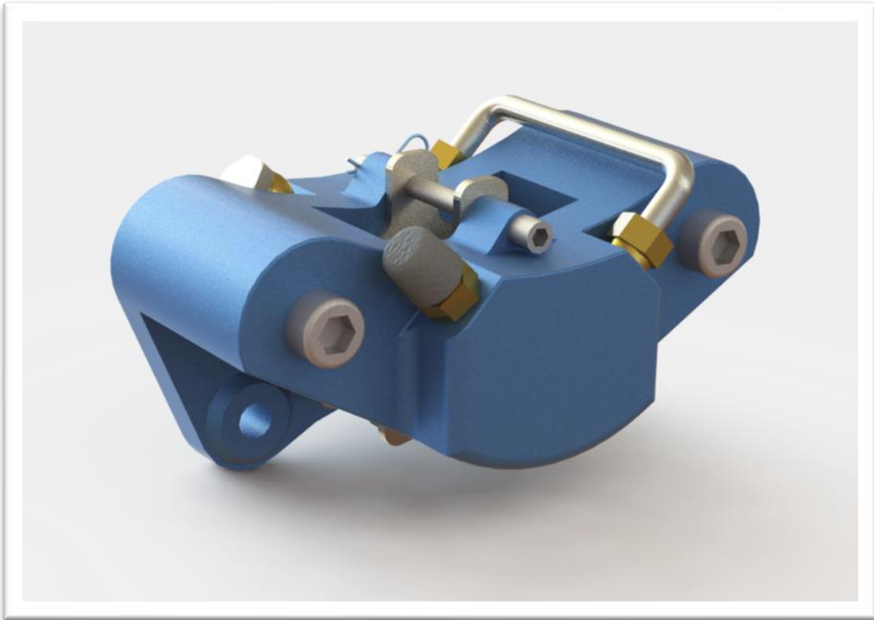
Design of a Compact Rear Brake Calliper for the QUTMS FSAE Electric Vehicle

Detailed Design Summary & Component Analysis

Group M5

EGH420 – Mechanical Systems Design

31st October, 2021



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1.1 Background & Scope

An automotive braking system has the fundamental function of allowing the driver to slow or completely stop the vehicle using a manual force input. The existing braking system on the QUT Motorsport (QUTMS) QEV3 electric race car consists of two hydraulic systems (front and rear) actuated by the driver via the brake pedal which includes a front/rear bias bar configuration. A brake fluid transmits pressure generated in the master cylinder at the pedal-end of brake lines to each wheel. Hydraulic disc-brake callipers on each wheel press brake pads against a steel brake rotor to generate friction, allowing kinetic energy from the motion of the car to be converted to heat. QUTMS has selected the Wilwood GP200 calliper as an off-the-shelf system for use on the two front wheels of the car; however, space constraints at the rear wheels mean that off-the-shelf callipers would not be suitable and would foul on the wheel. QUTMS subsequently engaged the engineering team with the task of designing a custom compact brake calliper suited for use on the QEV3 car. The aim of this project was to develop, analyse and communicate the design of a brake calliper that met the specific needs of the vehicle and interfaced with the existing braking system while complying with FSAE rules. The assumed use case for the calliper was assumed to be normal (up to maximum load) brake application in the forward direction in standard FSAE events.

The scope of this design was limited to development of the brake calliper as a sub-system of the existing mechanical system. As such, the calliper was required to mount directly to the upright assembly and interface with hydraulic lines feeding pressure to each wheel. The battery limits were brake rotors, hydraulic line and upright mounting point on the car; no components outside of this were scoped for modification. QUTMS specific requirements for the design were that it considers reasonable cost yet provide more than adequate stopping capability, have a focus on lightweight and be manufacturable using practical, modern methods. As specified by FSAE rules, the brake system must be capable of locking-up all four wheels during a stopping test at speed. This requirement was to form a key parameter in the design and analysis. Additionally, one design was required to be useable on both left- and right-hand sides of the car providing the same performance in either configuration. Maintainability was also important in the interest of rapid repair with minimal disassembly. An effective design was to make use of off-the-shelf parts where appropriate in the wider interest of practicality and cost efficiency for the team.

1.2 Design Summary

The final design was arrived at following selection from four conceptual designs and further refinement following systems level design completion. Consisting of two opposed pistons of 24 mm outer diameter, the conventional-calliper-inspired design makes use of two individual calliper halves which are held together with two M8 bolts in the configuration depicted in figure 1. The fixed calliper design was chosen over floating calliper designs based on reports of greater stiffness and even pad wear reported in the literature [1]. Each calliper half is to be machined from billet 7075 T6 aluminium, chosen for its near-optimal strength-to-weight and heat conductivity characteristics. The piston diameter was chosen with consideration of fundamental calliper design methodology developed by Seward [2] and manufacturer specifications for the chosen commercially available brake pads, the AP Racing CP4226D27-RX sintered disk brake pad. These pads were selected for their compact form, high-heat capability and history of use for FSAE car applications [3]. Opposed piston designs are proven to offer consistent, high accuracy brake performance due to symmetrical pad actuation and contact pressure balance, while offering small form factor capability, satisfying the performance-orientated needs of QEV3 [4]. The product of these design factors is a light-weight calliper at just 560 g inclusive of the brake pads and all associated fasteners and hydraulic hardware.

The calliper mounts directly to the QEV3 wheel upright via the inboard side bracket using two M8 mounting fasteners as indicated in figure 1. Detail images showing the calliper mounted to the upright can be found in Appendix A. Adequate clearance of at least 5 mm was maintained between the calliper assembly and all surrounding components, considering only the dimensions of the current wheel off-set used on the car. This compactness was achieved through careful consideration of the height at which calliper components sit above the external diameter of the brake rotor. The limiting factor was the structural connection between both halves so all other components, including the pistons and hydraulic fittings, were positioned below the effective outside diameter of the calliper body by using a “concave-top” shape. In comparison to callipers like the GP200, this form allowed pad retention hardware, hydraulic fittings, and fluid passage to be positioned closer to the rotor, making optimal use of space to meet the primary design requirements.

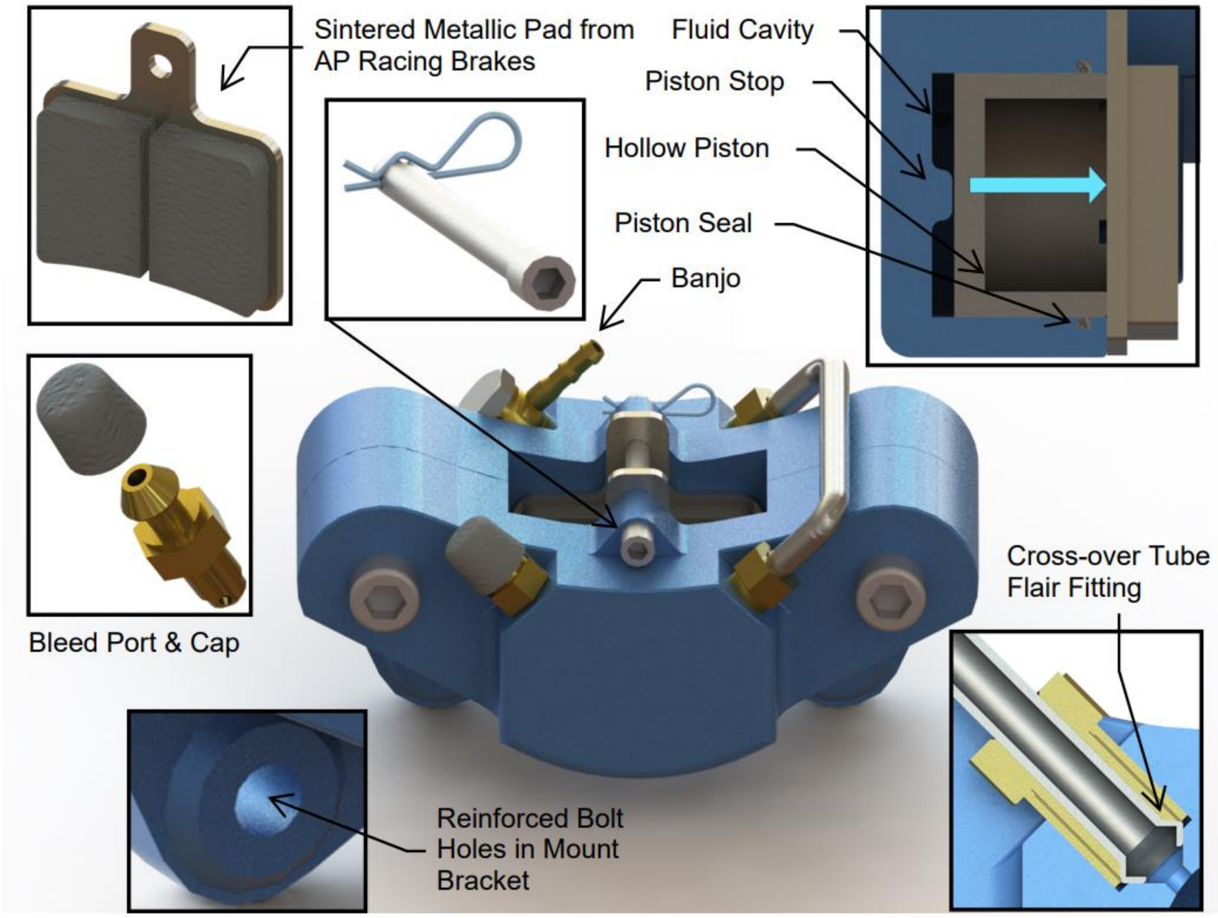


Figure 1 - Brake calliper design detail schematic.

