University of Southern Queensland

Faculty of Health, Engineering and Sciences

Continuously Variable Ratio Rocker Arms

A dissertation submitted by

Benjamin Schnebli

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Towards the degree of

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ENG4111 & ENG4112 Research Project

Limitations of Use

Certification

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.

Benjamin Schnebli

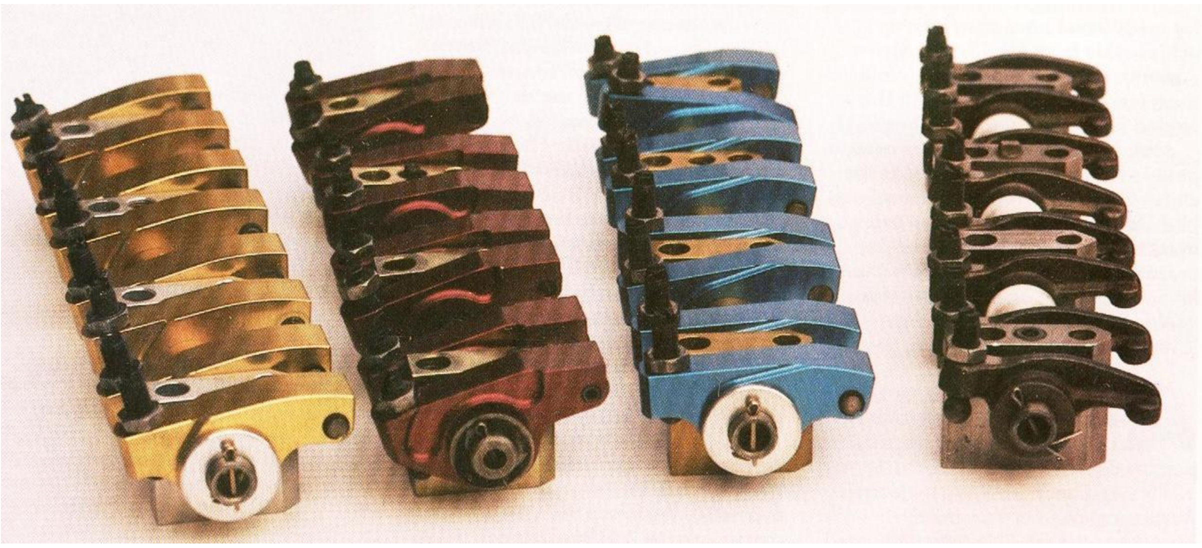
Student Number: [redacted]

Abstract

A design for continuously variable ratio rocker arms was to be considered as a personal project for a a-series engine. It is the desire for me to design a variable lift system to be used in older pushrod engines.

The purpose of this paper is to design a continuously variable valve lift that can be used with a british motor company a-series engine as for completion of ENG4111 and ENG4112 reasearch project. First current literature was conducted to find out what designs are available and how they are implenented. From this a methodology was created to generate a design for the A-series engine.

Finally the results are published showing how the design was established and a material outcome is conluded. Conclusions and further work is published along with a complete set of detailed and assembly drawings.



Aftermarket aluminium rockers and standard cast rockers (Calver special tuning-Rocker gear)

Contents

Abbreviations

VVL - variable valve lift

CVVL - Continuously variable valve lift

VCT - Variable cam timing

BMC - British motor company

BMW - Bavarian motor works

Dyno - Dynamometer

FWD - Front wheel drive

RWD - Rear wheel drive

Chapter 1 – Introduction

1.1 Background

The BMC A-series was first put into production in 1952 (MGCC 2017). This first production version of the engine was used to power motor cars and had a capacity of 803cc. The first car that this was used in was the Austin A30 from 1952 – 1956. The engine was also used in some early Morris minors in the same 803cc capacity. In 1957 the capacity was increased to 948cc by increasing the bore size from 2.28 to 2.477 inch (David Vizard, tuning the A-series engine pg 16). In 1958 the engine was used for the mini but in front wheel drive east west orientation. Although these engines are essentially the same there are some differences. The FWD engine has the gearbox housed in the sump and the gearbox is driven through a set of gears from the crankshaft. The first FWD engine was the same as the 948cc engine but with a reduced crankshaft stroke, this resulted in a capacity 848cc.

The BMC A-series engine was used in a multitude of different variants with later in 1964 the biggest version was produced with 1275cc capacity. The engines all have the same architecture and parts can mostly be shared.

When building a-series engines for minis, rocker ratio is of great importance. An example engine was capable of having the power curve shifted higher with a set of 1.3 ratio rockers compared with the standard 1.2 ratio rockers. The standard 1.2 ratio gave a lower RPM peak power and torque. The engine was a 1098cc A-series with 12g295 cylinder head, Graham Russell camshaft and Garrett GT2554 turbocharger.

For everyday driving where the engine spends most of its time at low RPM, a lower ratio generally will be more valuable. This is due to the fact that most of the time your engine will reside in the lower RPM range unless the car is used for racing. This can be exaggerated even more with different cam choices and by using ratios of 1.5 and higher. Removing and changing the rockers is not a hard task but requires removing the back cylinder head studs resulting in a risk of head gasket failure. Being able to easily change rocker ratio or even have it controlled for you by a computer while the engine is running would unlock some performance and drivability by being able to utilise the high and low power that the engine could develop.

1.2 Project Aim

The Project aim is to produce a design and drawings of a variable valve lift system that can be incorporated into the A-series engine. I could then use the drawings to manufacture a complete working VVL system for use with my own engine. The aim is to produce a complete working design within this paper. Including the following:

- General drawings and CAD models
- Detail and workshop drawings
- Design calculations for stress and verification

- Component materials
- Simulations and finite element analysis

This will give complete details for the project allowing the complete manufacture the variable ratio rocker system.

1.3 Scope of work

A list of the work that is performed for this project is listed below,

- Research on existing production and non production methods and designs of VVL is performed.
- Determine a design that would work with the A-series cylinder head.
- Determine the amount of variation of lift through research and simulations.
- Design the components for the VVL.
- Perform hand calculations and Finite element analysis (FEA) to ensure the design will be acceptable for use. This includes fatigue life and stress analysis.
- Verify the design
- Produce detail, assembly drawings and material specifications for manufacture.

1.4 Knowledge gap

Variable valve lift is no means a new innovation as it is currently used with a number of manufactures. Manufactures using this system in discrete (switchable between two lift heights) form are more common that continuously variable. From my research the major manufactures using the discrete method are listed below.

- Honda
- Nissan
- Toyota
- General motors
- Fiat
- Porsche
- Subaru
- Mitsubishi

Manufactures than I have found to be using a method of continuously variable valve lift are only BMW, Honda, Nissan, Toyota and fiat.

The introduction shows that there is a good advantage of using variable valve lift. There are a large number of systems available although a reliable continuously variable rocker arm system for use with a pushrod engine has not been successfully used or documented. Some patents show possible solutions for older pushrod engines although no reliable solutions can be found for the BMC A-series engine or even current use for any pushrod engines. The closest possible solution is that of William W. Entzminger (1989 patented) with his gear rack design. All the manufactures using a variable valve lift system are using in conjunction with an overhead camshaft engine particularly dual overhead camshafts. As pushrod engines are generally older designs and most being developed over 30 years ago when variable cam timing and valve lift was not commercially available.

The knowledge gap that will be filled is to design a continuously variable valve lift system to be designed for use on a BMC A-series engine. This is to be designed to use as many of the standard engine parts as possible to minimise modification and to be just as reliable in terms of maintenance as the original. The aim would be to fit in the existing rocker cover and to not appear different from the outside. This would be based on a variable ratio arm by moving the centre shaft to adjust the pivoting point. A full analysis to ensure a reliable and sturdy design is to be considered.

1.5 Study justification

The aim of this project is increase the potential of the A-series engine that could be used as an aftermarket alternative for enthusiasts and for circuit racing.

The use and implementation of the variable valve lift function within the rocker of a BMC a-series engine to maximise efficiency to give the best possible performance across the entire rev range. This project will focus solely on the mechanical design to full fill the requirements of Mechanical engineering Project courses. The interest in this project is from my own building of a 1976 Leyland mini which has a 1098cc turbocharged engine. I'm looking to increase the drivability and efficiency of the engine for me personally although there could be some aftermarket possibilities with the project.

1.6 Outline of the study

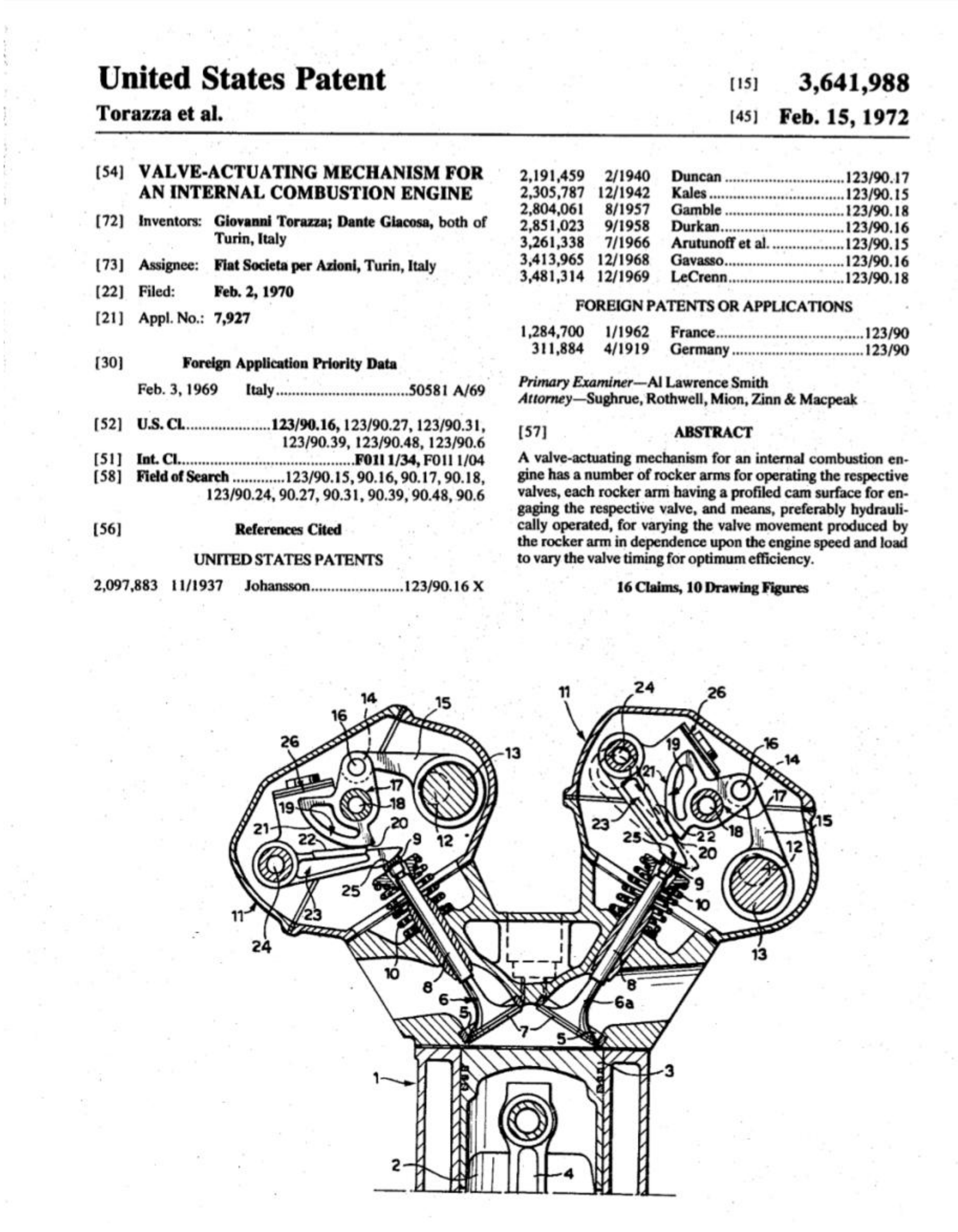
Engine manufactures spend huge amounts of time, research and development to increase the efficiency and performance capabilities of their engines. These innovations used in engines today will have the same gains if utilised in older engine designs. Variable valve lift was first Patented by Giovanni Torazza in 1972(free patent) (shown in figure 1) although the early systems used a stepped or discrete operation. This meant that they had a low and high setting that could be switched at certain rpm. Variable valve lift is commonly used with variable cam timing to ensure the best possible conditions throughout the rev range.
There are many different methods of changing the valve lift as it depends on how the valves are operated. Most common methods are by utilising different cam lobes with differing lift amounts. Other methods of variable valve lift use varying pivot points to increase and decrease valve lift. Many manufactures use different designs with different naming but all doing a similar task. Two main types of variable valve lift are;

- Discrete
- Continuous

Discrete variable valve is the most common type of system as fond in the knowledge gap. This means that the valve lift can be switched from high to low at a defined point in the rev range. This is done by Honda, Audi, Mercedes and Chevrolet (National academic press 2015) all using a similar technique of switching between two different camshaft profiles. Two cam profiles can be switched between, one being an economical design focused on the lower rev range and the latter on peak power. These systems increase valve lift and cam duration at a predefined point. This switching method is usually done by mechanical means controlled by an engine control unit (ECU).

The other method is a continuously variable valve lift which can be changed linearly throughout the entire RPM range for the engine. Continuously variable valve lift has only been used in engines since 2001 and was first used by BMW (National academic press 2015). The system was called Valvetronic and was used in conjunction with variable intake length and variable cam timing on both intake and exhaust camshafts. This was focused primarily on fuel consumption with the complexity of the system adding some valve train losses. Toyota and fiat use a similar system although fiat uses a hydraulic solenoid valve opposed to Toyota and BMW using an electric motor. All systems use an intermediate cam shaft to change pivoting angles thus increasing valve lift.

These two systems are common in modern engines but there are no currently available systems for non-overhead cam engines. This research proposal will plan a methodology to utilise this technology in older pushrod engines.



United States Patent [15] **3,641,988**

Torazza et al. [45] **Feb. 15, 1972**

[54] **VALVE-ACTUATING MECHANISM FOR AN INTERNAL COMBUSTION ENGINE**

[72] Inventors: **Giovanni Torazza; Dante Giacosa,** both of Turin, Italy

[73] Assignee: **Fiat Societa per Azioni, Turin, Italy**

[22] Filed: **Feb. 2, 1970**

[21] Appl. No.: **7,927**

[30] **Foreign Application Priority Data**

Feb. 3, 1969 Italy 50581 A/69

[52] **U.S. Cl.**.....................**123/90.16,** 123/90.27, 123/90.31, 123/90.39, 123/90.48, 123/90.6

[51] **Int. Cl.**...**F01l 1/34,** F01l 1/04

[58] **Field of Search** 123/90.15, 90.16, 90.17, 90.18, 123/90.24, 90.27, 90.31, 90.39, 90.48, 90.6

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Primary Examiner—Al Lawrence Smith
Attorney—Sughrue, Rothwell, Mion, Zinn & Macpeak

[57] **ABSTRACT**

A valve-actuating mechanism for an internal combustion engine has a number of rocker arms for operating the respective valves, each rocker arm having a profiled cam surface for engaging the respective valve, and means, preferably hydraulically operated, for varying the valve movement produced by the rocker arm in dependence upon the engine speed and load to vary the valve timing for optimum efficiency.

16 Claims, 10 Drawing Figures

Figure 1: Giovanni Torazza variable valve (patent 1972)

Chapter 2 – Literature Review

2.1 Introduction

There are many different types of A-series rockers. These include different versions of the manufactures and aftermarket replacement rockers. These rockers can be steel, cast or sintered steel for factory manufactured or Aluminium roller tipped and Aluminium full roller rockers for aftermarket. All types can be purchased in a number of fixed ratios but none have discrete or variable ratio available. This means that in order to find the best performing ratio dynamometer tuning would need to be performed using different ratios. This would be time consuming due to A-series engines rocker design. Removal requires taking the whole assembly off which uses the rear head studs thus needing a head gasket change. The new rockers would then require valve lash adjustment and the dyno tuning to be done again.

A book by David Vizard tuning the a-series engine does help with rocker ratio selection in chapter 11 part 3 “high lift rockers”. Again this favours the peak performance aspect and not the overall performance of the engine. David explains that high lift and high ratio do increase the engines ability to breath although can be detrimental to low speed output (David Vizard 1999, pg326). Once the engine speed has climbed the gain from the extra breathing will pay off this can be seen in figure 2.

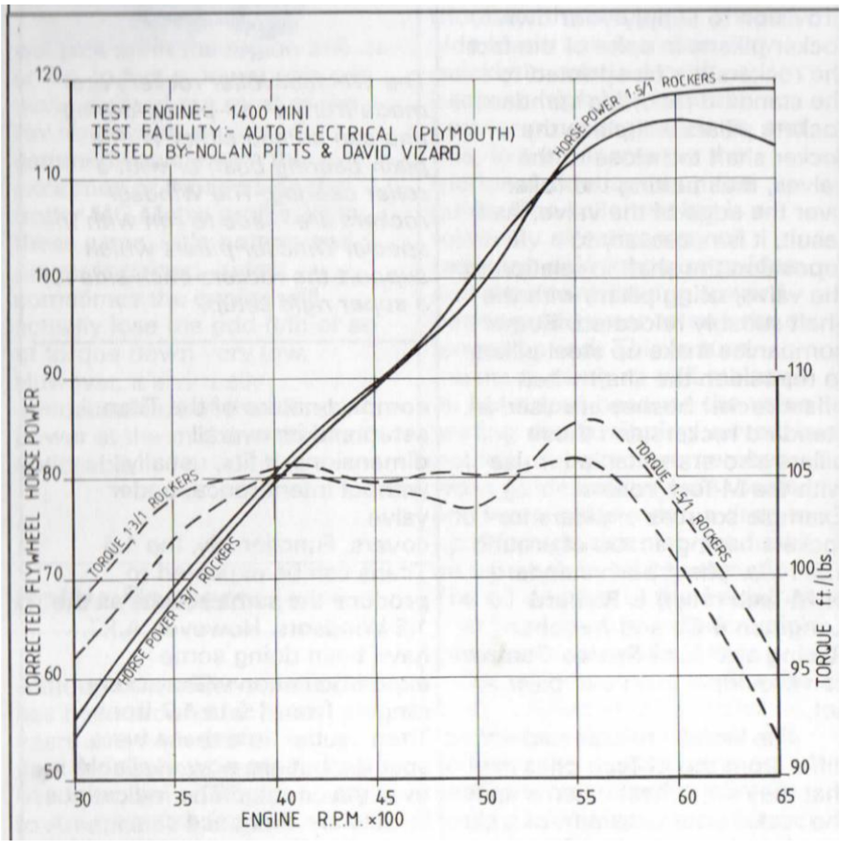


Figure 2: Difference of power and torque with rocker ratios 1.3 and 1.5, showing loss of low end torque with 1.5 ratio rockers over the smaller 1.3. (david Vizard)

Figure two compares two rocker ratios 1.3 and 1.5 ratios. This test is done with a 1400cc A-series engine with the only variable being the ratio of the rockers.

The two rocker ratios 1.3 and 1.5 with a 0.290” lift cam would give 0.377” and 0.435” lifts at the valve respectively. This equates to a difference of 0.058 or 1.47mm. A can be seen in figure two the 1.3 ratio has more torque and Hp to roughly 5250 RPM where the 1.5 ratio then out performs the 1.3. This graph positively shows that having a variable ratio would increase the area under the curve giving greater overall performance.

2.2 Understanding of volumetric efficiency

Volumetric efficiency is an important engine parameter and is defined as the ratio of the air going into the cylinder verses the actual cylinder capacity. This parameter is affected by valve timing, valve lift, intake and exhaust runner length, and intake and exhaust pressure (Shumei Yin 2017). Shumei thesis shows modelling using GT power software and Matlab modelling for different valve lifts. His finding show that increasing the valve lift shifts the volumetric efficiency curve higher in the rev range showing that increasing the valve lift as the engine speed increases can increase the engines volumetric efficiency. By having an increased efficiency meaning increased mass flow will allow for more performance and better economy.

2.3 Variable valve lift

Many car manufacturers produce engines with their own design of variable lift. Most common being the discrete method as mentioned in chapter 1.

Nissan Neo VVL utilises an extra cam lobe with higher lift in between two lower lift lobes. There are three rocker arms running on the lobes with the outside ones pushing the valves. At a certain RPM a solenoid is activated allowing oil pressure to push a small piston to lock all three rockers together; this in turn makes the valves open with the higher centre cam lobe (5523 motorsports).

General Motors (GM) have a similar method like Nissan which uses oil pressure on a piston to lock the rocker arm to utilise the third cam lobe. GM only uses this on the intake valve and is designed to increase fuel economy.

Most discrete methods of lift control use this design of an extra lobe and some sort of rocker switching. This works well for overhead cam applications but doesn’t have a continuous of infinite lift change. All parts are purely mechanical and suited to the particular engine which won’t allow tuning for a specific engine. All these designs work effectively for their application they are not suited for a pushrod motor.

Designs for continuously variable valve lift are much different in design and are less common than discrete from my research. Continuously variable valve lift combined with continuously variable cam timing is only been done by a few manufactures. These manufactures are BMW with their Valvetronic system, Hondas advanced VTEC (Paultan 2007), Nissan Variable valve event and lift and Toyota Valvematic. Honda uses a drum around the camshaft lobes with a small rocker built into it. By turning the drum this changes the point at which the rocker is pushed adjusting the ratio to change the valve lift. The two figures below taken from US patents for Honda’s Advance VTEC explain the operation.

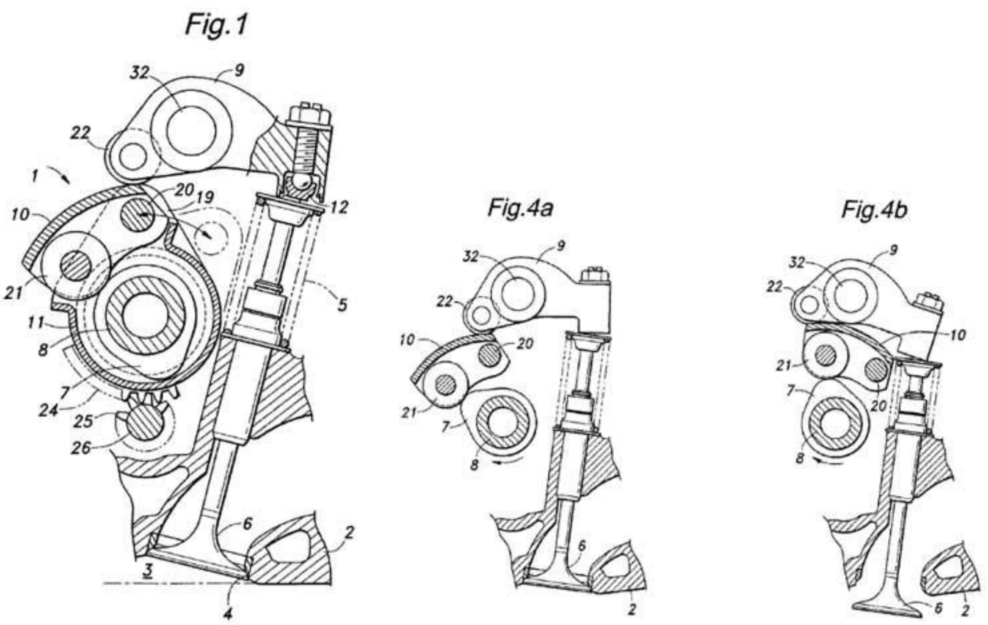


Figure 2.1: Cross sectional view of Honda's AVTEC continuously variable valve lift

BMW's Valvetronic system was developed before Honda's AVTEC and has been used in production since 2001 (BMW Blog 2016). BMW variable valve lift has complete range from no lift to full lift allowing the engine to be run with wide open throttle and control the engine just with valve lift. This uses a second camshaft with a set of intermediate rocker arms and is only fitted to the inlet valves.

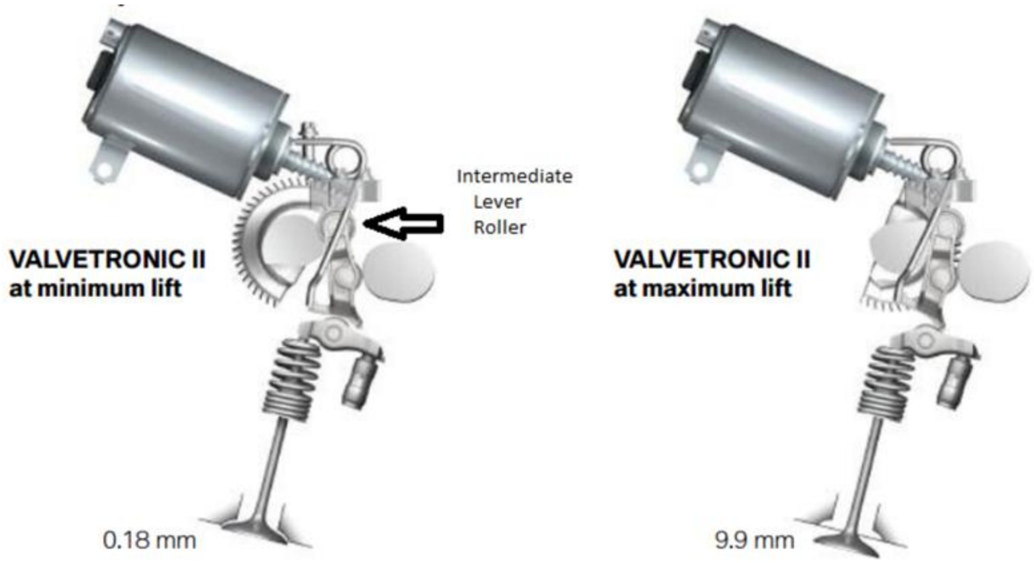


Figure 2.2: BMW Valvetronic valve lift (search auto parts 2017, how to repair BMW Valvetronic)

Both these systems are again system designed specifically for dual overhead cam engines making adaption to a pushrod engine relatively complicated.

2.4 Variable valve lift for pushrod engines

Research into variable lift rockers found multiple patents which are explained below and also found was some information on a product that was proved to be unreliable. No production or current variable ratio rockers could be found and particularly none for the BMC A-series engine.

David N. Vaseleniuch has a patent for a variable ratio rocker using a radiuses follower in the rocker. This allows continuous variable ratio from 1.6-2.11. The shows a good design for variable ratio although no methods of how the follower is moved and also how the valve clearance is accounted for. This method would make actuation complicated as the pivot to change the ratio is moving with the rocker. Details and diagrams are shown in the appendix in figure A1.

Another patent found was by William A Pohle which shows a variable lift rocker using a simple method of an eccentric screw for the pushrod seat. This solves David N. Vaseleniuch's issue of valve and rocker clearance although shows no method of actuation. From the look of the patent it is more so designed for fast adjustment of rocker ratios rather than variable ratio while the engine is operating. Details are shown in the appendix figure A2.

A company called Eaton manufactures a variety of finger followers for use with variable valve lift and or cylinder deactivation for cruising efficiency. These seem to be of similar design to that of the Honda VTEC and BMW valvetronic variable valve lift although only being of discrete adjustment. Eaton Claims improved fuel economy, performance and drivability with two modes of operation (Eaton 2019). Eaten explain that the operation is achieved by a dual lift rocker arm actuated by two cam profiles. More information for this twostep rocker design was found in a SAE technical paper which evaluates the design and development of a twostep rocker for an overhead cam engine (N. Hendriksma, T. Kunz and C. Greene., "Design and Development of a 2-Step Rocker Arm," SAE Technical paper 2007-01-1285, 2007). This design is extremely similar to the Eaton Finger follower although the SAE paper follows a more in depth assessment of the design and stress on the components.

Further research showed some interesting information on a Mopar forum for a company called Hot Rockers for which the only information I could source was that of the forum. The company seems to have disappeared although the product was of a continuously variable rocker arm. The post states the unreliability and high cost of the product.

Alternatives to variable rocker ratios from this research would be that of a variable lifter. Murl L Burton in 1996 Patent a variable duration hydraulic lifter. This design is based on a damper within the lifter. A damper responds to acceleration, thus higher acceleration i.e. faster engine speed result in more damping. Murl's patent explains that the "leak" varies the lift over engine speed. The downside to this is that it is of a fixed damping and thus needs to be removed to be changed. The damping coefficient would be linear with engine acceleration so not be "tuneable" for different speeds.

William W. Entzminger produced a design that was patented in 1989 which shows a possible design for a variable ratio lever arm (Appendix, figure A3). This design gives a continuously variable ratio via a toothed shaft inside a slot with a rack. This seems the most promising design although no prototype or working models can be sourced.

The literature review conducted shows allot of differing designs and research into the stress and analysis of the fulcrum point discrete design. No detailed analysis of the variable ratio arm could be sourced although a number of designs were found through past patentees.

The design and analysis of rocker arms that are fixed ratio is well documented. A paper by Dr. Goteti Chaitanya shows the fatigue of different materials for different ratios. He also explains the possibilities of lightweight composite material rocker arms over the conventional forged steel and aluminium alloys. Rocker arms are under large fluctuating loads and need to be strong enough to withstand a large fatigue life. A model is produced and finite element analysis preformed on three different materials including aluminium composite.

2.5 Understanding of valve lift

David Vizard explains that in a two valve per cylinder motor valve lift is of high importance. This is due to the valve size being limited by the bore size. In the smaller capacity A-series engines this is also an issue as they are of “under square design” meaning the bore size is smaller than the stroke. This is particularly bad in the original 1952 803cc which comprises of a 3 inch stroke but a small 2.28” bore meaning valve size is very limited by the bore. David Vizard continues to explain that flow will peak at valve lifts of around 35-45% of the valve diameter.

A typical 1098cc A-series engine would be equipped with 1.156” intake valves giving an approximate maximum flow of:

$$0.35 * 1.156 = 0.4046" \; to \; 0.45 * 1.156 = 0.5202"$$

This is of course looking at peak flow which is only a perfect value for a certain engine speed range and won’t be optimised throughout the entire engine operation speed. This also shows that at the standard ratio of 1.2 with a 0.290” lift cam only 0.348” lift would be achieved, not given maximum flow.

2.6 A-series Engine Builder

Graham Russell Engineering PTY, LTD. NORTH ROCKS, 2151 NSW, Australia

Further research was done by speaking with an A-series engine builder. Graham Russell is well known in the mini scene for his high performance engines and his personal cam designs. Graham races a mini in the historic class and has a great reputation for high quality parts.

Discussion with him about the project was made and he stated that all engines perform slightly different although generally don’t see a gain with higher lift until over 5500rpm. He explained that there can be a large gain in low engine speed torque using smaller valve lift although generally at the expense of high engine speed power. Cam profile parameters like lobe lift, duration and lobe separation all play a part in what actual lift works meaning that a dedicated discrete method would only for a specific engine combination and thus continuously variable system that could be tuned specifically would work best.

2.7 Summary

Research into the variable valve lift found a number of different possibilities for a design to suit the a-series engine, although to design this variable valve lift system rocker mechanics and stress need to be considered.

A list of other information that was used for the design is below.

- Rocker geometry- Rocker geometry by Jim Miller 2010

- Rocker forces- International journal of engineering sciences and research technology. Design and static structural analysis of a rocker am in an internal combustion engine. By Sachin Bacha , P. Swaminadhan and D. Deshpande 2018.

