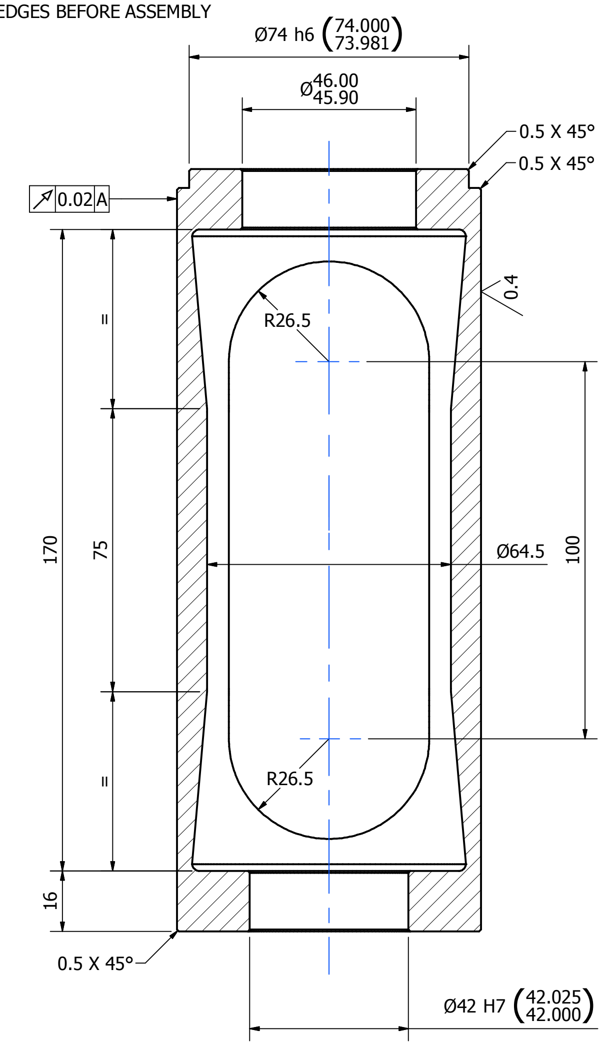
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

NOTE:
1. FOR DIMENSIONS MARKED "*" REFER HELICOIL MANUAL FOR CORRECT DETAIL ON INSERT INSTALLATION.
2. PISTON TO BE FREE OF BURRS & SHARP EDGES BEFORE ASSEMBLY

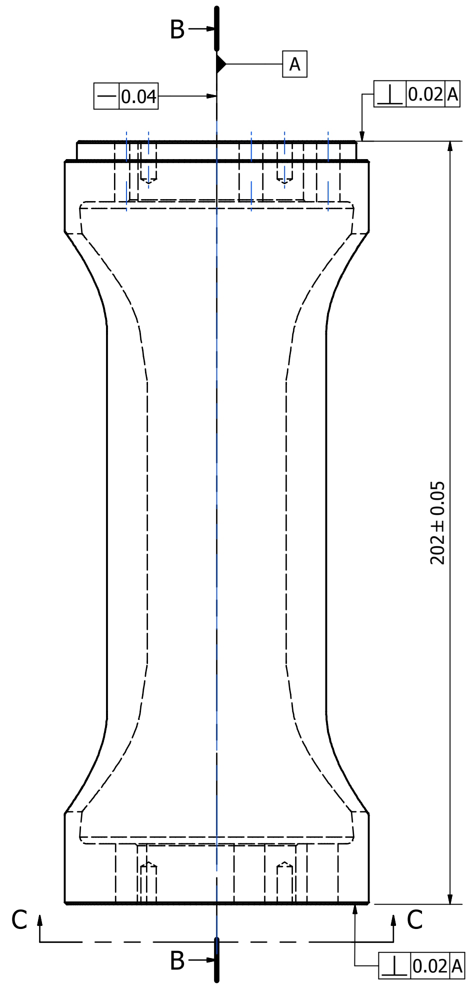


- TOP VIEW -

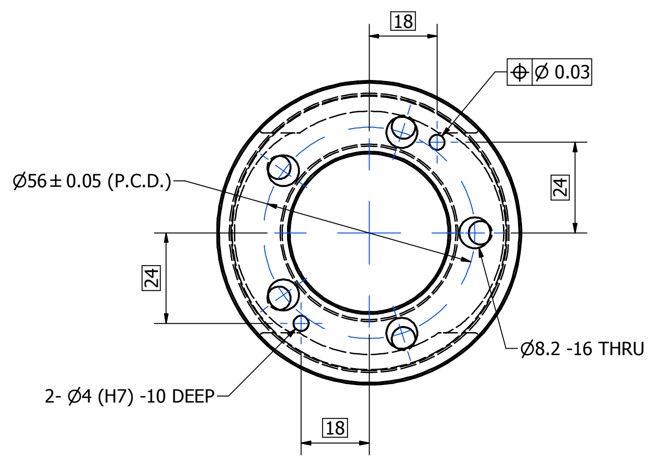




SECTION B-B
SCALE: 1 : 1



4111-M069 - DETAIL -
SCALE: 1 : 1



C
SCALE: 1 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M069	1	AS1866 4032-T6	PISTON EXTENSION	0.67 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN	THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN KBD	DATE 13/07/2014	CLIENT UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 13/07/2014	DESCRIPTION OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION PISTON EXTENSION
APPROVED	DATE	JOB No. ENG4111/2 — DRAWING No. 4111-M069 — 1 OF 1 — SIZE A1 — REV. 0

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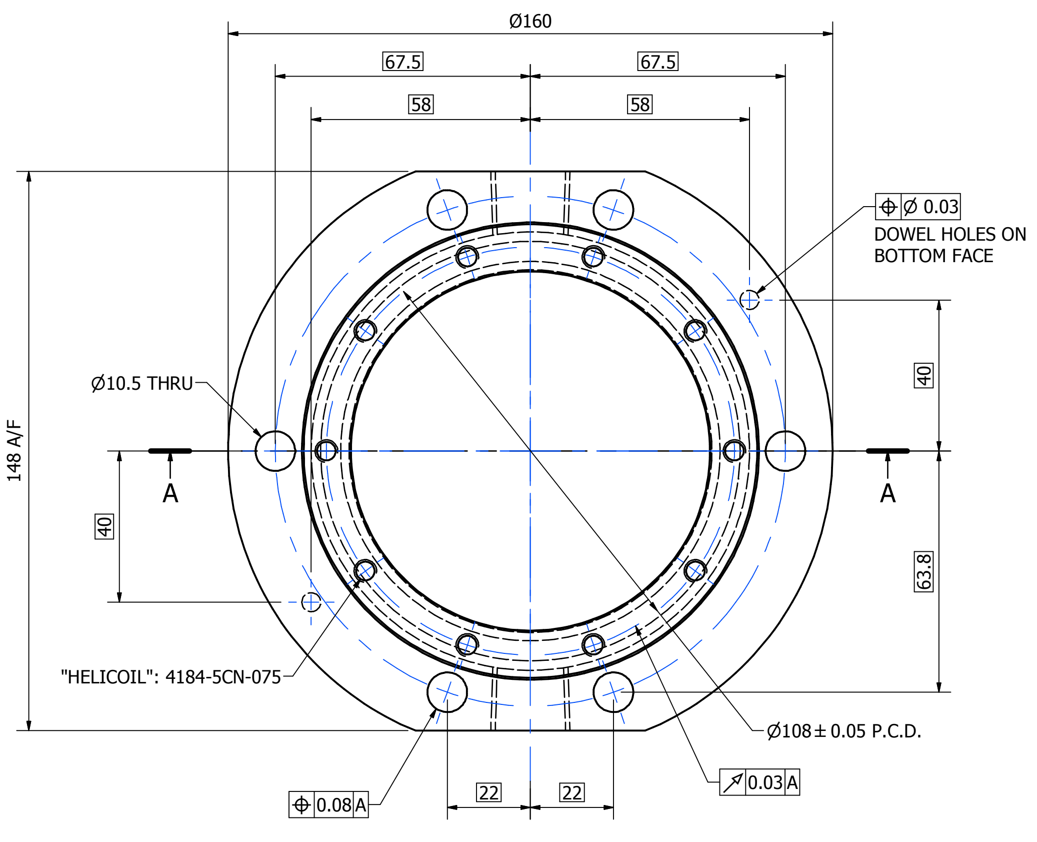
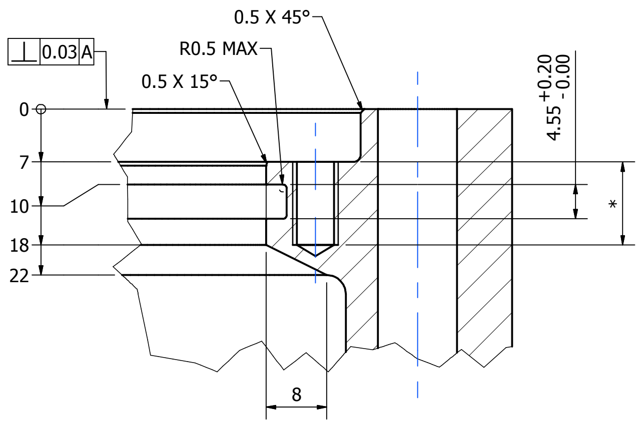
DO NOT SCALE - IF IN DOUBT ASK.