Fluid management design required a Banjo fitting to attach the brake line to the calliper per standard existing fittings on the car. The 8 mm Banjo was positioned on the inboard side of the calliper while a fluid bleed port was mounted to the outboard side to facilitate the least fluid-path obstruction and best use of space. Following an FMEA action, a dust-cap was added to the bleed port to avoid blockages caused by debris ingress. A brake fluid cross-over tube was selected over internally routed fluid passages as this reduced the possibility of loss of fluid containment at the interface between the calliper halves at a slight cost of calliper-wheel clearance. The tube assembly makes use of standardised “bubble flair” fittings and 3/16-inch steel tubing historically proven for use in automotive brakes [5]. Additional benefits include increased serviceability associated with readily available components and reduction in manufacturing costs which improve the value of this design to QUTMS overall vehicle performance.

In operation, fluid pressure transmitted to the rear of the pistons via the fluid channels exerts a perpendicular thrust force on each brake pad, by first displacing the pistons. Shown in Figure 2, an elastic piston seal prevents fluid from escaping while also deforming to provide restoring elastic force to avoid brake drag. Stops at the rear of the piston bore ensure the piston is not capable of blocking the fluid passages which could render the brakes inoperable. A H7/g6 clearance fit between the piston and bore was specified in accordance with ISO 286-1 to allow accurate location and sliding fit for precise operation while allowing for fluid film lubrication around the wet area of the piston bore. Additional failure-addressing features include the threaded pad retaining pin which incorporates an R-clip that ensures the brake pads cannot become displaced during operation while still allowing fast removal/installation capability. This is made possible by the open-top pad cavity which does not require removal of the calliper when replacing brake pads, improving overall maintainability. Adding to maintainability is the use of metric threaded fastening components and standard size hex-head tool compatibility. Low component count and ease of maintenance lends the design to a student-run team.

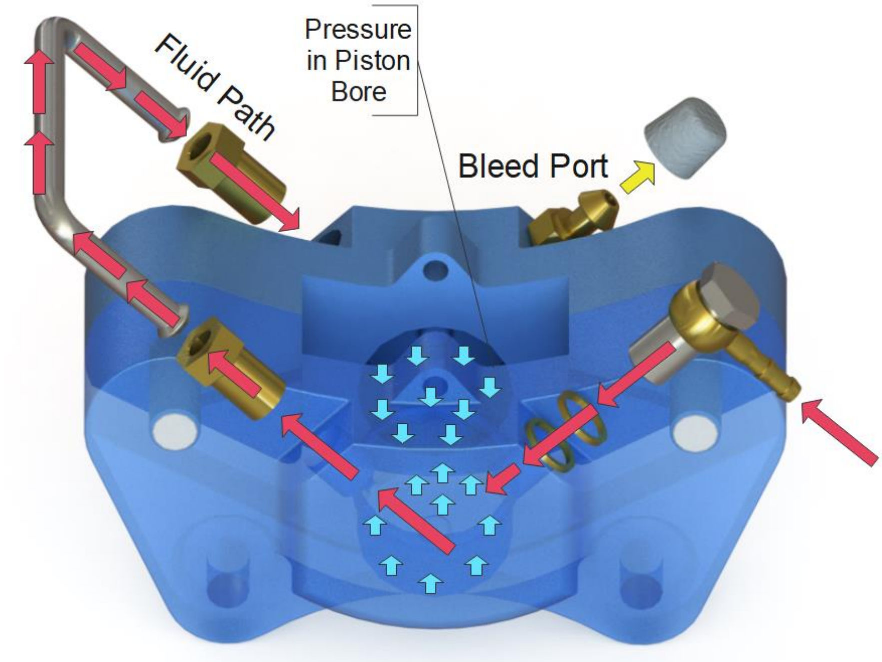


Figure 2 - Calliper internal fluid passages - exploded detail view with arrows indicating fluid flow.

Heat transfer from the braking surfaces to the brake fluid can cause brake fade as the fluid begins to vaporise and cause compressible pockets to form. Solid brake feel was considered important, not only for direct performance, but to provide the driver accurate and consistent feedback. To decrease heat conduction, the stainless-steel pistons were designed with minimal surface area in contact with the brake pad. Slots in the contact patch allow air to circulate through and cool the piston, limiting heat flux to the fluid. In absence of specification from QUTMS, the chosen brake fluid was DOT 5.1 for its high-temperature stability and less corrosive nature than DOT 4 or DOT 3 fluids. Additionally, to increase lateral rigidity of the calliper, the calliper bolts were positioned as close as practically possible to the centre axis of the piston bore. These factors in combination were able to increase the operating load range of the calliper.

1.3 Design Analysis & Validation

Analysis and validation of the calliper consisted fundamentally of both structural and thermal considerations based on the significance of these aspects for proper performance and longevity of the calliper. The calliper body was analysed using FEA static and fatigue simulations based on a maximum braking force required at 1513 N and maximum hydraulic fluid pressure of 9 MPa with extreme driver input. The braking force was determined based on the FSAE requirement for the brakes to be capable of locking up all four wheels [6]. A maximum stress of 211 MPa was observed under peak load and a factor of safety against fatigue failure at $5x10^5$ cycles of 1.7 was achieved using 7075 T6 aluminium for the halves. The locations of critical stress are shown in Figure 3. Analysis of the calliper bolts was carried out to verify the amount of pre-tension required to maintain frictional support between the calliper halves and at the mounting bracket. The titanium calliper bolts required a pre-tension of 3000 N and the mounting bolts, a tension of 750 N. Fatigue analysis of the calliper bolt group showed verified capability to outlast the calliper body with a FOS of 2.27 against fatigue failure. The design of these custom bolts was found to be sufficient for the integral structural role they play.

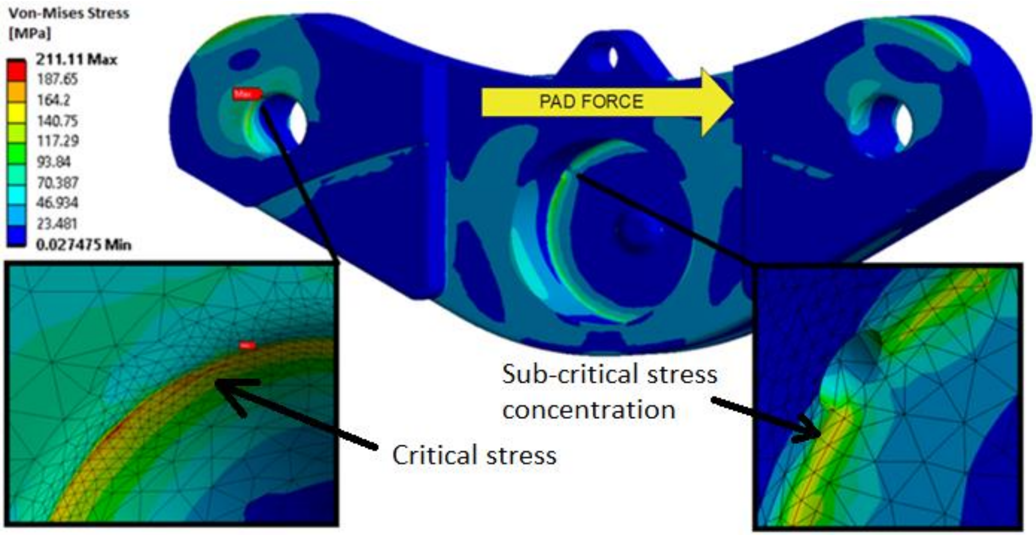


Figure 3 - Composite stress contour plot of brake calliper half (outboard).

Structural analysis of the piston showed a factor of safety of 2.2 against maximum pressure, validating the selection of 316 stainless steel material for the manufacture of this component. The pressure load of 9 MPa resulted in a peak stress of 102.45 MPa on the piston, shown in figure 4, well within the operating limit of the material. Further simulation of the pad-piston load interface was able to determine an adequate pressure distribution on the brake pad was achieved, allowing even and efficient wear of the pads against the rotor. Fillets added at key stress concentration points were shown to prevent high-stress at the piston while maintaining gaps for airflow and additional cooling capacity.

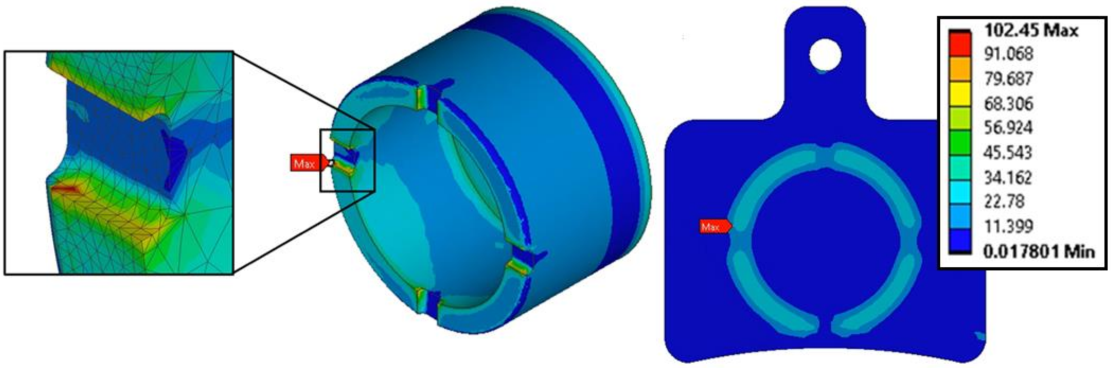


Figure 4 - Piston FEA stress contour structural analysis. Stress in units of MPa.

Thermal analysis was undertaken to evaluate the probable thermal load imparted to the calliper during worst-case-scenario stopping loads based on the QEV3 vehicle and its use cases. Importantly, the operating temperature and conductive response to heat generation at the brake pads was considered to ensure the brake fluid would be protected from extreme temperature which would otherwise cause the fluid to boil, leading to loss of performance. Thermodynamic principles and FEA thermal simulations were used, finding the peak temperature at the pad-rotor interface could reach 262°C. The rotor, brake pad, piston and calliper body were analysed in series with this thermal load applied to the pad/rotor. Given the chosen materials, a heat flux into the calliper system through the brake pads was found to be 1086 kW/m^2 and 352 kW/m^2 into the brake rotor. Immediately after extreme brake application, thermal energy was seen to flow into the calliper via the piston and disperse throughout the calliper body as seen in figure 5. The small contact area of the piston/pad interface was shown to be effective at limiting heat transfer and allowing the rotor and pads to bear the bulk thermal load. After five seconds, the heat was able to be dissipated such that the surfaces in contact with the brake fluid did not exceed 50°C. This was not able to account for repeated application; however, considering the boiling point of DOT 5.1 fluid is rated at 270°C, the calliper was shown to operate below this temperature even at peak heat input.