Going through the literature review didn't a exact solution for the design although shows potential methods. William W. Entzminger patented design is the mechanical principle that will be followed here. William did this by having the rocker position over the valve fixed and moving the shaft position relative the rockers centreline to change the ratio (William A. Pohle 1980). He used a rack and pinion design to hold the rocker in position while moving the shaft. Although usable and a more solid method will be used.

Chapter 3 – Research Design and Methodology

This chapter will cover the design considerations, procedures, tools & resources to design a set of continuously variable rocker arm for an A-series engine. The final outcome and results will be covered in chapter 5 with any design revisions.
The design that will be utilised for this continuously variable rocker will be of a sliding pivot point. This allows the ratio between the pushrod and valve to be changed and thus changing the valve lift. An overview of the design is shown below.

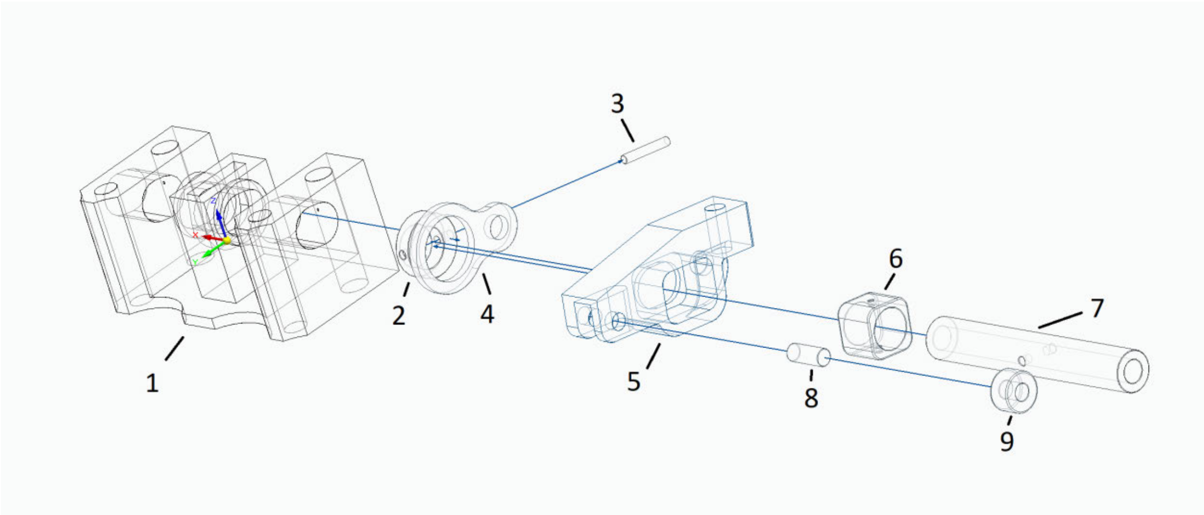


Figure 3: exploded diagram of components for a single rocker

The main rocker components are comprised of the following parts:

1. Rocker posts and actuating cam follower
2. Actuating cam
3. Actuating cam locking pin
4. Rocker linkage arm
5. Rocker body
6. Sliding bush
7. Main shaft
8. Roller tip pin
9. Roller tip

Each component will be individually assessed and made to fit the original BMC A-series cylinder head. Design has been made to be as simple as possible to make manufacturing and repair a simple task.

3.1 Methodology

The methodology that is going to be used to design the variable ratio rockers will be as follows.

- Using the research and simulation to find an optimal rocker ratio range, this will be compared with fundamental equations for maximum lifts in internal combustion engines.
- Design a variable rocker that can be used for the A-series engine within the ratios found.

- Perform Finite element analysis and hand calculations for stress analysis to find material specifications needed.
- Produce a prototype that can be used.
- Results and conclusions defined.

Simulation would be performed using an Engine simulation software package; the software used is Performance Trends engine analyzer pro V3.9. The software can calculate engine performance in power and torque with being able to change cam specification such as valve lift and valve rocker ratio. Kin yip chan explains the difference in engine simulations software stating that the four main commercial packages used are;

- Ricardo wave
- Lotus engine simulations
- AVL fire
- GT Power

Ricardo wave engine simulation package solves using the one dimensional form of the Navier-Stokes equations. This software can analyze the mass flow, pressure and energy within the engines manifold and cylinder head components.

Lotus engine simulation software is code developed by Lotus using combustion and heat transfer zero dimensional equations and fuel composition solver based on the engine input data. The software can predict gas flow, combustion and performance of internal combustion engines (Kin Yip Chan 2013). The input data is quite extensive and requires a lot of input to ensure minimal output error. I did manage to get a copy of Lotus engine simulations software although had uses limitations of single cylinder evaluation.

Most engine simulation software is industry specific making it hard to be able to use for individual purpose (Kip Yip Chan 2013). For this project Engine Analyzer Pro software will be used and can simulate different changes and output a range of data and is available for personal use. The free version was used in this instance. Output data includes:

- Cycle data
- RPM data
- Can simulate differing valve lift

For Valve lift calculations, equations from the internal combustion engine theory and practice by Taylor are used. To determine the maximum valve lift the inlet Mach index number is used. This is the ratio of the typical velocity to the inlet velocity.

Taylor determined that when comparing Mach index number with volumetric efficiency over 0.6 efficiency falls rapidly with increasing engine speed. So when determining valve lift for maximum engine efficiency inlet velocity should not exceed 3/5th the speed of sound (Mach 0.6). This will be solved for the entire engine speed range using MATLAB.

Designing of the actual rocker will be constrained to fit inside the standard cylinder head and rocker cover. This will be designed to use as many factory parts as possible and designed as simple as possible. Design will be done with hand sketches, then drawn in Solidedge ST10 and movement can be simulated to confirm design. This will be done in Solidedge ERA (Explode, render and animate) environment.

Finite element analysis will be performed as well as hand calculations on the stress of the components to ensure an accepted working life is met. FEA will be performed using Creo Parametric 4.0 using the solid models produced in Solidedge. These models will be saved as STP and IGES files to covert between the programs.

Material Selection process will be done after the hand calculations and FEA is preformed. Material selection will be done using easy to source materials and then they will be assessed for strength price and availability. Fatigue life will be a factor due to the fluctuating load that the rockers will be running with. Heat cycle could be a factor with composite material and will also be assessed. These factors will be done with a weight performance requirement system.

Machining process ability is also a performance requirement of the material; this can add or subtract costs during the manufacture and not just at the initial investment. These processing operations can have a major influence on the material selection process and need to be accounted for.

The components for the rocker arm will be broken down into the sections below.

- Rocker arm geometry and ratio
- Rocker arm design
- Rocker post design
- Variable actuation
- Material specification

Rocker arm geometry and ratio is determined by using the simulations along with the fundamental equations. This is compared with the knowledge from Graham Russell to establish the ratios that are used. This will be based on the 998-1275cc A-series engines and valve lifts to suit.

Rocker arm design is governed by the rocker ratio and the position of the valve and pushrod. These are fixed although heights can be adjusted to raise the rocker position. The rocker posts need to hold the rockers over the valve and also support the rocker shaft. This is fixed by the cylinder head casting bolts (detail drawing can be found in the appendix). Variable actuation can be performed in a number of ways as seen in the literature review. The only constraint here is overall physical size. This is determined by the rocker cover which dimensions also can be found in the appendices.

The design for these rocker arms will be follow some design considerations. The rockers will be designed to use as many of the existing components as possible without hindering or overcomplicating the system. The system will be designed at half lift for the maximum ratio to optimise the rocker and valve position.

Material specifications will be determined from the geometry and the forces one the components. Cost strength summery will conclude the outcome.

3.2 Forces

The whole rocker assembly is subject to alternating forces that come from rotating parts, cylinder pressure and valve spring pressure. All calculations are based from the forces and thus need to be calculated first. Below is a diagram of the rocker arm showing the forces acting on the whole system.

Forces acting on the rocker can be calculated by the following sets of equations by D. Raja Kullayappa from the research article Analysis and Optimization of Rocker arm (2017)

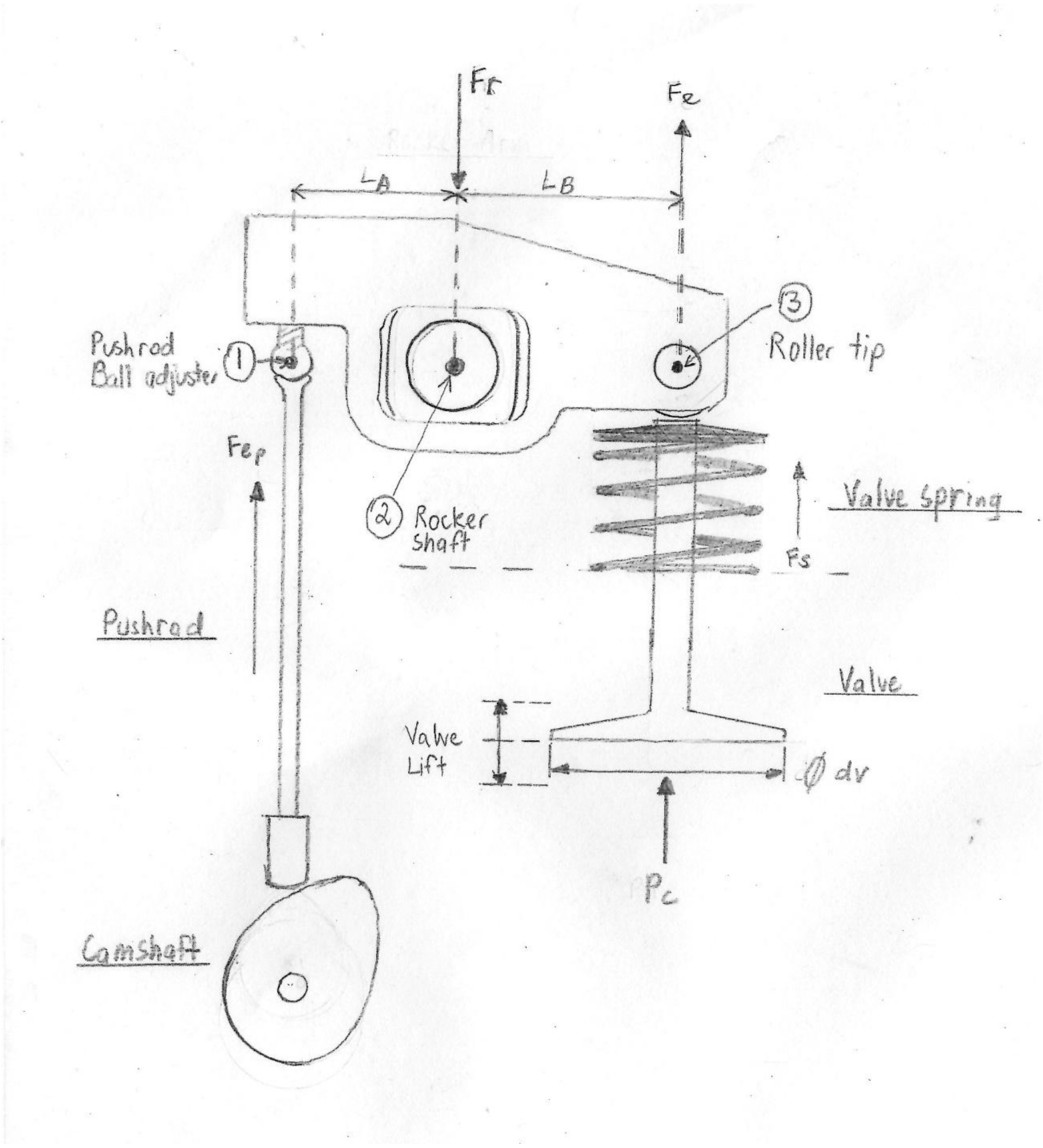


Figure 3.1: Valve train forces

Fe = Total load on rocker arm

P = total load on the valve

P1 = Gas load from cylinder

Fs = spring force

Fa = Acceleration force

W = Weight of valve

Pc = cylinder pressure

dv = valve head diameter

Ps = maximum suction pressure

t = time (s)

r = Half lift

h = full lift

K = spring rate

P_1 = spring preload

a = valve acceleration

$$\text{Forces on the rocker can be calculated using ; } Fe = P + Fs + Fa$$

$$(Fe = the\ roller\ tip, point\ 1\ on\ figure\ 3.1)$$

Where;

$$P = P1 + w$$

$$P1 = \frac{\pi}{4} \times dv^2 \times Pc$$

$$w = mass \times gavity$$

$$Fs = P1 = \frac{\pi}{4} \times dv^2 \times Ps - w$$

$$Fa\ = Engine \frac{RPM}{2} (camshaft\ rotates\ half\ speed)$$

$$t = \left(\frac{RPM}{2} / 60\right) \times 360$$

$$a = \omega^2 \times r = \left(\frac{2\pi}{t}\right)^2 \times r$$

$$Fa = m \times a + w + (P_1 + K \times h)$$

These equations will find the maximum forces on the valve tip with the pushrod side of the rocker having the force multiplied by the rocker ratio. This force will be a maximum at the highest ratio and on the inlet valve due to the greater diameter.

Push rod rocker force can be calculated by;

$$Fe_p = Fe \times Rocker\ ratio$$

3.3 Rocker posts

Rocker post design will be based on the original mounting within the cylinder head. This will utilise the cylinder head bolts and the rocker post studs. The post will also house the rocker shaft and will be used for the variable actuation. The rocker posts are subjected to two main forces; these are the extra tension on the rear bolts from cylinder pressure and upward force from the valve force. This force is transmitted through the centre from the rocker shaft into the sliding bearing. The bolts and studs need to be torque to specification this is supplied with the bolt manufactures or by using the workshop manual. Calculation needs to be done for bolt strength and bearing deformation on the post itself. The rocker posts need to hold to main shaft and also provide shaft movement to adjust the pivot point. Each post holds two rockers although only one rocker is activated at a time.

3.4 Bolts and studs

Lifting force Fr on the rocker posts is caused by the moment created by force Fc. This can be calculated by taking moments around the pushrod ball adjuster at point 1.

$$\sum M_1 = 0 = Fe \times (L_A + L_B) - Fr \times L_A$$

$$Fr = \frac{Fe \times (L_A + L_B)}{L_A}$$

Since distances L_A & L_B are variable distances calculation needs to be done at maximum and minimum ratio. The rocker posts consist of a stud and bolt and the force Fr is shared between the two.

The diagram below shows where the forces are present:

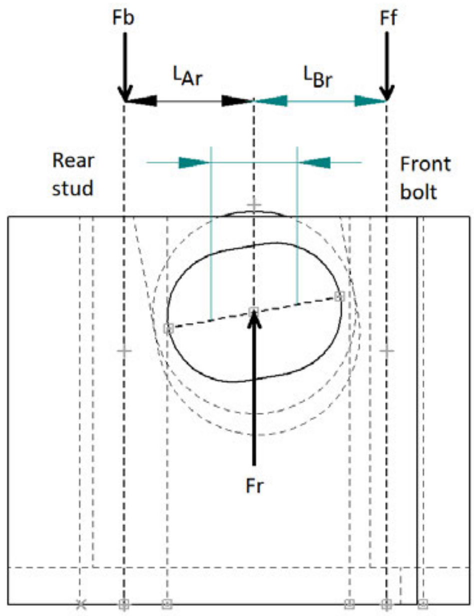


Figure 3.2: Rocker post forces

Lifting force Fr needs to be shared between the two bolts in the rocker post. The two bolts are governed in size. A factor of safety needs to be used to ensure failure won't occur. Using the valve lift for both minimum and maximum lift bolt stress can be calculated. Bolt specification is selected from this. Bolts will be determined for tensile loading only as shear force is assumed to be very small.

Nominal load will be; $Fr \times SF\ (safety\ factor)$. The bolts undergo fatigue loading with fluctuating tension and using the torque specifications the axial force for each bolt can be calculated;

$$T = 0.2F_i d$$

Where:

$$T = torque\ applied$$

$$F_i = Axial\ force$$

$$d = nominal\ major\ thread\ diameter$$

Fluctuating tension force will be from 0-Fr and individual bolt forces can be calculated by taking moments.

$$\sum M_{Fb} = 0 = Fr \times L_{Ar} - Ff \times (L_{Ar} + L_{Br}),$$

$$\sum M_{Ff} = 0 = Fb \times (L_{Ar} + L_{Br}) - Fr \times L_{Br}$$

Bolt initial tension produces root stresses of;

$$\sigma = \frac{F_i}{A_t} K_f$$

Using table the table below (page 452 Rober c. Juvinall) the fatigue stress concentration factor for steel threaded members can be calculated.

Hardness	**SAE Grade (unified Threads)**	**SAE Class (ISO Threads)**	**k_f Rolled Threads**	**k_f Cut Threads**
Below 200Bhn (annealed)	2 and below	5.8 and above	2.2	2.8
Above 200Bhn (hardened)	4 and above	8.8 and above	3.0	3.8

Table 1.0: Fatigue stress concentration factors on threaded members

The rear stud as shown in figure 8 also clamps the cylinder head, so is subjected to extra force than that of the front bolt. The extra cylinder force is calculated by;

$$F_{cylinder} = P \times A$$

Cylinder pressure can be directly taken from the simulation data and cylinder area is:

$$\pi \times cylinder\ radius^2 \times cylinder\ stroke$$

There are in total 9 studs so the force is shared between them equally in the 4 cylinders. Therefore:

$$F_{cylinder} = \frac{4}{9} P \times A$$

Bolt tension for the rear stud needs to be sufficient to ensure it is greater the cylinder force, if $F_{cylinder} = F_i$ Head gasket separation will occur.

3.5 Valve lift equations

With the ratio selected maximum valve lift can be calculated using the rocker movement kinematics. The diagram below shows the simplified moment. The lift from the cam pushing the rocker can be pictured as an offset four bar slider crank (Cairo University Scholars 2015). With the push rod being moved vertically while the rocker is the crank moving with radius r_a .

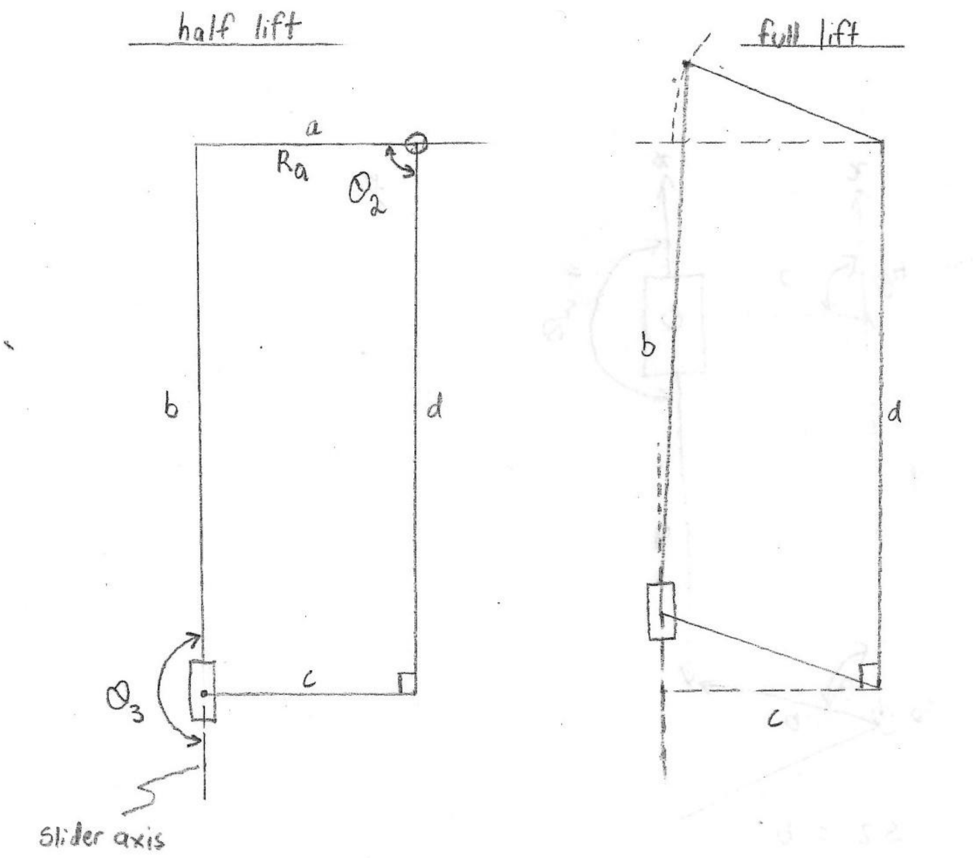


Figure 3.3: Valve lift movement kinematics

This needs to be done to accurately check valve lift as it is not all movment is completely transmitted to the valve due to losses in the movements. Rocker arm geometry is important to ensure that there is no wasted CAM information (Jim Miller 2010). The rocker arm is given linear movement which is converted to rotational movement and back to linear. This set of movements needs to have the correct geometry to supply the valve with the same information that the cam shaft is ground too. For the A-series engine this geometry is relatively simple for a fixed ratio as the valves are parallel to the pushrods although will be of some compromise as the pivot point adjusts. This section will find the geometry that best suits the ratios that are chosen.

The relationship between the two angles can be found using the equation:

$$d = acos\theta_2 - bcos\theta_3$$

Once θ_2 is found the valve lift can be found using:

$$Valve\ lift = \left[2 \times \sin\left(\theta_{2_{Full\ lift}} - 90°\right) \times R_b\right] - Valve\ clearance$$

Where $R_b = rocker\ ratio \times R_a$

Once valve lift is found Taylors mach index number can be used to find a theoretical valve lift vs. engine speed.

$$Using\ the\ Equation;\ \frac{A_p s}{A_i} = mean\ inlet\ velocity$$

Where;

$$A_p = piston\ Area$$

$$s = piston\ velocity$$

$$A_i = area\ of\ the\ inlet\ valve\ opening$$

$$From\ Tyalor\ pg171\ this\ gives;\ \frac{A_p s}{A_i a} = mach\ index\ number$$

3.6 Material specification

Material specifications will be based on the stress analysis and fatigue life due to fluctuating loading on the rocker arm. Minimum weight is of high importance here on the moving components to reduce an extra valve train loses.

Fatigue strength factors for a $10^6\ cycle\ strenght.$

$$S_n = S_n^1 C_L C_G C_S C_T C_R$$

Where; $C_L = load\ factor$

$C_G = Gradient\ factor$

$C_S = Surface\ factor$

$C_T = Temperature\ factor$

$C_R = Reliability\ factor$

Only 10^6 cycle strength will be considered so all components are designed for an infinite life. Formulas used for fatigue are covered in more detail in Machine component design 2016, Robert C.Juvinal, Kurt M. Marshek chapter 8 pages 313-371.

Using the Ashby diagrams with the strength vs. cost and strength vs. density and material candidate can be selected. Figure 8 and 9 show the two Ashby diagrams that will be used.

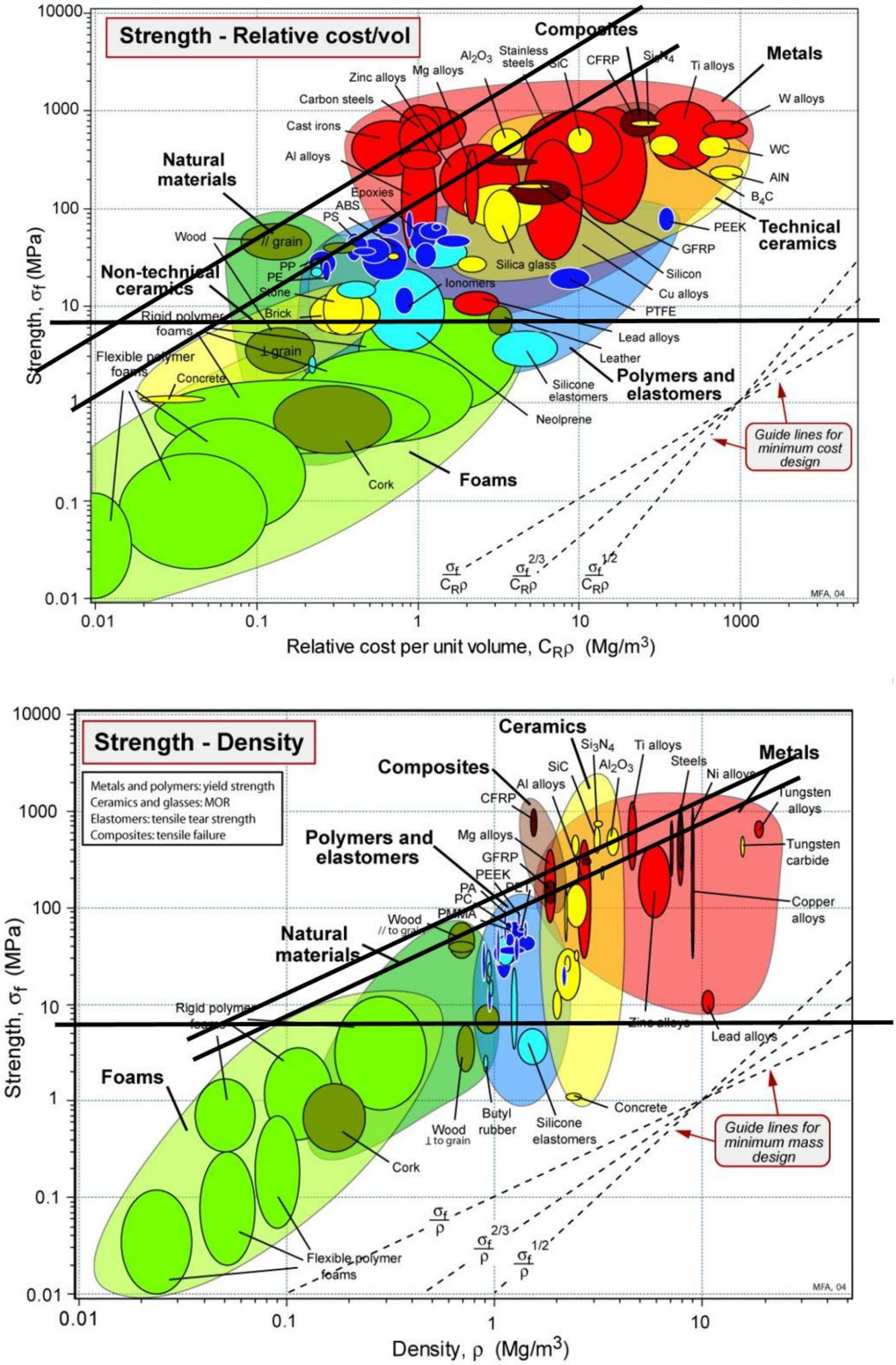


Figure 3.4 & 3.5: Strength vs. Relative cost and strength vs. Density (Canfield joints, material selection

Chapter 4 – Results

To determine the ratio that the rockers will work within was determined by four resources. The four resources that were used to determine this ratio are as followed:

- Simulations
- Calculations
- Literature
- Experts

4.1 Engine simulations

Engine simulations were performed using performance trends software called engine analyser 3.9B. This software was used mainly because of cost but it did have some limitations listed below. This software is focused on V8 engines although does have some base 4 cylinder models that can be used. The cylinder head inputs are specific for individual runners where the A-series engine is Siamese port. Limitations found with engine analysis included:

- No Siamese port option
- Exhaust length and size
- Valve lift has no physical limitation

Initial tests were done with a stock 1098cc engine with specifications for the engine taken from Leyland mini workshop manual and flow and performance specifications from David Vizards tuning the A-series engine. The engine simulations were run with 1.1 – 2.0 rocker ratios, the max hp and torque for each sample rpm was found. Rpm sample were taken every 250 RPM from 1500 to 6250. The average HP and torque for the standard ratio is then compared with a varying ratio. The varying ratio was computed in excel using the maximum value of hp and torque for each RPM sample.

Initial results showed some benefit to differing rocker ratio. If the ratio was increased, peak power could be increased by 6.5% and also allow the engine to run to a higher RPM than previously with the smaller lift. The simulations do prove that if the ratio constantly changed throughout the rev range some improvements could be made overall giving a greater area under the torque and hp curve.

Using average torque and HP a comparison between standard 1.25, 1.7 ratio and variable ratio can be seen.

	Average HP	Average Torque (lb.ft)
1.25 ratio	37.809	53.625
1.7 ratio	39.47575	55.19725
Variable 1.1-2.0	40.021	55.833

Table 1.01: Average power from simulations

This simulation is also comparable to the results from Fig. 11.8 PG326 David Vizard Tuning the A-series engine. This is shown in figure 2 in the literature review. It was noted that the ratio could be increased to 2.2 and would still output higher HP figures. This would be unrealistic for real situations due to spring coil bind and extreme rocker angles. The valve opening area would also exceed the intake area resulting in no further gain in airflow.

Results proved comparable with the workshop manual stating 60lb.ft @2500 rpm and 50hp @5100 rpm. Standard rocker ratio stated is 1.25.

Engine analyser showed 4% lower than the stated power for the standard 1098. Simulations were done for standard (1.25), 1.4, 1.6, 1.7, 1.8 ratios and 2.0, results show in table below up to 1.7. Further ratios were not testing due to physical limitations as of actual space; rocker angles and issues of wear and reliability could play a factor. This was noted above that the software would go to 2.2.

Ratio	Hp (rpm)	Torque (rpm)
1.25	47.97 (5000)	60.2 (2500)
1.4	49.59 (5250)	60.15 (2500)
1.6	50.88 (5500)	59.83 (2750)
1.7	51.425 (5500)	59.77 (3500)

Table 1.02: Initial power figures from simulations

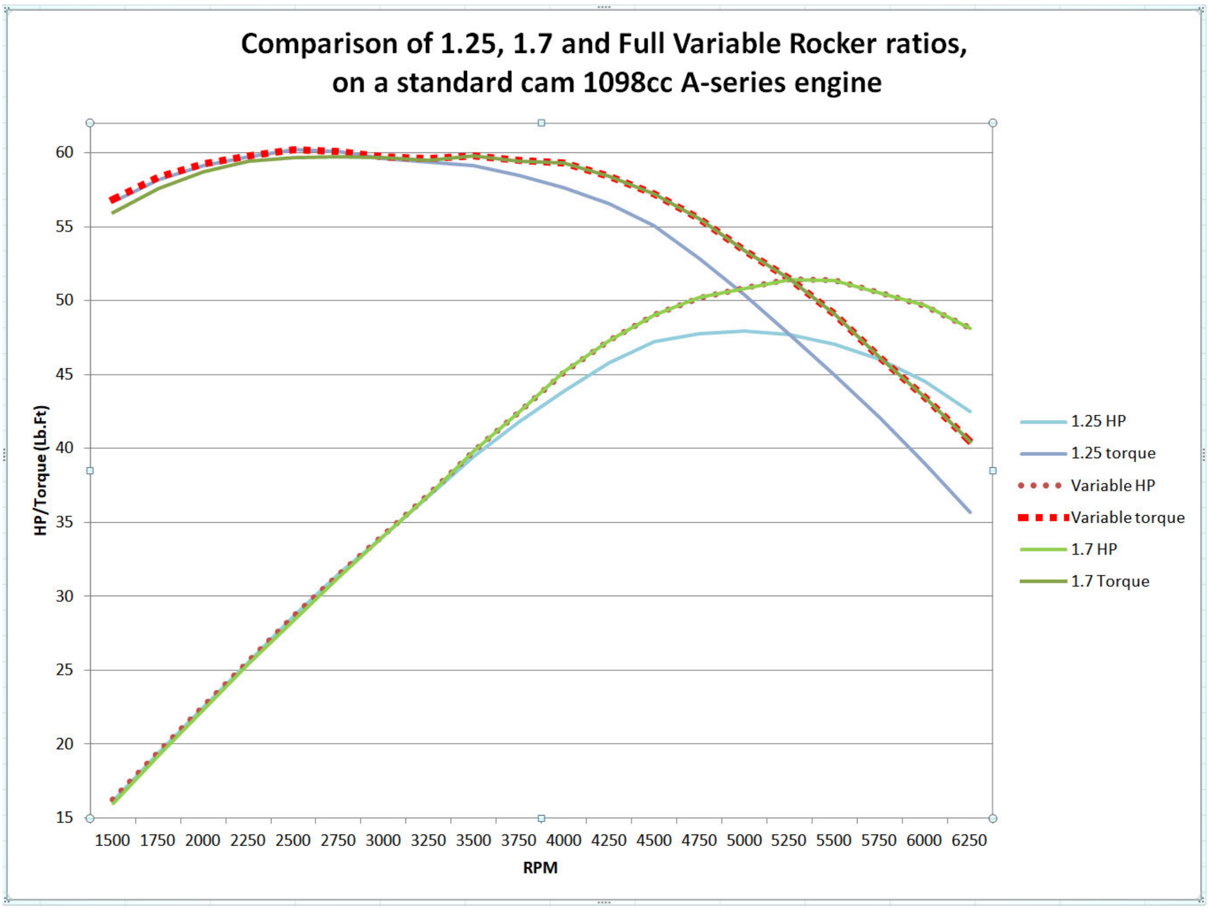


Figure 4.01: 1098cc standard cam engine simulations (engine analyser 3.9B)

The variable ratio shows that maximum engine performance can be utilised over the entire rev range rather than a specific are with a fixed ratio.