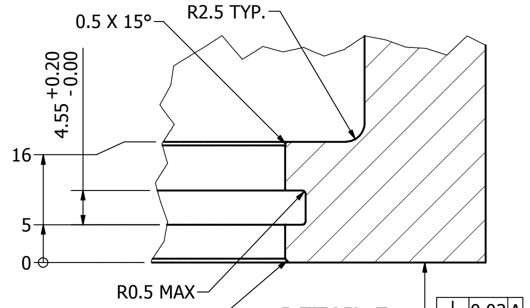
4111-M072 - TOP VIEW -

SCALE: 1 : 1



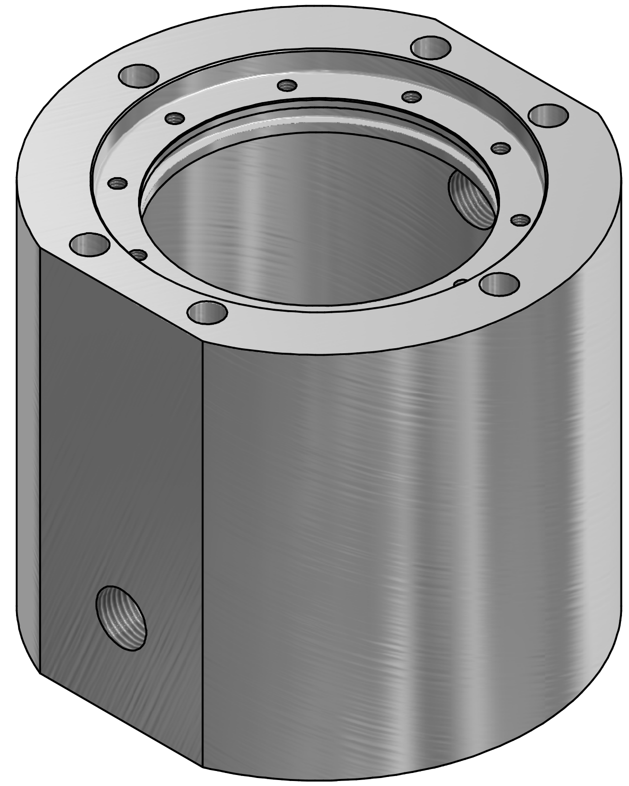
DETAIL C

SCALE: 2 : 1



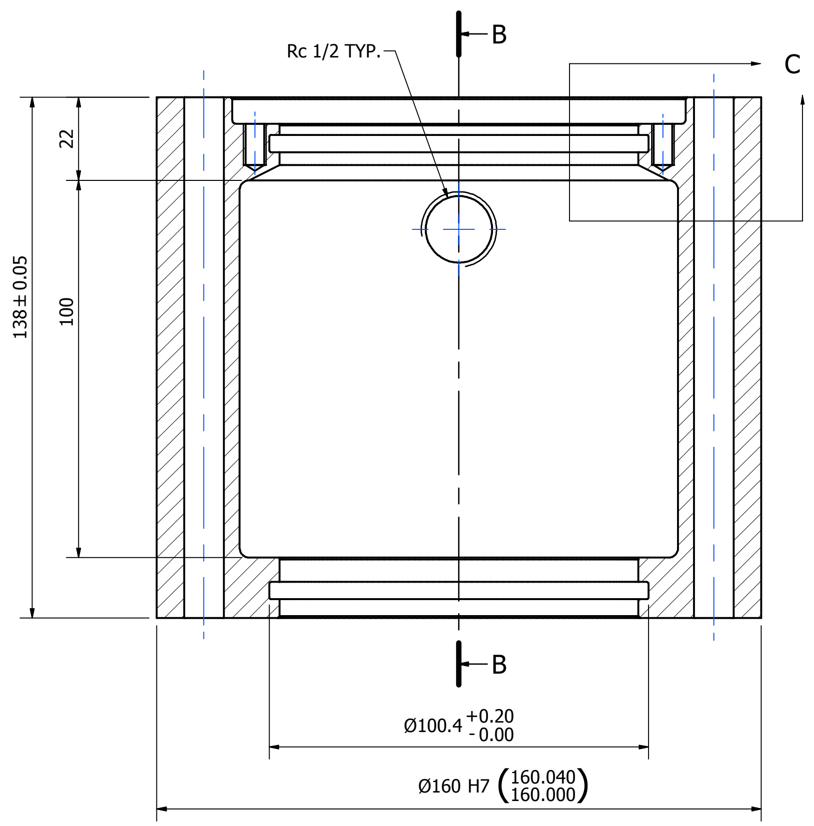
DETAIL F

SCALE: 2 : 1



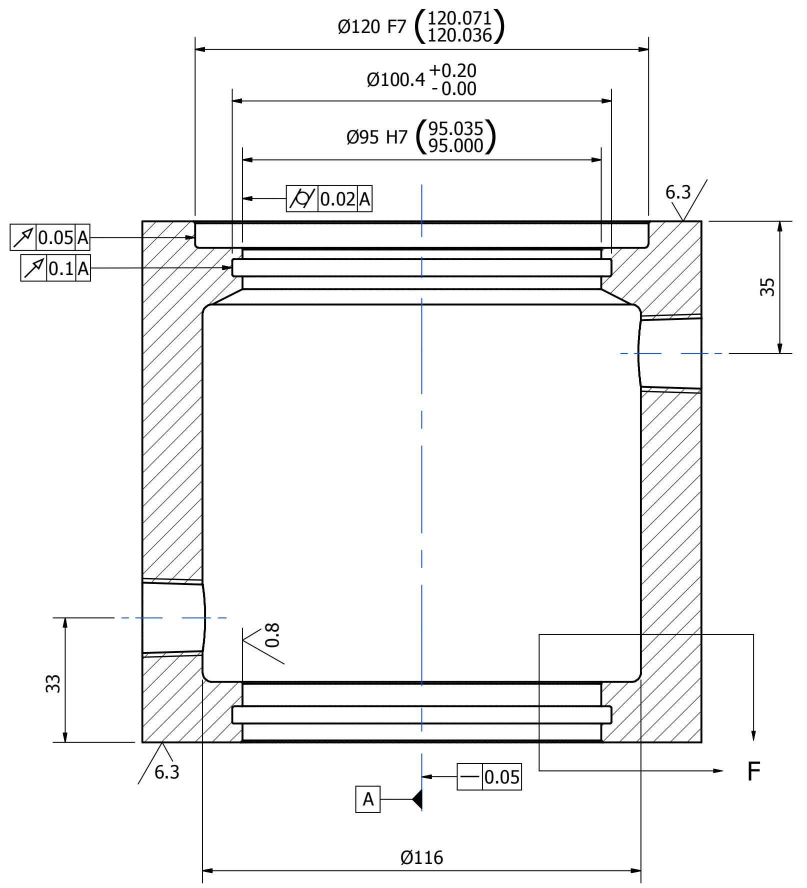
4111-M072 - ISOMETRIC VIEW -

SCALE: 1 : 1



SECTION A-A

SCALE: 1 : 1



SECTION B-B

SCALE: 1 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M072	1	AS1865 -2011T6	UPPER BARREL	3.6 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	06/08/14	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

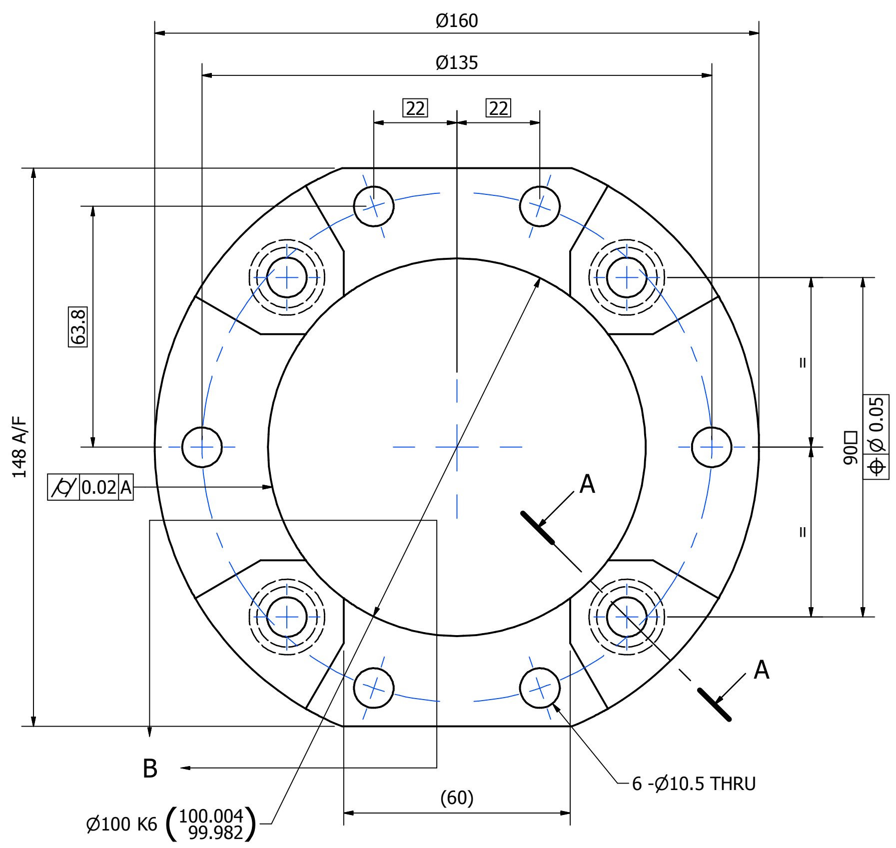
THE UNIVERSITY OF SOUTHERN QUEENSLAND.

		DATE	
DRAWN	KBD	29/07/2014	CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED	KBD	29/07/2014	DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
CHECKED			DRAWING DESCRIPTION: UPPER BARREL DETAIL
APPROVED			

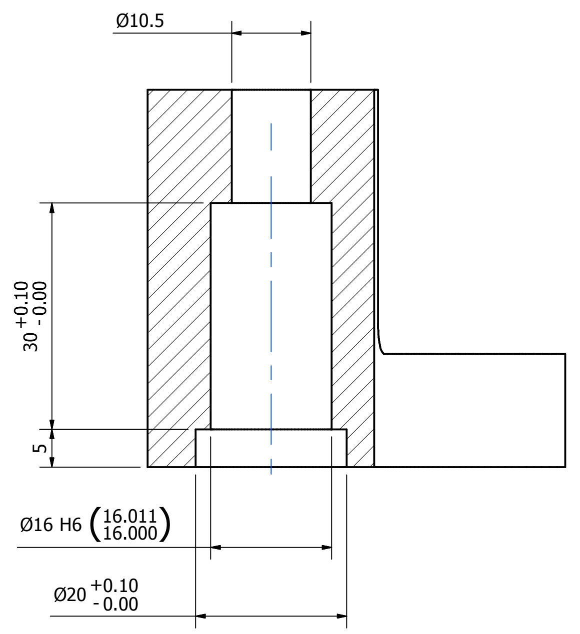
JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M072	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M072.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M072.idw

DO NOT SCALE - IF IN DOUBT ASK.