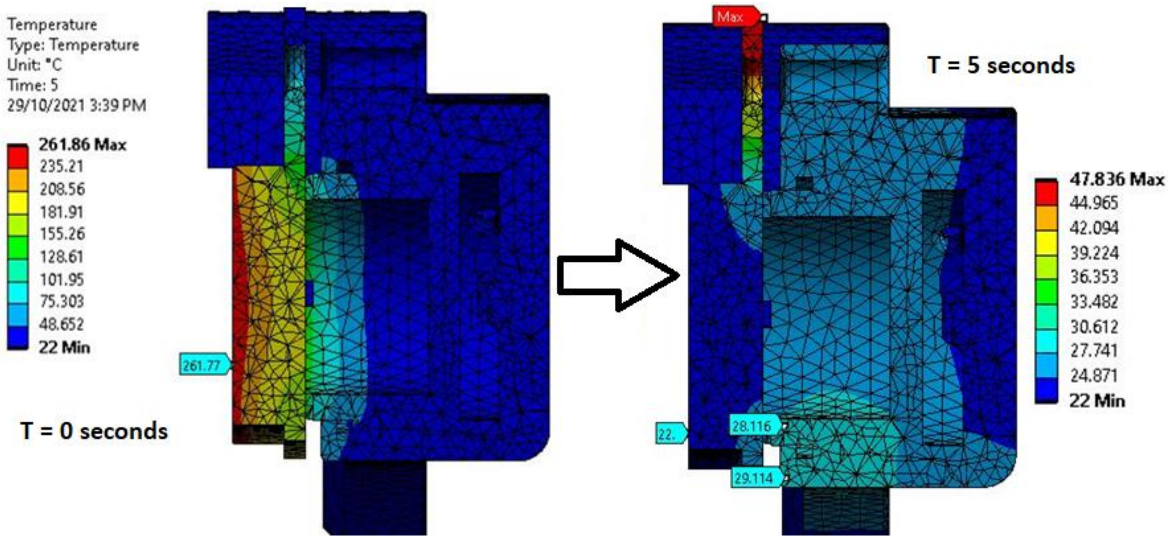


Figure 5 - Calliper temperature gradients at 0 seconds and 5 seconds after braking.

Minor analysis of the pad retaining pin and retaining clip was able to justify the limited structural requirements of this component. Using structural analysis to verify pre-tensile and free-pad bearing loads, the pin was shown to operate with a safety factor greater than 10. The R-clip selected was considered suitable based on evidence of its use in similar circumstances and preference over a split-pin which was heavier.

1.4 Conclusions and Recommendations

In summary, the calliper design presented here was shown to fulfill the compact and lightweight requirement of QUTM's QEV3 car. The low-profile body and careful placement of hydraulic fittings ensured that the calliper could easily fit within the confined wheel envelope while maintaining adequate stiffness for drivability. In particular, the air-insulated hollow piston design was able to reduce heat transfer compared to solid designs. The cross-over tube feature allows simple maintenance and overall design reliability while moving fluid away from direct thermal loads. Based on the verifications made during analysis, the designed system was deemed operable and safe.

Future analysis should consider more advanced thermal simulations to evaluate the ability for the design to cope with heat from extended periods of braking. In prolonged braking scenarios it is possible that cooling may be inadequate and could be improved with additional surface area modifications or a dedicated cooling system. Additional future work should include weight optimisation of the calliper body. Further refinements and removal of material in lower-stress areas of the calliper body may be able to reduce overall weight while maintaining the same performance, improving the value of the calliper to the team.

2 Systems Level Design Summary

This section presents a summary of all systems level design analyses that were undertaken throughout the design lifecycle. Some analysis elements have changed since being conducted in the previous report and the following shall be considered current for the final design. The systems level load analysis has changed due to the decision to increase the piston diameter from 20 mm to 24 mm and is reflected in Appendix B. This was done for the sake of employing standard size seals and has the added effect of lowering the required system pressure to achieve the same braking force. The key calculations and worst-case static loads are discussed. The second design change was opting for a fluid cross-over tube mounted to the calliper halves rather than the internal fluid passages that featured in the previous design. This decision was made to reduce complications with machining and sealing internal fluid passages.

2.1 Summary of Systems Analysis, DFM, DFA and FMEA

The systems level analysis was able to establish the peak load case in normal operation of the brake calliper to which the design was to be analysed for. Limiting factors of the load case were identified as the fluid pressure resultant from extreme driver input and tractive force between the tyre and the ground in a peak stopping deceleration instance. Weight transfer from rear to front was assumed based on the known centre of gravity (COG) of QEV3 and downforce created by the aero package was neglected in favour of a safety factor of 2. The FOS was deemed sufficient to account for additional load caused by greater tractive force from velocity-dependant downforce on the car. A high-level calculation summary can be found in Appendix B to support changes made since the previous report was submitted. In summary, the calliper is to withstand fluid pressure of at least 9 MPa, which was assumed as the highest possible fluid pressure a driver may exert on the rear brake fluid system based on previous studies [7, 8]. The peak thrust force imparted to the calliper body was found to be ~ 1500 N at the limit of tyre traction under braking. Although the pad manufacturer specifies a nominal frictional coefficient between the brake pad and a steel rotor of $\mu_{pad} = 0.55$ [3], a conservative value of $\mu_{pad} = 0.35$ was assumed to account for wet conditions, worn pads or high heat operation. This complete load case was considered conservative enough to cover the most demanding environment the calliper could see in use.

The Design for Manufacturing (DFM) analysis highlighted the rough costs and procedures associated with manufacturing of the total calliper assembly. To reduce manufacturing costs and complexities multiple components are off-the-shelf products, including the brake pads, pad retaining pin and clips, external tubing, piston seals, and fluid fittings. The brake calliper was designed to outlast the QEV3 vehicle, thus machining was primarily chosen rather than casting as the large initialisation costs and complexities associated with cast parts would be underutilised. The DFM table, most importantly, highlighted that the calliper bodies required high operation count due to the intricate geometry and number of features. CNC machining was assumed which significantly increased the overall design rating. Consequently, the team was able to further simplify the design by replacing internal ducts with external piping and removing additional features on the outer piston housing.

The DFA presented in report 2 followed the classification system aligned with Boothroyd's handling and insertion table using alpha and beta to determine which category and manipulation each component required. The DFA study concluded that the assembly time for the entire set up, including mounting the assembly to the upright, was approximately 170 seconds. With several significant assumptions, this is understood to be subject to a variety of factors not considered in this report in detail. It was assumed that the components are new with no debris or cleaning required with all necessary tools easily accessible. Furthermore, the tool acquire time (TA) was estimated on the basis that a technician that has all the required tools prepared before the job is undertaken and does not have to leave the workstation to find tools. This could be the same as having a tool kit for each technician per wheel corner at the racetrack.

Report 2 conducted a thorough failure modes and effects analysis (FMEA) which highlighted foreseeable failure modes associated with the concept brake calliper in operation. A recurring action was increased safety factor, owing to the heightened possibility of failure points being triggered if a spike in heat or mechanical load is experienced. To ensure higher confidence in the design, a minimum FOS of 1.5 was aimed for. Appropriate and targeted material selection featured as an important action for many failure items, thus, the analysis detailed later included material selection for critical components. Some other suggestions that were actioned in the revised design were: External sealed fluid channel to avoid leakage between calliper, dust cap on the bleed port to reduce blockages from debris and a standard R-clip rather than the vulnerable clip design originally proposed. To improve thermal performance, the FMEA found that a refined piston design would decrease thermal flux to the fluid. The most appropriate findings were implemented in the final design.

2.2 Risk Assessment Study

A risk assessment was undertaken for the brake calliper design and the hierarchy of controls was implemented to mitigate potential risks as was deemed necessary and achievable. It was assumed that the brake calliper would be assembled and installed by members of QUT Motorsport. Maintenance would also be carried out by QUT Motorsport participants with the availability of appropriate training and expertise on tool operation and installation of the calliper. The assumption was also made that the calliper would be installed on the car only in the configurations intended. The car was assumed to only operate within a controlled environment associated with FSAE events and the driver wears all appropriate safety gear associated with regular driving protocol for the QEV3 car. Experienced and appropriately qualified drivers are assumed to be the only persons operating the vehicle. The scope of this risk assessment encapsulated the following:

- Assembly of the brake calliper unit and attachment/installation on the vehicle by the team.
- Regular standard servicing required to maintain the calliper unit, including
    - Replacement of worn brake pads,
    - Cleaning in and around the calliper,
    - Bleeding the brake fluid from the calliper, and
    - Disassembling the brake calliper for the purpose of an overhaul.
- Normal operation of the car (and brakes) as would be seen in any of the FSAE events.

The completed risk assessment matrix in Appendix C should be referred to for all identified risks, originating hazards and proposed controls analysed here. Upon completion of the risk assessment, the following key measures were proposed to reduce the risk risks associated with the brake calliper designed here:

- Specification of bolt torque for calliper bolts and mounting bolts to provide adequate support to calliper. It was also recommended that moderate thread locking compound (such as LocTite) be used by QUT Motorsport when assembling the calliper as this will lessen the chances that a bolt would loosen during operation of the vehicle.
- Internal fluid channel routing was recommended to be eliminated in favour of external fluid routing via a tube and standardised fittings. This was concluded based on the risk of calliper failure causing loss of vehicle control and possible injury and monetary losses.
- PPE including long-sleeve t-shirt, gloves and safety glasses was recommended to be always worn by maintenance personal when working on the calliper due to the risk posed by metallic parts, forces and chemical hazards posed by the handling and disposal of DOT 5 or 5.1 brake fluids.