As high lift ratio rockers are not something that would be used with a standard engine another simulation was performed using the Graham Russel RE-13 camshaft and a 12g295 cylinder head. These are common modifications and improve the engines performance and airflow capabilities.

The second simulations were done using the same engine capacity but using the re-13 camshaft which has specifications of - $maximum\ lift - 0.290"$ / $7.366mm\ and\ Duration\ 276°$. The compression ratio was also changed to 9.5:1 to better suit the camshaft.

Results were similar to the standard 1098cc engine but the rocker ratio had a much higher effect on the engine torque and hp.

The transition point or crossover point between the ratios also changed meaning that a continuously variable system would give a much smoother transition.

Figure 4.02: 1098cc standard cam engine simulations (engine analyser 3.9B)

From the figure 4.02 it can be seen that the 1.25 rockers give around 4% increase in torque in the lower rpm. Through the transition point the ratio variation actually takes about 1500rpm and this is where continuously variable will out-perform discrete ratio rockers. The 1.25 ratio performs best to 4250rpm where the ratio can slowly change to 1.7 by 5750rpm. Different capacity or differing engine and cam combinations all would have diverse needs. The Variable ratio for this simulation would change the valve lift from 9.2075mm to 12.522mm.

4.2 Mach index ratio calculations

Using Taylor's inlet mach index vs. volumetric efficiency a theoretical valve lift profile can be created.

For a standard 1098cc engine with standard specification, the valve lift was plotted in excel using a mach index maximum of 0.6. This ensured the volumetric efficiency is around 80%.

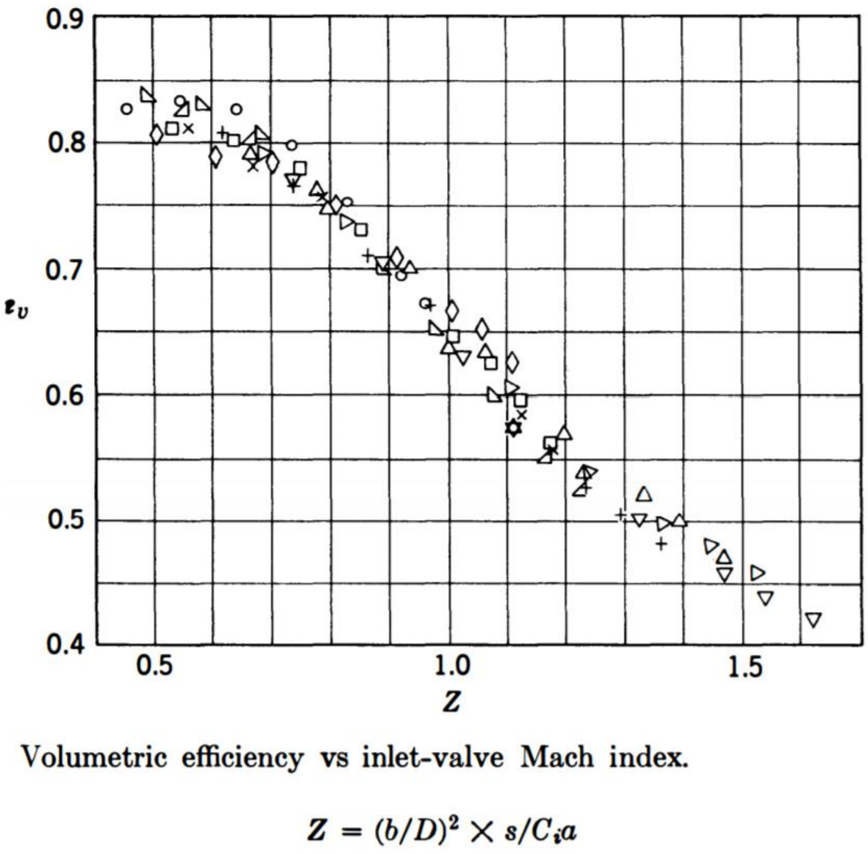


Figure 4.03: Volumetric efficiency vs. inlet-valve Mach index (Taylor pg174).

Using the equation; $Z = \left[\frac{b}{D}\right]^2 \times s/c_i \times a$

Where;

$$b = bore\ diameter$$

$$D = inlet\ valve\ diameter$$

$$s = mean\ piston\ speed$$

$$a = velocity\ of\ sound\ at\ inlet\ temperature$$

$$c_i(\text{inlet} - \text{valve flow coefficient}) = 1.45 \times \frac{lift}{valve\ diameter}$$

Example calculation for 1098cc engine;

Bore (mm)	64.59
Stroke (mm)	83.72
Valve diameter (mm)	29.2862

Maximum Lift (mm)	7.9248
Inlet temperature (°C)	50
Maximum engine speed (rpm)	5100

Table 1.03: 1098cc engine specifications

$$c_i = 1.45 \times \frac{7.9248}{29.2862} = 0.392$$

$$a = 331.4 + (0.6 \times 50°C)$$

$$s = 5100 \times 2 \times 83.72/6000$$

$$Z = \left[\frac{64.59}{29.2862}\right]^2 \times 14.23/0.392 \times 361.4$$

$$Z = 0.488$$

This shows in stock form with the cam having 0.250" lift and 1.25 ratio rockers the mach index number is still below 0.6 and can be improved with increased lift. Using the simulations as a guide the Engine speed was increased to 5500rpm to find the increased theoretical rocker ratio. Without changing the mach index number valve lift can be increased to 8.55mm giving a rocker ratio of 1.35.

Using the mach index ratio showed that only small valve lifts were needed until higher RPM, the actual valve lifts calculated were lower than the simulations. Using the mach index calculation above a valve lift vs engine speed graph was created;

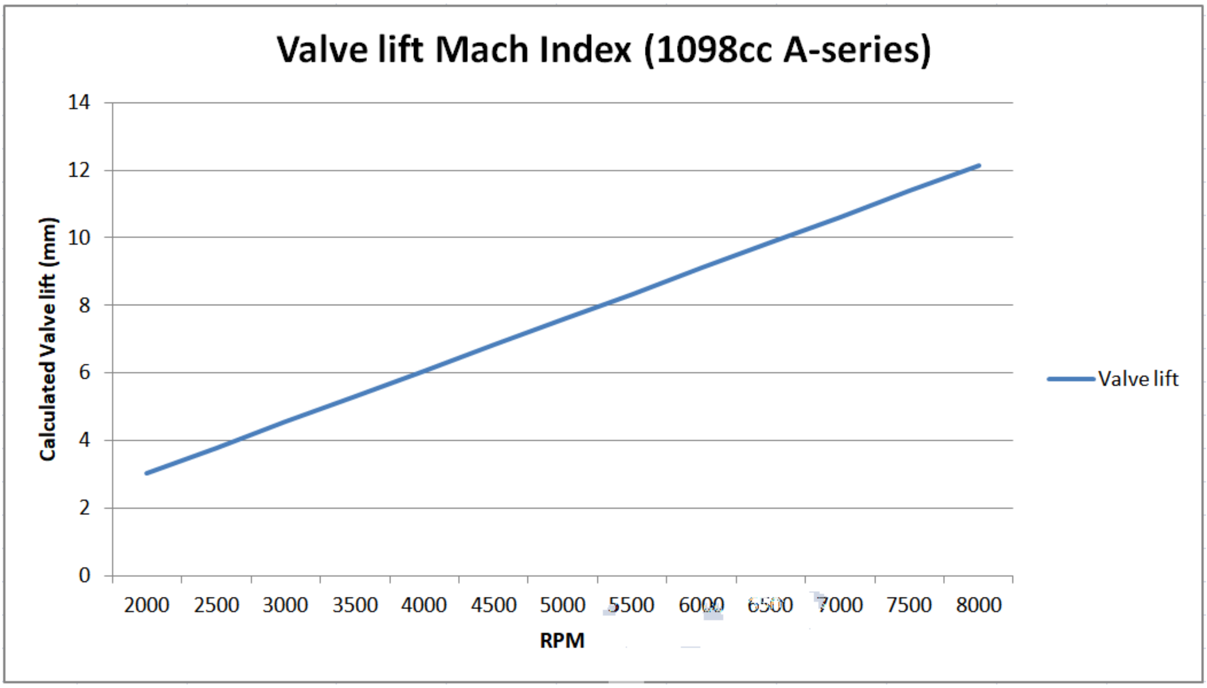


Figure 4.04: Mach index ratio, calculated valve lift for 1098cc A-series

This calculated valve lift showed around 8.7mm lift at 5750rpm which would only be 1.18 ratio as opposed to the simulations showing that lifts of 12mm or more are acceptable by 5750rpm. Taylor also states that valve lifts of over 0.25 valve diameter are rarely used.

4.3 Literature

To find ideal ratios the literature used was David Vizard tuning the A-series engine. David Vizard states that Valve lift is very dependent on engine components (pg 321 tuning the a-series engine). This confirms the findings above from the simulations with engine analyser. Vizard does his comparisons using mainly the larger A-series engines (1275cc and above) and does explain that there can be a large loss with the 1.5 ratio in the smaller engines in the lower RPM. The comparisons are mainly using 1.5 ratio although does explain that the 1.7 ratio was in experimental stages during the time of the publication and has potential for an increase in volumetric efficiency. Vizard also states the same as Taylor with 0.25 x valve diameter = max lift, although he describes that this will only work if there is 100% efficiency. This is not obtainable and that flow capabilities continue to increase to 0.35 x valve diameter and even more.

4.4 Experts

The final decision process for finding the ideal ratio was to speak with Graham Russell from Russell Engineering PTY LTD. Graham has had years of experience in the field of building and re-building standard and performance A-series engines. Graham stated to a good range of variation would be 1.3-1.6 ratio.

4.5 Ratio chosen

From the four resources it was determined that the ratio needed is very engine dependent. To make a design that can be used in wide varieties of capacity and combination it was decided to use a ratio from just lower than standard 1.2 to 1.7. This will cater for large varieties of engines and give a good advantage of low RPM torque and high RPM Horsepower. The simulations, literature and experts all showed similar answers although Taylors Mach index ratio gave anomalous results.

As cam lift will change the rocker geometry the rockers will be designed around a single cam and could be suited to other cams available. The cam the design will be based on is the Russel Engineering RE-13 cam as this is a common upgrade and has a maximum lift similar to other performance camshafts available. This is based on a performance view in mind which would be the main use of the rockers. If tuning for more economy a system similar to BMW valvetronic would be better. The valvetronic system allows throttle control by reducing valve lift to zero although was not the aim for this project.

4.6 Forces

First forces on the rocker are calculated:

$$P = P1 + w$$

$$P1 = \frac{\pi 35^2}{4} \times 1.379\ (IMEP\ 200psi)$$

Using the gross indicated mean effective pressure (IMEP) which is the mean effective in cylinder pressure acting on the piston. From the simulations this value peaked at around 180Psi.

Values can reach up to 200psi depending on engine combinations. Valve head diameter can be as large as 35mm, therefore;

$$P = 1326.753N$$

Typical production inlet valve weight is around 57g with larger aftermarket valves generally weighing less.

$$w = 9.81 \times 0.057$$

$$p = 1326.753 + 0.5592 = 1327.312N$$

Initial spring force;

$$Fs = \frac{\pi}{4} \times 35^2 \times 0.02 = 19.242N$$

Force due to acceleration of the valve;

Assuming maximum 8000rpm, camshaft RPM = $\frac{8000}{2} = 4000$

Angle turned by camshaft = $\frac{4000}{60} \times 360 = 24{,}000\ deg/sec$

Time take for valve to fully open and close; using a typical aftermarket cam of 276° total crank duration.

$$t = \frac{138}{24000} = 0.00575s$$

Using 0.290" lift cam with 1.7 lift ratio;

$$a = \left[\frac{2\pi}{0.00575}\right]^2 \times 0.0062611 = 7476.093m/s^2$$

$$Fa = 0.057 \times 7476.093 + 0.5592 = 426.697N$$

Total force acting; (C-AEA526 8000rpm valve springs spring constant 87.6N/mm, 6.68mm pre-load)

$$Fe = P + Fs + Fa + (P_1 + K \times h) = 1772.692 + (585.168 + 1096.945)$$

$$\boxed{Fe = 3454.805N\ (total\ load)}$$

4.7 Bending stress rocker

Bending stress can be calculated analytically using the force acting on the rocker.

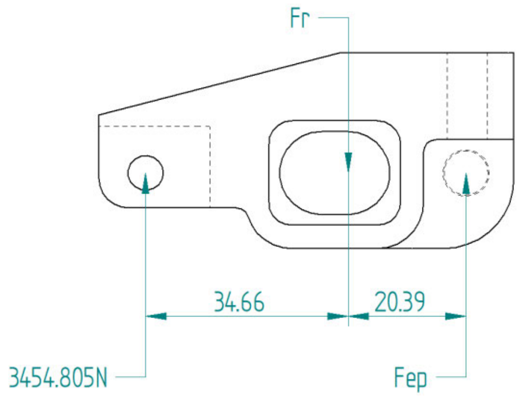


Figure 4.05: Rocker force diagram

$$\sum M\ Fep = 0 = (-3454.805 \times 0.05505) + (Fr \times 0.02039)$$

$$\boxed{Fr = \frac{190.187}{0.02039} = 9327.465N}$$

$$\sum Fy = 0 = 3454.805 - 9327.465 + Fep$$

$$\boxed{Fep = 5872.66N}$$

$$Bending\ stress = \sigma = \frac{My}{I}$$

Taking the moment around Fr;

$$\boxed{M = 3454.805 \times 0.03466 = 119.74Nm}$$

Moment of inertia and rocker shaft;

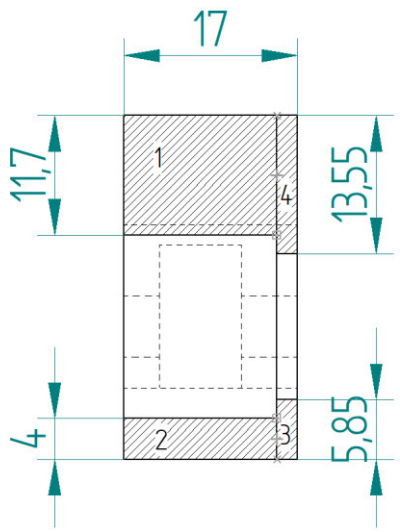


Figure 4.06: Section view of rocker for moment of inertia

$$I_1 = \frac{1}{12}bh^3 = \frac{1}{12}15 \times 11.7^3 = 2002.016$$

$$I_2 = \frac{1}{12}15 \times 4^3 = 80$$

$$I_3 = \frac{1}{12}2 \times 5.85^3 = 33.37$$

$$I_4 = \frac{1}{12}2 \times 13.55^3 = 414.64$$

$$I_{total} = \sum [I_i + A_i {d_i}^2]$$

$$\bar{y} = \frac{\sum Ay}{\sum A_i} = \frac{5691.565}{234.3} = 24.291$$

$$I_{total} = 27429.963mm^4$$

$$\boxed{\sigma = \frac{119743.54 \times 24.291}{27429.963} = 106.04Mpa}$$

This maximum calculated bending stress is located at the base of the rocker arm directly below the rocker shaft when a 1.7 arm ratio is used.

For more detailed equations refer to appendix.

4.8 FEA rocker

Using the loads calculated above the model was tested in Creo Simulate to verify the design. Using the maximum loads and having the rocker arm positioned at maximum lift a number simulations were performed. The rocker laod case was done by constraining pushrod adjusting bolt on the radius and having the calculated loads on the roller tip and sliding bush. These were usied as bearing loads with the roller tip having the 3.45kN load and the sliding bush -9.32kN. The model was meshed using a maximum mesh of 2mm and a 1mm mesh for the surface corners. The mesh is shown below in figure 4.7.

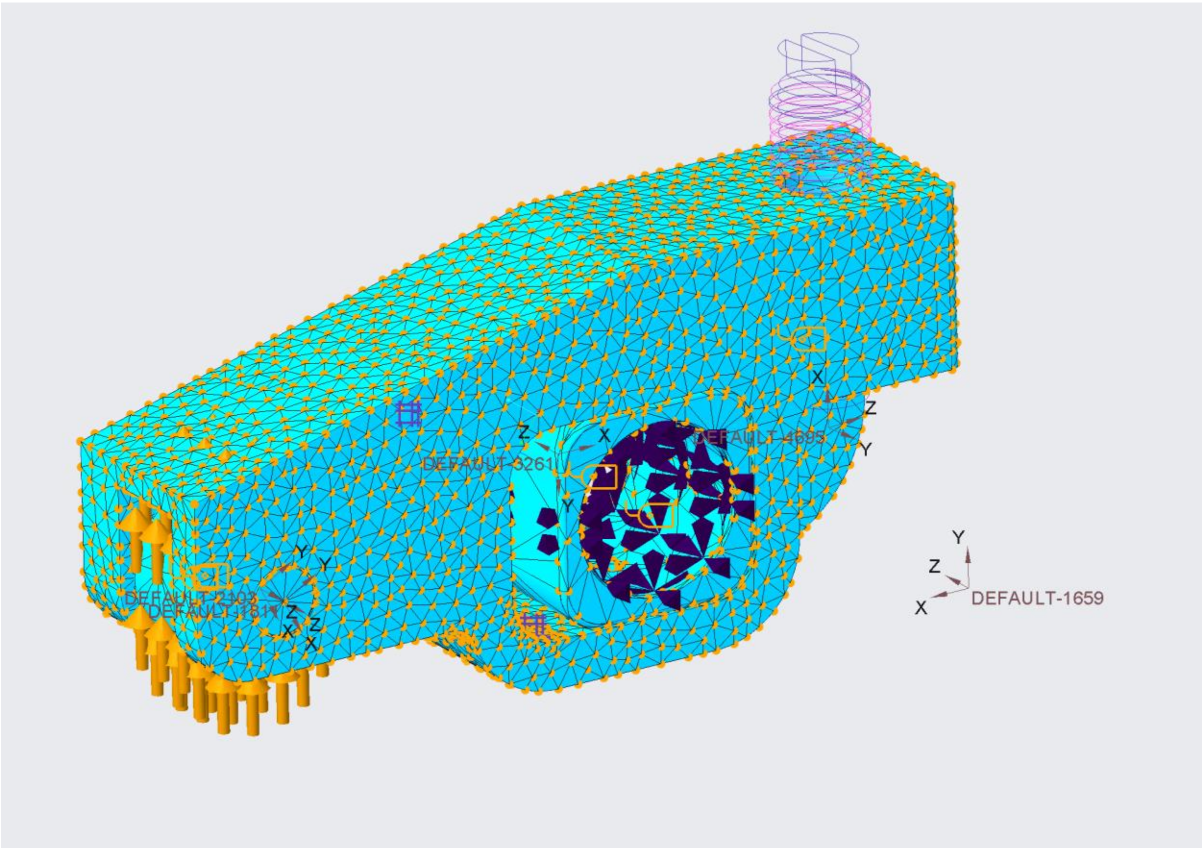


Figure 4.07: load case 1 meshing.

Surface contact treatments were used, and the sliding bush was modelled as bonded for the lower surfaces and free for the remaining. The roller tip was also modelled as bond to the shaft but free for the roller faces. This was done to imitate the working loads as precise as possible.

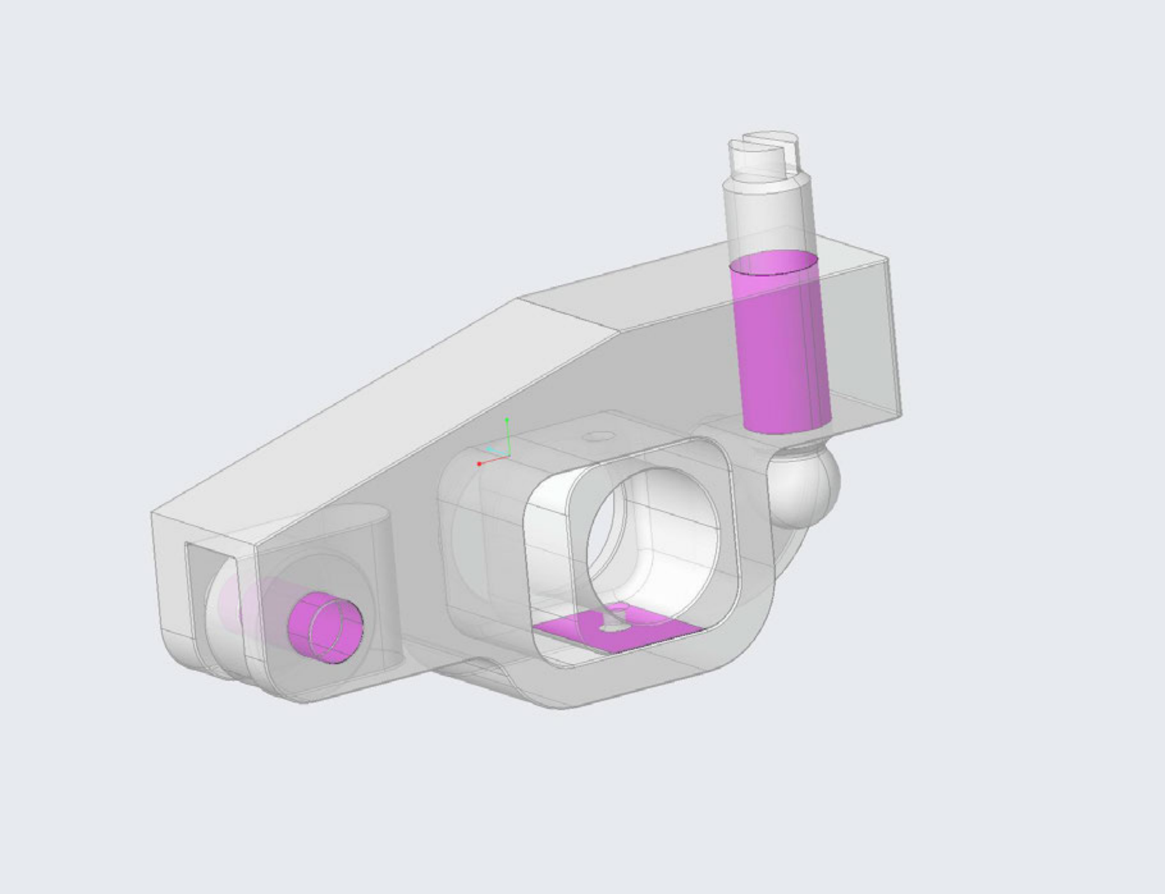


Figure 4.08: Creo Simulate rocker contact surfaces

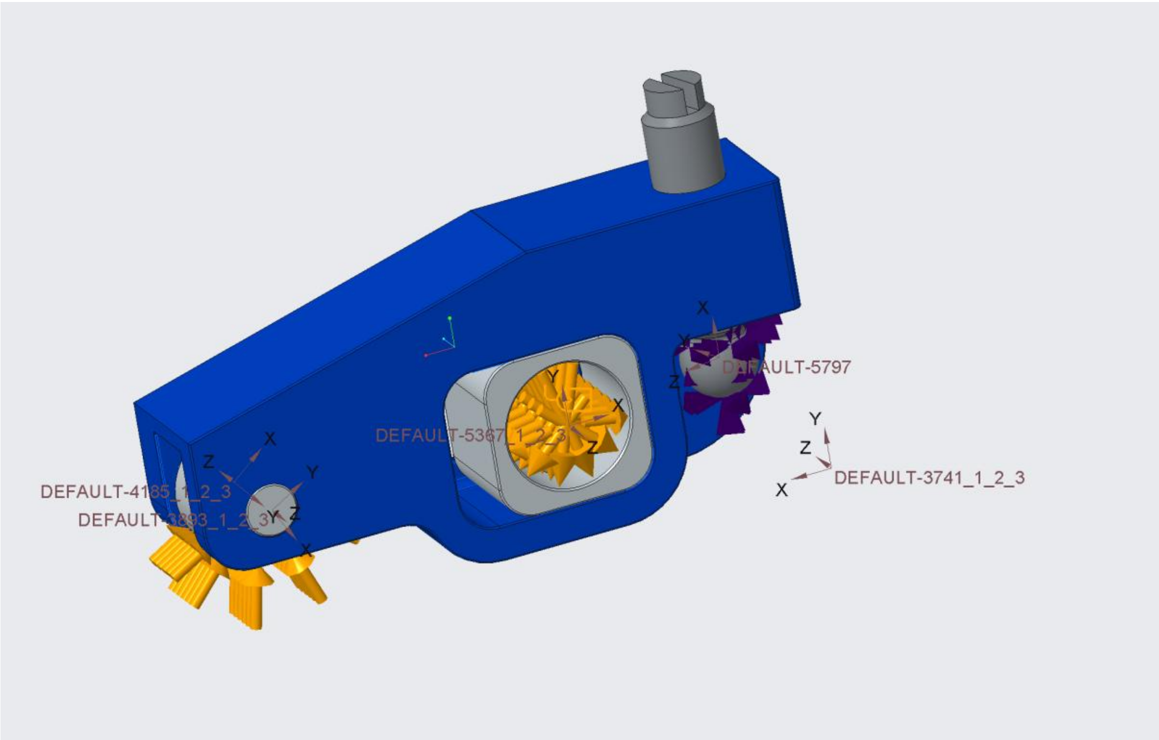


Figure 4.09: Rocker loads and constraints

The analysis was run and checked to see if results were consistent with the calculated bending stress. This was located at 27mm from the pushrod end and 8.5mm in the width. Results were comparable with the calculated bending stress.

Calculated bending stress (σ)	106.04Mpa
FEA Von-Mises stress (σ)	106.966Mpa
FEA High stress area A (σ)	431.7Mpa
FEA High stress area B (σ)	365.57Mpa

Table 1.04: Rocker calculated stresses

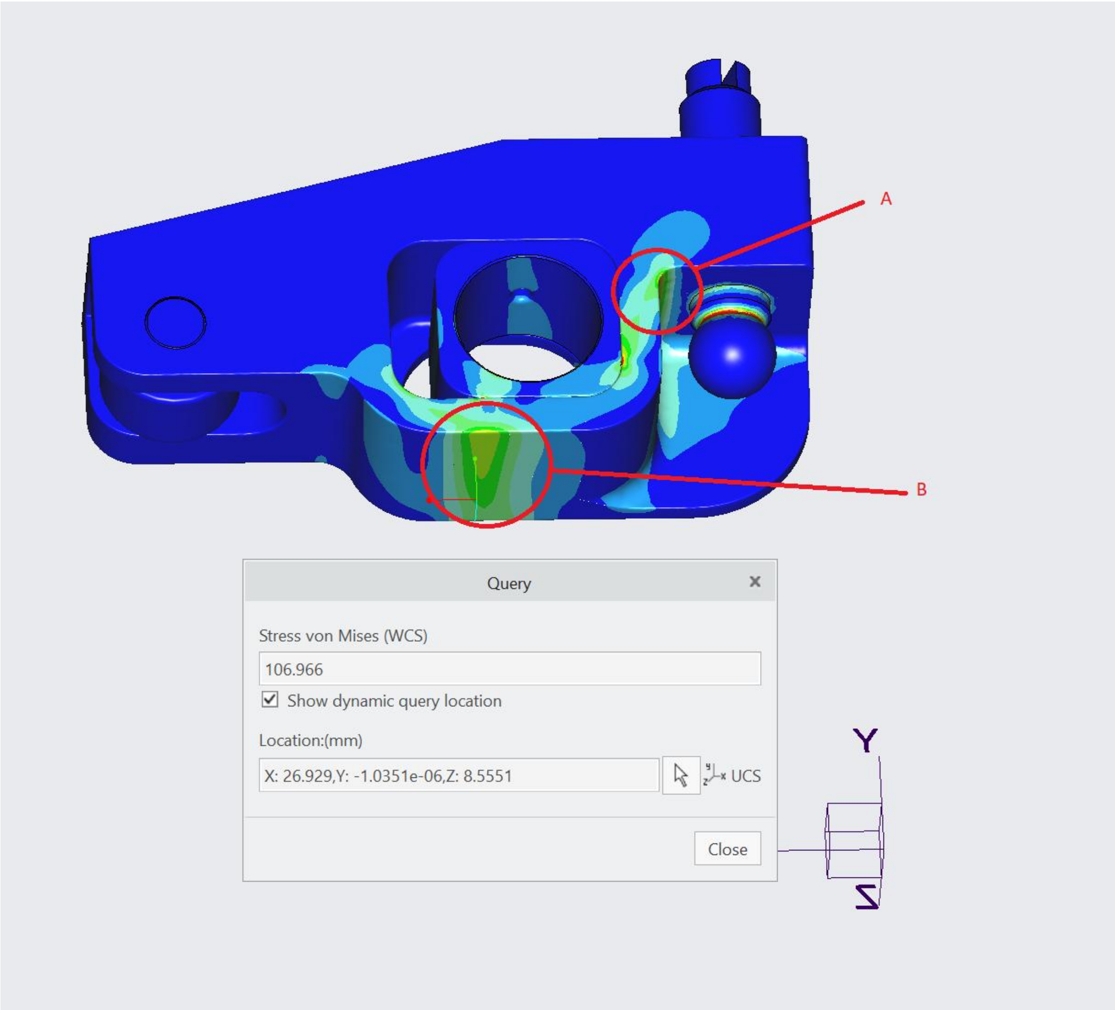


Figure 4.1: FEA results comparison for rocker

Although material specifications had not been calculated at this stage it was decided to modify the rocker geometry to reduce the stress areas A and B to a lower value. The area A could use a much bigger radius without affecting the rocker movement or pushrod adjusting bolt. This was decided to modify the radius to 6mm from the original 2mm. Area B was modified to have a complete radius rather than a flat bottom, this added material in the critical load are which was of high stress. Both modifications were of similar design to a Harland Sharp aftermarket 1.35 ratio rockers which can be seen below.