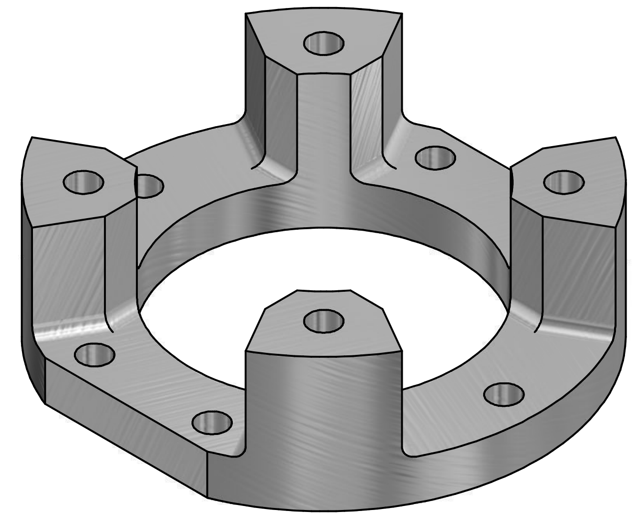


- TOP VIEW -



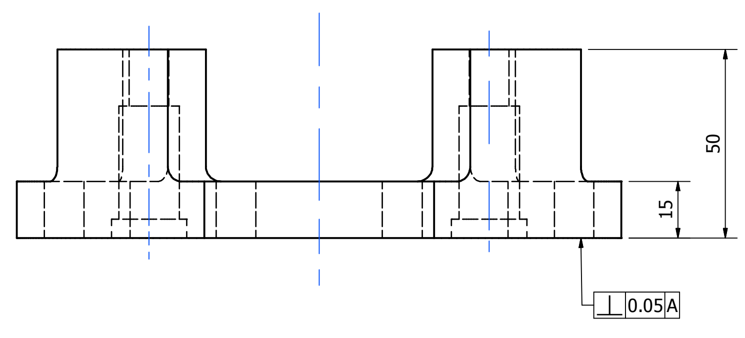
SECTION A-A

SCALE: 2 : 1



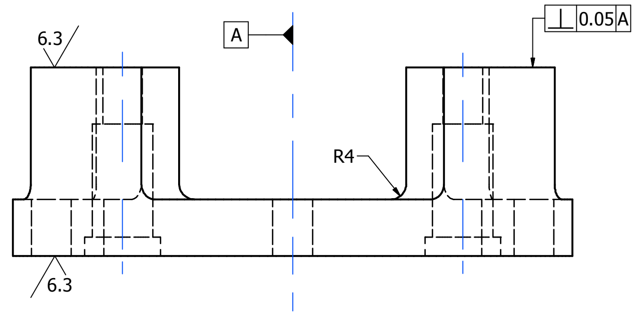
4111-M075 - ISOMETRIC VIEW -

SCALE: 1 : 1

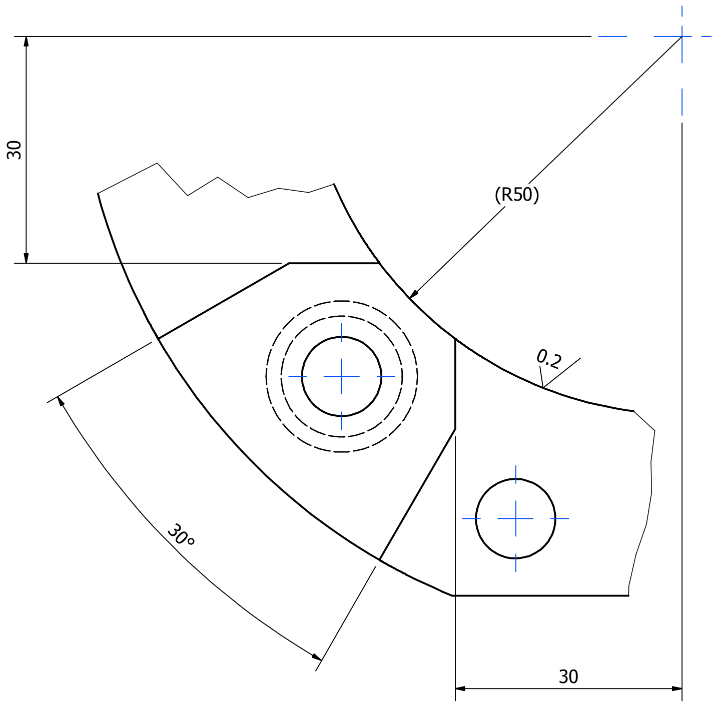


4111-M075 - DETAIL -

SCALE: 1 : 1



- SIDE VIEW -



DETAIL B

SCALE: 2 : 1

IMPORTANT: DO NOT FABRICATE UNTIL FINAL OPTICAL BORE DESIGN IS COMPLETE

MECHANICAL NOTES

- ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.
- WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005
- ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1
- ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.
- ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.
- HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.
- GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992
- MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.
- LIMITS AND FITS TO COMPLY WITH AS1654-1995.
- GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.
- SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

NOTE:
1. COLLAR TO BE FREE OF BURRS & SHARP EDGES BEFORE ASSEMBLY
2. THREADED FERRULE TO BE FITTED USING "LOCTITE" 680 RETAINING COMPOUND.

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M075	1	AS1865 -2011T6	OPTICAL ACCESS COLLAR	0.8 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	06/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN	DATE	DESIGNED	DATE	CHECKED	DATE	APPROVED	DATE
KBD	29/07/2014	KBD	29/07/2014				

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: OPTICAL ACCESS COLLAR DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M075	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M075.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M075.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

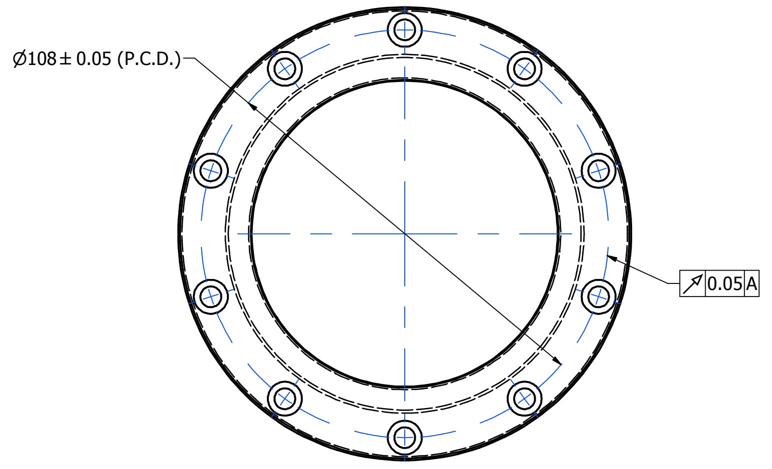
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

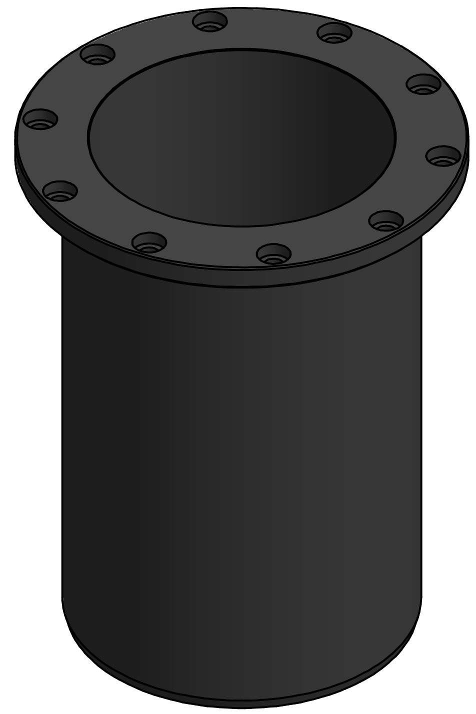
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

NOTES:

1. CASTING TO BE FREE OF CRACKS OR POROSITY
2. STRESS RELIEVE FINISHED COMPONENT
   - HEAT FROM A COLD OVEN TO 550°C - 580°C
   - SOAK FOR 2+ HRS AT TEMPERATURE
   - SWITCH OFF OVEN AND ALLOW TO COOL TO 100°C BEFORE REMOVAL
3. HONE CROSS HATCH PATTERN USING A SILICON CARBIDE HONE (45-65° PATTERN) NOTE "LAY" DIRECTION
4. WIPE BORE CLEAN AFTER HONING USING A CLEAN LIGHTLY OILED CLOTH (MORE THAN ONCE IF REQ'D).

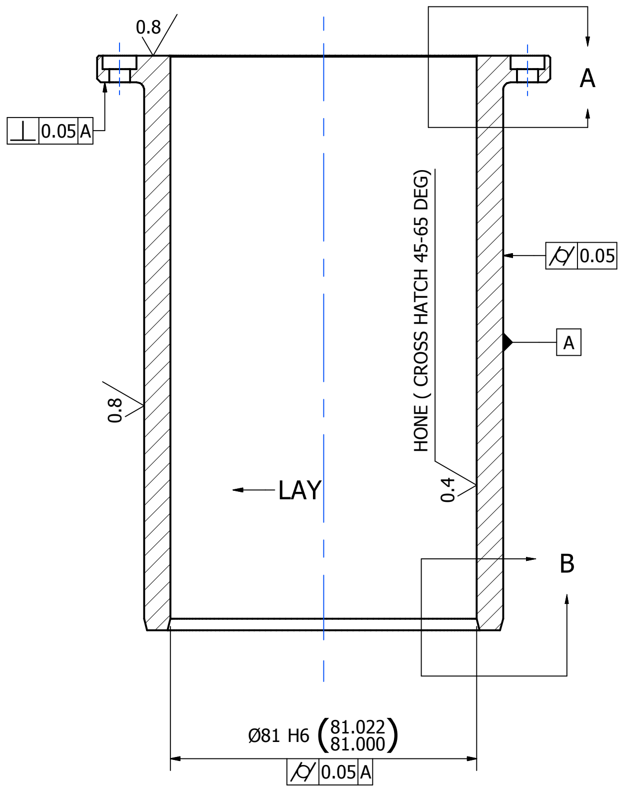


- TOP VIEW -



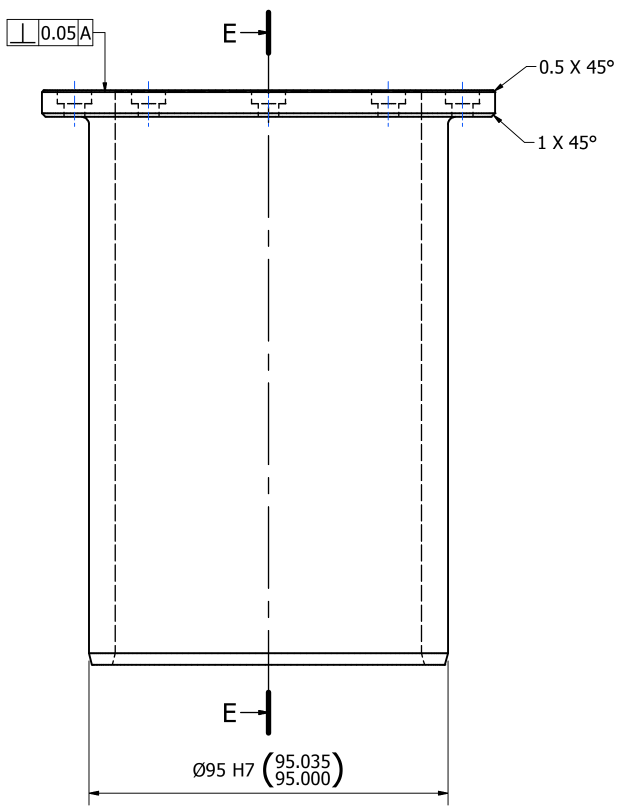
4111-M078 - ISOMETRIC VIEW -

SCALE: 1 : 1



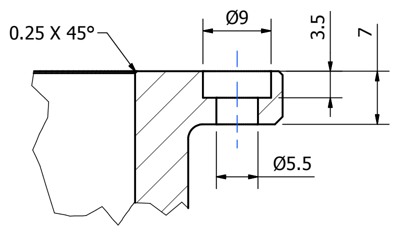
SECTION E-E

SCALE: 1 : 1



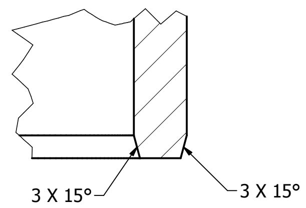
4111-M078 - DETAIL -

SCALE: 1 : 1



DETAIL A

SCALE: 2 : 1



DETAIL B

SCALE: 2 : 1

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M078	1	2P Cast Iron	UPPER CYLINDER SLEEVE	2.3 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	06/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD DATE: 13/07/2014