- Calliper bolt orientation was recommended to face outboard on the car to reduce the risk of personnel injuring hands if bolt force was to "let go" during loosening or tightening by avoiding other components in the direct path of the tool.

2.3 Triple Bottom Line (TBL) Analysis

A triple bottom line (TBL) analysis of the calliper design was carried out to evaluate the lifecycle of the overall calliper system. A lifecycle analysis was followed by cost-benefit analysis and finally an ethics evaluation. The details of these are outlined in the proceeding sections.

2.3.1 Life Cycle Analysis

Several high-quality materials compose the calliper design. These include synthetic rubber and metals such as titanium and steel which are to be sourced through companies that ultimately extract the material from the earth and process it to the form required for manufacture. Therefore, the design directly and indirectly contributes to the energy and resource intensive practices associated with purchasing raw material and off-the-shelf products required premanufacturing. Figure 6 illustrates the high-level lifecycle of the materials used to make the calliper, as well as its disposal after use. Raw metal billets were all assumed to undergo similar processing and were grouped as such. The rubber and elastomer used in the dust cap and piston seal respectively, were grouped also due to similar manufacturing methods. Off-the-shelf parts (all other metal components) were grouped as their manufacture went further than the parts designed here.

The impact factor matrix in Table 1 shows a summary of the calculated values generated through SolidWorks sustainability reports with appropriate customisation. The sustainability report concluded values for the Carbon Footprint (KG), Energy Consumption (MJ), Water Acidification (KG) and Water Eutrophication (KG) generated within stages such as material, manufacturing, distribution, and end-of-life. Each material is extracted from the ground and processed using various refining processes that produce CO_2 as the major detrimental effect. Then, the material would be distributed and manufactured into the respective components. Distribution will prominently consist of postal due to the quantity and size of the design which generates CO_2, depleting natural resources and generating greenhouse effect. The team will explicitly control manufacturing of components such as the inboard and outboard calliper bodies, calliper bolts and pistons distributing the respective material to the proposed manufacturing outlet. While off-the-shelf products will be independently distributed and manufactured by third party companies, however, QUTM contributes their respective impacts and consequences outlined in the impact matrix.

Stage 1 represents the extraction process. Sub-stages 1a and 1b displays the material values for the metal and rubber components respectively which are associated with stage 1c. Thus, 1c stages have been covered in preceding stages and ignored within table 1. Stages 2 and 3 have been combined to represent the processing and manufacturing stages of the LCA using manufacturing values for metal and rubber. Stage 4 represents the distribution values to accurately portray the design. It is assumed that the products have been exclusively sourced from Australia to reduce travelling distance. With travelling distances unknown and quite arbitrary, the default 970km was set for all components. It is likely the total travelling distance is inflated compared to the actual travelling distance. In addition, it is likely that some components share distributors which would reduce actual travelling distance. Thus, values may be overinflated and not accurate. Stage 5,6,7 and 8 display the after-life values of the design. In addition, composite materials such as the brake pad were simplified to rubber which may have an underlying impact on the values. Overall, table 1 displays a significant amount of energy consumption required by the design with a large percentage being from rubbers processing. This energy has been defined by the sustainability report as non-renewable being fossil fuels. Thus, the design significantly depletes and burns fossil fuels indirectly. Furthermore, manufacturing of rubber material produces significant Co2 seen in stages 2b and 3b. Overall, the table displays realistic and relatively tame sustainability values.

The team has decided to mostly source off-the-shelf products, where possible, which can be more efficiently produced, minimising costs and impacts. In addition, the team has decided to source locally to reduce distribution costs and related impacts from shipping. All components should be exported to one destination being the proposed manufacturing outlet or QUT. The manufacturing procedures required by the team consists of operations such as CNC, machine, grind, polish, and lathe. These operations require high amounts of energy in the form of electricity which will be likely sourced through fossil fuels with its associated impacts on the depletion of the ozone and natural resources. In addition, all manufacturing operations have poor material utilisation but are most ideal due to the required quantity and quality of the parts.

During the life cycle of the calliper, regular service and final disposal is required. All these procedures possess no significant environmental impacts, other than potential leaks and spillages of associated oils and fluids which could contaminate the environment locally. It is proposed that all metal components can be recycled with only the pads and rubber components forming landfill waste. Overall, due to the quantity of callipers to be produced, the discussed environmental impacts are low.

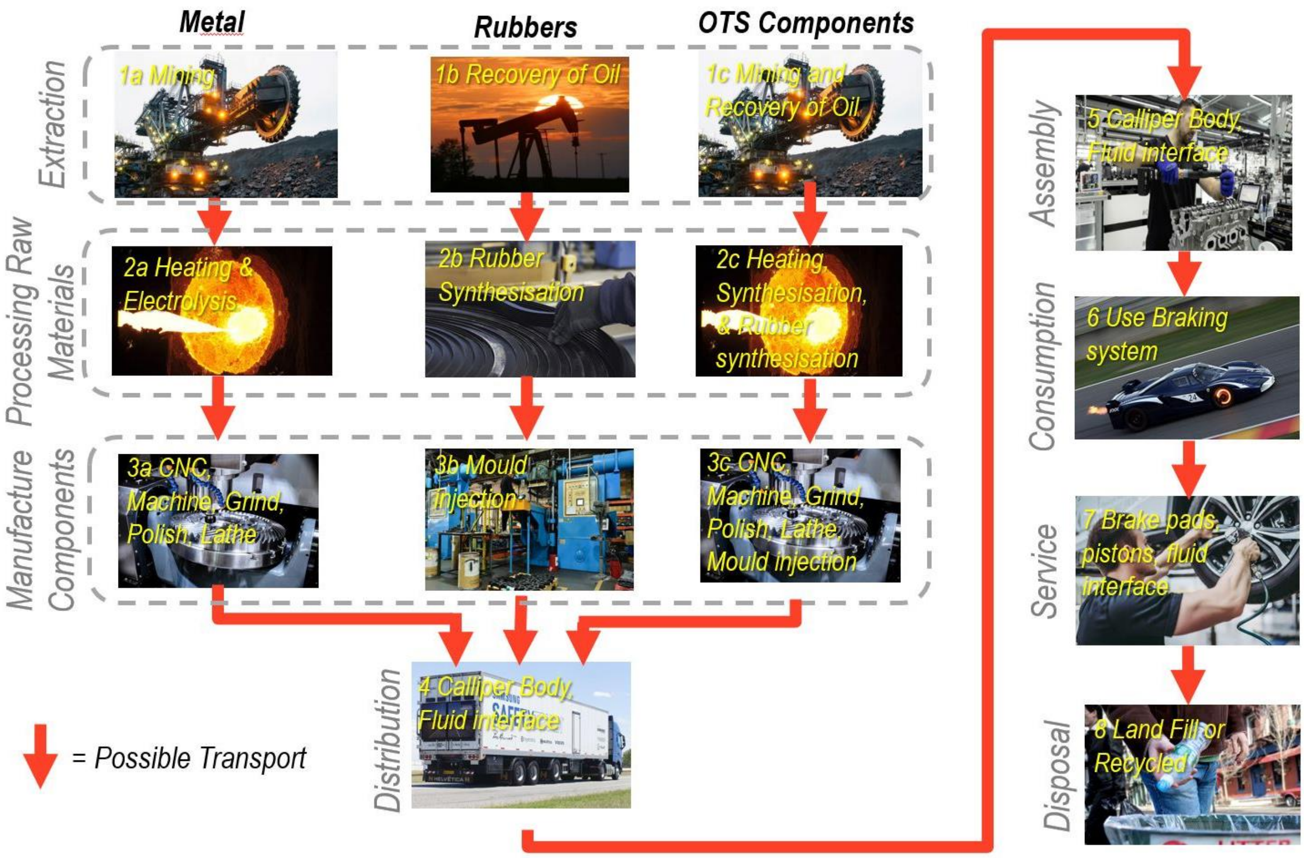


Figure 6 - Life cycle assessment flow chart.

Table 1 - Impact factors matrix for calliper LCA.

Stage	1. Global Warming	2. Mineral and Fossil Fuel Depletion (i.e. abiotic depletion)	3. Land Transformation / Occupation and Biodiversity	4. Water Resource Depletion	5. Eutrophication	6. Acidification	7. Eco-Toxicity	8. Photo-Chemical Smog	9. Ozone Depletion	10. Ionising Radiation	11. Human Toxicity	12. Respiratory Effects	13. Nuisance	14. Indoor Air Quality
1a.	10.57 KG Co2	129.104 MJ	Remove land which affects eco-system	Used to strip debris	.007 KG	.073 KG	INSIGNIFICANT	UNLIKELY	ASSOCIATED 1	UNLIKELY	UNLIKELY	INSIGNIFICANT	Noise	UNLIKELY
1b.	1.12e-1 KG Co2	2.818 MJ	Occupies land	INSIGNIFICANT	0.0000253 KG	.000242 KG	Toxic emssions and Spills	UNLIKELY	ASSOCIATED 1	UNLIKELY	Localised Affects	Localised Affects	Localised Affects	
1c. Associated with 1a. and 1.b														
2a. & 3a.	.83 KG Co2	7.4 MJ	Manufacturing Facilities ocupy Land	Used to strip debris and lubrication	.0026 KG	.073 KG	INSIGNIFICANT	UNLIKELY	ASSOCIATED 1	UNLIKELY	UNLIKELY	Micro Particles	Noise	Micro Particles
2b. & 3b.	21.6 KG Co2	224 MJ	Manufacturing Facilities ocupy Land	Used to strip debris and lubrication	.0052 KG	.01 KG	INSIGNIFICANT	UNLIKELY	ASSOCIATED 1	UNLIKELY	UNLIKELY	Micro Particles	Noise	Micro Particles
2c. & 3c. are associated with a. & .b														
4	.049 KG C02	.7 MJ		INSIGNIFICANT	5.1e-5e-5 KG	.0002 KG	Toxic Emissions	UNLIKELY	ASSOCIATED 1	UNLIKELY	UNLIKELY	INSIGNIFICANT	UNLIKELY	UNLIKELY
5, 6, 7 & 8	.41 KG Co2	4.56 MJ		INSIGNIFICANT	0.000304 KG	.0021 KG	Improper Use can be significant	UNLIKELY	ASSOCIATED 1	UNLIKELY	INSIGNIFICANT	INSIGNIFICANT	UNLIKELY	UNLIKELY