Figure 4.11: Comparison of Harland sharp rocker showing large radius in critical areas

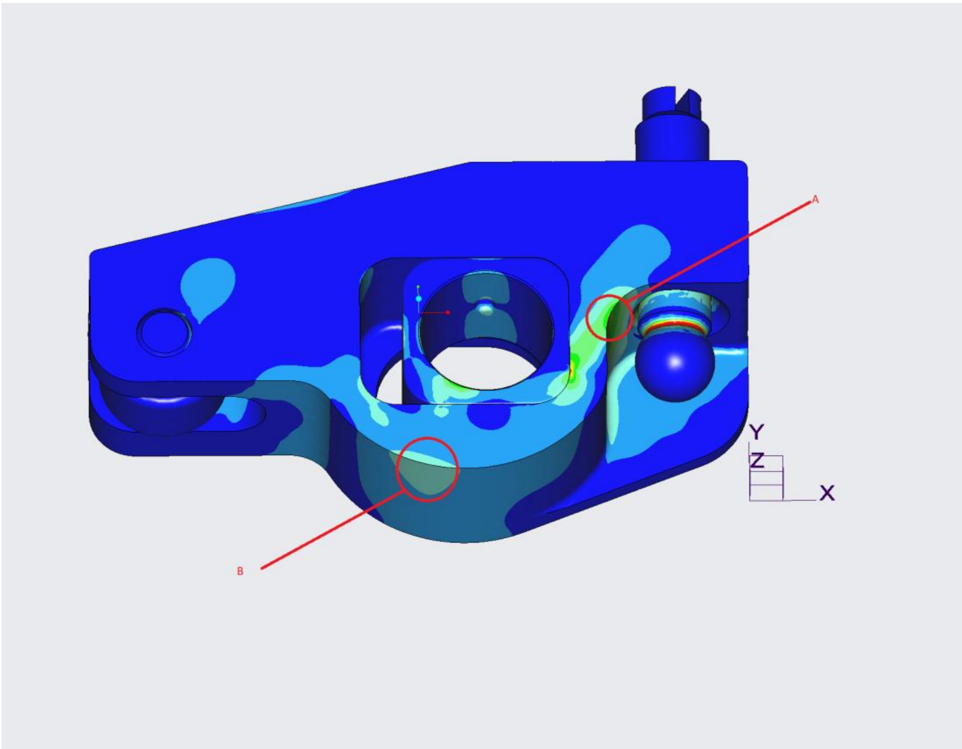


Figure 4.12: FEA results for modified design (V2)

After changing the design in the areas A and B the FEA simulation was repeated with the same loading conditions, constraints and contact points. This reduced the stress dramatically to a level that was thought to be of a more acceptable level although further refinement of location A was done. The results are tabulated below.

FEA High stress area A (σ)	275.9Mpa
FEA High stress area B (σ)	118.6Mpa

Table 1.05: Rocker forces modified design (V2)

The radius was again enlarged in area A to try to reduce the stress to under 200MPa. Unfortunately the webbing could not be mirrored as the head of the pushrod would foul. Because of this the radius was further enlarged to 10mm. This still allows adjustment and pushrod movement on the ball.

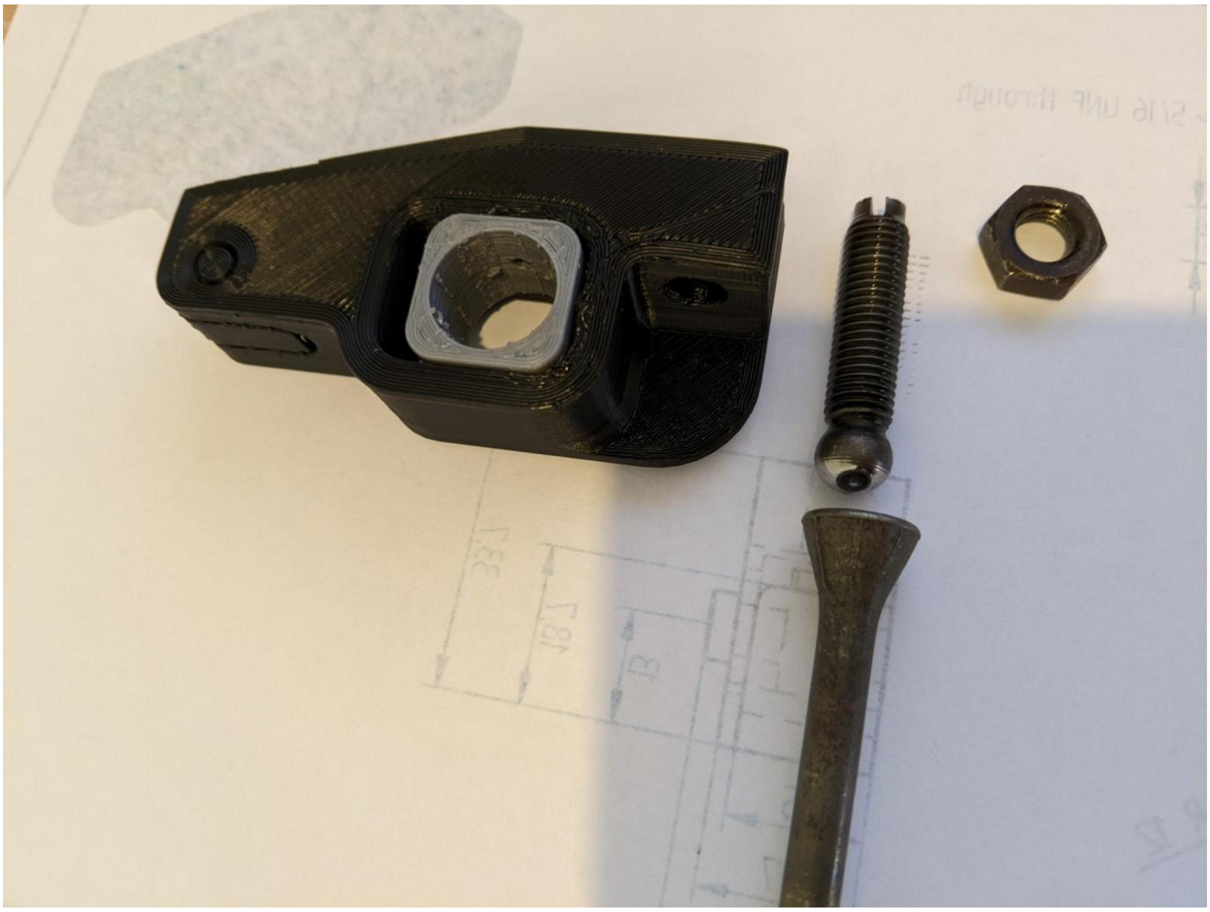


Figure 4.13: Pushrod and pushrod adjuster bolt being the shape limiting factor

FEA was repeated again with the same conditions with the larger 10mm radius, this further reduced the stress point. The high stress in the pushrod adjustment bolt is ignored as this part is a factory manufactured part that is not being re-designed. The results from the third variance of the rocker are below.

FEA High stress area A (σ)	190.6MPa
FEA High stress area B (σ)	118.6Mpa

Table 1.06: Rocker forces modified design (V3)

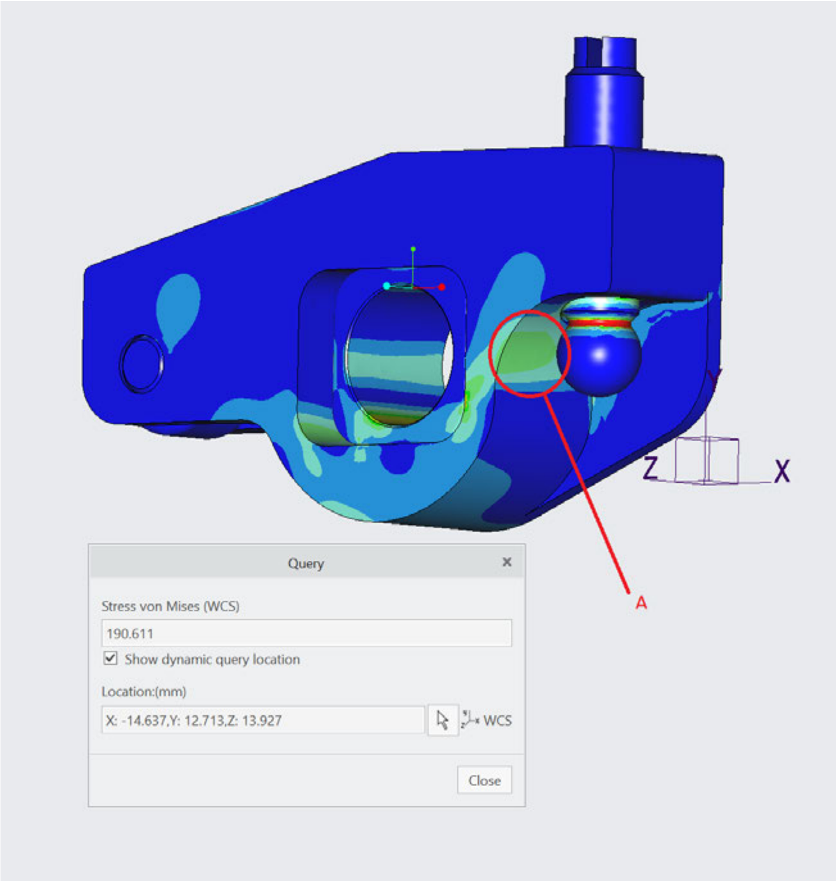


Figure 4.14: FEA results with further modifications (V3)

4.9 Rocker Material analysis

The material for the rocker needs to satisfy some general conditions. These conditions include fatigue loading, temperatures up to 140°C and ease of manufacture. Corrosion resistance is not an issue as the components have oil lubrication and also subject to splash lubrication.

The standard rockers come in 3 different materials, Pressed steel, cast steel and sintered steel. Aftermarket rockers are generally aluminium and are made from a 6061 or 7075 series alloy.

Below is a materials table for the material selection (pg831 Juvinall):

Material	**Ultimate** S_u (MPa)	**Yield** S_y (MPa)	**Machining (A= best, D=worst)**	**Cost (A=cheapest, D=Costly)**
6061-T6 Aluminium	310	275	C	C
7075-T6 Aluminium	570	505	B	D
1020 Carbon steel	448.2	330.9	B	A
1040 Carbon steel	620.5	413.7	A	B
4140 Alloy steel	1020.4	655	B	C

Table 1.07: material strengths and parameters for selection

Aluminium alloys listed are T6 heat treaded while the steels are all rolled state as this is how they would be commonly purchased from steel suppliers. Cast, sintered and pressed steel are not considered for this application due to shape constraints and ease of manufacture. From the Ashby diagrams carbon steels represent the best cost per strength ratio and also allow for infinite life with fatigue loading. Aluminium alloys have a similar cost but slightly reduced strength over the carbon steels and can only have calculated fatigue strength of $5x10^8$ cycles. For the steels a true endurance limit a 10^6-cycle strength can be calculated.

Fatigue strength is calculated with:

$$S_n = S_n^1 C_L C_G C_S C_T C_R$$

Where:

$$S_n^1 = 0.5 \times S_u\ (Steels)$$

$$S_n^1 = 0.4 \times S_u\ (Aluminium\ Alloys)$$

$C_L = load\ factor$

$C_G = Gradient\ factor$

$C_S = Surface\ factor$

$C_T = Temperature\ factor$

$C_R = Reliability\ factor$

Aluminium 6061 sample calculation:

$S_n^1 = 0.4 \times S_u = 310 \times 0.4 = 124MPa$

$C_L = Bending = 1.0$

$C_G = 10mm - 50mm = 0.9$

$C_S = (figure\ 8.13\ pg323\ Juvinall) = 0.8$

$C_T = < 840°\mathrm{F} = 1.0$

$C_R = 90\% = 0.879$

$$S_n = 78.48Mpa$$

The calculated strengths are tabled below:

Material	$S_n\ (MPa) 90\%\ reliability$	$S_n\ (MPa) 50\%\ reliability$
6061-T6 Aluminium	78.48	89.28
7075-T6 Aluminium	144.3	164.16
1020 Carbon steel	141.83	161.35
1040 Carbon steel	196.35	223.38
4140 Alloy steel	322.89	367.344

Table 1.08: Calculated fatigue strengths for rocker arms

From the stress analysis and the fatigue strength calculation it can be seen that for the conditions calculated the aluminium alloys would fail during fatigue before the $5x10^8$ cycles was met. This only leaves the carbon steels as a choice for the rockers. The two steels that would be acceptable for the 10^6-cycles would be 1040 and 4140.

The material for the rockers would be 1040 steel or equivalent as the 4140 is an unnecessary strength that would be more expensive and labour intensive to machine. The 1040 carbon steel gives a factor of safety of:

$$FOS = \frac{196.35}{190.6} = 1.03\ (90\%\ reliability)$$

4.10 Rocker Post stresses

Using the force $-Fe = Fr$ as the load the two maximums were calculated.

$$\sum M_1 = 0 = 9327.465 \times 11.88 - Ff \times 28.5$$

$$Ff = 3888.08N$$

$$\sum Fy = 0 = 9327.465 - 3888.08 - Fb$$

$$Fb = 5439.385N$$

Bearing area stress was calculated using $\sigma = \frac{Fr}{A_B}$

$$\sigma = \frac{9327.465}{243.1}$$

$$\sigma = 38.37Mpa$$

Bolt and stud torque from the Leyland workshop manual:

Location	**Torque setting**	A_t
Rear studs 3/8’’	42ft/lb / 56.945Nm	$56.645mm^2$
Front bolts 5/16”	25ft/lb / 33.896Nm	$37.42mm^2$

Table 1.09: Fastener torque settings

Bolt axial force (static);

$$F_{i,rear} = \frac{56.945}{0.2 \times 0.00952}$$

$$= 29.908kN$$

$$F_{i,front} = \frac{33.896}{0.2 \times 0.00794}$$

$$= 21.345kN$$

Bolt root stress, A_t taken from table 10.1 (page 413 Rober c. Juvinall);

$$\sigma_{rear} = \frac{29908}{56.645}$$

$$\sigma_{rear} = 529.813Mpa\ (static\ load)$$

$$\sigma_{front} = \frac{21345}{37.42}$$

$$\sigma_{front} = 570.42Mpa\ (static\ load)$$

Fluctuating load added, stress with rolled threads $k_f = 3.0$;

$$\sigma_{rear} = 288.09Mpa$$

$$\sigma_{front} = 311.71Mpa$$

Cylinder pressure force on rear studs;

$$F_{cylinder} = \frac{4}{9} \times 378.24$$

$$F_{cylinder} = 168.11N\ (each\ stud, fluctuating\ load)$$

$$\sigma_{rear} = \frac{168.11}{56.645} \times 3$$

$$\sigma_{rear} = 8.903Mpa$$

Total forces;

$$\boxed{\sigma_{rear\ total} = 826.806Mpa}$$

$$\boxed{\sigma_{front\ total} = 882.13Mpa}$$

Shear stress bolt surface;

$$\sigma_{shear,rear} = \frac{29900}{183.59} = 162.86Mpa$$

$$\sigma_{shear,front} = \frac{21345}{103.67} = 205.89Mpa$$

4.11 FEA posts

The FEA on the rocker posts was done similar to the rocker by using the calculated valves and comparing the results. The axial bolt loads were used on surface regions on18mm for the rear stud and 14mm for the front bolt. The applied loads were -29.9kN and -21.3 respectively. A bearing load was used for the reaction from the rocker shaft and this was 5439.4N. Results show average stress around bearing surface to be close to calculated average.

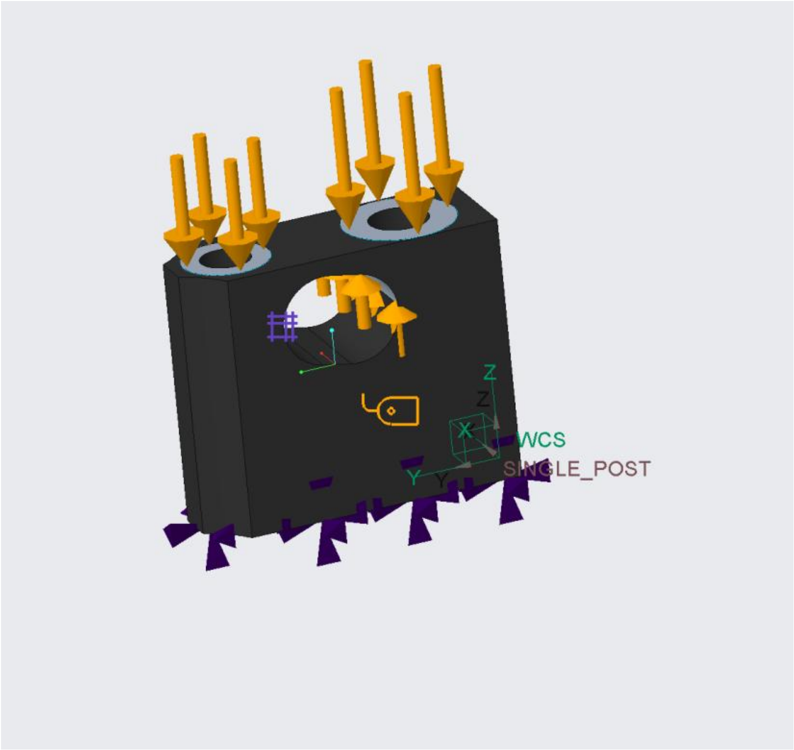


Figure 4.11: Rocker loads and constraints

Below is a table of results and areas of interest. Area C is localised around the hole that breaks into the stud hole and is ignored.

Calculated average bearing stress (σ)	38.37Mpa
FEA point stress over bearing face (σ)	51.7Mpa
Shear stress back bolt(σ)	162.86Mpa
Shear stress front bolt(σ)	205.89Mpa
FEA bolt rear (Max) (σ)	171.95Mpa
FEA bolt front (Max) (σ)	218.6Mpa
FEA area A	235.1Mpa
FEA area B	205.3Mpa
FEA area C	546.1Mpa

Table 1.1: Rocker post calculated stresses

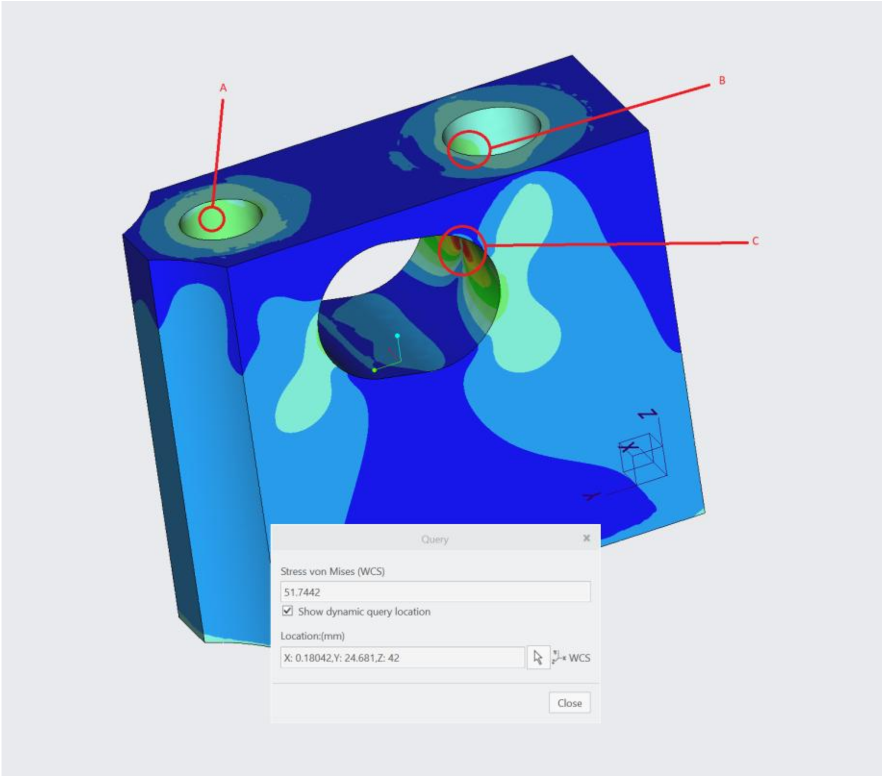


Figure 4.12: FEA results comparison for single post

4.12 Materials

For the posts the same materials used for the rocker analysis are used. The alternating stress on the posts is caused by the rocker shaft giving a bearing stress to the slotted section of the post. The rocker posts are under the same conditions of temperature up to 140°C and again ease of manufacture. These posts also receive oil lubrication and subject to splash oil. The posts will also be designed for maximum life so $5x10^8$ cycles for aluminium and for the steels a true endurance limit of 10^6-cycle strength can be calculated.

Fatigue strength is calculated with:

$$S_n = S_n^1 C_L C_G C_S C_T C_R$$

Where:

$$S_n^1 = 0.5 \times S_u \ (Steels)$$

$$S_n^1 = 0.4 \times S_u \ (Aluminium\ Alloys)$$

Aluminium 6061 sample calculation:

$S_n^1 = 0.4 \times S_u = 310 \times 0.4 = 124MPa$

$C_L = Axial = 1.0$

$C_G =< 10mm = 0.8$

$C_S = (figure\ 8.13\ pg323\ Juvinall) = 0.8$

$C_T = < 840°\text{F} = 1.0$

$C_R = 90\% = 0.879$

$$S_n = 69.76MPa$$

The calculated strengths are tabled below:

Material	$S_n\ (MPa) 90\%\ reliability$	$S_n\ (MPa) 50\%\ reliability$
6061-T6 Aluminium	69.76	79.36
7075-T6 Aluminium	128.27	145.92
1020 Carbon steel	126.07	143.42
1040 Carbon steel	174.53	198.56
4140 Alloy steel	287.01	326.528

Table 1.08: Calculated fatigue strengths for rocker posts

From the stress analysis and the fatigue strength calculation any material in the table 1.08 would meet the material loading conditions. As the rocker post has a bearing surface carbon steel is thought to be the preferred material. The material chosen for the rocker post is 1020 steel with allowing for the option of being able to carburize the bearing surfaces.

$$FOS = \frac{126.06}{51.7} = 2.44\ (90\%\ reliability)$$

4.13 Post fastener materials

From Table 10.5 (pg435 Juvinall) using a SAE class 12.9;

$S_n^1 = 0.5\ \times\ S_u = 1220 \times 0.5 = 610MPa$

$C_L = axial = 1.0$

$C_G =< 10mm = 1.0$

$C_S = (figure\ 8.13\ pg323\ Juvinall) = 0.7$

$C_T = < 840°\text{F} = 1.0$

$C_R = 90\% = 0.879$

$$S_n^1 = 375.33MPa$$

For rear stud 3/8 UNF;

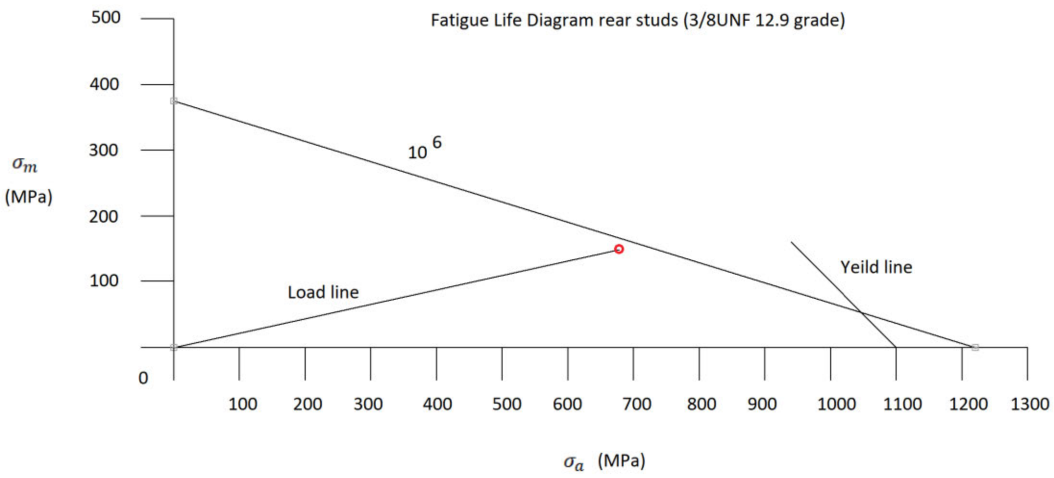
$$\sigma_{rear\ min} = 529.813MPa$$
$$\sigma_{rear\ max} = 826.806MPa$$

$$\sigma_m = 678.31MPa\ , \sigma_a = 148.5MPa$$

For front stud 5/16 UNF;

$$\sigma_{rear\ min} = 570.42MPa$$
$$\sigma_{rear\ max} = 882.13MPa$$

$$\sigma_m = 726.275MPa\ , \sigma_a = 155.85MPa$$



$$\frac{\sigma_a}{S_n} + \frac{\sigma_m}{S_u} = \frac{1}{FOS}$$

$$Rear\ stud\ FOS = 1.05$$

$$Front\ stud\ FOS = 0.989$$

Rear stud design is fine for designed load and fatigue although front stud falls just short. Changing to a bigger stud would require drilling and tapping the cylinder head which would make installation more difficult. The second option would be to lower the initial tightening torque to reduce static load. By reducing initial tightening torque to 30Nm the static stress would be lowered to $504.86MPa$. New loads would be:

$$\sigma_{front\ min} = 504.86MPa$$
$$\sigma_{front\ max} = 816.57MPa$$

$$\sigma_m = 660.72MPa\ , \sigma_a = 155.85MPa$$

$$New\ FOS = 1.045$$

This shows the importance of torques settings on bolts as extra tightening tension could potentially lead to failure. A FOS over 1 has been deemed acceptable for this design due to maximum loadings and an infinite fatigue life with 90% reliability has been used.

4.14 Testing

Once the design was finalised the final models and drawings were produced which can be seen in Appendix A.4. From these models a 3D printed prototype was made to confirm the design. The model was simplified to only two rockers as the printer max size is 235x235mm. This was sufficient for testing and also reduced printing time.

The rocker arm was printed with green 1.6mm PLA+ with a layer height of 0.2mm. These proved to be acceptable for visual prototypes but were not very structural and too weak for the rocker posts. The rocker posts were printed with silver ABS 1.6mm filament; this was sliced using Ultimaker Cura 4.5 using a layer height of 0.12mm. Print time for this component was 16h 30min. The roller tip, roller tip shaft, actuating cam and sliding bush were printed with the same settings. The rocker linkage arm was laser cut from 3mm acrylic as was much stronger for a thin part. The rocker shaft was machined from 1020 9/16" bright mild steel to make the assembly usable.

Figure 4.13: 3D printed prototype for testing

For testing of the variable rocker assembly, they were bolted to a 1098cc A-series engine with a 12g295 cylinder head. The valve spring was removed as the spring force would break the plastic prototype. First the maximum cam lift was measured; this was measured using a dial indicator directly on the pushrod while turning the motor over.

After maximum cam lift was measured valve lift was measure on the rocker at maximum and minimum ratios using the variable shaft position. Due to not being able to use the valve spring valve lift had to be measure on the tip of the rocker.

	Measured lift	Calculated ratio
Camshaft	6.67mm	
minimum ratio	8.04mm	1.205:1
Maximum ratio	10.82mm	1.62:1

Table 1.09: Calculated rocker ratios

It is noted that the plastic did give some slight inaccuracies due to flex and also not being able to measure direct valve lift.

Figure 4.14: Measuring maximum cam lift

Figure 4.15: Measuring maximum valve lift

Chapter 5 – Conclusions

5.1 Introduction

This report was set out to provide a complete design for fitting of continuously variable ratio rockers to an A-series engine. The report has fulfilled a design that has been analysed to perform both for strength and mechanical working

5.2 Conclusions

Overall this project has met the project specification with a working model and detailed drawings for the manufacture of the continuously variable ratio rocker arms. Unfortunately due to time constraints a full manufactured prototype was not able to be manufactured. The 3d model was successfully run in Solidedge ERA (Explode, Render and animate) with all components working as expected.

In summery the components of the project specification were met and further work is detailed below. The project specification can be seen in chapter 8.

5.3 Further work

There are a few components that would need to be added to further work for this project. The components of this design that are not included are:

- Variable actuation control
- Variable actuation control system
- Components manufactured in material specifications
- Dynamometer testing on an A-series engine

5.4 Variable actuation control

For variable actuation control, the movement of the main shaft needs to have some rotation control. From the literature review the current VVL control systems are actuated by either hydraulic of electrical. For a hydraulic system to work it can use the engines oil system which can supply oil pressure of around 60psi. Although this system could be made quite simple and compact it doesn't suit a continuously variable motion. The hydraulic system could rotate the shaft by using a lever arm with a small cylinder but infinitely variable control would become difficult. This system would suit a discrete variation as per its use with Honda VTEC and Nissan VVL.

An electrical controlled system with a stepper motor would be an ideal setup for the variable actuation. Stepper motors have precise movement control and high torque at low RPM (MCMA 2020). This is idea due to the movement from low lift to max lift is only 90° but needs to be precise. Positioning at particular points between max and min would also be possible allowing for completely continuous variation. From some quick research a NEMA 17 stepper motor would be physically possible to fit although further work for actuating torque and mounting would need to be done. Refer to Appendix A5.1 for data sheet. NEMA 17 stepper motors are also run from a 12V system which is ideal for most cars.

5.5 Variable actuation control system

For the variable actuation control system there are numerous programmable microcontrollers available. For the selection process identification for the inputs and outputs and also the memory needed to process and store (ARM 2020). For this project the only output would be the stepper motor but there would be a few inputs. Inputs could be:

- Crank RPM, Rotary encoders, photoelectric or magnetic rotational.
- Throttle position (TPS)
- Manifold absolute pressure (MAP)
- Rocker shaft home positions

A map based from these inputs would need to be developed to change the position of the stepper motor based on the inputs. The microcontroller would also need user friendly software that could be developed so that the system could be tuned without needing the knowledge of the programming language. A USB or similar interface for tuning would need to be added to the board.

5.6 Components manufactured in material specifications

For testing purposes a full working prototype in the materials specified needs to be manufactured. This can then be run to ensure all sizes and the tolerances are acceptable. Manufacturing a prototype will show any problems with machining due to complex part design.

Manufacturing would be done using 2 axis CNC lathes and 3 axis CNC machining centres. This keeps cost down and also allows for simplified machining.

3D models can be saved as IGES files and imported into Mastercam for tool path creation.

5.7 Dynamometer testing on an A-series engine

Once a complete prototype is machined using the material specified and full setup with the programmable microcontroller and stepper drive could be tested. This would be done on a 1098cc A-series at Graham Russell Engineering as they have a water brake dynamometer setup that could be utilised.

Chapter 6 – Risk Assessment

This project has some risks involved during and beyond the completion of this dissertation. These risks will be identified, evaluated and managed with a risk management control. The risks will be categorised into two types, that of during and post project.

During project risk identification and evaluation;

1. Using 3d printer for prototype models - Heated table 110°C, heated extruder 240°C and rapidly moving axis and motors with exposed drive belts.
2. Machining tools - Milling machine and Center lathe, both have risk of hot shavings, heavy objects and moving parts.
3. Fitting – Cylinder head with compressed springs
4. Fitting – cylinder head weight and sharp edges
5. CNC machinery – Heavy objects moving parts and hot shavings.