DESIGNED: KBD DATE: 13/07/2014

CHECKED: DATE:

APPROVED: DATE:

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: LOWER CYLINDER SLEEVE DETAIL

JOB No.: ENG4111/2 DRAWING No.: 4111-M078 1 OF 1 SIZE: A1 REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M078.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M078.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

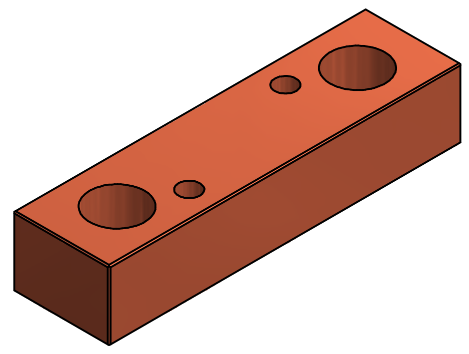
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

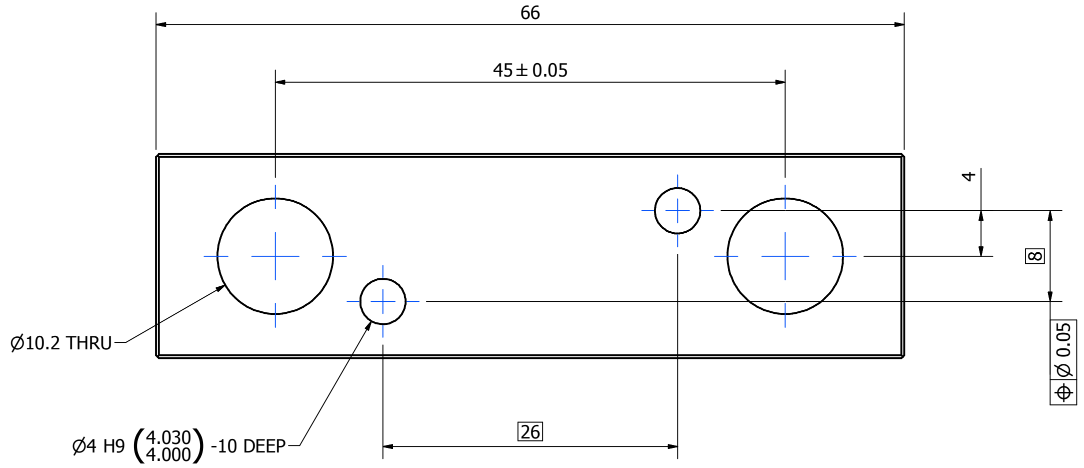
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

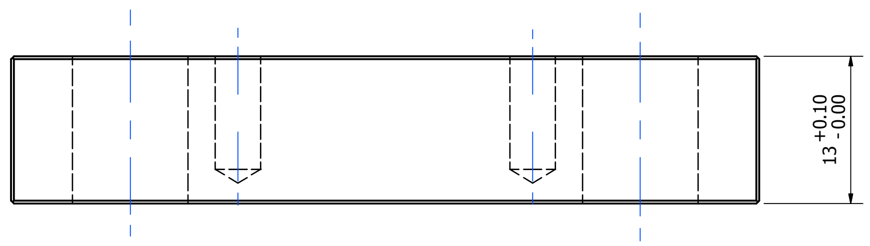


4111-M080 - ISOMETRIC VIEW -

SCALE: 2 : 1

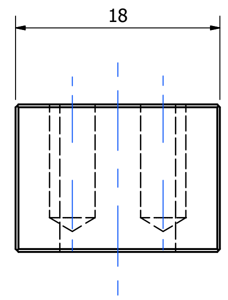


- TOP VIEW -



4111-M080 - DETAIL -

SCALE: 3 : 1



- END VIEW -

4111-M080	1	SD170	CRANK BALANCE WEIGHT	0.22 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN: KBD	DATE: 13/07/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 13/07/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	CRANK BALANCE WEIGHT DETAIL
APPROVED	DATE	JOB No. ENG4111/2	DRAWING No. 4111-M080 1 OF 1 SIZE A1 REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M080.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M080.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

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HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

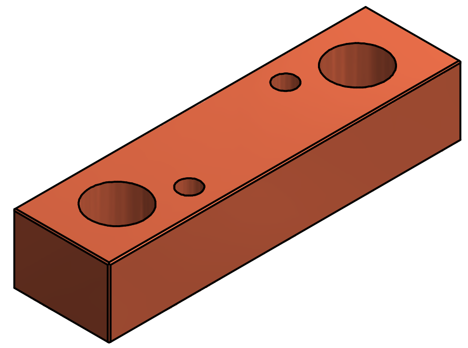
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

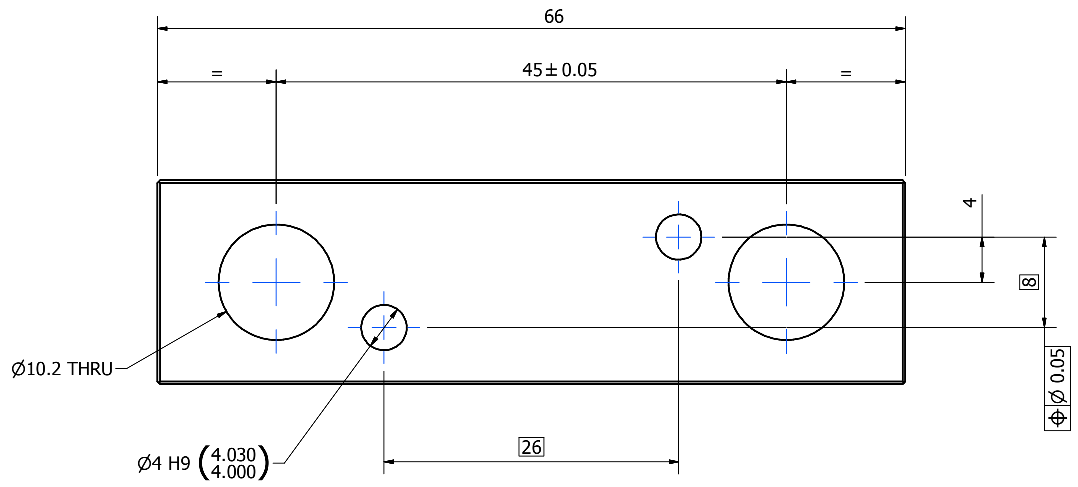
SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

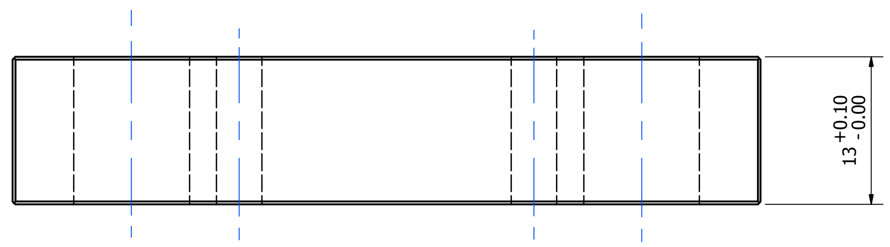


4111-M081 - ISOMETRIC VIEW -

SCALE: 2 : 1

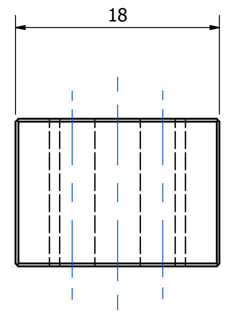


- TOP VIEW -



4111-M081 - DETAIL -

SCALE: 3 : 1



- END VIEW -

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M081	1	SD170	CRANK BALANCE WEIGHT	0.219 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	3.2

ALL DIMENSIONS IN MILLIMETRES UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN
DRAWN	KBD — DATE 13/07/2014
DESIGNED	KBD — DATE 13/07/2014
CHECKED	DATE
APPROVED	DATE