2.3.2 Cost-Benefit Analysis

Cost-benefit of the design refers to the value of the outcomes with consideration of the costs associated with achieving them. Costs associated with the design encapsulate the following as identified in the previous DFM analysis of the system and the cost breakdown detailed later in this report:

- Manufacturing costs associated with the production of designed parts. These include the calliper body, pistons, pad retaining pin, calliper bolts and cross-over tube. Manufacturing costs were assumed to be borne by QUTM through QUT or their sponsor's manufacturing services.
- Procurement of off-the-shelf parts, including spares and replacement parts such as brake pads that are expected to be replaced at routine intervals due to wear. See purchasing list for details.
- Interval costs associated with maintenance and time taken to complete maintenance. This was assumed to be non-monetary for the purpose of this discussion and could be considered relative to other calliper designs.
- Disposal cost of spent, broken or redundant brake calliper or parts at the end of its lifetime. This was also considered a relative cost, not quantified due to the small nature and likely negligible impact of disposal.

The overall cost of producing the calliper was found to be $939.90 (AUD), detailed in section 3.3. This includes the materials and manufacturing cost of designed parts and procurement costs of off-the-shelf parts. The interval costs for the calliper will be primarily the cost of replacing brake pads. The brake pads cost $37.44 per set. At one set per wheel, this cost would be $74.88 each time the pads need replacing. Other maintenance costs may include replacement of the brake fluid periodically. In this scenario, it was assumed that the cost of brake fluid would be insignificant, and the frequency of replacement could be considered independent of the calliper design. Thus, cost of fluids was excluded. In the event of a failure of a component or after significant use requiring preventive inspection, costs associated with component replacement on an as-needed basis. Time costs associated with assembly, installation and maintenance of the calliper were considered non-monetary because QUT Motorsport (QUTM) is run by students who are not paid for their time.

The brake calliper was designed specifically for the current QUTM FSAE electric vehicle, QEV3. This proprietary design has no intent to generate direct revenue, being manufactured solely for QUTM use. Therefore, success of the design is not determined by the perceived profits/revenue but rather the capital costs and the return gained in performance and additional social and indirect economic benefits. The Wilwood GP200 proposed for the front brake and the currently used Wilwood PS1 callipers cost $238 and $207.94 respectively [9]. Although these are much less expensive than the design here, no off-the-shelf calliper systems were deemed able to fulfil the space envelope requirements of QEV3. The QUTM team is interested in keeping costs low; however, the capability to manufacture this design to meet their specific needs is also a factor that benefits the team and its capability. It was also recognised that much of the cost of the calliper, particularly manufacturing of designed parts, would possibly be mitigated due to input from the many team sponsors.

The brake calliper design presents indirect social and economic benefits to QUT and the QUTM team. The improved braking system is likely to enable greater vehicle performance, tailored to the specific needs of the car and the driver. Additionally, there is increased potential for development and proof-of-concept for the team's capability to design and produce their own parts as this is something that QUTM is interested in pursuing. This may also lead to positive attention from sponsors and the FSAE, potentially leading to additional sponsorships and/or support to further the team's capability for future development. In summary, a Net Present Value could not be attained for the calliper due to the non-monetary benefits being significant. However, it was concluded that the performance of the calliper would be comparable to its competitors and the value provided by the design would likely outweigh the higher capital cost.

2.3.3 Ethics Evaluation

The calliper design and system/component lifecycle were evaluated in terms of ethics as required by the body of Engineers Australia. The EA Code of Ethics was referred to and each of the four guidelines was addressed with regards to the project and designed system. The scope of the ethics evaluation was limited by the scope of the project which saw limited coverage of some ethical aspects that would be more pertinent to projects on a commercial scale.

2.3.3.1 "Demonstrate Integrity"

Throughout the project, the team made conscious efforts to operate based on a well-informed conscience by considering a wide breadth of human factors involved with the design of the brake calliper for the QEV3 car. By considering the safety and useability of the design for the driver, manufacturer and servicing personnel, the team has discerned areas where the design could prove hazardous and addressed these.

Within the team, members acted impartially and had regular discussions surrounding aspects of the design and the analysis/verification processes. Honesty was upheld by giving detailed and comprehensive reasoning for decisions made, including undertaking standard analysis of systems-level design and component-level design such that external parties shall be able to understand the decisions made here. The team respected obligations to provide QUTM with all requested documentation completed to a high degree of accuracy. All persons were considered equal in importance and value, including those in the design team and those interacting with the design, using appropriate judgement and design tools such as the previous Risk Assessment analysis.

2.3.3.2 "Practise Competently"

The project/design team consciously developed the skills and knowledge necessary to complete each stage of the design process. This involved learning and implementing analysis techniques and developing skills in the use of modelling software. Members sought peer input for their tasks and gave constructive feedback where necessary to ultimately improve the team's final product. Members practiced within their areas of competency (mechanical engineering discipline); however, greater care should be taken in the future to recognise specific competencies of each group member as this would allow greater confidence in the work undertaken by each individual.

No material or results were falsified and were presented accurately to the best of the group's knowledge. Legal requirements, standards (FSAE) and QUTM guidelines were considered to ensure the calliper design met the required specification to be considered safe and operable on the QEV3 car. No tasks or activities were undertaken that would be considered outside of the qualifications held by the group.

2.3.3.3 "Exercise Leadership"

The team upheld the reputation and trustworthiness of the engineering practice by selectively verifying the most critical design aspects and presenting the results in the proceeding sections of this report. Specifically, aspects of the design that were identified as vague or lacking were improved in suitable ways in the interest of facilitating ease of use to the end user. An example of this was remediation of identified failure modes from the FMEA analysis.

The team engaged in discussion surrounding the selection of a final design from four concept designs. Communications and deliberations were made honestly and with consideration of the broad range of design outcomes. Reasonable efforts were made to communicate with the stakeholders, QUT Motorsport, to ensure sufficient and accurate information was gathered to support the effective design of the brake calliper. An example of this was email and verbal communications made to identify key dimensions and operating conditions of the car to support the selection of appropriately sized pistons for the calliper. Failure to do so may have led to poor or insufficient brake performance.

2.3.3.4 "Promote Sustainability"

As stated previously, the stakeholders were consulted with regards to their needs which was able to allow effective design of the calliper solution. Selection of materials and consideration of design safety was summarised in a TBL analysis and the flow-on effects to the environment and communities was considered. Implementing measures to mitigate potential risks such as contamination from brake fluid was done to limit the potential for detriment to the environment as part of the design.

In completing a comprehensive list of analysis techniques, the design was changed/improved where improvements were identified in accordance with the obligation for engineers to deliver sustainable outcomes that consider economic, environmental and social consequences. The calliper was made simple and maintainable so that it may not compromise QUT Motorsport's capabilities to compete in the event of a minor failure or general wear and tear.

3 Manufacturing Specifications and Costing

A technical detail drawing set was produced for the design which can be found under Appendix I. The drawing numbers and their associated description is outlined in table 2. The manufacturing procedures for designed parts is outlined in the following section, as is the purchasing list and cost breakdown for two callipers as is required to service both rear wheels. Drawings ending in 001-008 were components designed as part of this project while those ending in 101-110 are off-the-shelf items. Dimensions of off-the-shelf items are subject to change due to possible variations in products sourced from suppliers.

Table 2 - Drawing numbering index.

DRAWING NUMBER	DESCRIPTION
H420M5-DWG-001	Brake Calliper System Assembly
H420M5-DWG-002	Brake Calliper System Exploded View
H420M5-DWG-003	Calliper Body Detail – Inboard half
H420M5-DWG-004	Calliper Body Detail – Outboard half
H420M5-DWG-005	Piston Detail
H420M5-DWG-006	Calliper Bolts Detail
H420M5-DWG-007	Pad Retaining Pin Detail
H420M5-DWG-008	Cross-Over Tube Detail
H420M5-DWG-101	Piston Seal
H420M5-DWG-102	R-Clip
H420M5-DWG-103	Mounting Bolts
H420M5-DWG-104	Brake Pads - CP4226D27
H420M5-DWG-105	Tubing Nut for Cross-over Pipe
H420M5-DWG-106	Bleed Screw
H420M5-DWG-107	Bleed Screw Dust Cap
H420M5-DWG-108	Banjo Fitting
H420M5-DWG-109	Banjo Bolt
H420M5-DWG-110	Copper Washer for Banjo

3.1 Manufacturing Procedures

A general manufacturing procedure was specified for the manufacture of all designed components for the brake calliper. The procedures are intended to indicate a sequence of processes for taking a specified material stock and producing the finished component. Key part features and geometry are specified as is the desired machine for the operation. This is to give an overview only and was not intended to be a direct instruction with enough detail to capture every small element. A detailed manufacturing procedure, listing machine parameters and variables like cutting speeds was considered outside the scope of work as this would be completed by QUT Motorsport or a third-party manufacturer prior to the manufacturing step. The proceeding sub-sections present the desired material, machine process and sequence of processes to be undertaken for manufacture of each component designed.