Control management;

1. When using the 3d printer enclosure needs to stay closed until table and extruder nozzle have sufficiently cooled this temperature is outputted on the LCD screen so is clear to see. Hands and hair to keep clear of moving axis and motors which can be done by keeping enclosure closed.
2. When using machining tools there is a requirement of PPE. Wearing long parts and long sleeves but no loose clothing when operating the machines. Safety glasses at all time and hearing protection when necessary. Steel cap boots at all times when using the machining tools as there is a risk with heavy objects. Hands clear of spinning chuck/tools and tables and lead screws.
3. When fitting valve springs safety glasses should be worm due to the compressed springs. Correct valve spring compressor should be used and steel cap boots worn.
4. When carrying the cylinder head care should be taken due to the heavy weight of the cast steel design. Safety boots should be worn. Gloves can be worn to prevent sharp edges cutting fingers.
5. When using CNC machinery correct operating producers should be followed. This includes understanding all door locks and safety, keeping away from moving machine components and wearing appropriate PPE.

Post project risk identification and evaluation;

6. Change in engine tune – Emissions change and poor running if not tuned
7. Fitting – Moving parts with sharp edges.

Control management;

6. When fitted to the engine the engine should be tuned accordingly to ensure correct running and air fuel mixture is met. Emissions testing should be done when fitted on vehicles newer than 1976 in accordance to the state registrations laws. New south Whales emission laws

can be found at: https://www.rms.nsw.gov.au/about/environment/air/emission-standards.html. Incorrect running at worst could lead to engine failure.

7. When the rocker assembly is fitted to the engine care should be taken due to moving parts and sharp edges on components. Correct torque specifications should be met for all fastened hardware.

Chapter 7 – Resource Analysis

This project will use a number of recourses to complete the task these are outlined below. These are in no particular order.

- Computer software – Three software packages will be used for the completion of the project.
  -Solidedge ST10 is software I have personally so is free to use for the modeling.
  -Crea Parametric 4.0 can be used with my student license and is free for use.
  -Engine Analyzer Pro will need to be purchased a basic version is $129 or pro version $499.
- CNC machinery for prototypes can be used after work hours free of charge; I also have access to a small centre lathe, drilling equipment and press personally.
- 3d printer will be purchased for initial prototype
- Materials for prototypes with need to be purchased
- Engine that will be used for the testing is my own 1098cc engine.

These resources and estimate prices and tallied in the table below

Resource	Quantity	Cost	Availability
Test engine	1	-	Any time
Solidedge ST10	-	-	Anytime
Creo Parametric 4.0	-	-	Anytime
Engine Analyzer	-	$129-$499	Anytime
Machinery • Lathe • Milling machine • Press • Drilling machine	-	-	CNC machines available after work hours on request. Manual machines anytime
material	-	Allowance $400	Need to be ordered
3D printer	1	$263.46	Anytime

Table 1.0: Resource availability and costing chart

Details for resources

Machinery and material will be sourced at: *Zamco Engineering PTY LTD 1/15 Cranegie Pl Blacktown 2148*

Engine analyser software will be downloaded from: *https://performancetrends.com/Engine-Analyzer-Pro.htm*

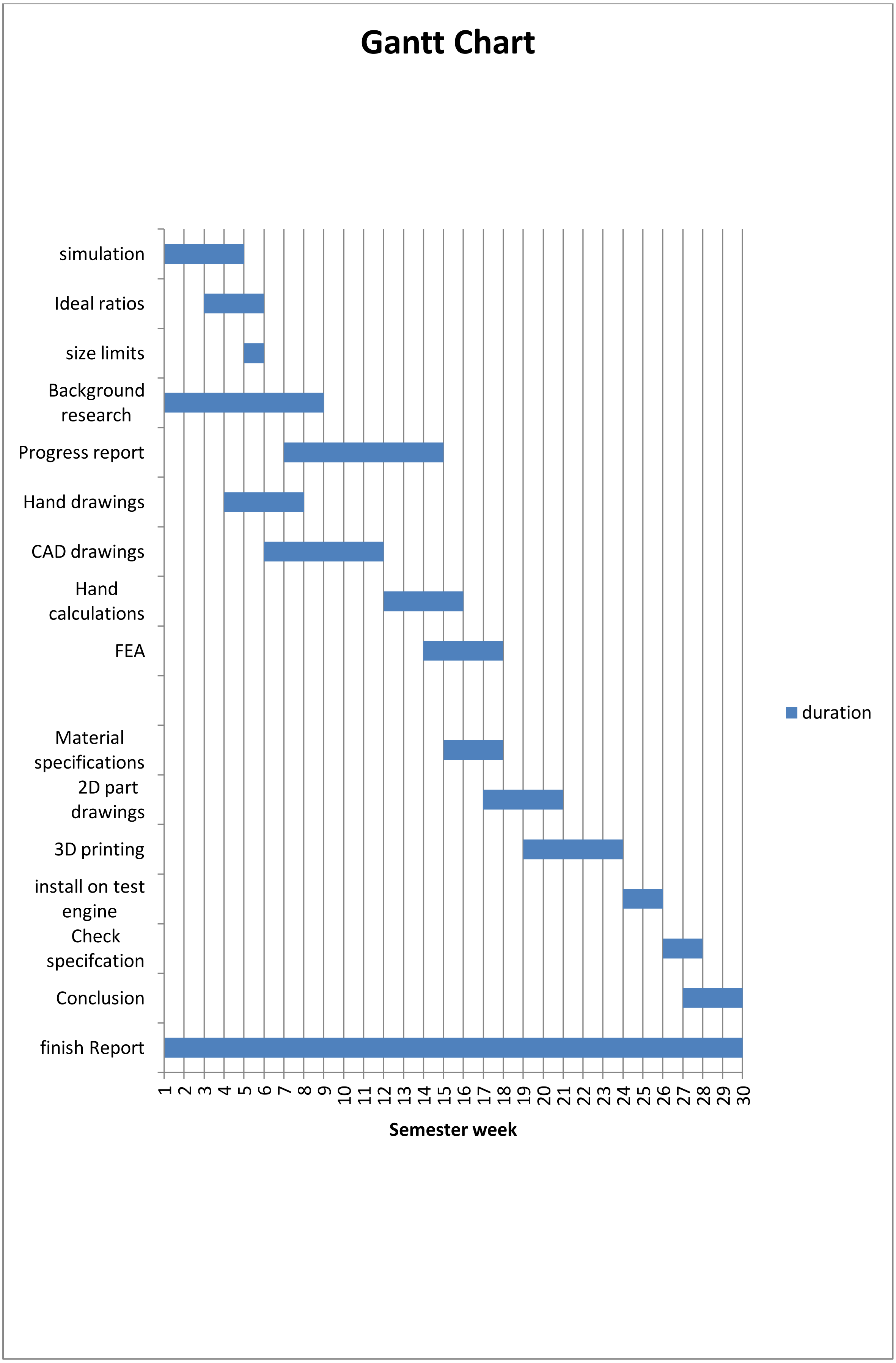
3D printer was sourced from Banggood on the 3/3/20

Chapter 8 - Project Timelines

Phase	Data collection (Semester 1)
1	Simulation using 1098cc a-series motor specifications
1.1 1.2	Find ideal ratios that can be used Size limits
1.3	Background research
1.4	Progress report
	Design
2.	Hand designs and sketches of rocker
2.1	Cad drawing of new design and existing rocker
2.2	Hand calculations of rocker forces and stresses
2.3	FEA analysis of new design and existing
	Determine material and part drawings (semester 2)
3.	Material specifications
3.1	2d part drawings for manufacture
	Manufacture prototype
4.	3D print working prototype
4.1	Install on test engine
4.2	Check specifications (maximum and minimum lift)
	Results
5.	Conclusions assessed and documented
5.1	Finish report
Extra	If time persists
6.	Control system
6.1	Prototype to material specifications
6.2	Dyno testing

Table 1.0: Project timeline

The Project timeline is to ensure that the project outline can be met and that it is achievable over the two semester of research project part 1 & 2. The projected timeline is shown below in a Gantt chart. Project begins Semester 1 2020 and finishes end of semester 2. The Gantt chat does not show mid semester break or midyear break. Finish report is work involved throughout both semesters and will be the final phase of the timeline.


Gantt Chart
simulation
Ideal ratios
size limits
Background research
Progress report
Hand drawings
CAD drawings
Hand calculations
FEA
Material specifications
2D part drawings
3D printing
install on test engine
Check specifcation
Conclusion
finish Report
duration
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30
Semester week

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Appendices A.1

Project Specification

ENG4111/4112 Research Project

Project Specification

For: Benjamin Schnebli

Title: Continuously Variable Ratio Rocker Arms

Major: Mechanical Engineering

Supervisors: Chris Snook

Enrolment: - ENG4111 – EXT S1, 2020
- ENG4112 – EXT S2, 2020

Project Aim: The project aim is to design and produce a continuously variable ratio rocker arm for use with the BMC a-series engines

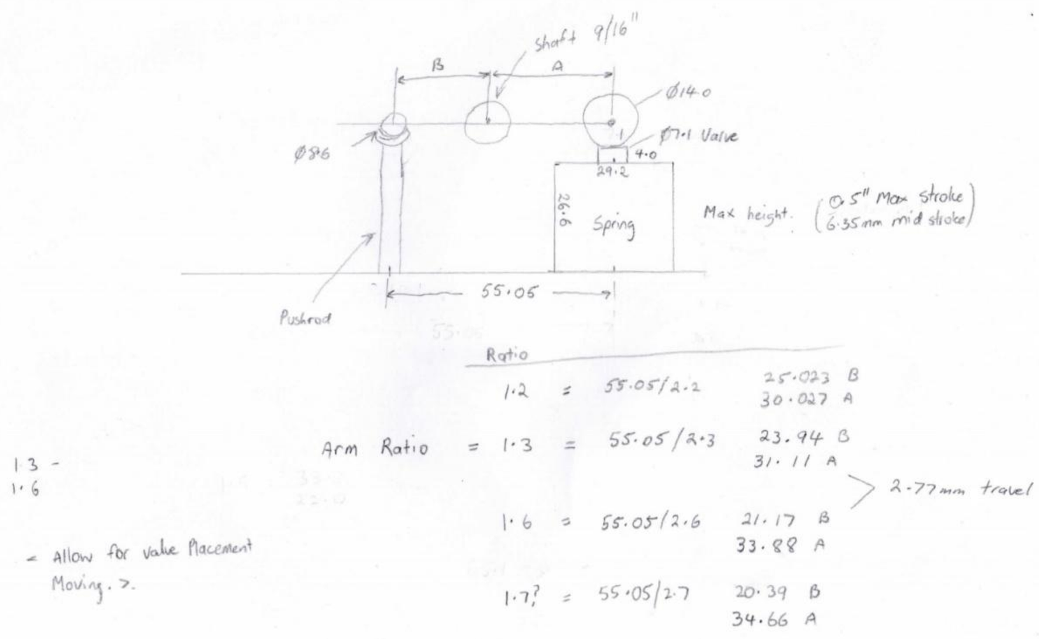
Programme: **Version 1, 15th March 2020**

1. Find the useable rocker ratios for the A-series engine
2. Examine existing rockers and evaluate design and strength
3. Create a specification for the design
4. Design a variable rocker to withstand the specification developed.
5. Perform Finite element analysis and hand calculations for stress analysis to find material specifications needed.
6. Make a prototype that can be tested

If time and resources persists

7. Design a system to control the rockers
8. Test a working prototype

Basic Layout/ Arm ratio calculations



Appendix A.1.1: Hand sketch of basic layout for rocker ratio calculation.

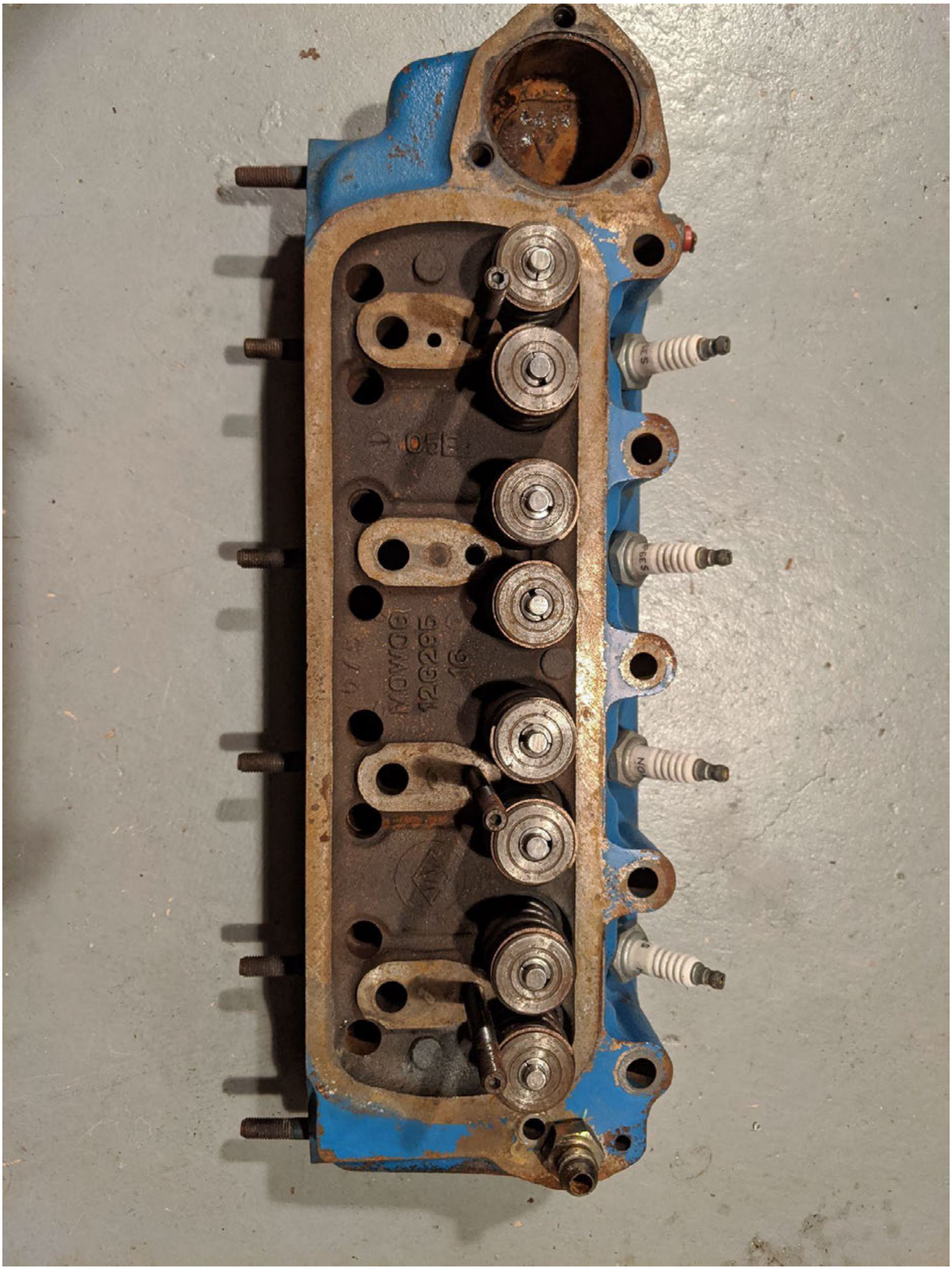
GENERAL DATA

The following figures are taken from the latest manufacturing specifications and drawings available at the time of publication, but the manufacturers, in accordance with their policy of constant product improvement, reserve the right to vary the specifications contained in this manual without notice. Where tolerances are not shown, the figure quoted must be regarded as nominal.

	848 cc. 850 Saloon and Van	998 cc. Mini Minor, Deluxe, Mini Matic, Van, Moke, BMC Moke and Cooper	997 c.c. Cooper	1098 cc. Mini 1100 Saloons and Vans, BMC Moke, Clubman Saloon	1275 cc. Cooper 'S', Moke, Clubman 'G.T.'
ENGINE					
Type symbol	8AM and 8Y Saloon and Van	9YE — Saloon and Van, 9YA — Deluxe, 9AN—99H—Mini Matic, 9FA and 9Y — Cooper, 9YB — Moke, 9YH — BMC Moke.	9F	10Y 10YJ—Std. Saloon & Van 10YC and 10YM — Saloon 10YF — BMC Moke 10YN — Mini K 1000—1100 Saloon & Van 1001 — Clubman Saloon	9F/SA/Y Cooper 'S' 1200 — Clubman G.T. 12Y — G.T. or Moke 1204 Moke
Number of cylinders and valve operation	4 cylinder OHV pushrod operated.	= = =	= = =	= = =	= = =
Bore	2.478" (62.94 mm)	2.543" (64.588 mm)	2.458" (62.43 mm)	2.595 2.543" (64.588 mm)	2.780" (70.60 mm)
Stroke	2.687" (68.26 mm)	3.00" (76.20 mm)	3.20" (81.33 mm)	3.296" (83.72 mm)	3.20" (81.33 mm)
Capacity	51.7 cu. in. (848 cc)	60.96 cu in (998 cc)	60.87 cu. in. (997 cc)	67 cu in (1098 cc)	77.9 cu. in. (1275 cc)
Firing order	.. 3. 4. 2.	= = =	= = =	= = =	= = =
Compression ratio — high compression	8.3 : 1	8.3 : 1, 9.0 : 1 Cooper & Mini Matic	9.0 : 1	8.5 : 1	9.75 : 1 Cooper 'S' and Clubman G.T.
— low compression	7.8 : 1	7.8 : 1 Cooper, 7.6 : 1 Moke	8.3 : 1	7.5 : 1	8.8 : 1 Moke
Capacity of combustion chambers	24.5 cc	24.5 cc, 22.4 cc Mini Matic, 28.3 c.c. Cooper	26.1 cc.	1.59 cu. in. (26.1 c.c.)	1.306 cu. in. (21.4 cc)
Torque	44 lb. ft. (6.08 kg. m) @ 2900 r.p.m.	52 lb. ft. (7.27 kg. m) @ 2700 r.p.m. 57 lb. ft. (7.88 kg. m) @ 3000 r.p.m. H.C. Cooper 56 lb. ft. (7.74 kg. m) @ 2900 r.p.m. L.C. Cooper	54 lb. ft. (7.65 kg. m) @ 3600 r.p.m. (H.C.) 53 lb. ft. (7.32 kg. m) @ 3500 r.p.m. (L.C.)	60 lb. ft. (8.29 kg. m) @ 2500 r.p.m.	79 lb. ft. (10.9 kg. m) @ 3000 r.p.m. Cooper 'S' and Clubman G.T. 69 lb. ft. (9.6 kg. m) @ 2500 r.p.m. Moke
Brake horse power	34 @ 5500 r.p.m.	40 & 42 Mini Matic @ 5250 r.p.m. 55 @ 5800 r.p.m. Cooper H.C. 52 @ 6100 r.p.m. Cooper L.C.	55 @ 6000 r.p.m. (H.C.) 52 @ 6000 r.p.m. (L.C.)	50 @ 5100 r.p.m.	78 @ 5800 r.p.m. Cooper 'S' and Clubman G.T. 65 @ 5250 r.p.m. Moke
R.A.C. rating	9.8 h.p.	10.3 h.p.	9.64 h.p.	10.3 h.p.	12.4 h.p.
Engine idle speed (approx.)	500 r.p.m.	500 r.p.m. (650 r.p.m. in Neutral — Mini Matic)	500 r.p.m.	= = =	= = =
Oversize bore: first	+ .010" (0.254 mm)	+ .010" (.254 mm)	+ .010" (0.254 mm)	+ .010" (.254 mm)	+ .010" (.254 mm)
maximum	+ .040" (1.016 mm)	+ .020" (.508 mm)	+ .040" (1.016 mm)	+ .020" (.508 mm)	+ .020" (.508 mm)

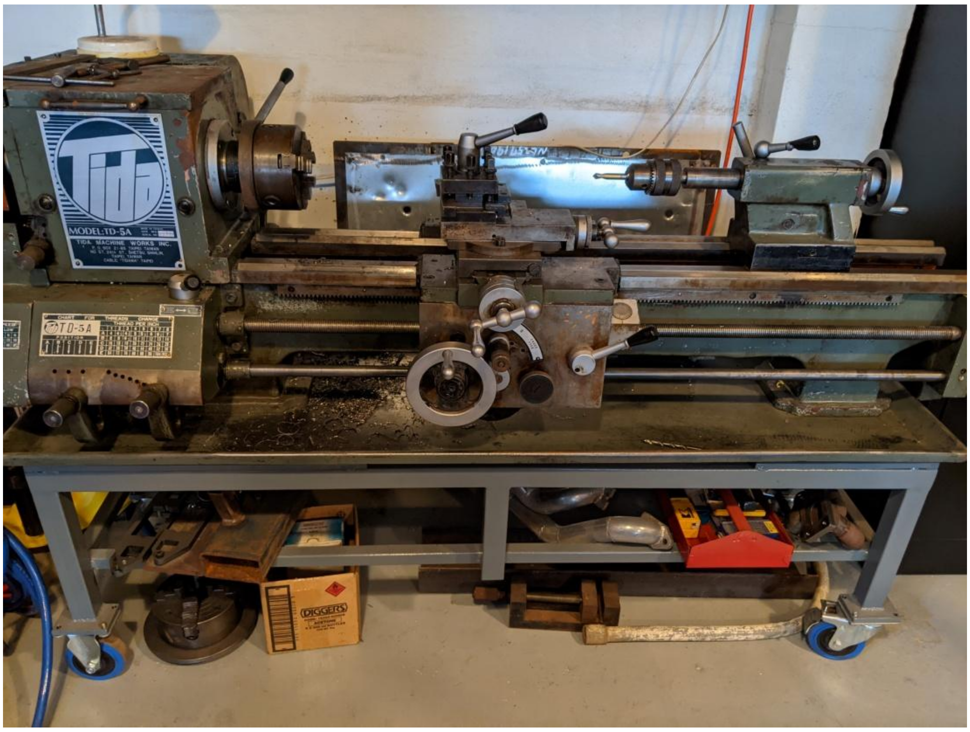
= = = denotes common to model listed in preceding column.

Appendix A.1.2: A-series engine general data (Leyland workshop manual)



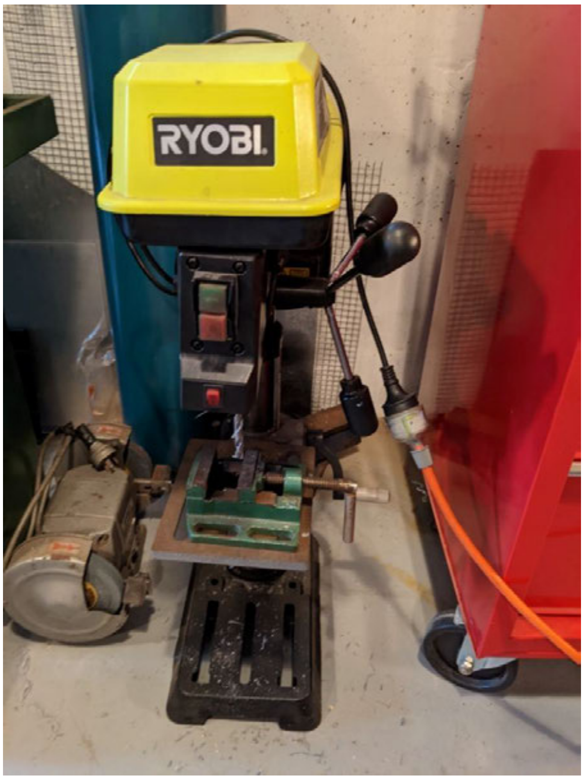
Appendix A.1.3: 12g295 cylinder head from 998cc A-series

Appendices A.2 – Equipment and measuring



Appendix A.2.1: Centre lathe for prototype parts and hand tools above

Appendix A.2.2: Hydraulic press for removing valve springs



Appendix A.2.3: drill press

Appendix A.2.4: Hand tools

Appendices A.3 - United States patents

US 20070151532A1

(19) **United States**
(12) **Patent Application Publication** (10) **Pub. No.: US 2007/0151532 A1**
Vaseleniuck (43) **Pub. Date: Jul. 5, 2007**

(54) **VARIABLE RATIO ROCKER ASSEMBLY**

(76) Inventor: **David N. Vaseleniuck**, Denver, NC (US)

Correspondence Address:
TREGO, HINES & LADENHEIM, PLLC
9300 HARRIS CORNERS PARKWAY
SUITE 210
CHARLOTTE, NC 28269-3797 (US)

(21) Appl. No.: **11/619,403**

(22) Filed: **Jan. 3, 2007**

Related U.S. Application Data

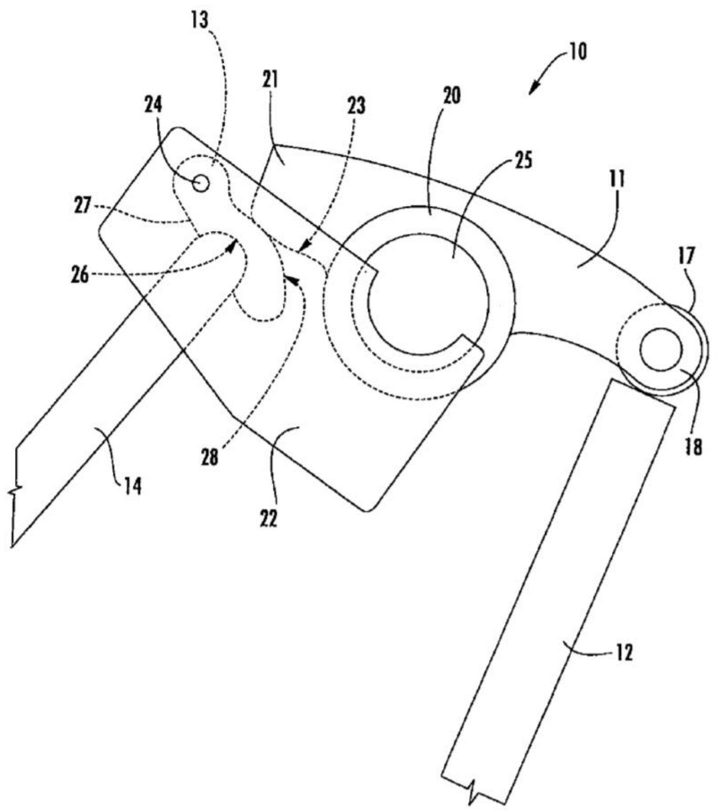
(60) Provisional application No. 60/756,154, filed on Jan. 4, 2006.

Publication Classification

(51) **Int. Cl.**
F01L 1/34 (2006.01)
F01L 1/18 (2006.01)
(52) **U.S. Cl.** **123/90.16**; 123/90.39; 123/90.41

(57) **ABSTRACT**

A variable ratio rocker assembly for increasing the amount of air entering a combustion cylinder. The rocker assembly includes a rocker arm adapted to move a valve between a closed position and an open position, a radiused follower adapted to interact with a portion of the rocker arm, and a push rod for moving the follower in response to a camshaft. The follower causes the rocker arm to move the valve between the closed position and the open position.



Appendix A.3.1: David N.vaseleniuck Unites states patent for a Variable ratio rocker assembly 2007.

United States Patent [19]

Pohle

[11] **4,182,290**

[45] **Jan. 8, 1980**

[54] **VARIABLE RATIO ROCKER ARM**

[76] Inventor: **William A. Pohle**, P.O. Box 684, Hallandale, Fla. 33009

[21] Appl. No.: **918,061**

[22] Filed: **Jun. 22, 1978**

[51] **Int. Cl.**[2] **F01L 1/18**

[52] **U.S. Cl.** **123/90.39**; 123/90.45; 74/519

[58] **Field of Search** 123/90.39, 90.41, 90.42, 123/90.43, 90.45, 90.46, 90.47, 90.15, 90.16, 90.36; 74/519, 559

[56] **References Cited**

U.S. PATENT DOCUMENTS

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1,654,020	12/1927	Schmidt	123/90.39
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2,322,173	6/1943	Spencer	74/559
3,189,011	6/1965	Briggs	123/90.45
3,251,350	5/1966	Thompson	123/90.41
3,911,879	10/1975	Altmann	123/90.39
4,077,369	3/1978	Buehner	123/90.39

FOREIGN PATENT DOCUMENTS

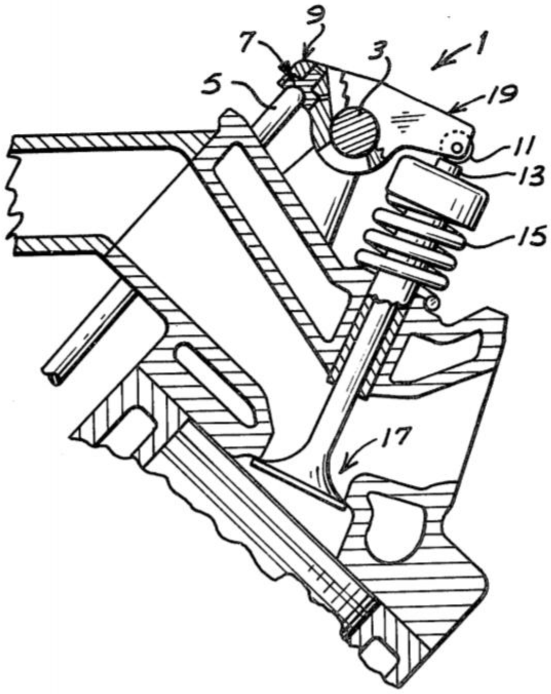
733651	3/1943	Fed. Rep. of Germany	123/90.39

Primary Examiner—Charles J. Myhre
Assistant Examiner—Jeffrey L. Yates
Attorney, Agent, or Firm—Crickenberger and Moore

[57] **ABSTRACT**

The amount of lift of a valve lifter rocker arm is determined by a rotatably mounted valve pushrod seat having an eccentrically positioned depression for receiving the pushrod end. The seat is rotatably adjusted to one of two predetermined positions to increase or decrease the leverage ratio of the rocker arm. A screwdriver slot in the seat member permits easy adjustment without removing the rocker arm, and a set screw locks the seat member in the desired position.

5 Claims, 4 Drawing Figures



Appendix A.3.2: William A. Pohle United states patent for Variable ratio rocker arm 1980.