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: CRANK BALANCE WEIGHT DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M081	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M081.ipt
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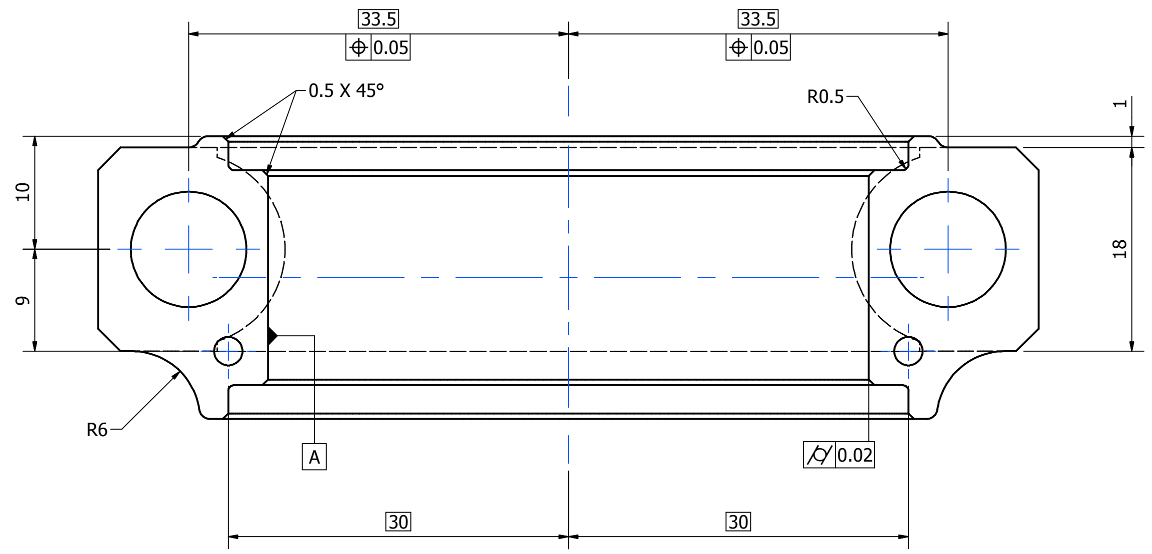
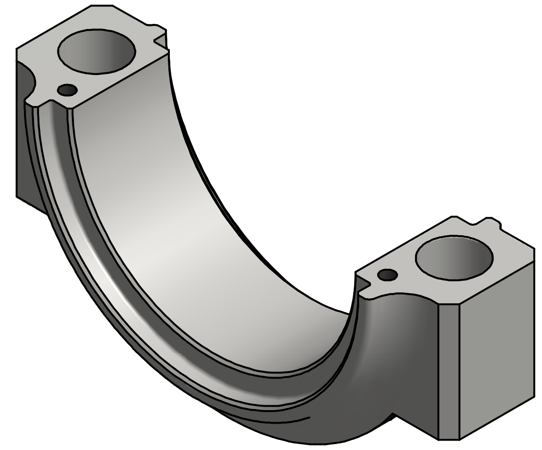
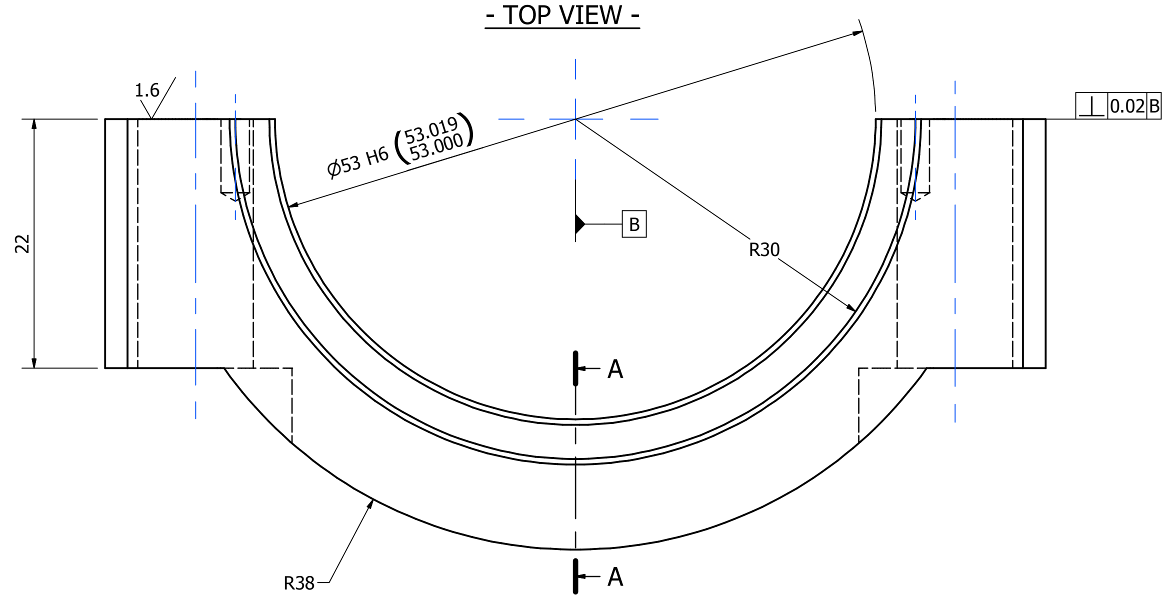
DO NOT SCALE - IF IN DOUBT ASK.



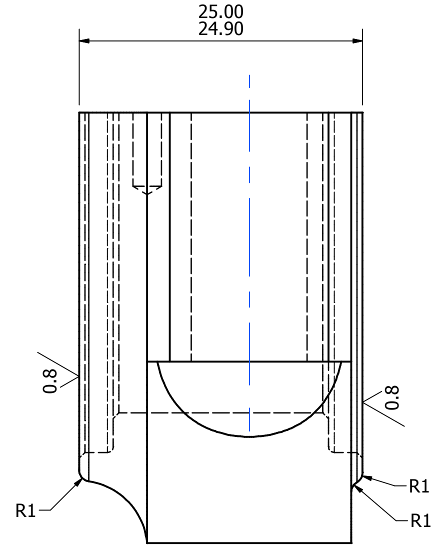
- TOP VIEW -



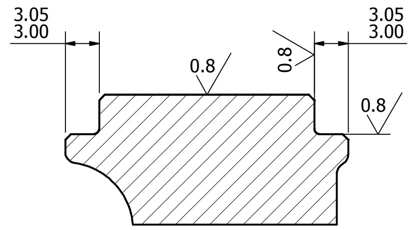


4111-M091 - DETAIL -

SCALE: 3 : 1

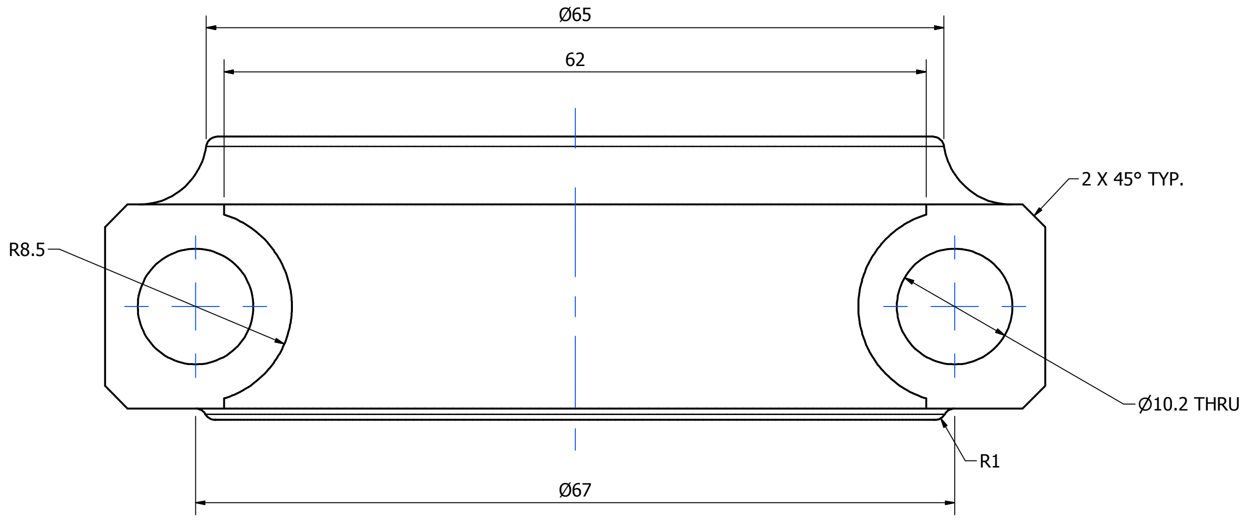


- SIDE VIEW -



SECTION A-A

SCALE: 3 : 1



- BOTTOM VIEW -

NOTE:

1. DIMENSIONS SHOWN THUS: TO BE FINISH MACHINED WITH CAP ASSEMBLED TO CONNECTING ROD. FIXINGS (BOLTS) TO BE LIGHTLY OILED & TORQUED TO 55Nm FOR MACHINING.
2. CONROD ASSEMBLIES TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A021 +/- 0.03kg, +/- 0.05mm
3. ALL SHARP EDGES TO BE REMOVED
4. ALL CORNERS / RADII TO BE 0.5mm UNLESS NOTED.
5. ALL PARTS TO FIT TOGETHER WITHOUT FORCING

HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- NITRIDE AFTER FINAL MACHINING

MACHINING:
- ROUGH MACHINE TO 3mm ABOVER FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.
- NOTE THE DIRECTION OF FINISH LAY ◄►

SURFACE TREATMENT:
- SURFACE TO BE "BLUED" ON COMPLETION (REFER 4111-A001 FOR DETAILS)

IMPORTANT:
- CRANKSHAFT IS OFFSET AXIALLY 2.5mm TOWARDS REAR OF CRANKSHAFT TO ALLOW FOR SAME OFFSET IN CONROD

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M091	1	AS1444 -4340	CONROD BIG END SADDLE	0.19 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

SCALE: AS SHOWN		CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DRAWN: KBD	DATE: 3/08/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
DESIGNED: KBD	DATE: 3/08/2014	DRAWING DESCRIPTION	BIG END CAP DETAIL
CHECKED	DATE		
APPROVED	DATE	JOB No.: ENG4111/2	DRAWING No.: 4111-M091 1 OF 1; SIZE: A1; REV.: 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-M091.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M091.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

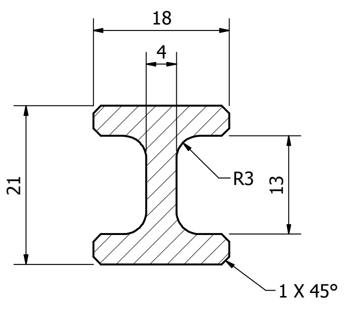
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

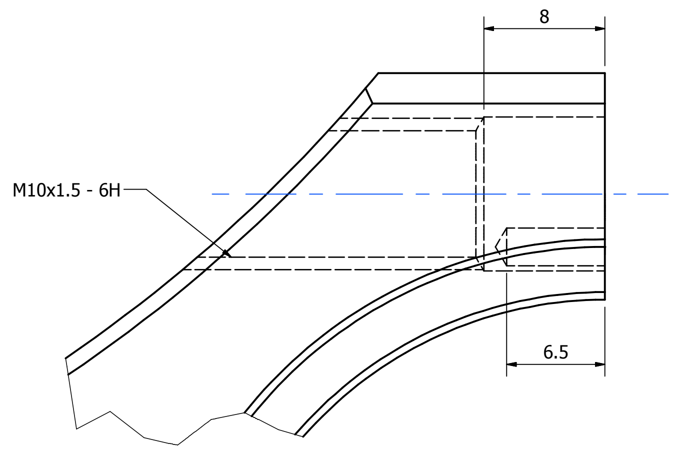
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

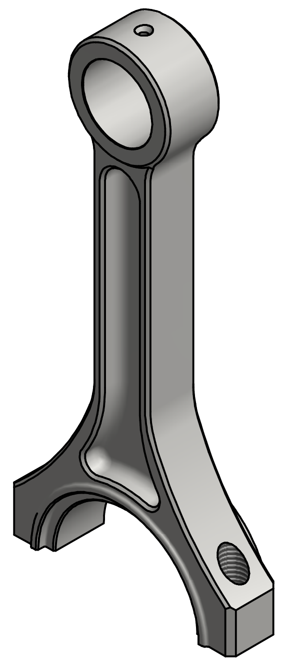
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS



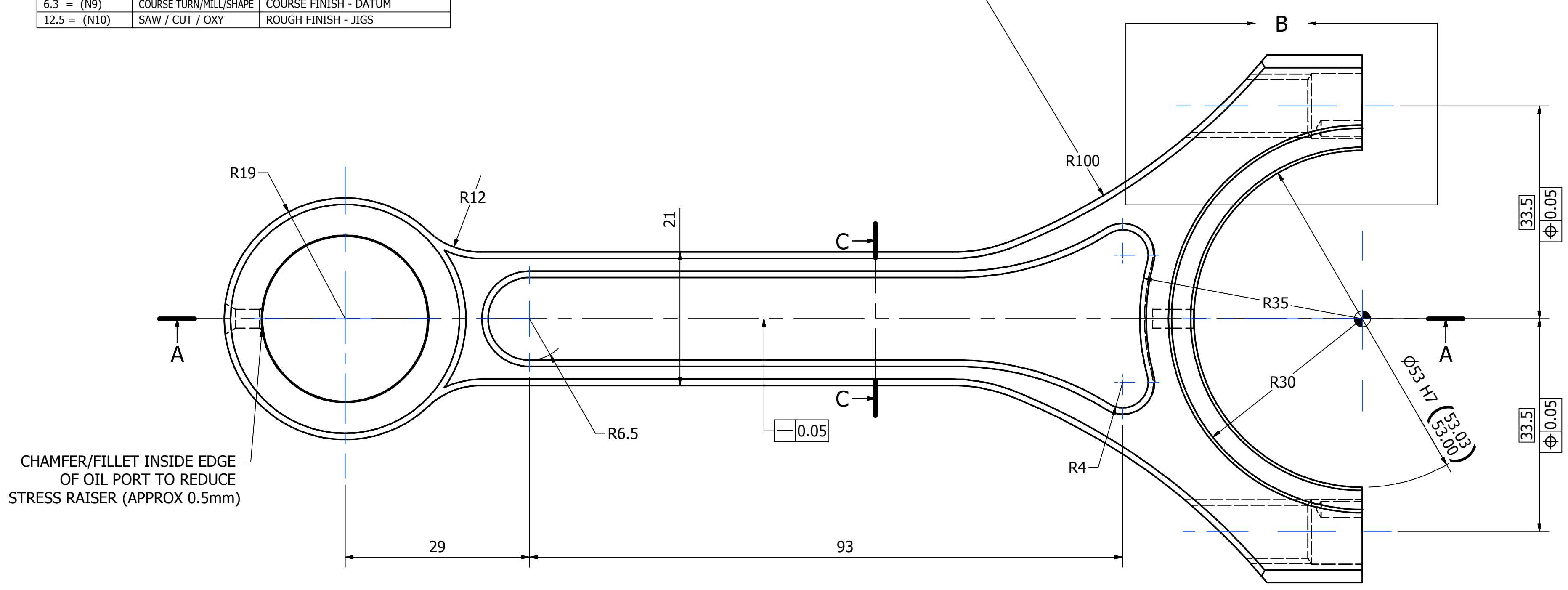
SECTION C-C
SCALE: 2 : 1



DETAIL B
SCALE: 4 : 1



4111-M092 - ISOMETRIC VIEW -
SCALE: 1 : 1



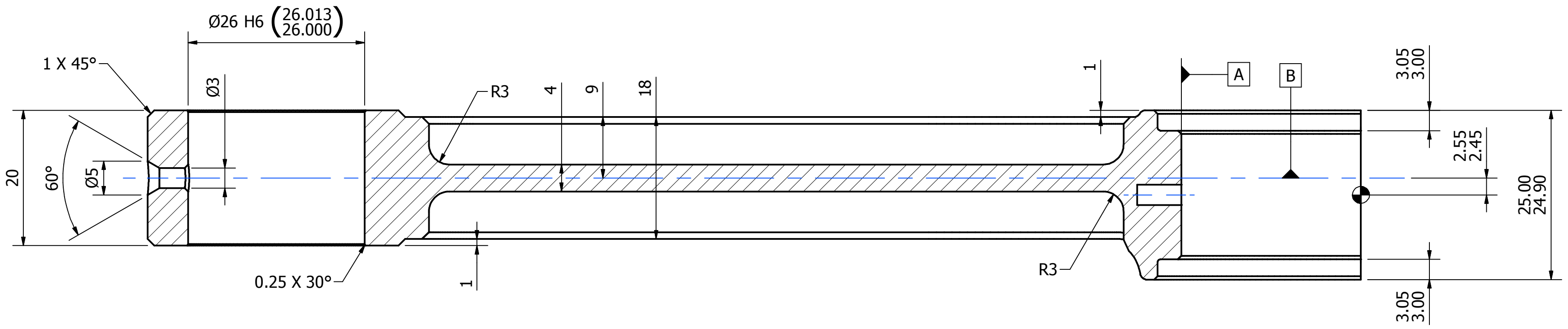
4111-M092 - DETAIL -
SCALE: 2 : 1

NOTE:
1. DIMENSIONS SHOWN THUS: TO BE FINISH MACHINED WITH CAP ASSEMBLED TO CONNECTING ROD. FIXINGS (BOLTS) TO BE LIGHTLY OILED & TORQUED TO 55Nm FOR MACHINING.
2. CONROD ASSEMBLIES TO BE BALANCED TO MASS & C.O.G. SHOWN IN 4111-A021 +/- 0.03kg, +/- 0.05mm
3. ALL SHARP EDGES TO BE REMOVED
4. ALL TIGHT CORNERS / RADII TO BE IDENTIFIED AND ADDRESSED.
5. ALL PARTS TO FIT TOGETHER WITHOUT FORCING

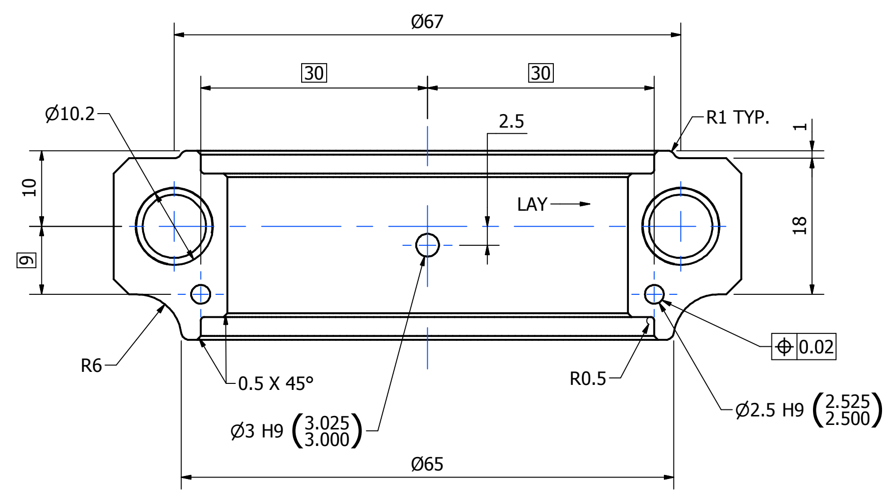
HEAT TREATMENT:
- NORMALIZE AS SPECIFIED IN 4111-A001
- TEMPER AS SPECIFIED IN 4111-A001 ("T" CONDITION)
- HEAT TO APPROX 150°C AND DIP IN USED SUMP OIL TO TREAT SURFACE UPON COMPLETION.

MACHINING:
- ROUGH MACHINE TO 3mm ABOVE FINAL DIMENSIONS & HEAT TREAT TO TO "T" CONDITION.
- SURFACE FINISHES TO 0.2Ra TO BE POLISHED IN OPPOSITE DIRECTION TO GRINDING.
- NOTE THE DIRECTION OF FINISH LAY ◄►

IMPORTANT:
- CRANKSHAFT IS OFFSET AXIALLY 2.5mm TOWARDS REAR OF CRANKSHAFT TO ALLOW FOR SAME OFFSET IN CONROD



SECTION A-A
SCALE: 2 : 1



- END VIEW -

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-M092	1	AS1444 -4340	CONROD	0.48 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm ..... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

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REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	03/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

DRAWN	DATE
KBD	13/07/2014
DESIGNED	DATE
KBD	13/07/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: CONROD DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-M092	1 OF 1	A1	0

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D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-M092.idw

DO NOT SCALE - IF IN DOUBT ASK.

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

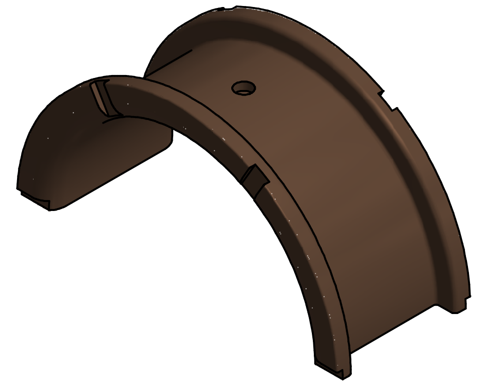
MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

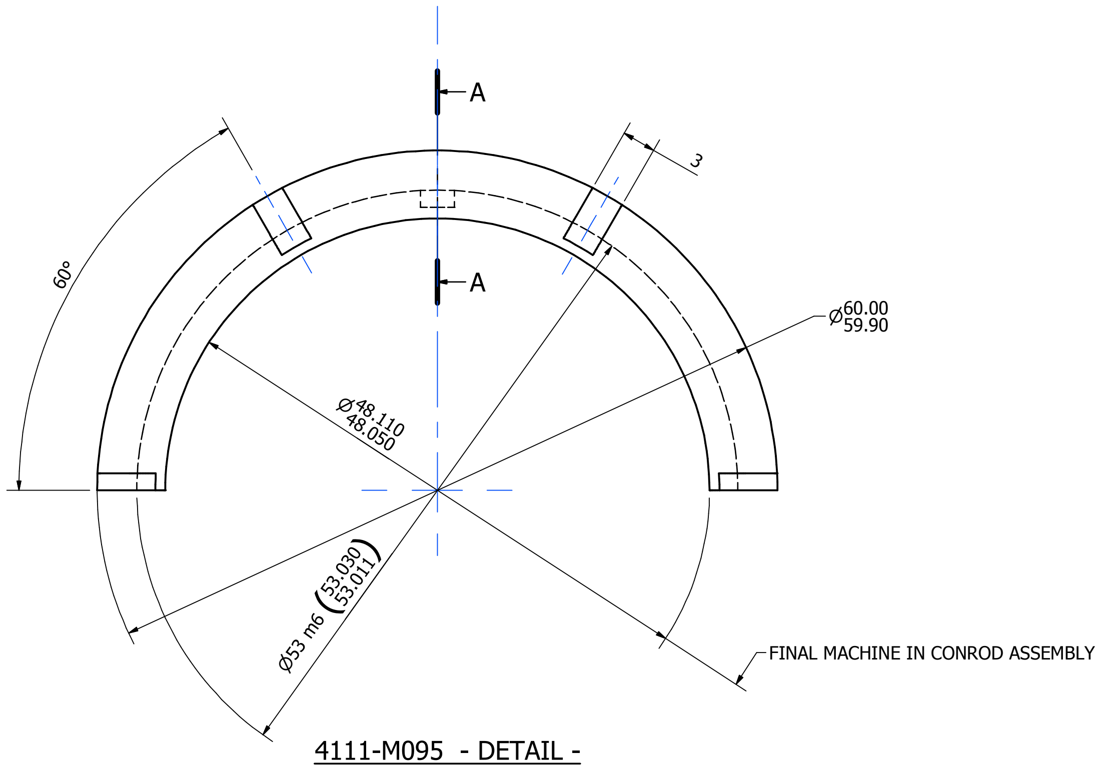
GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