3.1.1 Calliper Body – Inboard Half

Stock material: 7075 T6 Alloy billet, 100mm x 100mm x 100mm blank

- Operation 1
    - Use 5-axis CNC machine to cut out pre-programmed manufacturing steps
    - Firstly, mill away excess material until external profile is achieved.
        - Operation 1a
            - Bore two 8mm x 8mm holes on back side of caliper opposing at 90 degrees to each other (Refer to drawing H420M5-DWG-003)

        - Operation 1b
            - Add 2mm X 45° conical chamfer to bottom of holes completed in Operation 1a
        - Operation 1c
            - Bore one 2mm x 10mm hole at bottom of holes completed in operation 1b
        - Operation 1d
            - Bore two parallel 8mm x 21.2mm holes on “wings” of calliper 80mm apart for the calliper bolts (Refer to drawing H420M5-DWG-003).
        - Operation 1e
            - Bore one 4mm x 10.2mm hole at the top centre of the calliper body for the brake pad retaining pin (Refer to drawing H420M5-DWG-003)
        - Operation 1f
            - Bore piston cavity as per drawing H420M5-DWG-003 using boring tool in CNC machine
- Operation 2
    - Use Tap and die tools to create internal threads for holes created in operations 1a, 1d, and 1e
- Operation 3
    - Grind all contact surfaces in pad cavity and both joining faces. Deburr all sharp edges with 0.5 mm chamfer or fillet
- Operation 4
    - Polish all external surfaces and anodize component
- Operation 5
    - Hone cylinder bores to final tolerance and surface finish Ra = 0.4 µm.

3.1.2 Calliper Body – Outboard Half

Stock material: 7075 T6 Alloy billet, 100mm x 100mm x 100mm blank

- Operation 1
    - Use 5-axis CNC machine to cut out pre-programmed manufacturing steps
    - Firstly, mill excess material until external profile is achieved.
    - Operation 1a
        - Bore two 8mm x 8mm hole on back side of caliper opposing at 90 degrees to each other (Refer to drawing H420M5-DWG-004)
    - Operation 1b
        - Add 2mm X 45° conical chamfer to bottom of holes completed in Operation 1a
    - Operation 1c
        - Bore one 2mm x 10mm hole at bottom of holes completed in operation 1b
    - Operation 1d
        - Bore two parallel 8mm x 21.2mm holes on “wings” of calliper 80mm apart for the calliper bolts (Refer to drawing H420M5-DWG-004).
        - Bore an additional two 8mm x 21.2mm holes on “wings” of calliper 80mm apart for the upright attachment holes (Refer to drawing H420M5-DWG-004).
    - Operation 1e
        - Bore one 4mm x 10.2mm hole at the top centre of the calliper body for the brake pad retaining pin (Refer to drawing H420M5-DWG-004)
    - Operation 1f
        - Bore piston cavity as per drawing H420M5-DWG-004.
- Operation 2

  - Use Tap and die tools to create internal threads for holes created in operations 1a, 1d, and 1e
- Operation 3
  - Polish all external surfaces and anodize component.
- Operation 4
  - Hone cylinder bores to final tolerance and surface finish Ra = 0.4 µm.

3.1.3 Pistons

Use 15mm x 24mm cylindrical steel bar

- Operation 1
  - Use 5-axis CNC machine to cut out pre-programmed manufacturing steps
  - Operation 1a
    - Bore 12mm x 19mm hole in the center from the end of the bar
  - Operation 1b
    - Add 0.3mm fillet at top edge (Refer to drawing H420M5-DWG-005)
  - Operation 1c
    - Add four 2.5mm incisions to the top edge (refer to drawing H420M5-DWG-005)
- Operation 2
  - Grind all external surfaces using fine metal grinder. Do not grind internal cavity. Clean up any small rough edges or mistakes caused by the CNC machine. Add 0.5 mm chamfer/fillet to deburr.
- Operation 3
  - Grind and polish piston exterior surface to final tolerance and polish to Ra = 0.2 µm

3.1.4 Calliper Body Bolts

Use 12mm x 48.8mm cylindrical bar stock

- Operation 1
  - Use lathe to turn diameter down to 8mm for 42.4mm of overall shank length
- Operation 2
  - Add 1mm X 45° conical chamfer to bottom of bold (Refer to drawing H420M5-DWG-006)
- Operation 3
  - Add 1mm fillet to top of bolt (11.4mm end)
- Operation 4
  - Use Tap and die tools to create external thread on 8mm diameter section
- Operation 5
  - Grind bolt head (11.4mm end) to remove any burrs.

3.1.5 Pad Retaining Pin

Use 5.5mm x 51.47mm cylindrical steel bar

- Operation 1
    - Use lathe to turn diameter down to 4mm for 46.47mm of overall length
- Operation 2
    - Use drill press to create 1.2mm hole at end of pin (Refer to drawing H420M5-DWG-007)
- Operation 5
    - Grind bolt head to final dimension
- Operation 6
    - Cut thread section.

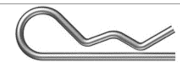
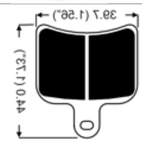
3.1.6 Cross-Over Tube

Material: Stock 3/16-inch AISI 304 stainless steel hydraulic fluid tubing, 88 mm length

- Operation 1 – bending
    - Form the two 90° bends in the tube using a tubing bending tool and a 2.6 mm radius bending mandrel.
- Operation 2 – Flaring ends (bubble flair)
    - Assemble both flare nuts on to the tube piece in the correct configuration with the thread orientated towards the open ends of the tube.
    - Produce bubble flair at each end of the tube using a purpose-made flaring tool for 3/16-inch metal tubing. The flare produced should have any burrs removed

3.2 Purchasing List Table

Table 3 - Purchasing list for off-the-shelf parts and supplies.

Part	Number required	Item number	Price (AUD)	Supplier	Contact phone/ website INTL.	Image
Brake line	0.5m	318BLStainless	6.66	Pirtek	13 42 22	
Flare nut	2(2 Pack)	AF373	26.26	RCE performance	3299 5570	
R-clip	1(2 Pack)	I/N:2420762	4.82	Bunnings	3716 9000	
Piston Seal	1(4 in kit)	073-000819	36	Sparesbox	Sparesbox.com.au	
Brake nipple	1(2 pack)	SSBN7-2Z1BK	5.93	ebay	ebay.com	
Brake dust cap	1(4 Pack)	P27-4PACK	3.2	Aliexpress	aliexpress.com	
Copper washer	1(6 in set)	BH069	11.98	Supercheap auto	3274 6311	
Brake pad	1 (4 pack rear set)	CP4226D27	85.09	mailordercarparts.co.uk	mailordercarparts.co.uk	
Banjo fitting	2	RWF-720-06ABK	72	ebay	ebay.com	

Part	Number required	Item number	Price (AUD)	Supplier	Contact phone/ website INTL.	Image
Braided brake lines rear	1 (pairs)	BLGKTECH/R32	94.5	Gktech	https://au.gktech.com/	
Loctite for mounting bolt	1	IN:1560362	15.58	Bunnings	3716 9000	
M8 Mounting Bolt	1(8 in set)	90108-MAS-E01	64.37	Titan Classics	titanclassics.com	
Calliper Machine cost	1	Outsourced	293.22	Sponsor or external	Through sponsors or externally	
Piston Manufacturing	1	Outsourced	47.06	Sponsor or external	Through sponsors or externally	
Pin Manufacturing	1	Outsourced	3.45	Sponsor or external	Through sponsors or externally	

3.3 Cost Summary

The calliper components were broken down into individual parts with part prices from a supplier, or a manufacturing cost. The manufacturing costs were estimated from SolidWorks and the reports for each part can be seen in APPENDIX F through to H. For each part purchased through a supplier, the lowest cost with comparable quality was selected. This was inclusive of shipping or the assumption the part would need to be picked up from a store within the local vicinity with no extra shipping cost. Additionally, for the components such as the calliper body, which would require extensive machine work, the cost for manufacturing was averaged using the cost template used through SolidWorks. A higher cost was used as economy of scale would not be achievable for a bespoke component with a limited production run. Materials were selected based on chosen materials detailed later in this report. Furthermore, it was assumed that the machine file would be prepared such that minimal alterations to the CNC manufacturing is required saving extra operator cost for the calliper body.

A full costings table detailing the options and final chosen item is included in Appendix D. The off-the-shelf parts prices totalled $422.17, and the manufacture of the calliper body (x2) was $416.73. The cost to produce the pistons was $94.12 for two stainless steel machined items and $6.90 for the two pad retaining pins for a total price of **$939.9 AUD.** The summary of these costs can be seen in Appendix E and in figure 7 below. Both calliper halves had similar pricing for manufacturing with $157.57 and $136.46. The extra cost for the outboard half was due to the drilling and tapping for the fluid portions of the calliper. Costs for off the shelf bolts for the calliper halves were considered into the cost summary, however the team would aim to produce these components as this would give further control on loads and quality that was specified such as rolled or cut threads and would impact the bolts tension requirements. For this summary, the calliper body bolts were included as off-the-shelf.

Part name	Part description	Supplier	Price	Length/ No. of	QTY	freight	Selected
brake line	cut per meter	Pirtek	6.66			0	6.66
flare nut	Stainless steel inverted Flare tube nut	RCE performance	13.13	2	2	0	26.26
R-clip	3/8 x 1/2' Zinc plated R clip - 6 pack	Bunnings	4.82	6	1	0	4.82
Brake nipple	M8 x 1.0 mm brake bleed screw	ebay	5.93	4	1	0	5.93
Brake dust cap	4 pieces brake dust cap	Aliexpress	3.2	4	1	0	3.2
Copper washer	copper washers	Supercheap auto	5.99	3	2	0	11.98
brake pad	39.7 x 44	mailordercarparts.co.uk	37.44	1	2	10.21	85.09
caliper bolt (body)	m8x45mm can do both rear calipers with this pack		65	8	1	12.32	38.66
caliper bolt (to hub)	m8x45mm		65	8	1	12.32	38.66
banjo fitting	M8 to 6An Banjo fitting	ebay	31.67	1	2	8.66	72
braided brake lines rear	rear braided brake lines	Gktech	79	2	1	15.5	94.5
Locktite for mounting bolt	Non-permanent adhesive threadlocker	Bunnings	15.58	1	1	0	15.58
Caliper Machine cost	Both caliper body sides - performed by CNC		208.37	2	1	0	416.73
Piston Manufacturing	Stainless steel - CNC			2	1	0	94.12
Pin Manufacturing	Steel - CNC			2	1	0	6.9
			Total cost for rear caliper x 2				939.9

Figure 7 - Cost summary of all required off-the-shelf parts.