U.S. Patent Jun. 27, 1989 Sheet 1 of 3 **4,841,799**

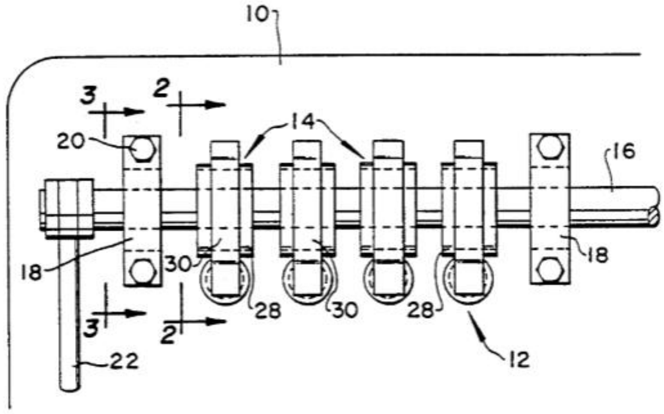


FIG. 1

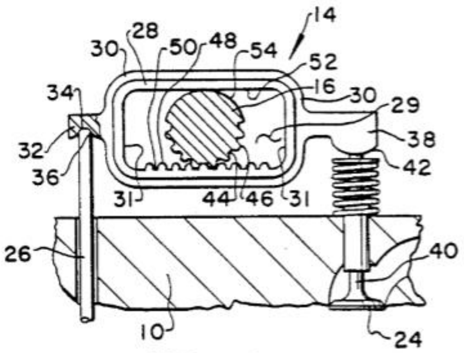


FIG. 2

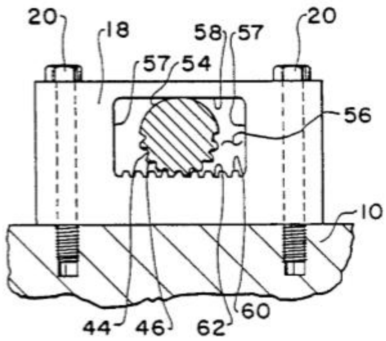


FIG. 3

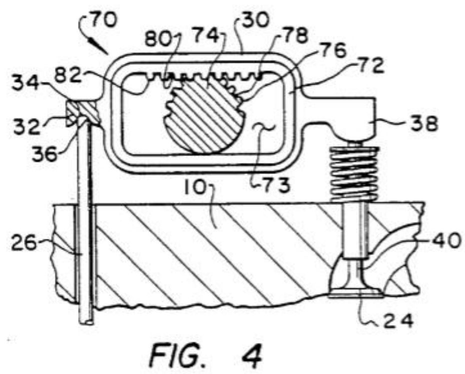


FIG. 4

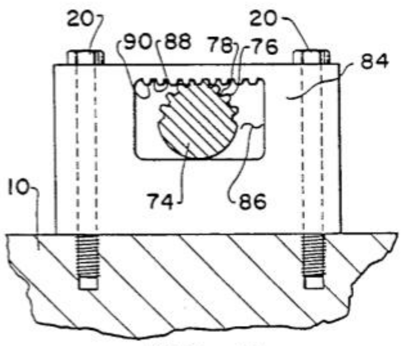


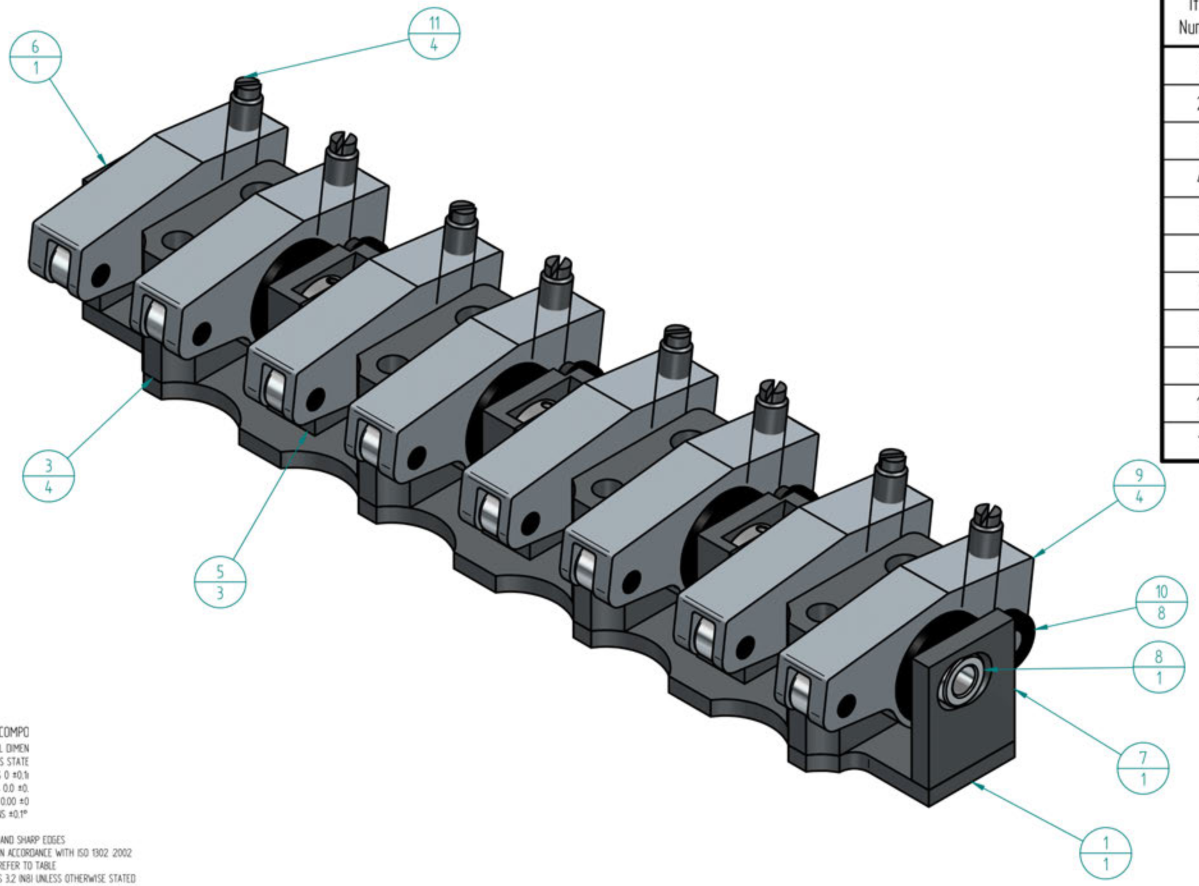
FIG. 5

Appendix A.3.3*: William W. Entzminger 1989 variable ratio rocker.*

Appendices A.4 – Detailed drawing

Compiled below is a list of the complete assembly and Detail drawings formatted for A3.

- Complete assembly
- Complete assembly B
- Main plate assembly
- Cam post assembly
- Rocker assembly
- Main shaft assembly
- 4mm dowel
- 12mm dowel sleeve
- Adjusting bolt
- Cam eccentric
- Cam post main
- Cam post side
- End post
- End post opposite
- Main post
- Main shaft
- Mounting plate
- Rocker LH
- Rocker RH
- Rocker linkage arm
- Roller tip pin
- Roller tip
- Sliding bush



Item Number	Description	Quantity
1	Mounting plate	1
2*	12mm dowel sleeve	8
3	Main posts	4
4*	4mm dowel	10
5	Cam post assembled	3
6	End post	1
7	End post opposite	1
8	Shaft with cam assembly	1
9	Lh rocker assembled	4
10	Rocker linkage arm	8
11	Rh rocker assembled	4

MACHINED COMPO
1. UNLESS STATED ALL DIMEN
2. TOLERANCES UNLESS STATE
LINEAR DIMENSIONS 0 ±0.1
LINEAR DIMENSIONS 0.0 ±0.
LINEAR DIMENSIONS 0.00 ±0
ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1: 1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

Item Number	Desription	Quantity
1*	cam post main	1
2*	cam post side	1
3*	M3x12 Socket head cap screw	3

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

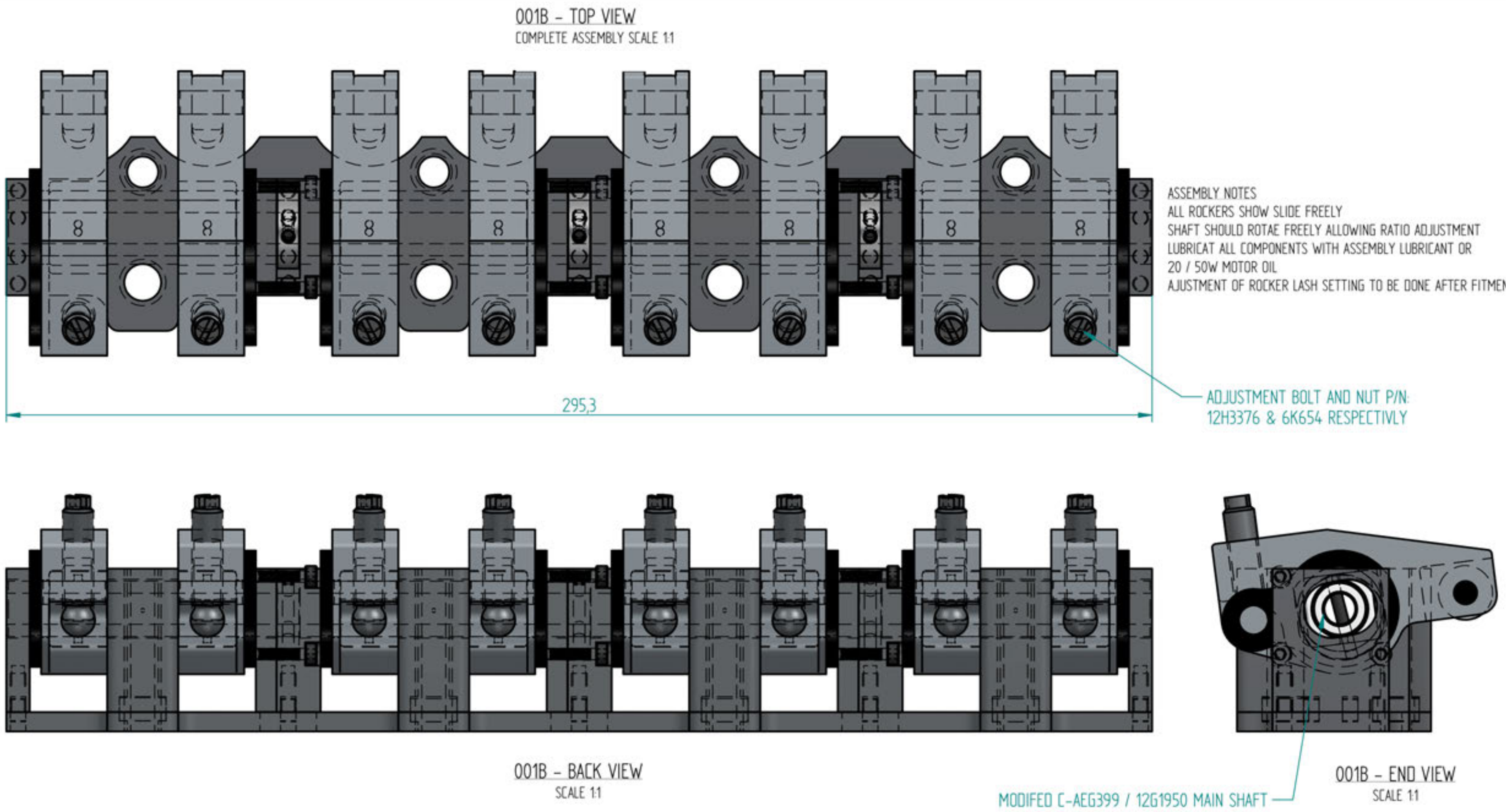
UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	001A	A

FILE NAME: Complete assembly.dft

SCALE: | WEIGHT: | SHEET 1 OF 1



ASSEMBLY NOTES
ALL ROCKERS SHOW SLIDE FREELY
SHAFT SHOULD ROTAE FREELY ALLOWING RATIO ADJUSTMENT
LUBRICAT ALL COMPONENTS WITH ASSEMBLY LUBRICANT OR
20 / 50W MOTOR OIL
AJUSTMENT OF ROCKER LASH SETTING TO BE DONE AFTER FITMENT

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	001B	A

FILE NAME: Complete Assembly-B.dft

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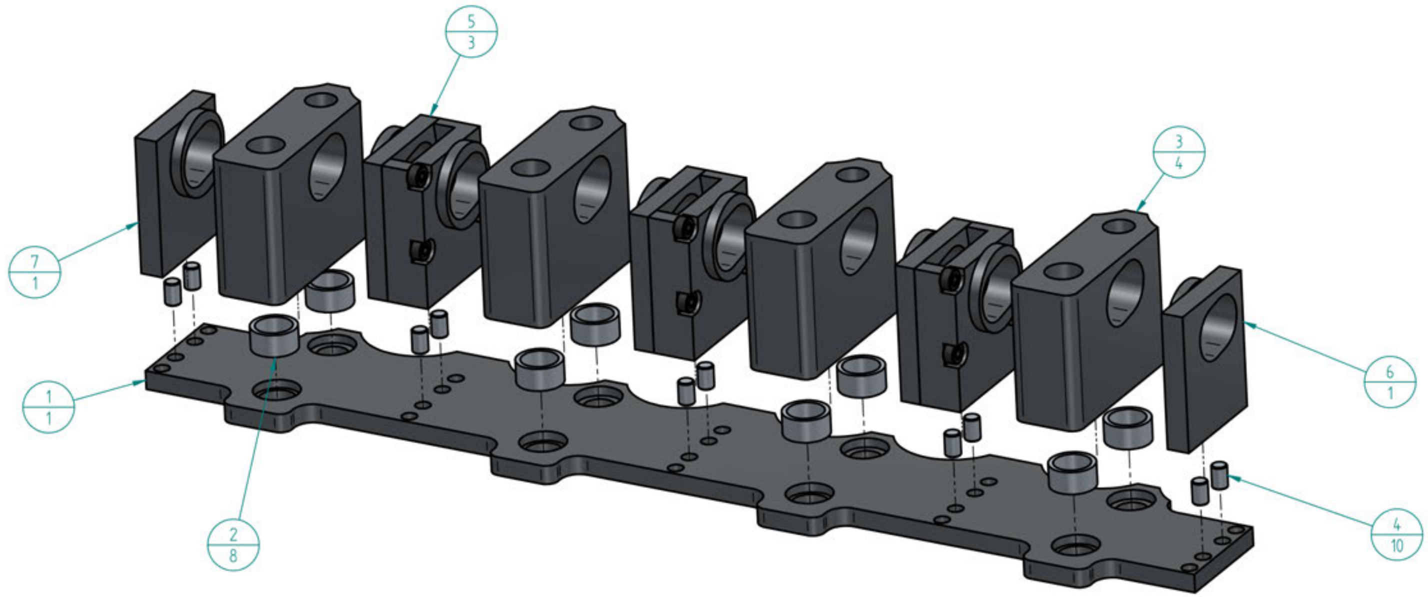


004 – EXPLODED VIEW
MAIN BASE ASSEMBLY SCALE 1:1

ASSEMBLY NOTES
DOWELS AND SLEEVES SHOULD BE A LIGHT TAP IN FIT
ALL FACES TO BE CLEANED AND FREE OF ANY GRIT BEFORE FITMENT

MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

Item Number	Description	Quantity
1	Mounting plate	1
2	12mm dowel sleeve	8
3	Main posts	4
4	4mm dowel	10
5	Cam post assembled	3
6	End post	1
7	End post opposite	1

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

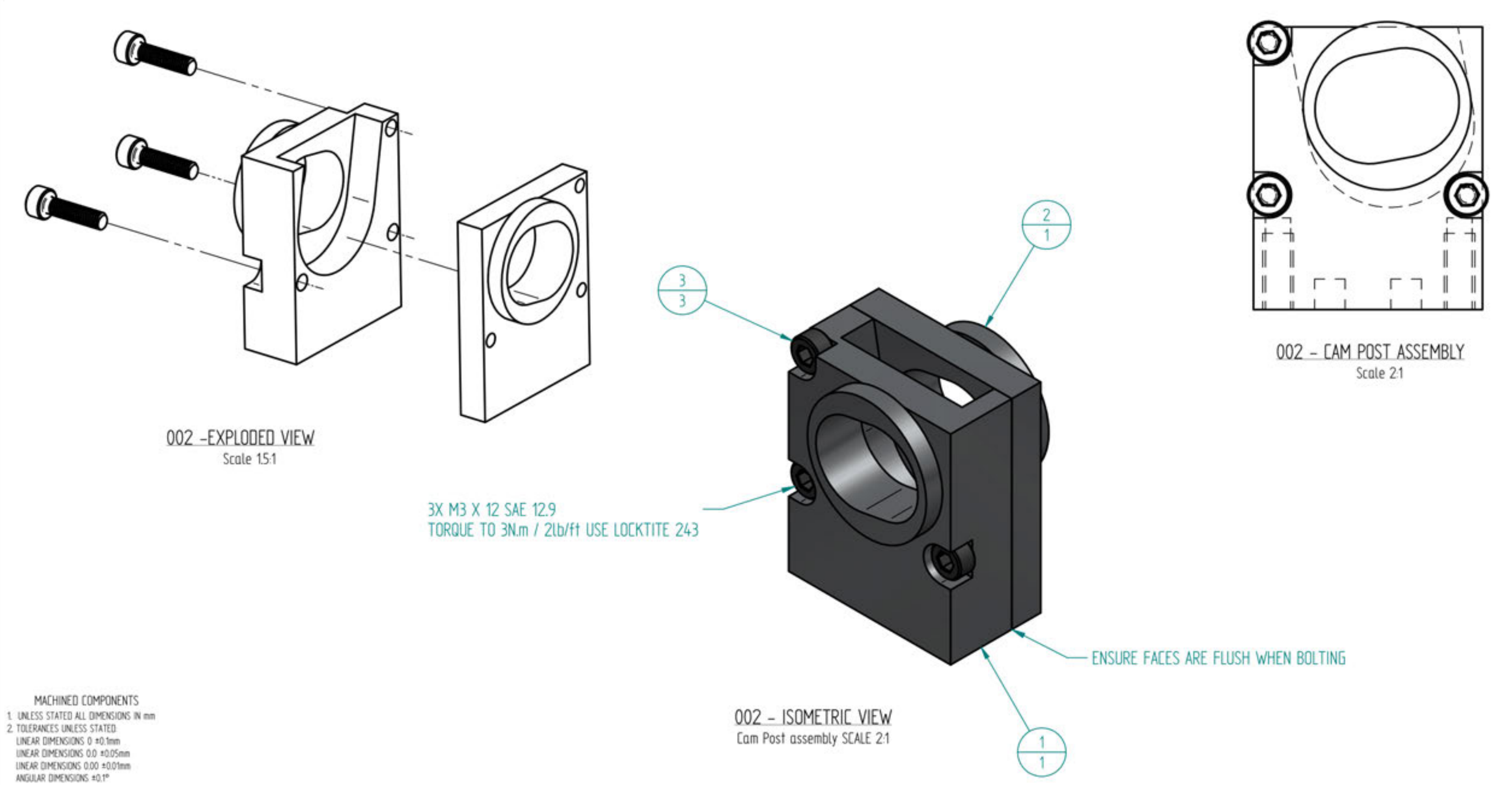
UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	004	A

FILE NAME: Main Base Assembly.dft

SCALE: | WEIGHT: | SHEET 1 OF 1



MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

Item Number	Desription	Quantity
1	cam post main	1
2	cam post side	1
3	M3x12 Socket head cap screw	3

	NAME	DATE
DRAWN	Benjamin S	13/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE: CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE: A3 | DWG NO: 002 | REV: A

FILE NAME: Cam Post Assembly.dft

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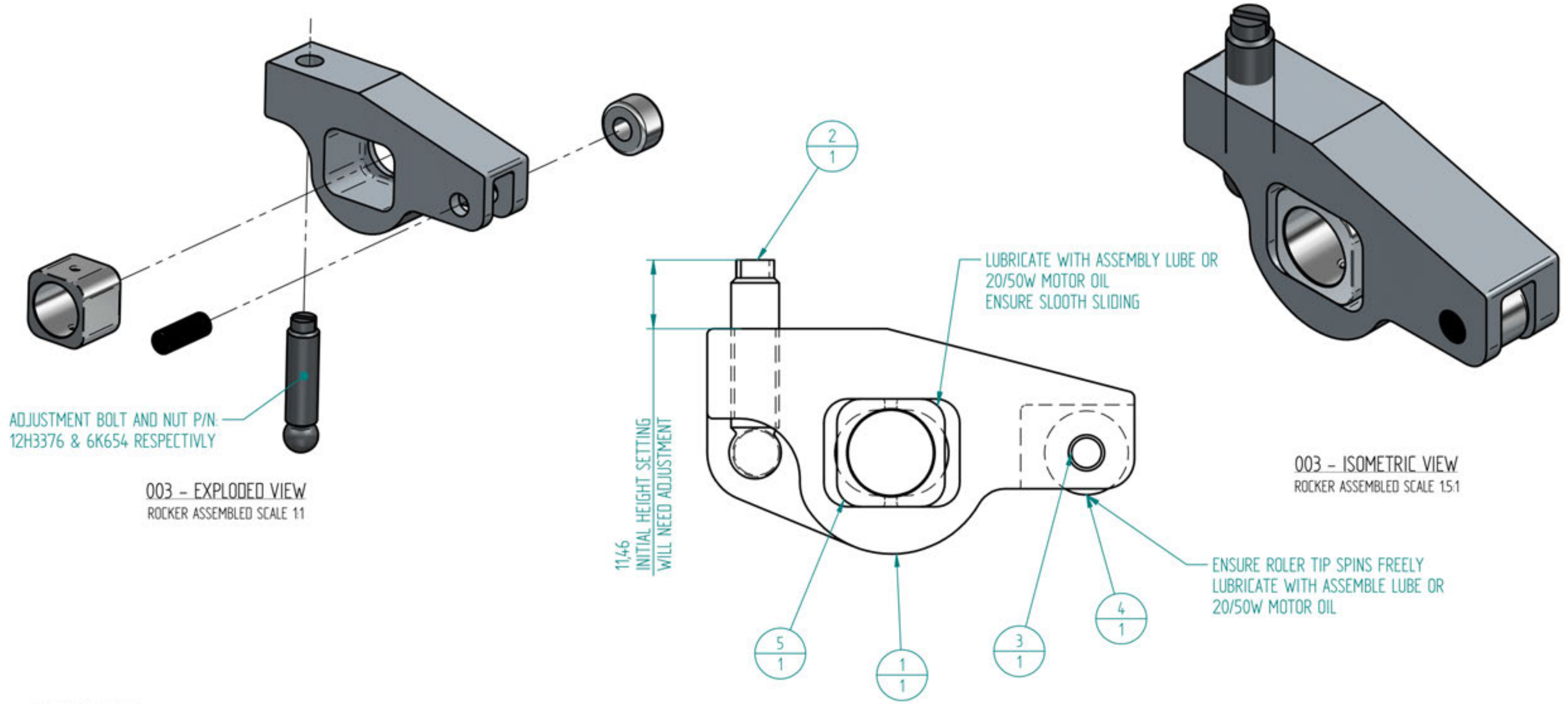

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED:
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 :1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

Item Number	Description	Quantity
1	Rocker LH / RH	1
2	Adjusting bolt p/n 12H3376	1
3	Roller tip pin	1
4	Roller tip	1
5	Sliding bush	1

NOTES
THIS ASSEMBLY DRAWING SHOWS LH ROCKER
ASSEMBLY OF THE RH ROCKER IS IDENTICLE

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE: CONTINUOUSLY VARIABLE RATIO ROCKER ARMS			
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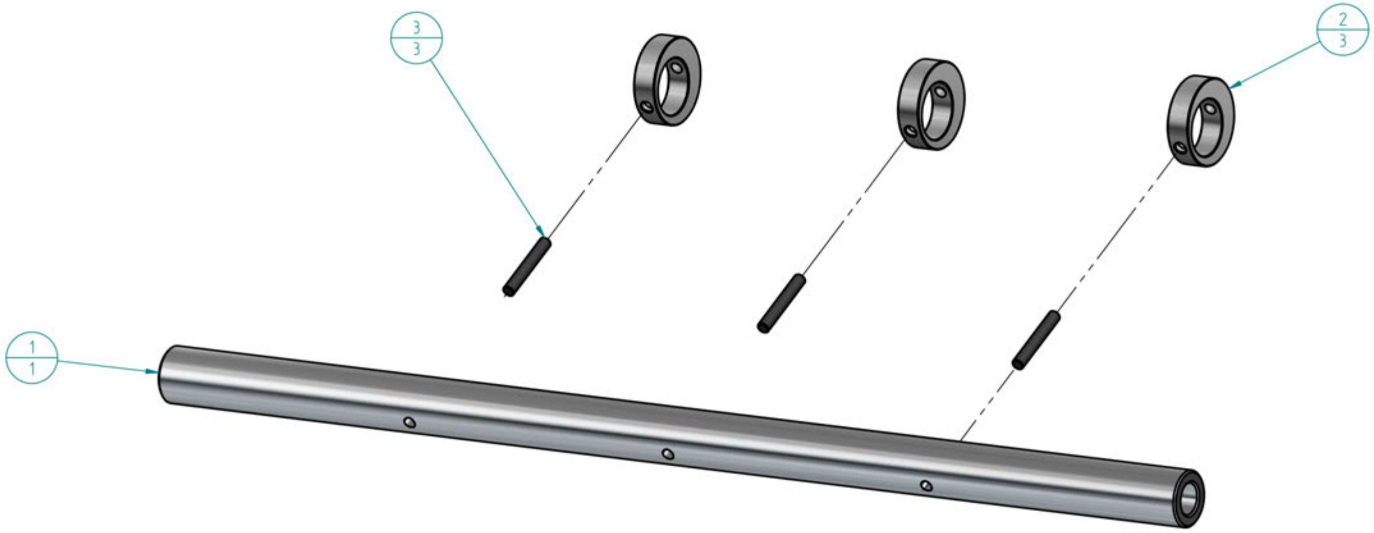


005 – EXPLODED VIEW

MAIN SHAFT ASSEMBLY SCALE 1:1

ASSEMBLY NOTES

CAMS NEED TO BE FITTED TO CAM POSTS BEFORE SLIDING SHAFT THROUGH

ALL FACES TO BE CLEANED AND FREE OF ANY GRIT BEFORE FITMENT

FOR MODIFICATION OF MAIN SHAFT REFER TO DRAWING 005

MACHINED COMPONENTS

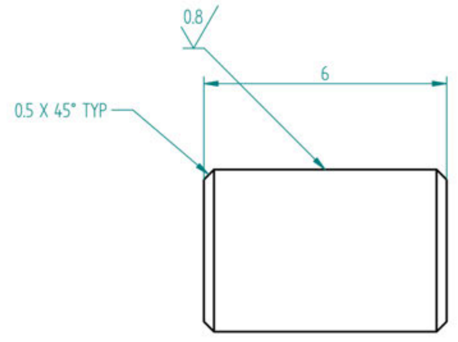
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED:
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 :1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED		
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

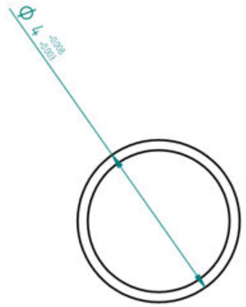
Item Number	Description	Quantity
1	Main shaft p/n 12A1950 or C-AEG399	1
2	Eccentic cam	3
3	3mm roll pin	3

	NAME	DATE	UNIVERSITY OF SOUTHERN QUEENSLAND
DRAWN	Benjamin S	15/10/20	
CHECKED			TITLE: CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
ENG APPR			
MGR APPR			
USQ			SIZE: A3 · DWG NO: 005 · REV: A
			FILE NAME: Main Shaft with cams Assembly.dft
			SCALE: · WEIGHT: · SHEET 1 OF 1

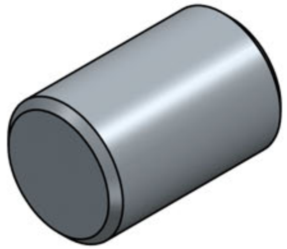
MACHINING NOTES
MATERIAL: EN36A HARDENED AND GROUND
HARDNESS RC-60
ALTERNATIVE STANDARE HARDENED ALLOY STEEL DOWEL



BSVR 001- SIDE VIEW
4MM DOWEL PIN SCALE 10:1



BSVR 001- END VIEW
4MM DOWEL PIN SCALE 10:1



BSVR 001- ISOMETRIC VIEW
4MM DOWEL PIN SCALE 10:1

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

	NAME	DATE
DRAWN	Benjamin S	13/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

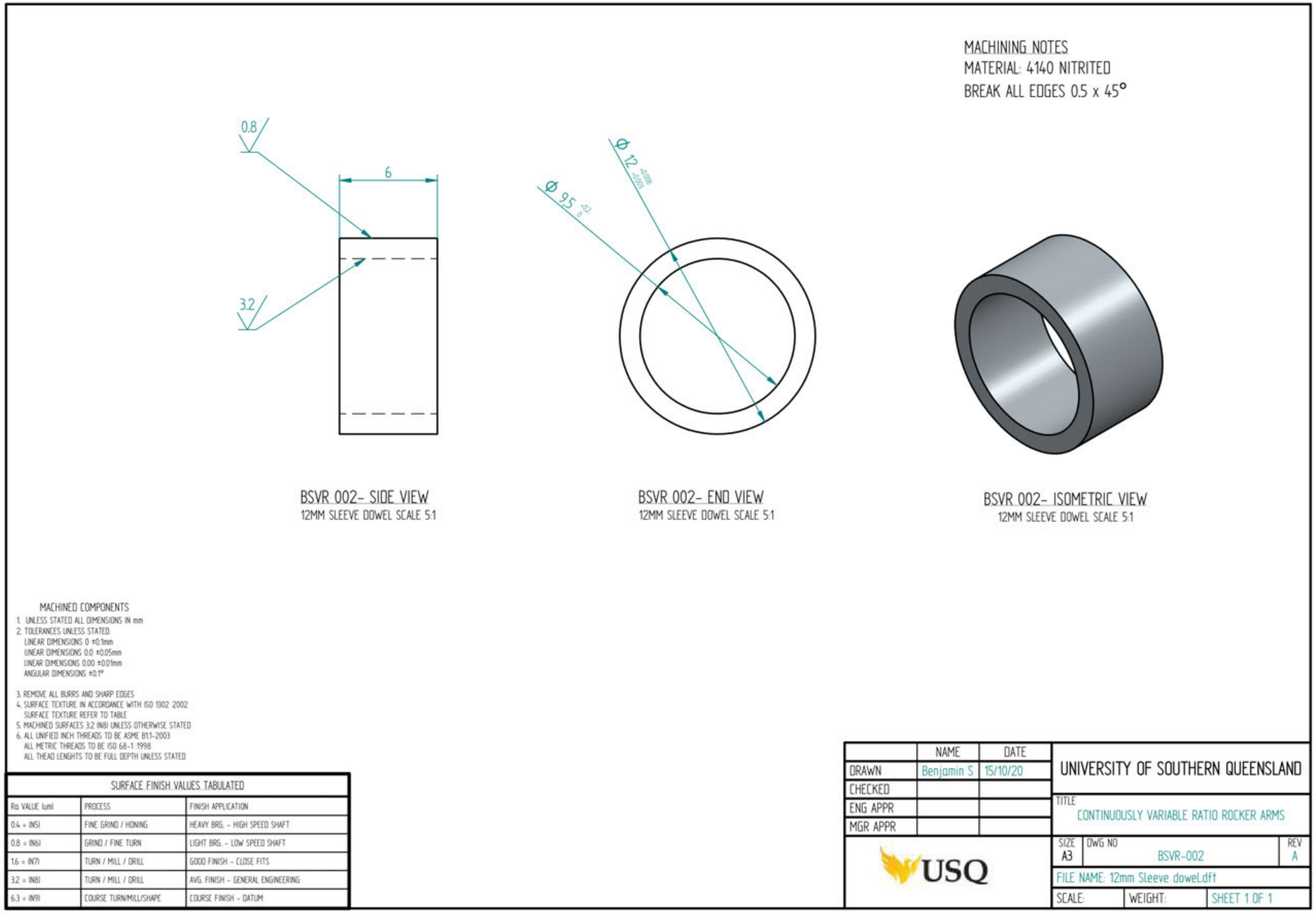
UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