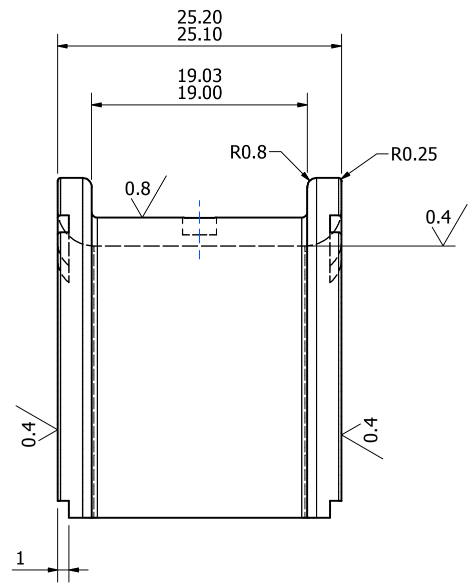
Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

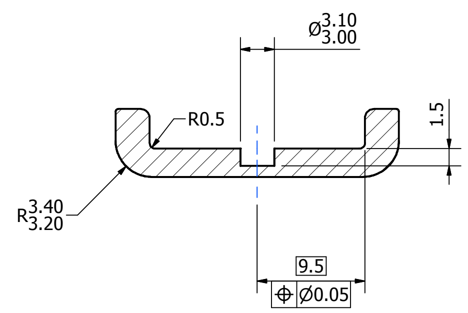


4111-M095 - ISOMETRIC VIEW -
SCALE: 2 : 1



4111-M095 - DETAIL -
SCALE: 3 : 1





SECTION A-A
SCALE: 3 : 1

4111-M095	2	954 - AL. BRONZE	BIG END BEARING SHELL	0.047 kg
PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS

PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		
QA CHECKLIST	SIGNED	DATE

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	10/08/2014	
0	ISSUED TO USQ	30/10/2014	

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PH: +61 7 4631 2100
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EMAIL: study@usq.edu.au

SCALE AS SHOWN		THE UNIVERSITY OF SOUTHERN QUEENSLAND.
DRAWN KBD	DATE 10/08/2014	CLIENT UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED KBD	DATE 10/08/2014	DESCRIPTION OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION LITTLE END BEARING
APPROVED	DATE	JOB No. ENG4111/2 · DRAWING No. 4111-M095 · 1 OF 1 · SIZE A1 · REV. 0

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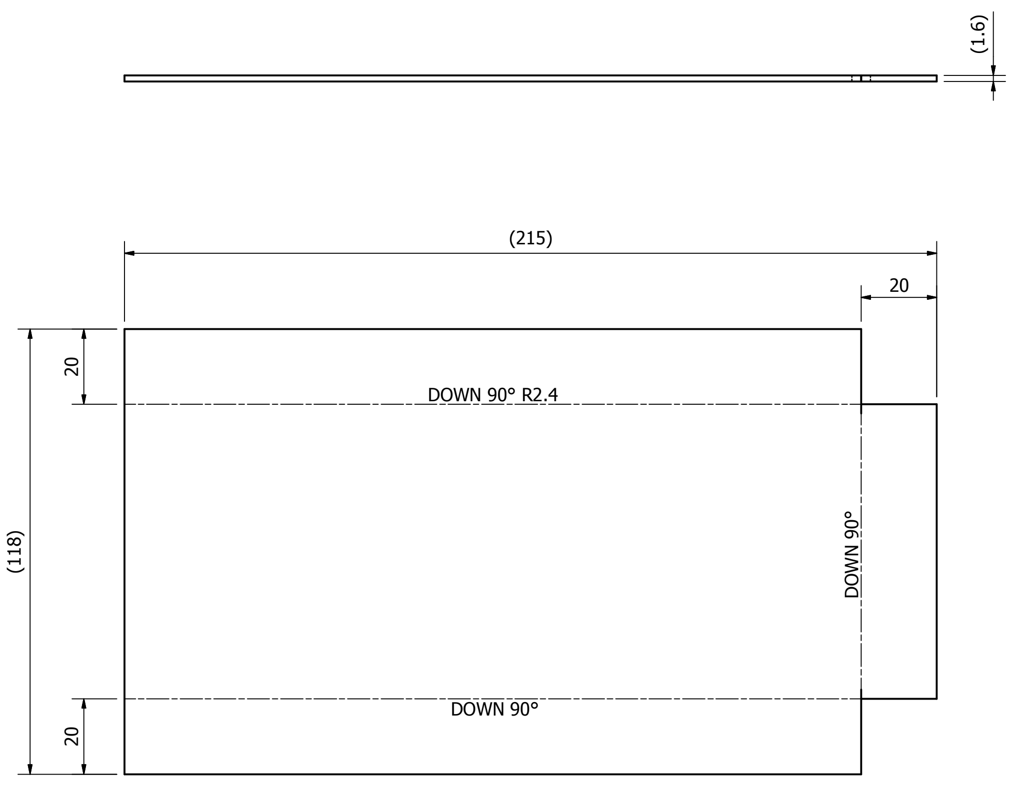
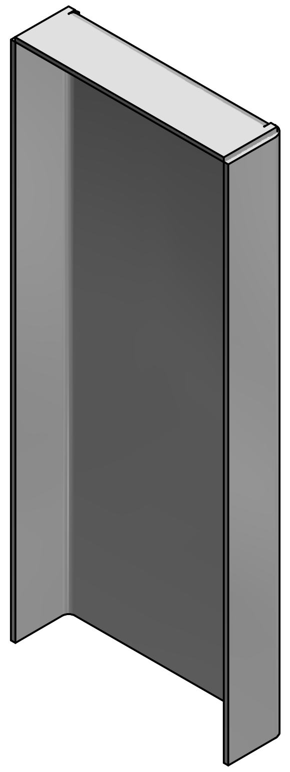


4111-S001 - FLAT PATTERN -
SCALE: 1 : 1



4111-S001 - ISOMETRIC VIEW -
SCALE: 1 : 1

PART NUMBER	MATERIAL	SIZE	DESCRIPTION	QTY	MASS	KEYWORDS
4111-S001	AS1734 6061T6	1.6 mm	OIL COVER	1	0.1 kg	411-S004; 4111-CUT FILE

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	09/08/2014	
0	ISSUED TO USQ	30/10/2014	

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University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

DRAWN: KBD	DATE: 9/08/2014	CLIENT	UNIVERSITY OF SOUTHERN QUEENSLAND
DESIGNED: KBD	DATE: 9/08/2014	DESCRIPTION	OPTICAL ACCESS ENGINE DESIGN
CHECKED	DATE	DRAWING DESCRIPTION	OIL COVER DETAIL
APPROVED	DATE		

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JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-S001	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-S001.ipt
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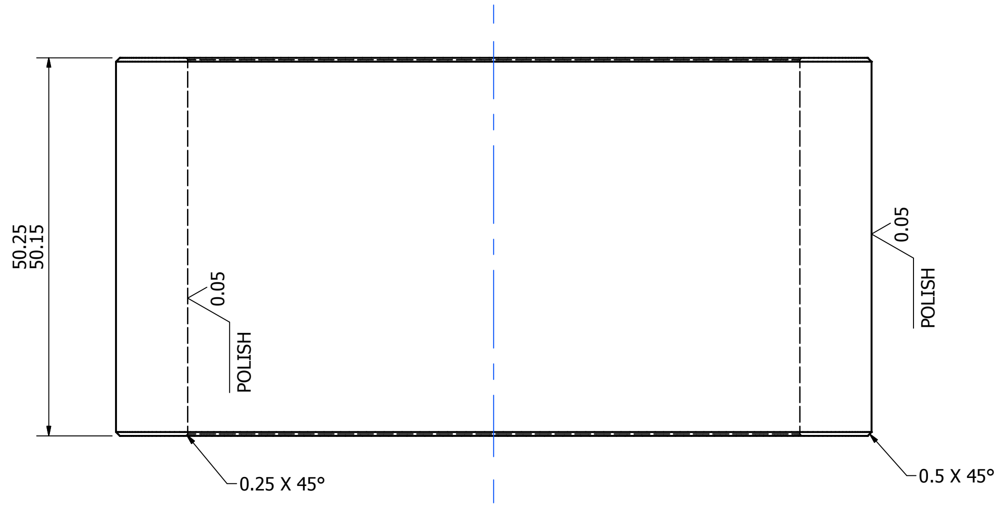
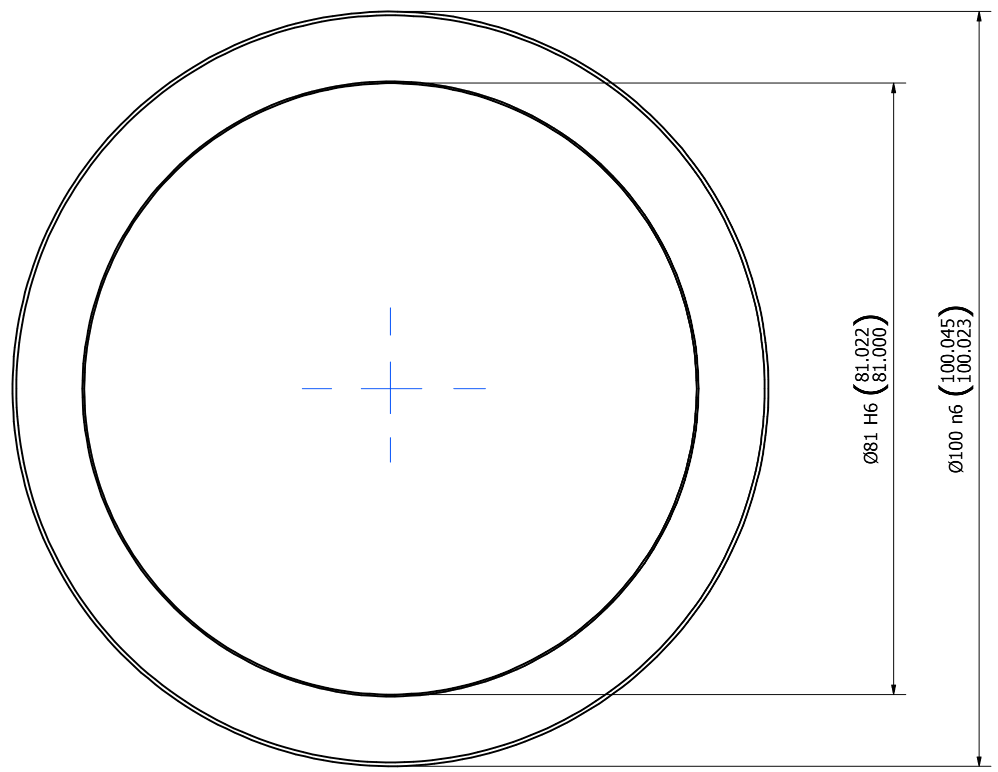