Due to the teams' sponsor relationships with various companies throughout Brisbane and via various networks, the final price would likely be reduced if the manufacturing cost could be mitigated by employing several companies' expertise for the component. Additional reductions in the final cost would be expected with the partners who are associated with the team currently with companies such as Australian American Racing (AAR) for milling or machine work, Calm Aluminium for the supply of the aluminium stock for the calliper body and Laser Central to perform the finishing service such as the tapping of the brake threads, deburring and drilling of the mounting holes as this could reduce CNC time of all components. Overall, this consideration would allow the manufacturing to become more viable and additional focus on sustainability could be allowed for as the sponsors could be working within higher environmental standards.

4 Detailed Design – Component Analysis Appendix

In this section, a comprehensive analysis of key system components is presented, detailing validation and detailed design reasoning behind the final assembly as submitted. The importance of component-level analysis for the design of the brake calliper is high due to the high-performance nature of its use. Comprehensive analysis was done with the aim of verifying the capability for the calliper to meet its functional expectations while optimising lightweight and cost-efficient characteristics. Where opportunities for improving the design were identified, these were noted as recommendations or directly implemented on the design. Overall, the proposed design was shown to meet the requirements set out following the systems-level analysis presented previously and, in some cases, exceeds the performance requirements.

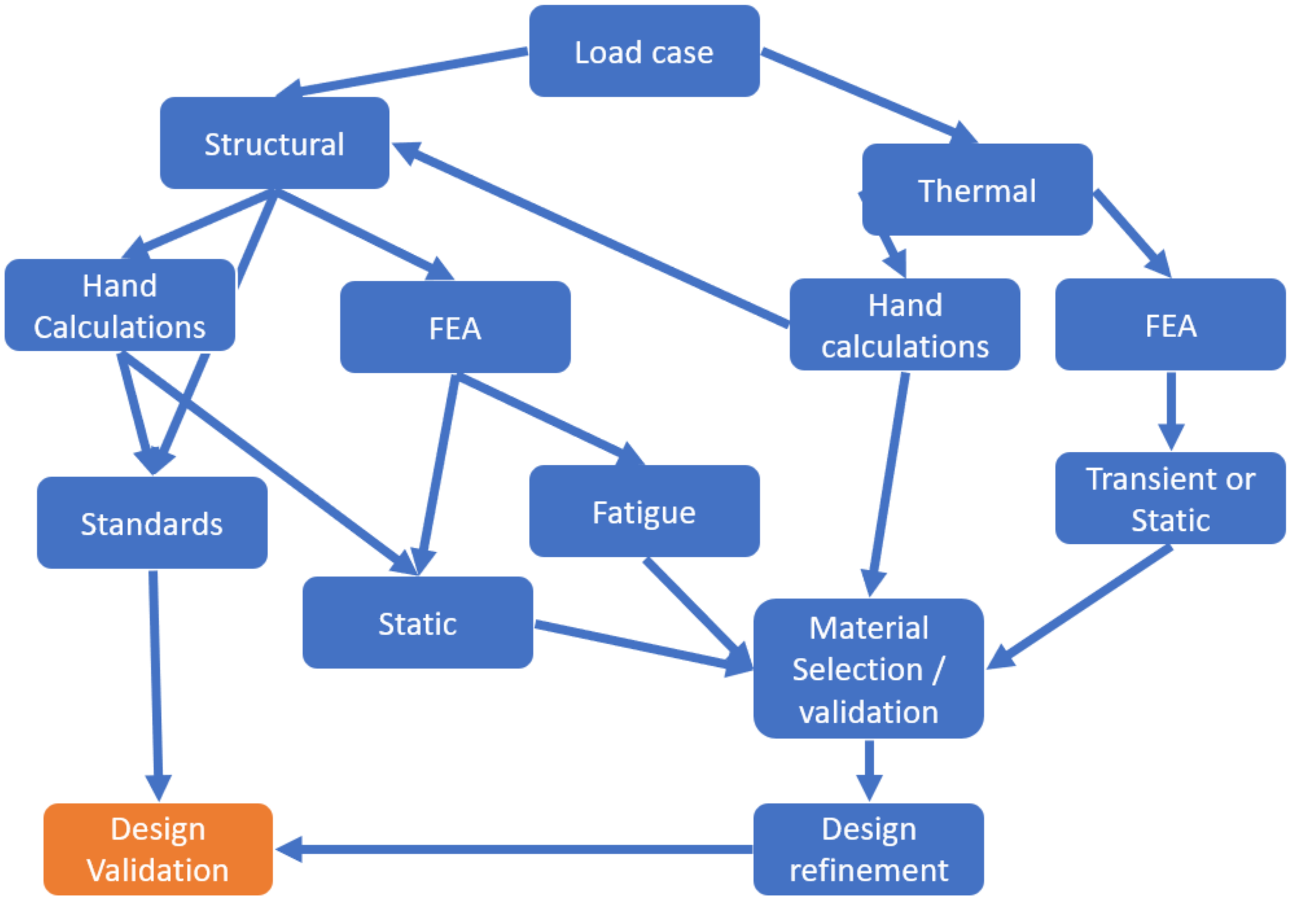


Figure 8 - Analysis methodology relationships.

The rear brake calliper must perform its role of converting kinetic energy to heat as a sub-system of the car's brake system. The two overarching analysis types performed were structural analysis and thermal analysis, each consisting of hand calculations, justifications and FEA modelling. Structural analysis was used to determine the ability for each component to withstand failure considering the worst-case loading scenario foreseen in its use and service lifetime. Thermal analysis allowed heat management characteristics of the design to be quantitatively analysed and verified as all thermal considerations made previously were on a qualitative, best-judgement basis. Figure 8 illustrates the relationships between each analysis methodology and its applicability to the brake calliper system. The calliper components were partitioned into four analysis sub-sections; the calliper body, which is the main component wholly designed in this project (including cross-over tube as an extension of the body), the bolt groups, pads and pistons, and full-system thermal performance. This partitioning is illustrated in figure 9, with each analysed component shown as an individual part. Excluded from the analysis was the R-clip, banjo assembly, bleed port and piston seal. Aside from the piston seal, these items were standardised, and selection was verified only by compatibility based on manufacturer specifications.

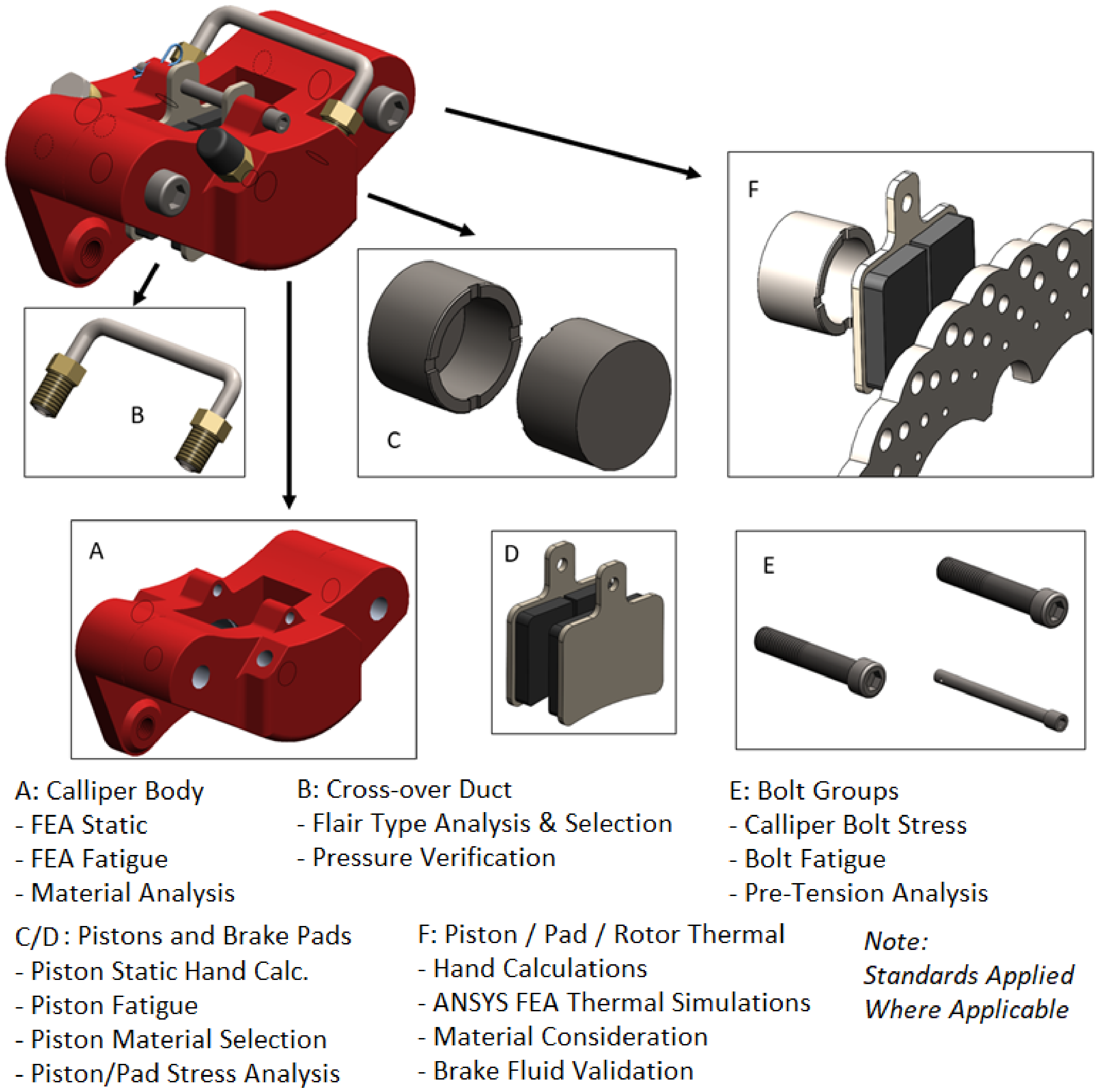


Figure 9 - Partitioning of component analysis and interactions.