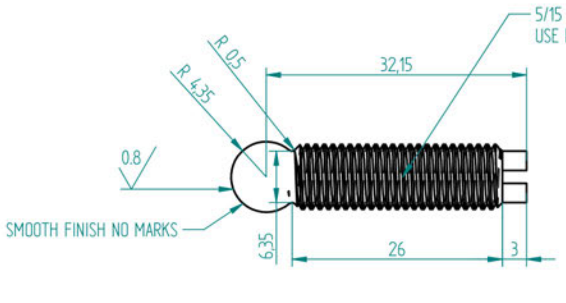
SIZE	DWG NO	REV
A3	BSVR-001	A

FILE NAME: 4mm Dowel Pin.dft

SCALE: | WEIGHT: | SHEET 1 OF 1

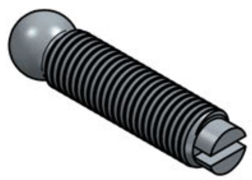

MACHINING NOTES
MATERIAL: 4140 NITRITED
BREAK ALL EDGES 0.5 x 45°
0.8
6
3.2
Ø 9.5
Ø 12
BSVR 002- SIDE VIEW
12MM SLEEVE DOWEL SCALE 5:1
BSVR 002- END VIEW
12MM SLEEVE DOWEL SCALE 5:1
BSVR 002- ISOMETRIC VIEW
12MM SLEEVE DOWEL SCALE 5:1
MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED:
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1 :1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED
SURFACE FINISH VALUES TABULATED
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
NAME
DATE
DRAWN
Benjamin S
15/10/20
CHECKED
ENG APPR
MGR APPR
USQ
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE
A3
DWG NO
BSVR-002
REV
A
FILE NAME: 12mm Sleeve dowel.dft
SCALE:
WEIGHT:
SHEET 1 OF 1

MACHINING NOTES
MATERIAL: 4140 NITRITED
OR USE P/N 12H3376 WITH 6K654 NUT



BSVR 003– SIDE VIEW
ADJUSTING BOLT SCALE 2:1

BSVR 003– END VIEW
ADJUSTING BOLT SCALE 2:1



BSVR 003– ISOMETRIC VIEW
ADJUSTING BOLT SCALE 2:1

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED:
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 :1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED		
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

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

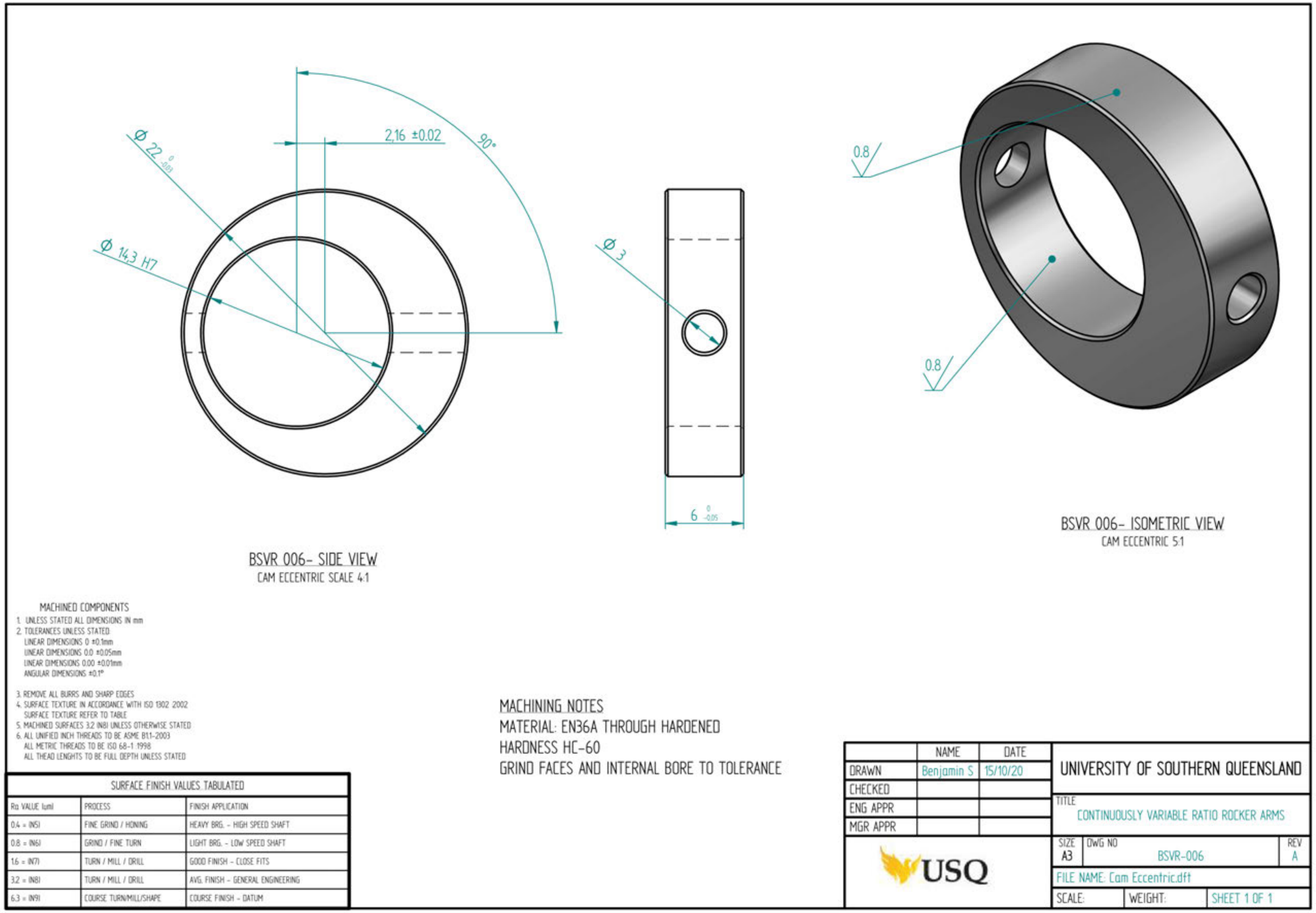
UNIVERSITY OF SOUTHERN QUEENSLAND

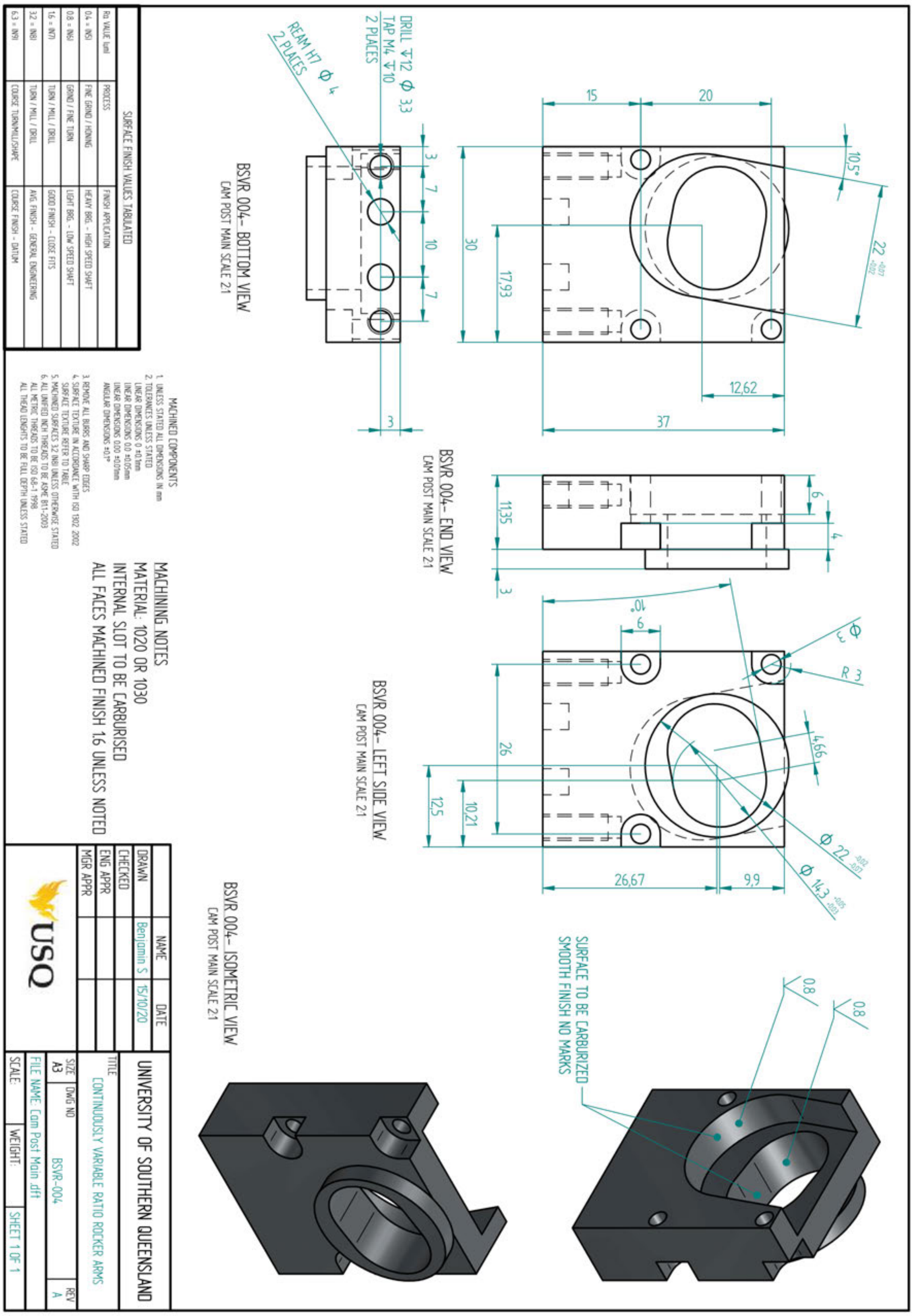
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

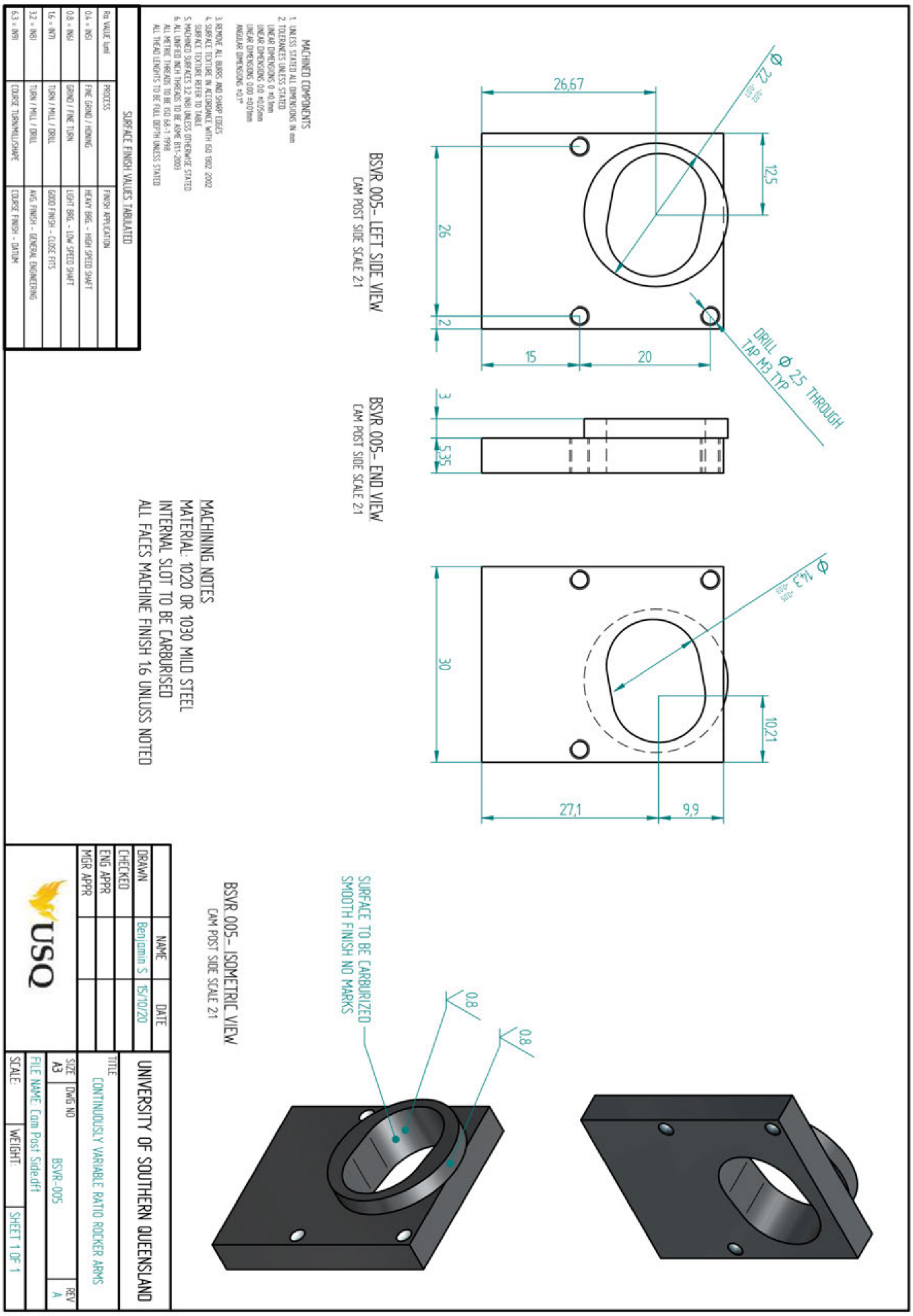
SIZE	DWG NO	REV
A3	BSVR-003	A

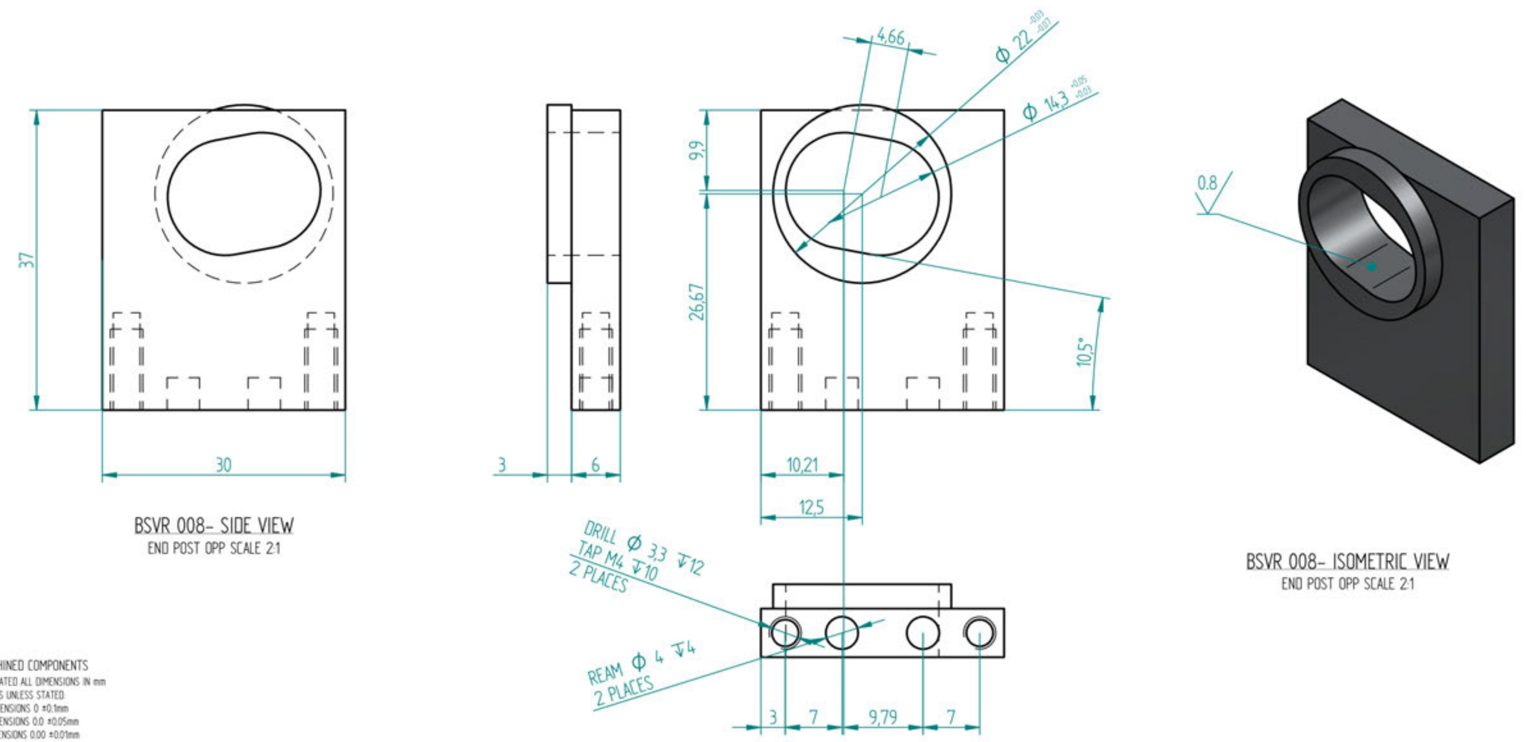
FILE NAME: Adjusting bolt.dft

SCALE:	WEIGHT:	SHEET 1 OF 1


Ø 22 0 -0.03
Ø 14,3 H7
2,16 ±0.02
90°
Ø 3
6 0 -0.05
0.8
0.8
BSVR 006- SIDE VIEW
CAM ECCENTRIC SCALE 4:1
BSVR 006- ISOMETRIC VIEW
CAM ECCENTRIC 5:1
MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1 1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED
SURFACE FINISH VALUES TABULATED
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
MACHINING NOTES
MATERIAL: EN36A THROUGH HARDENED
HARDNESS HC-60
GRIND FACES AND INTERNAL BORE TO TOLERANCE
NAME DATE
DRAWN Benjamin S 15/10/20
CHECKED
ENG APPR
MGR APPR
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE A3
DWG NO BSVR-006
REV A
FILE NAME: Cam Eccentric.dft
SCALE:
WEIGHT:
SHEET 1 OF 1


DRILL ⌴12 Φ 3,3
TAP M4 ⌴10
2 PLACES
REAM H7 Φ 4
2 PLACES
BSVR_004- BOTTOM VIEW
CAM POST MAIN SCALE 2:1
BSVR_004- END VIEW
CAM POST MAIN SCALE 2:1
BSVR_004- LEFT SIDE VIEW
CAM POST MAIN SCALE 2:1
BSVR_004- ISOMETRIC VIEW
CAM POST MAIN SCALE 2:1
SURFACE TO BE CARBURIZED
SMOOTH FINISH NO MARKS
MACHINING NOTES
MATERIAL: 1020 OR 1030
INTERNAL SLOT TO BE CARBURISED
ALL FACES MACHINED FINISH 1.6 UNLESS NOTED
MACHINED COMPONENTS
SURFACE FINISH VALUES TABULATED
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
NAME
DATE
DRAWN
Benjamin S
15/10/20
CHECKED
ENG APPR
MGR APPR
USQ
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE
A3
DWG NO
BSVR-004
REV
A
FILE NAME: Cam Post Main.dft
SCALE:
WEIGHT:
SHEET 1 OF 1


MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 2002
SURFACE TEXTURE REFER TO TABLE
SURFACE FINISH VALUES TABULATED
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
BSVR_005_ LEFT SIDE VIEW
CAM POST SIDE SCALE 2:1
26,67
12,5
26
2
15
20
DRILL Ø 2,5 THROUGH
TAP M3 TYP
BSVR_005_ END VIEW
CAM POST SIDE SCALE 2:1
3
5,35
MACHINING NOTES
MATERIAL: 1020 OR 1030 MILD STEEL
INTERNAL SLOT TO BE CARBURISED
ALL FACES MACHINE FINISH 1.6 UNLUSS NOTED
30
Ø 14,3
10,21
27,1
9,9
BSVR_005_ ISOMETRIC_VIEW
CAM POST SIDE SCALE 2:1
SURFACE TO BE CARBURIZED
SMOOTH FINISH NO MARKS
0.8
0.8
NAME
DATE
DRAWN
Benjamin S
15/10/20
CHECKED
ENG APPR
MGR APPR
USQ
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE
A3
DWG NO
BSVR-005
REV
A
FILE NAME: Cam Post Side.dft
SCALE:
WEIGHT:
SHEET 1 OF 1



MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1: 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

MACHINING NOTES
MATERIAL: 1020 OR 1030 MILD STEEL
INTERNAL SLOT TO BE CARBURIZED
ALL FACE MACHINE FINISH 1.6 UNLESS NOTED

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

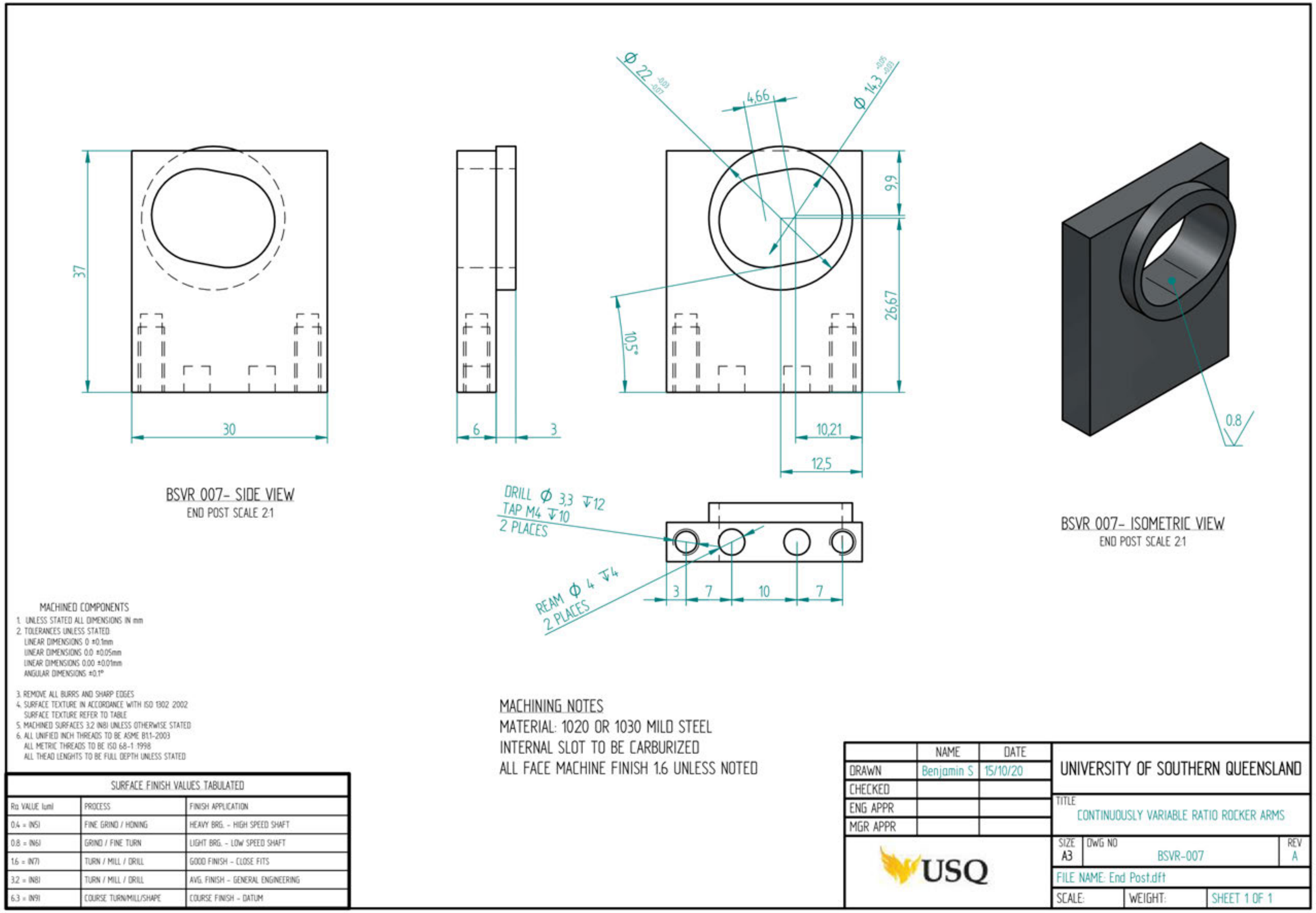
UNIVERSITY OF SOUTHERN QUEENSLAND

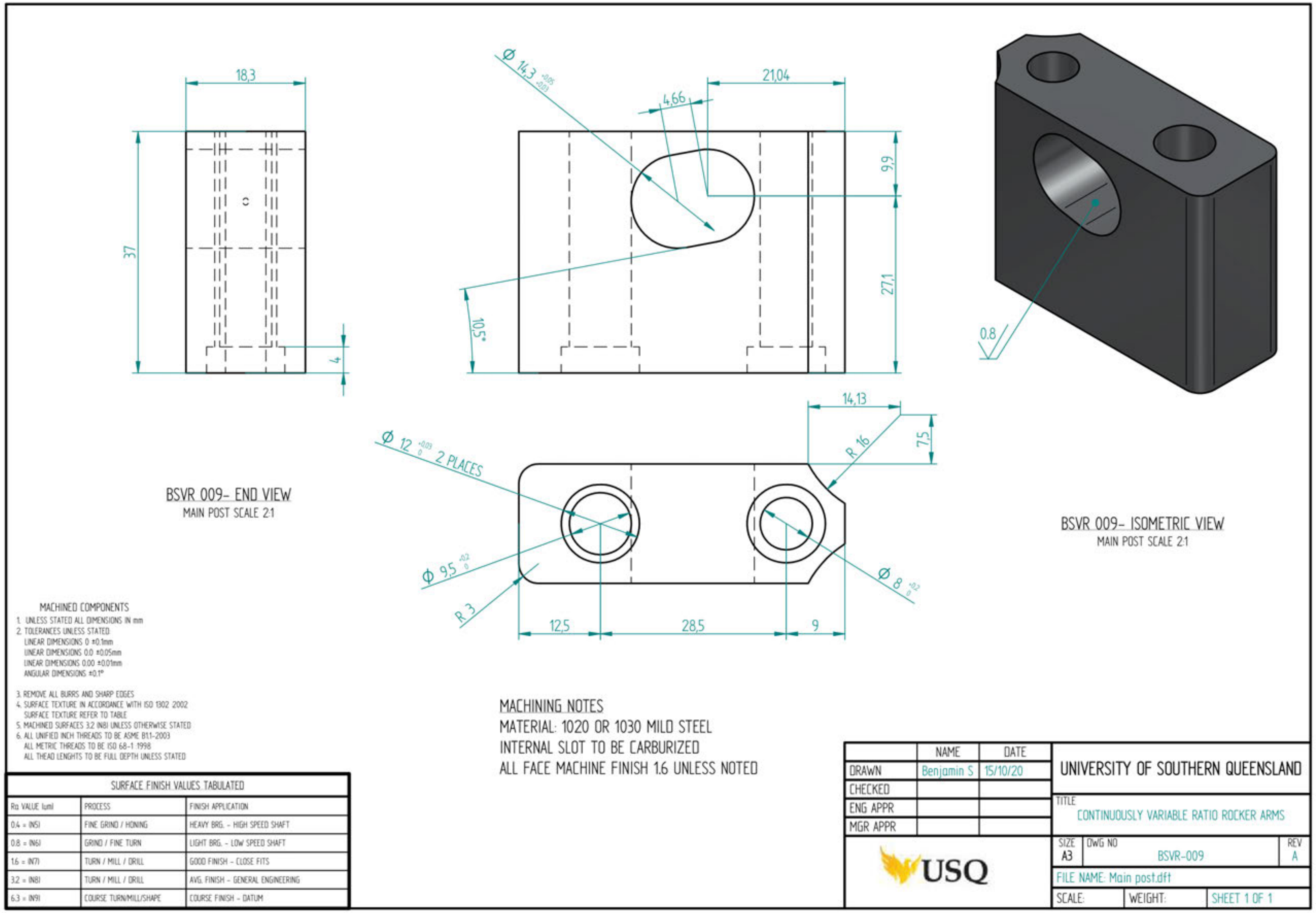
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

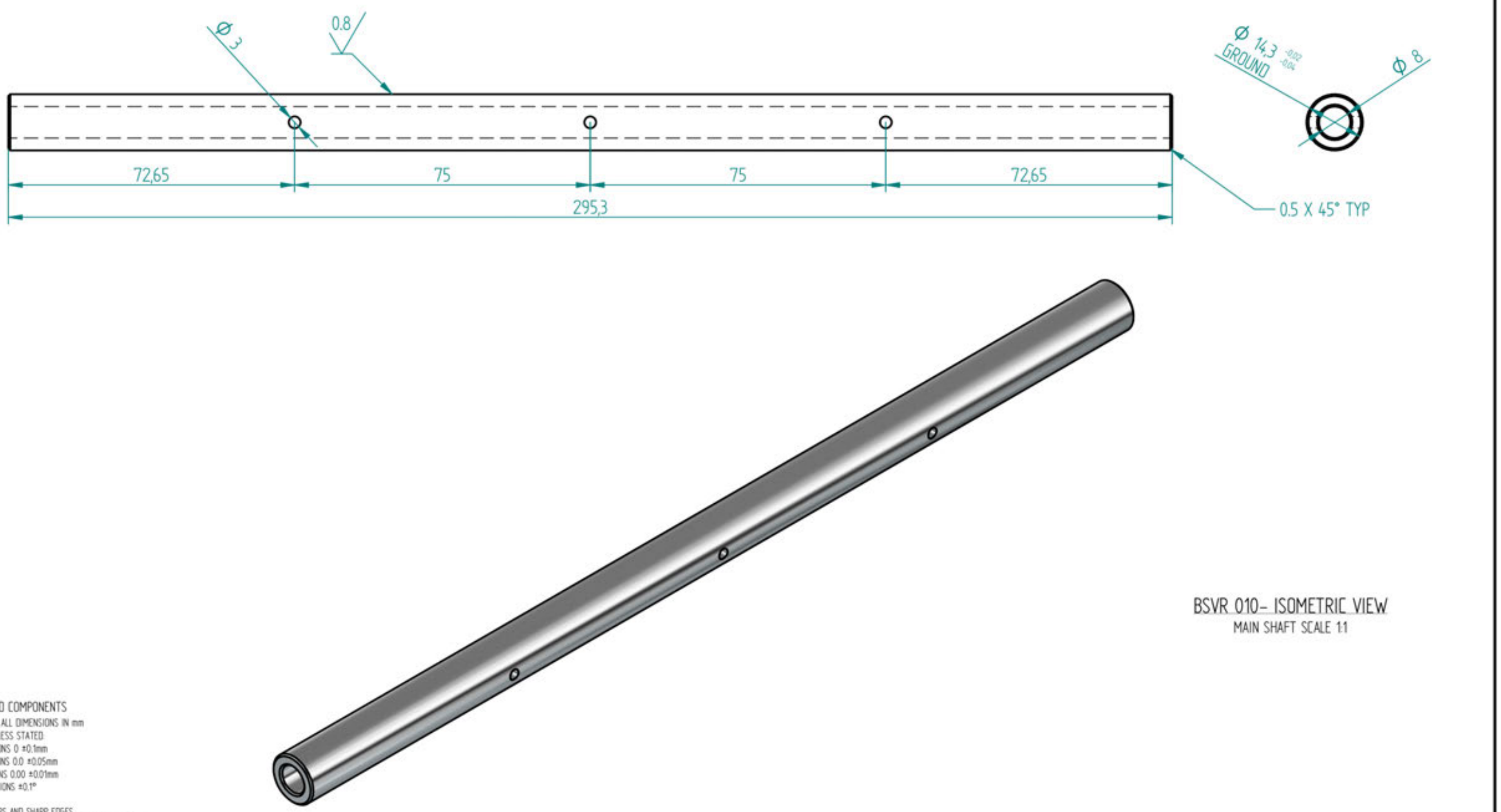
SIZE	DWG NO	REV
A3	BSVR-008	A

FILE NAME: End Post Opposite.dft

SCALE: | WEIGHT: | SHEET 1 OF 1


Ø 22 -0,03 -0,07
4,66
Ø 14,3 +0,05 +0,03
37
30
6
3
9,9
26,67
10,5°
10,21
12,5
0,8
BSVR 007- SIDE VIEW
END POST SCALE 2:1
DRILL Ø 3,3 ↧12
TAP M4 ↧10
2 PLACES
REAM Ø 4 ↧4
2 PLACES
3
7
10
7
BSVR 007- ISOMETRIC VIEW
END POST SCALE 2:1
MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED:
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1 :1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED
MACHINING NOTES
MATERIAL: 1020 OR 1030 MILD STEEL
INTERNAL SLOT TO BE CARBURIZED
ALL FACE MACHINE FINISH 1.6 UNLESS NOTED
SURFACE FINISH VALUES TABULATED
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
NAME | DATE
DRAWN | Benjamin S | 15/10/20
CHECKED
ENG APPR
MGR APPR
USQ
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE A3 | DWG NO BSVR-007 | REV A
FILE NAME: End Post.dft
SCALE: | WEIGHT: | SHEET 1 OF 1