- SIDE VIEW -

4111-X001 - ISOMETRIC VIEW -

SCALE: 1 : 1

IMPORTANT: DO NOT FABRICATE UNTIL FINAL OPTICAL BORE DESIGN IS COMPLETE



4111-X001 - DETAIL -

SCALE: 2 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUM	QTY	MATERIAL	DESCRIPTION	MASS
4111-X001	1	FUSED SILICA (QUARTZ)	OPTICAL BORE	0.3 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE: AS SHOWN

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DRAWN	DATE
KBD	4/08/2014
DESIGNED: KBD	DATE: 4/08/2014
CHECKED	DATE
APPROVED	DATE

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND

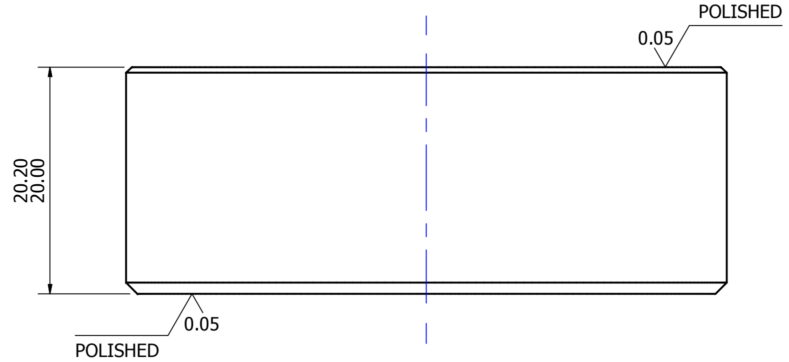
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN

DRAWING DESCRIPTION: OPTICAL ACCESS SLEEVE DETAIL

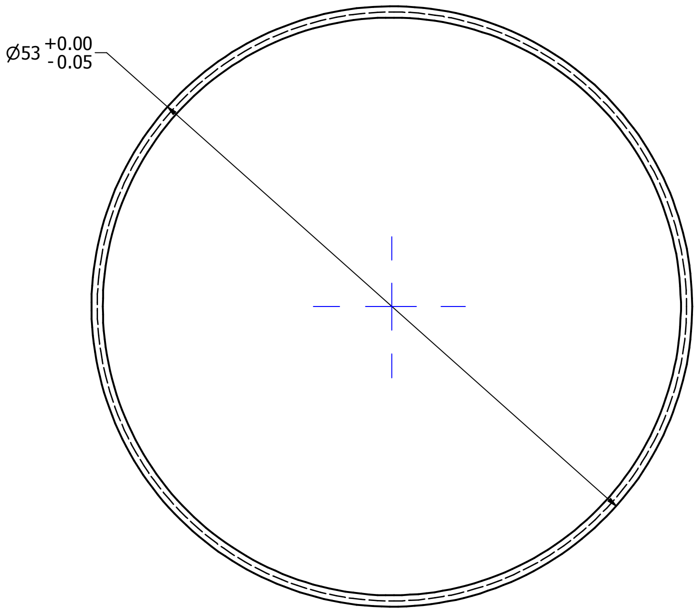
JOB No. ENG4111/2 | DRAWING No. 4111-X001 | 1 OF 1 | SIZE A1 | REV. 0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-X001.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-X001.idw

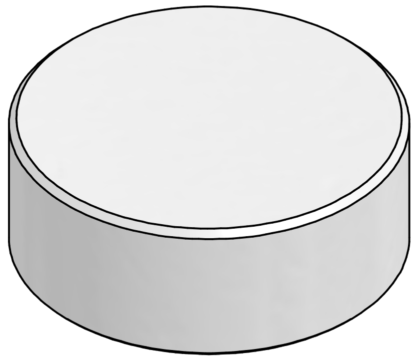
DO NOT SCALE - IF IN DOUBT ASK.



- SIDE VIEW -



4111-X002 - DETAIL -
SCALE: 3 : 1



4111-X002 - ISOMETRIC VIEW -
SCALE: 2 : 1

MECHANICAL NOTES

ALL DIMENSIONS ARE IN MILLIMETRES UNLESS NOTED OTHERWISE.

WELDING SYMBOLS ARE TO COMPLY WITH AS1101.3-2005

ALL WELDS AND WELD PREPARATION TO COMPLY WITH WITH AS/NZS1554.1

ALL WELDS ARE TO BE 6mm CONTINUOUS FILLET UNLESS NOTED OTHERWISE.

ALL SHARP EDGES AND CORNERS ARE TO BE REMOVED UNLESS NOTED OTHERWISE.

HOT ROLLED STEEL SECTIONS TO COMPLY WITH AS3679-2010 GRADE 300 UNLESS NOTED OTHERWISE.

GEOMETRIC TOLERANCES TO COMPLY WITH AS1100.201-1992

MACHINE SYMBOLS TO COMPLY WITH AS1100.201-1992.

LIMITS AND FITS TO COMPLY WITH AS1654-1995.

GENERAL DIMENSION TOLERANCES TO COMPLY WITH AS1100.201-1992.

SURFACE FINISH VALUES TABULATED BELOW.

Ra VALUE (um)	PROCESS	FINISH APPLICATION
0.4 = (N5)	FINE GRIND / HONING	HEAVY BRG. - HIGH SPEED SHAFT
0.8 = (N6)	GRIND / FINE TURN	LIGHT BRG. - LOW SPEED SHAFT
1.6 = (N7)	TURN / MILL / DRILL	GOOD FINISH - CLOSE FITS
3.2 = (N8)	TURN / MILL / DRILL	AVG. FINISH - GENERAL ENGINEERING
6.3 = (N9)	COURSE TURN/MILL/SHAPE	COURSE FINISH - DATUM
12.5 = (N10)	SAW / CUT / OXY	ROUGH FINISH - JIGS

PART NUMBER	QTY	MATERIAL	DESCRIPTION	MASS
4111-X002	1	FUSED SILICA (QUARTZ)	OPTICAL ACCESS WINDOW	0.1 kg

QA CHECKLIST	SIGNED	DATE
PROJECT MANAGER APPROVAL		
MAT / EQUIP COMPLIANCE		
PRE WELD / ASSEMBLY CHECK		
POST WELD / ASSY CHECK		
VISUAL WELD INSPECTION		
EQUIP. PRE COMMISIONING		

GENERAL TOLERANCES UNLESS INDICATED

PREPARATION AND ASSEMBLY	MACHINING
< 25mm .... ±1.0mm	0 .... ±0.1mm
25 - 600mm .... ±1.5mm	0.0 .... ±0.05mm
>600mm .... ±2.0mm	0.00 .... ±0.01mm
ANGLE .... ±1°	.... 3.2

ALL DIMENSIONS IN MILLIMETERS UNLESS NOTED OTHERWISE

REVISION HISTORY

REV	DESCRIPTION	DATE	CHECKED
A	PRELIMINARY ISSUE	04/08/2014	
0	ISSUED TO USQ	30/10/2014	

UNIVERSITY OF SOUTHERN QUEENSLAND
University of Southern Queensland
West St. Toowoomba - 4350 QLD.
PH: +61 7 4631 2100
Website: www.usq.edu.au
EMAIL: study@usq.edu.au

SCALE	AS SHOWN		
DRAWN	KBD	DATE	4/08/2014
DESIGNED	KBD	DATE	4/08/2014
CHECKED		DATE	
APPROVED		DATE	

THE UNIVERSITY OF SOUTHERN QUEENSLAND.

CLIENT: UNIVERSITY OF SOUTHERN QUEENSLAND
DESCRIPTION: OPTICAL ACCESS ENGINE DESIGN
DRAWING DESCRIPTION: OPTICAL ACCESS WINDOW DETAIL

JOB No.	DRAWING No.		SIZE	REV.
ENG4111/2	4111-X002	1 OF 1	A1	0

D:\Inventor Data\ENG4111 DESIGN PROJECT\Parts\4111-X002.ipt
D:\Inventor Data\ENG4111 DESIGN PROJECT\IDW Files\4111-X002.idw

Appendix E: Materials List

List of Materials Used and Properties

Note: Certain materials may not have been used in the final design.

Material properties used for fatigue analysis may differ slightlyfrom this table

Material Properties List

Material Name	Supplier	Standard	Tensile Strength Mpa	Yield Strength Mpa	Youngs Modulus Gpa (>100 deg C)	Poisson's Ratio	Density g\cm^3	Thermal Conductivity (W/m.K)	Specific Heat j/(kg.K)	Thermal Exp Co-eff 10^6/K (100 deg C)	Behavior	Other/Comments
4340	Bohler	AS1444	1000	700	205	0.29	7.85	42	460	11.1	Isotropic	Crankshaft & Conrod Steel
4140	Bohler	AS1444	1010	740	205	0.29	7.85	42	460	11.1	Isotropic	Shaft Steel
1020	Bohler	AS1444	410	230	205	0.29	7.87	51.9	486	11.7	Isotropic	Misc Bright Steel
EN36 A	Bohler	AS1444	930	635	205	0.29	7.85	34	460	11.1	Isotropic	Gear Material
Assab-01	Bolher	N/A					7.81	32	460	6.3	Isotropic	Silver Steel
2P CI	Bohler	N/A	220		90	0.29	7.15				Isotropic	Cast Iron Sleeve Material
GR 250	Bluescope	AS3678	345	250	220	0.28	7.86	56	460	12	Isotropic	Plate material (Flywheel etc.)
6061-T6	Capral	AS1734	260	240	68.9	0.33	2.7	167	896	23.6	Isotropic	Plate Aluminium
4032-T6	Unknown	AS1866	379	317	78.6	0.34	2.68	138	850	18	Isotropic	Optional Piston Material
2618-T61	Unknown	AS1866	440	372	74.4	0.33	2.76	147	875	20.6	Isotropic	Optional Piston Material
2011-T6	CAPRAL	AS1866	280	240								Upper & Lower Barrel Material
SAE841 (Oilite)	Blackwoods/BSC	N/A	96.5				6.4				Isotropic	Max Pressure Velocity 1.75 Mpa.m/s -Sintered Bronze Bearing Material
Densalloy SD170	Unknown	N/A	827	552			16.85				Isotropic	Balance Weight Material
Tetron C	Dotmar	N/A	14				2.17			114	Isotropic	Max operating Temp = 310 deg C (4hrs) Top Piston Compression Ring Material
Tetron B	Dotmar	N/A	16				3.9			134	Isotropic	Max operating Temp = 310 deg C (4hrs) Piston Guide Material (Slide Bush)
Viton	BSC	N/A	15.2		0.0073		1.77				Isotropic	O-ring material (optional)
Nitrile	BSC	N/A	14								Isotropic	O-ring material (optional)
954 -Al. Bronze	Bohler	N/A	515	205	115	0.318	7.45	58.6	375	16.5	Isotropic	Bearing Material
HSQ -300	Heraeus	N/A	50	N/A	72.5	0.17	2.203	1.38	772	0.51		Quartz Plate & Tube