BSVR 009- END VIEW
MAIN POST SCALE 2:1
18,3
37
4
Ø 14,3 +0,05 +0,03
4,66
21,04
9,9
27,1
10,5°
14,13
7,5
R 16
Ø 12 +0,03 0 2 PLACES
Ø 9,5 +0,2 0
Ø 8 +0,2 0
R 3
12,5
28,5
9
0,8
BSVR 009- ISOMETRIC VIEW
MAIN POST SCALE 2:1
MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1 :1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED
MACHINING NOTES
MATERIAL: 1020 OR 1030 MILD STEEL
INTERNAL SLOT TO BE CARBURIZED
ALL FACE MACHINE FINISH 1.6 UNLESS NOTED
SURFACE FINISH VALUES TABULATED
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
NAME
DATE
DRAWN
Benjamin S
15/10/20
CHECKED
ENG APPR
MGR APPR
USQ
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE
A3
DWG NO
BSVR-009
REV
A
FILE NAME: Main post.dft
SCALE:
WEIGHT:
SHEET 1 OF 1



BSVR 010– ISOMETRIC VIEW
MAIN SHAFT SCALE 1:1

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1: 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

MACHINING NOTES
MATERIAL: 4140 OR 4340 NITRIDED
P/N 12A-1950 OR C-AEG399 THEN DRILL HOLES AS PER DRAWING

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

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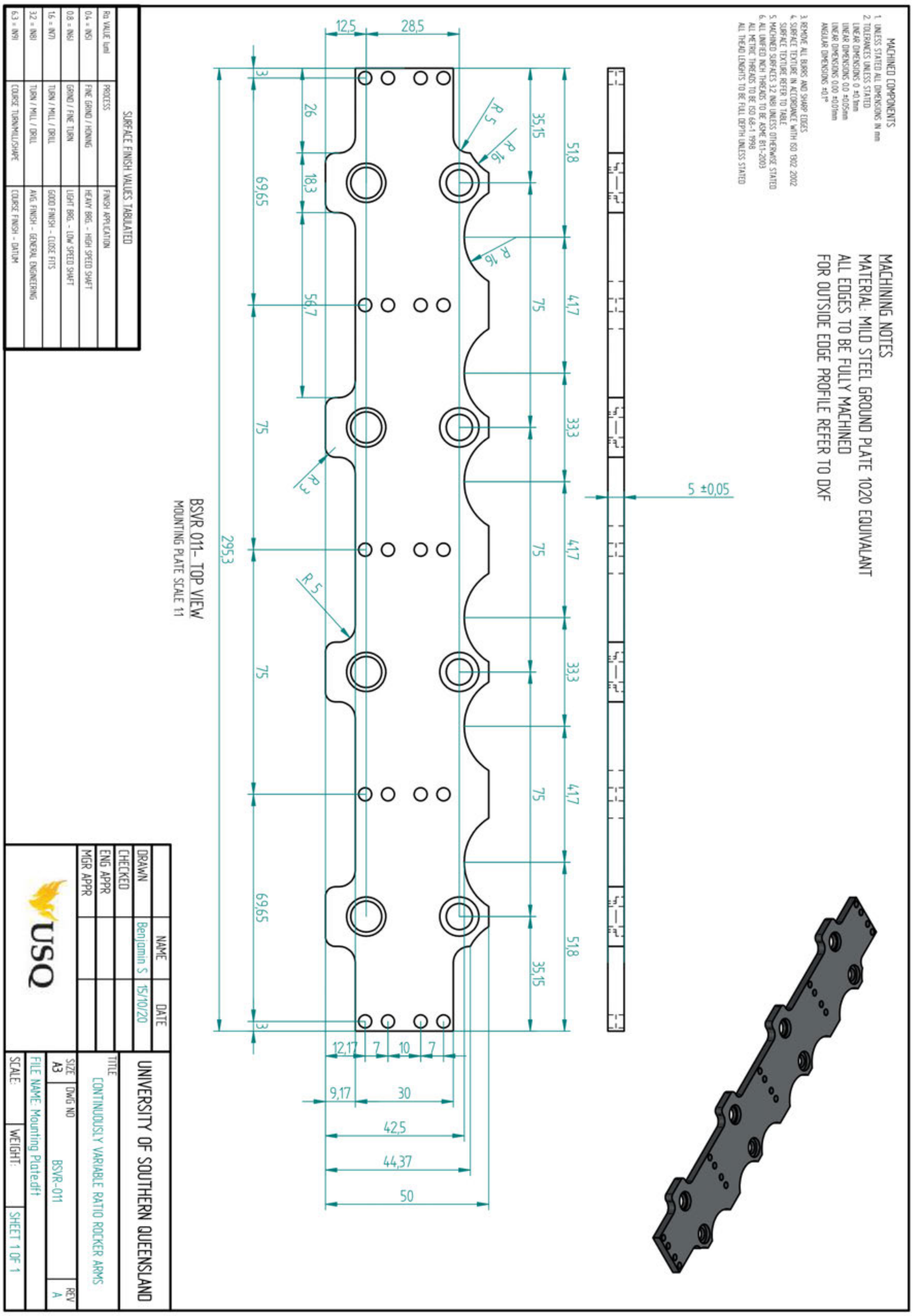
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

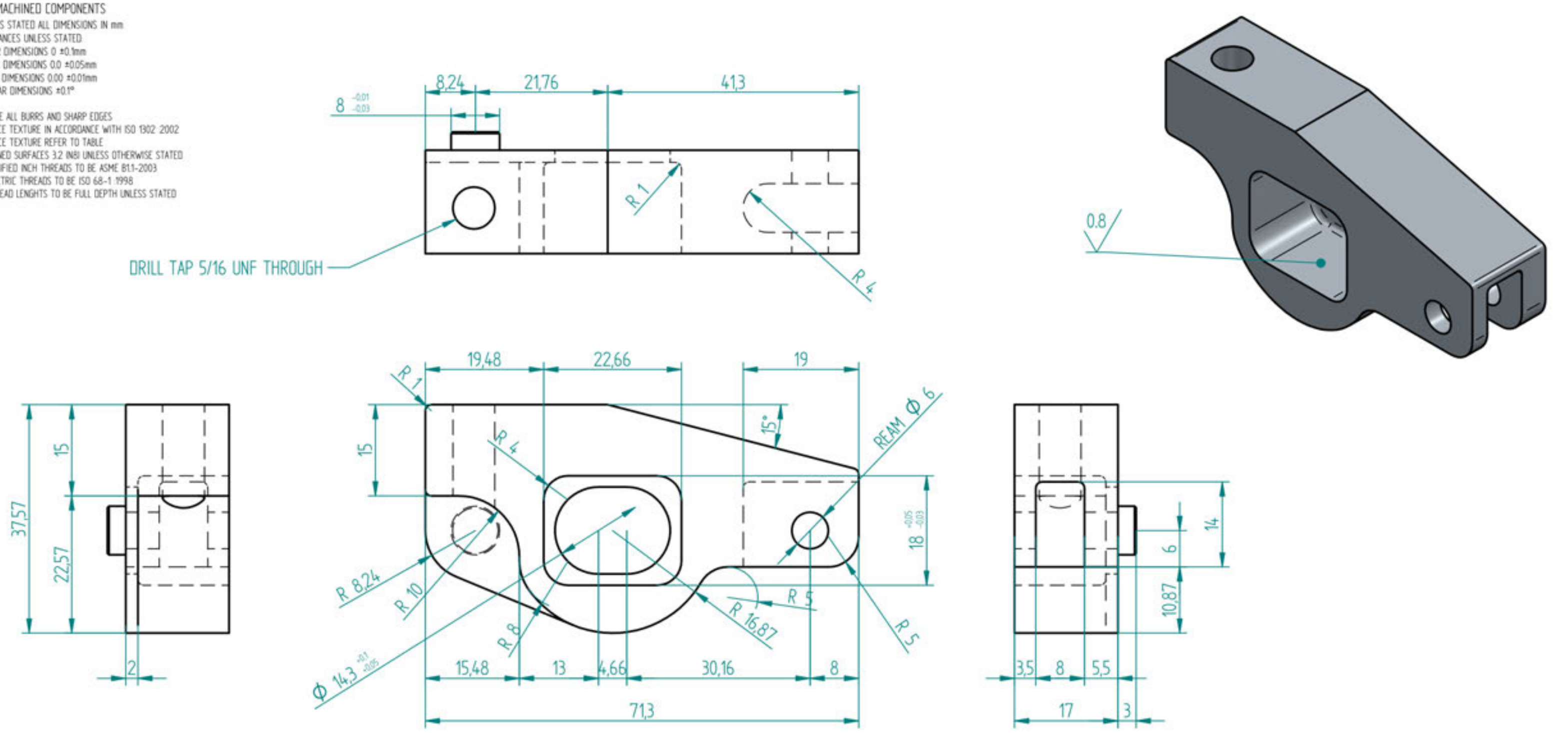
SIZE	DWG NO	REV
A3	BSVR-010	A

FILE NAME: Main Shaft.dft

SCALE: | WEIGHT: | SHEET 1 OF 1


MACHINING COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL METRIC THREADS TO BE ISO 68-1: 1998
ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL THEAD LENGTHS TO BE FULL DEPTH UNLESS STATED
MACHINING NOTES
MATERIAL: MILD STEEL GROUND PLATE 1020 EQUIVALANT
ALL EDGES TO BE FULLY MACHINED
FOR OUTSIDE EDGE PROFILE REFER TO DXF
5 ±0,05
BSVR_011-_TOP VIEW
MOUNTING PLATE SCALE 1:1
SURFACE FINISH VALUES TABULATED
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
NAME | DATE
DRAWN | Benjamin S | 15/10/20
CHECKED
ENG APPR
MGR APPR
USQ
UNIVERSITY OF SOUTHERN QUEENSLAND
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS
SIZE A3 | DWG NO BSVR-011 | REV A
FILE NAME Mounting Plate.dft
SCALE: | WEIGHT: | SHEET 1 OF 1

MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1: 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED



BSVR 012- SIDE VIEW
ROCKER LH SCALE 1.5:1

MACHINING NOTES
MATERIAL:1040 OR 1045 MILD STEEL
ALL EDGES TO BE FULLY MACHINED

SURFACE FINISH VALUES TABULATED

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

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

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TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

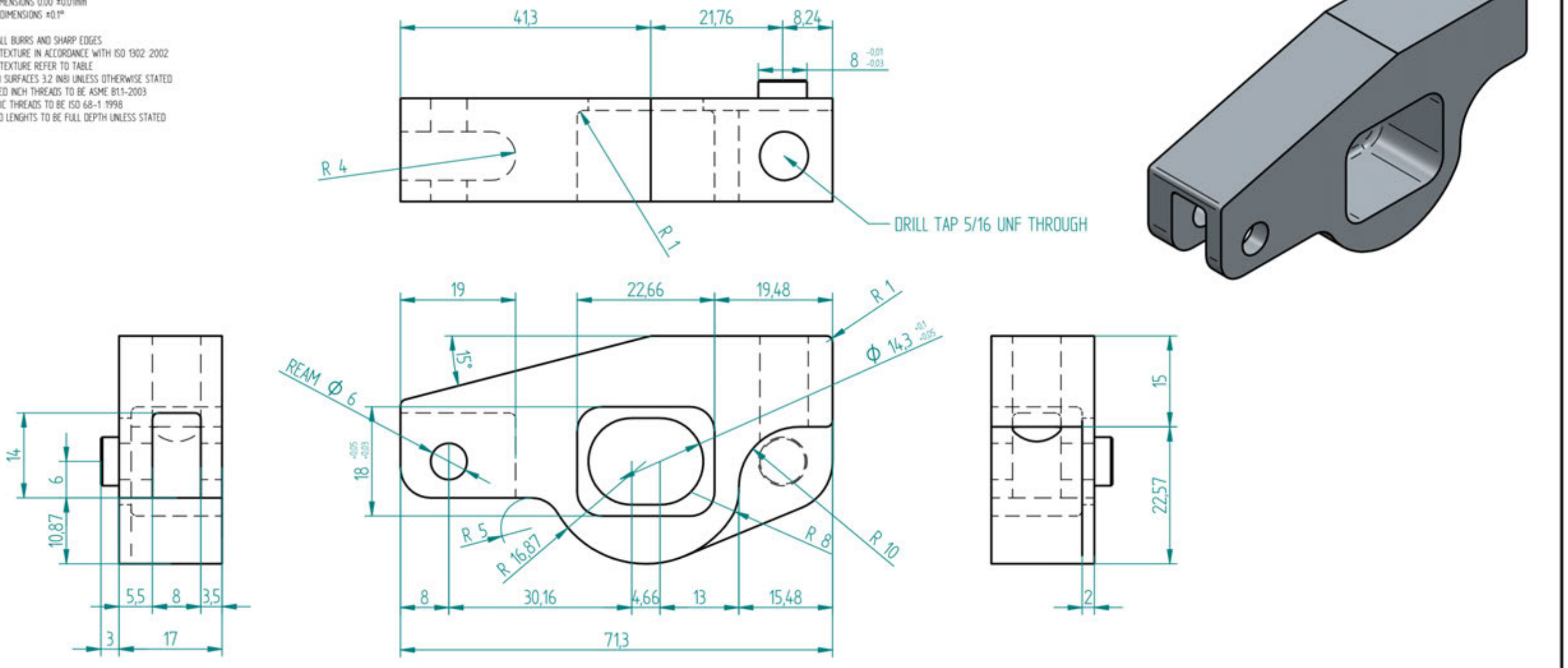
SIZE	DWG NO	REV
A3	BSVR-012	A

FILE NAME: Rocker LH.dft

SCALE: | WEIGHT: | SHEET 1 OF 1

MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED.
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1: 1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED



BSVR 013- SIDE VIEW
ROCKER RH SCALE 1.5:1

SURFACE FINISH VALUES TABULATED		
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

MACHINING NOTES
MATERIAL:1040 OR 1045 MILD STEEL
ALL EDGES TO BE FULLY MACHINED

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

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TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	BSVR-013	A

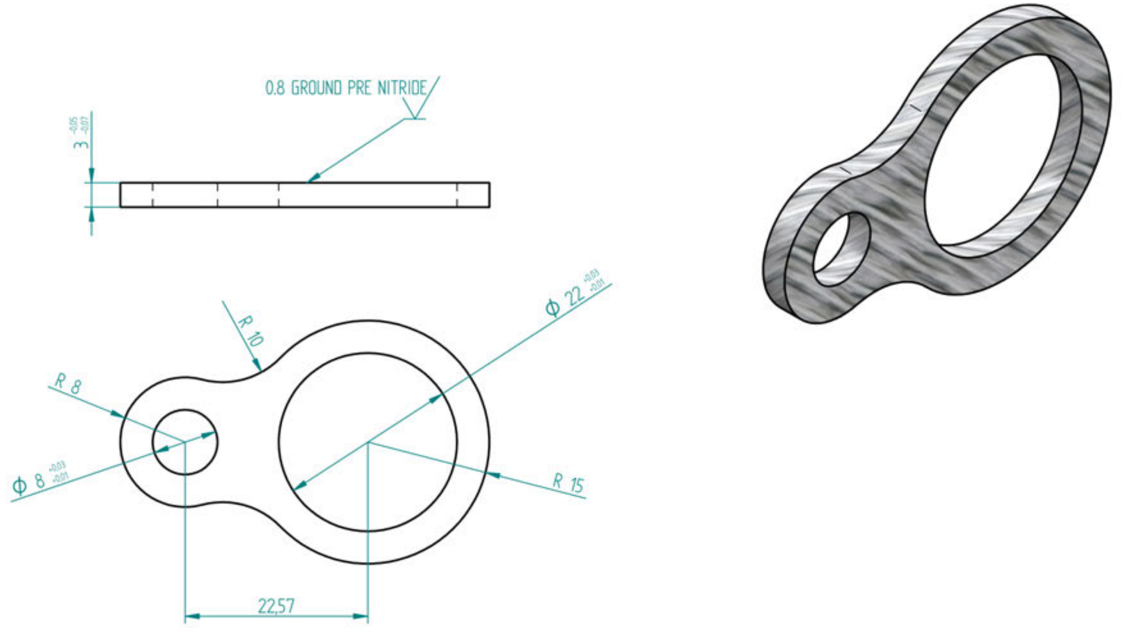
FILE NAME: Rocker RH.dft

SCALE:	WEIGHT:	SHEET 1 OF 1

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302: 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1: 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED



BSVR 014– SIDE VIEW
LINKAGE ARM SCALE 2:1

SURFACE FINISH VALUES TABULATED		
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

MACHINING NOTES
MATERIAL: 4140 OR 4340 NITRTIDED
ALL EDGES TO BE FULLY MACHINED UNLESS NOTED

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

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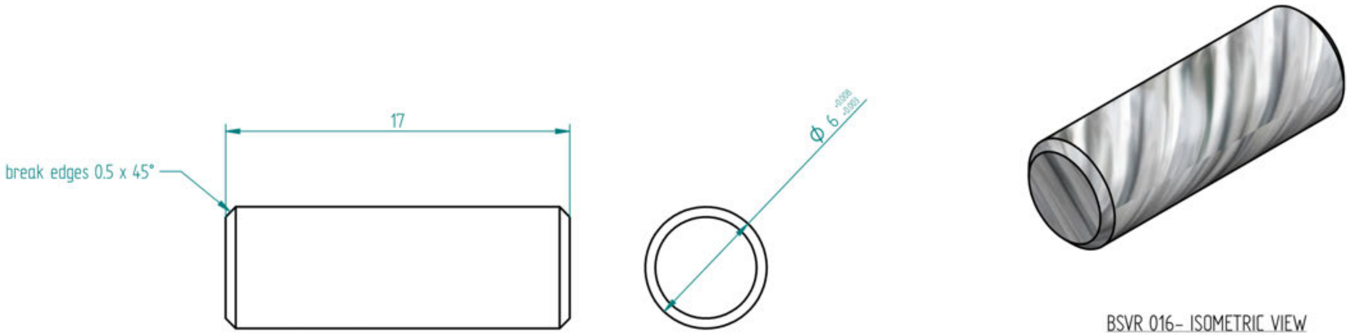
TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	BSVR-014	A

FILE NAME: Rocker Linkage Arm.dft

SCALE: | WEIGHT: | SHEET 1 OF 1



BSVR 016– SIDE VIEW
ROLLER TIP PIN SCALE 5:1

BSVR 016– ISOMETRIC VIEW
ROLLER TIP PIN SCALE 5:1

MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 :2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED		
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

MACHINING NOTES
MATERIAL: EN36A
HARDENED TO HC-60 THEN CENTRELESS GRIND OD
ALTERNATIVLY USE OF 6MM HARDENED ALLOY
STEEL DOWEL IS ACCPTABLE

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

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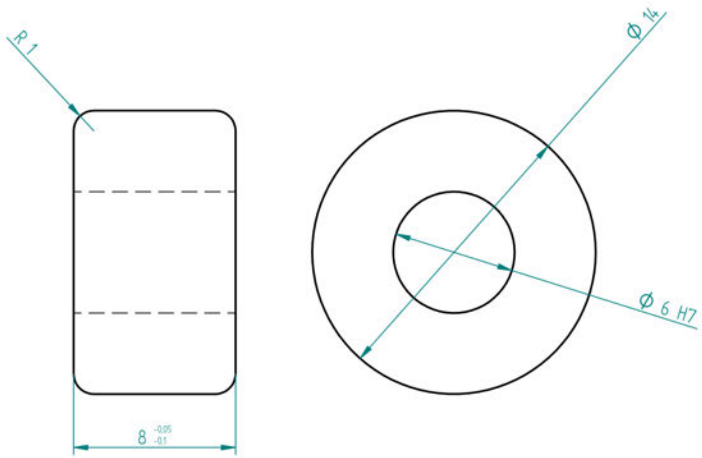
TITLE: CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	BSVR-016	A

FILE NAME: Roller Tip Pin.dft

SCALE: | WEIGHT: | SHEET 1 OF 1



BSVR 015– SIDE VIEW
ROLLER TIP SCALE 5:1

BSVR 015– ISOMETRIC VIEW
ROLLER TIP SCALE 5:1

MACHINED COMPONENTS
1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
LINEAR DIMENSIONS 0 ±0.1mm
LINEAR DIMENSIONS 0.0 ±0.05mm
LINEAR DIMENSIONS 0.00 ±0.01mm
ANGULAR DIMENSIONS ±0.1°

3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 2002
SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
ALL METRIC THREADS TO BE ISO 68-1 1998
ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

MACHINING NOTES
MATERIAL: EN36A
HARDENED TO HC-60 THEN FINISH GROUND
OUTSIDE DIMENSION TO BE GOVEREND BY VAVLE CLEARANCE

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

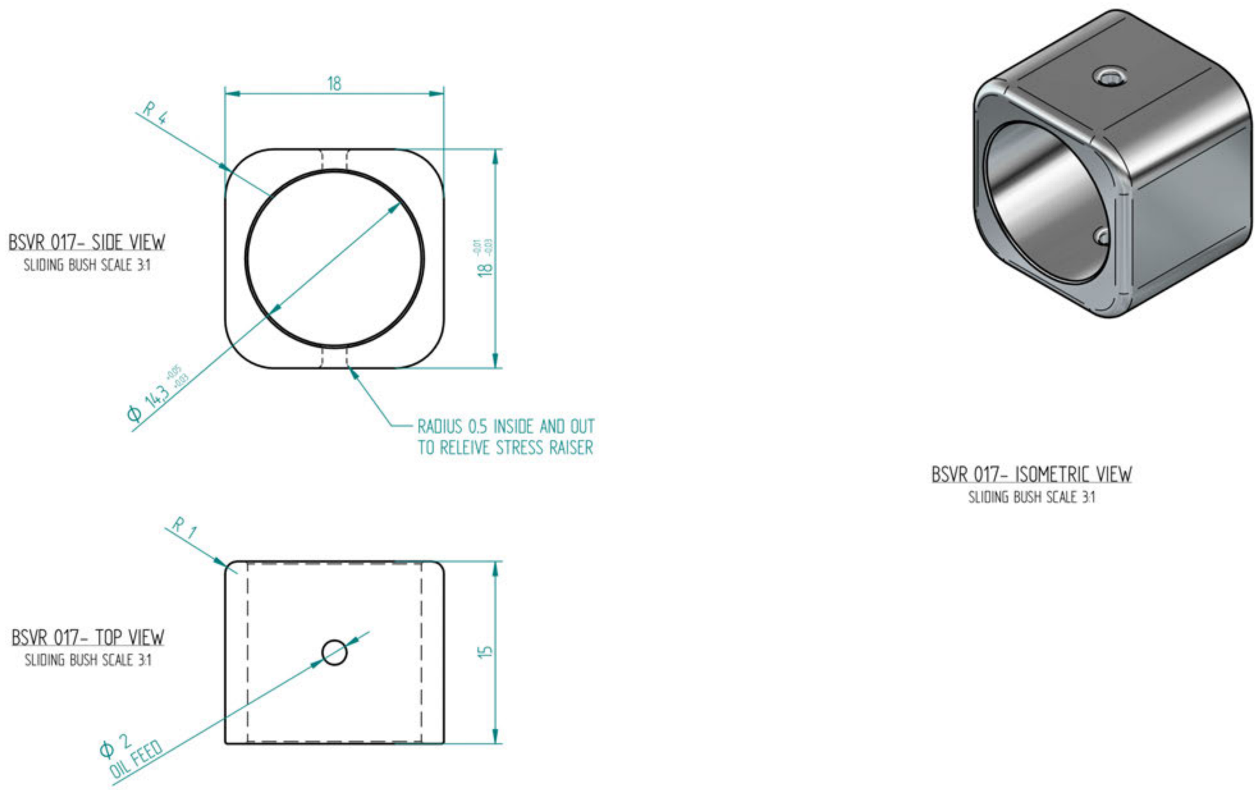
UNIVERSITY OF SOUTHERN QUEENSLAND

TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	BSVR-015	A

FILE NAME: Roller Tip.dft

SCALE: | WEIGHT: | SHEET 1 OF 1



MACHINED COMPONENTS

1. UNLESS STATED ALL DIMENSIONS IN mm
2. TOLERANCES UNLESS STATED
   LINEAR DIMENSIONS 0 ±0.1mm
   LINEAR DIMENSIONS 0.0 ±0.05mm
   LINEAR DIMENSIONS 0.00 ±0.01mm
   ANGULAR DIMENSIONS ±0.1°
3. REMOVE ALL BURRS AND SHARP EDGES
4. SURFACE TEXTURE IN ACCORDANCE WITH ISO 1302 2002
   SURFACE TEXTURE REFER TO TABLE
5. MACHINED SURFACES 3.2 (N8) UNLESS OTHERWISE STATED
6. ALL UNIFIED INCH THREADS TO BE ASME B1.1-2003
   ALL METRIC THREADS TO BE ISO 68-1 1998
   ALL THEAD LENGHTS TO BE FULL DEPTH UNLESS STATED

SURFACE FINISH VALUES TABULATED

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

MACHINING NOTES
MATERIAL: 4140 OR 4340 NITRIDED
MACHINE ALL FACES SURFACE FINISH 1.6 UNLESS NOTED

	NAME	DATE
DRAWN	Benjamin S	15/10/20
CHECKED		
ENG APPR		
MGR APPR		

USQ

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TITLE
CONTINUOUSLY VARIABLE RATIO ROCKER ARMS

SIZE	DWG NO	REV
A3	BSVR-016	A

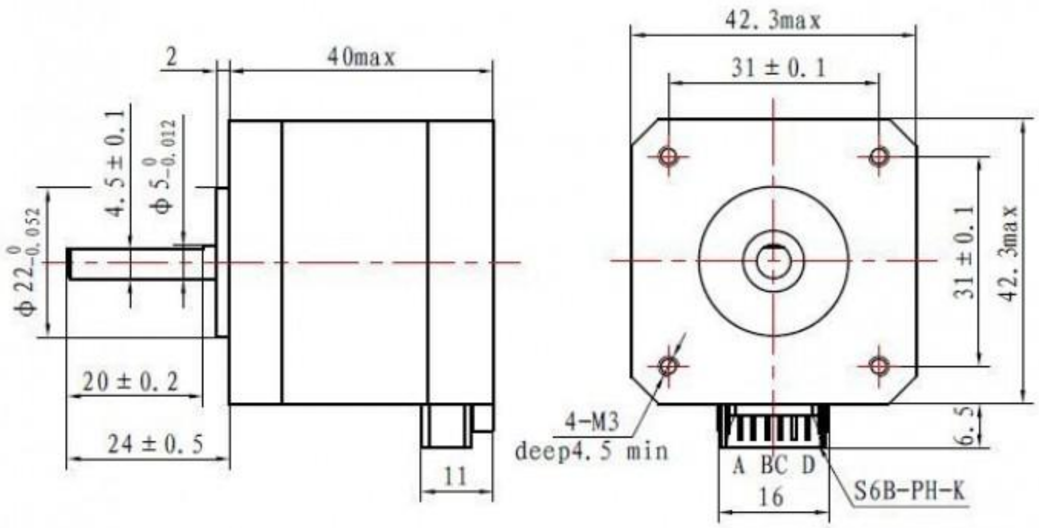
FILE NAME: Sliding Bush.dft

SCALE: | WEIGHT: | SHEET 1 OF 1

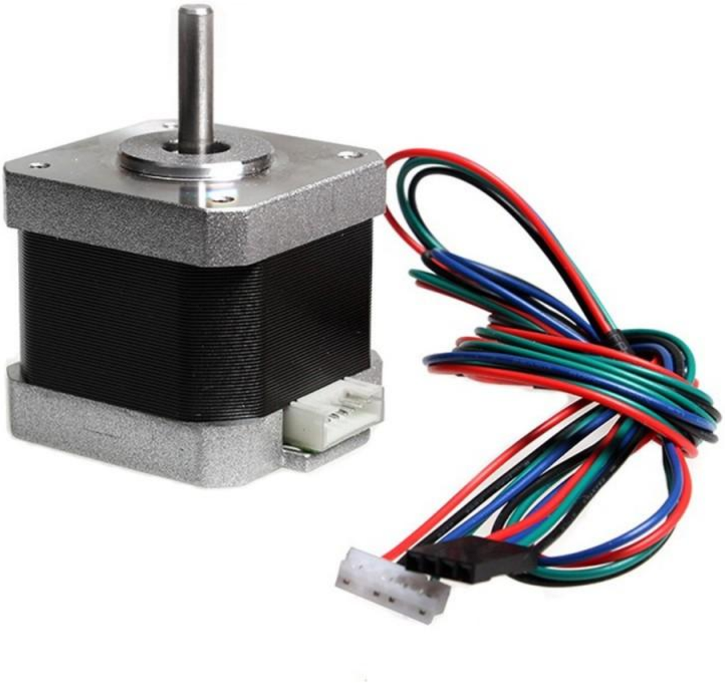
Appendices A.5 – Further work

HYBRID STEPPING MOTOR 42SHD0217-24B Casun®

General specification		Electrical specification	
Step angle	1.8°	Rated voltage	3.75V
Number of phase	2	Rated current	1.5A
Insulation resistane	100MΩmin. (500V DC)	Resistance per phase	2.5Ω ± 10%
Insulation class	Class B	Inductance per phase	5.0mH ± 20%
Rotor inertia	57g.cm^2	Holding torque	500mN.m
Mass	0.24kg	Detent torque	15mN.m



Appendix A.5.1: NEMA 17 data sheet (AUS 3d)



Appendix A.5.2: NEMA 17 stepper motor (AUS 3d)