Both calliper body halves make up the body as a component which was isolated as shown in item A in figure 9. A static worst-case load scenario was assumed for the static analysis of the body as this allowed the associated stresses and deflection to be evaluated. From the results, structural integrity could be evaluated, and a material was selected based on compatibility with the stress among additional requirements. Deflection was of interest as this is known to affect the feeling of "stiffness" fed back to the driver. Lower deflection was desirable. Fatigue analysis was then undertaken assuming a reasoned peak practical loading case (Appendix B) to validate that the body could survive a prescribed lifetime. This was important to ensuring reliability of this integral structural member and the analysis concluded this was achieved. An extension to the calliper body in terms of fluid routing, the cross-over tube design was analysed and validated to ensure compliance with expected pressures. This involved primarily simple numerical justifications as it is an off-the-shelf item.

The two bolt groups are the calliper bolts (figure 9, E) and the mounting bolts (not pictured) which are critical to supporting operational loads. While considered an off-the-shelf item that is bound by the design of the vehicle uprights, the mounting bolts were analysed for required pre-tension as validation of this parameter would mean that frictional support and nominal stresses could be maintained with confidence. The calliper bolts were also analysed for pre-tension requirements by means of hand calculations and static FEA; but fatigue analysis was also applied to ensure the bolts could withstand the fluctuating component of tension induced by clamping pressure. The bolt sizing and grade was validated for the design. An addition to bolt analysis, the pad retaining pin design was validated despite its loading being non-load-bearing.

The brake pad and piston were analysed under one task for structural integrity. This began with structural analysis of the piston, a project-designed component which was undertaken to validate the geometry of the component with respect to the maximum static load case. Hand calculations were done to establish and validate the baseline loads and stresses followed by numerical analysis via FEA in ANSYS. This technique allowed simulation of the full component geometry with boundary conditions emulating the real-world constraints on the piston. After initial analysis, modifications were suggested if/where required to reduce stress concentration and possibility of failure. Prior to conducting fatigue analysis, a material selection study was undertaken to select an appropriate material for the piston given the calculated static stresses and thermal considerations. Fatigue analysis was then conducted on the piston using the constrained FEA model. This aimed to verify the performance of the piston and its ability to resist fatigue failure as a primary mechanism. Finally, the pad/piston interaction was analysed using FEA to simulate the pressure profile imparted to the brake pad when pressure is applied by the driver. By evaluating the pressure distribution on the pad friction material face, conclusions could be drawn regarding evenness of wear of the brake pad. A larger pressure “footprint” was thought to provoke more even pad wear. The brake pad was not itself analysed for structure as the use case was assumed to sit within the manufacturer design specifications.

The final analysis undertaken involved detailed thermal analysis of the brake calliper body, piston, rotor, and pad with the aim of establishing capability of the calliper to remain at a suitable working temperature during high-load braking scenarios. Such scenarios when large amounts of heat are generated include the FSAE endurance event where repeated and sustained braking occurs. A worst-case scenario heat generation case was assumed, whereby the car would stop from 27 m/s at the highest possible rate. It was also assumed that the primary path for heat flow was from the brake rotor to the brake pad, to the piston and into the fluid and calliper body. The main aim of keeping the calliper cool was to verify it could maintain a temperature below the vaporisation point of the brake fluid. Failure to do so would result in “spongey” brake feel and poor stopping performance. Hand calculations were used to form a basic model of heat flux from the braking surface into interfacing components and finally the fluid. This was followed by FEA thermal simulations using ANSYS Static Thermal to verify the heat performance over the immediate timeframe. Validation of material suitability and limitations was discussed as part of the analysis based on perceived operating temperatures seen in worst-case heat loads.

4.1 Calliper Body Analysis (William Whigham, n10232869)

Analysis of the calliper body assumed both halves (two components) as they are essentially one entity in the design. The first analysis on the calliper body considered the maximum static load as summarised in section 2.1 and Appendix B. There are two primary loads and one support load that were considered in this analysis; fluid pressure and pad thrust force and support force from the mounting bolts. The highest static pressure case was assumed to occur at multiple yet infrequent instances in the service life of the brake calliper but would likely occur less than 100 times over the life of the vehicle/system as it is in no way a practical or effective use of the brakes. In most cases, the driver uses enough pressure to be near or on the edge of maximum traction as this provides optimal performance on-track. The magnitude of this pressure was calculated to be approximately 2.5 MPa. An exception to this is a FSAE event in which the braking ability of the car is evaluated by driving in a straight line and locking up all four wheels [6]. The maximum thrust force from the pads was a frequent occurrence as ideally the driver aims to use braking force at the tractive limits. According to the calculation in Appendix B, the magnitude of this force is 757 N per brake pad, or 1514 N in total. The allocation of these loads is communicated in Figure 10.

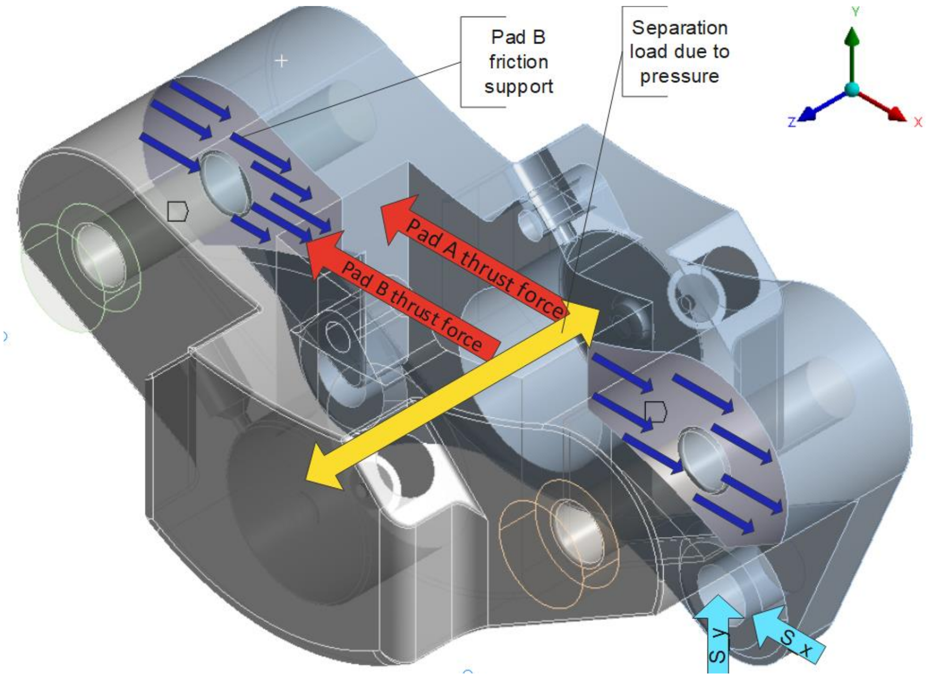


Figure 10 - Load diagram for calliper body structural analysis.

Due to the complexity of this component, hand calculations were limited to the determination of reasoned maximum loads. These calculations and parameters are detailed in Appendix B. Following a search for applicable standards, it was concluded that application of a standardised calculation was not practical, nor was there applicable standards available for FSAE brake calliper design. The following assumptions and simplifications were made for a finite element model (FEM/FEA) in ANSYS Workbench used for stress analysis of the calliper body:

- Only loads directly resultant from nominal brake operation while driving was considered. Thus, all net lateral loads (Z-direction) are zero.
- The calliper bolts, as an interacting component, were used to constrain the body halves as this would provide simpler and more accurate simulation and results respectively. It should be noted that stress analysis of the bolts themselves is outside the scope of this section.
- To simplify the model, it was assumed that the pad thrust force acted over the whole area of the support face, although it would be concentrated by the pad edge in practice.

- The pad retaining pin, as a non-structural component, has negligible effect on calliper body stress and strain characteristics and was excluded from the model.
- Weight loads of the calliper body and interfacing components is negligible.
- Friction force associated with the pistons and piston seals is negligible.

The bolt pre-tension required to maintain frictional support between the two calliper halves under maximum braking load was found using a factor of safety of 1.2. This was needed here for an approximate bolt pre-tension parameter in the FEM and is detailed in the following calculation. A friction coefficient between two machined aluminium faces was assumed as 1.05 based on a text book value [10]. Considering two bolts split the load from the outer brake pad equally, the support load from each connection is ~ 380 N in the x-axis (frictional) and 2036 N in the z-axis (separational). The bolt tension needed was found to be 3000 N.

Force required for frictional support component: $F_{fSupport} = 380 \times 1.05 = 400\ N$

Bolt force required: $F_{TBolt} = 1.2 \times (400 + 2036) = 3000\ N$

4.1.1 FEA Model Development

The FEA model used a simplified geometry to improve efficiency and reduce complexity of the model to the most important features. Simplifications included incorporating minimal geometries for the two bolts and removal of the seal groove. Applied force/pressure and support loads applied to the model are detailed in Figure 11. The cylindrical support was used on the bolt hole, bearing the assumption of worst-case scenario where frictional support between the calliper and upright fails. The following list of contacts were applied to the model bodies to simulate their real interactions:

1. Frictional contact between the two calliper halves, with μ = 1.05
2. Frictional contact between the bolt head and calliper body (steel-aluminium) with μ = 0.5 [10]
3. Frictionless contact between non-threaded portion of bolt and calliper body bolt hole
4. Bonded contact where thread section of bolt interacts with female threads inside calliper half.

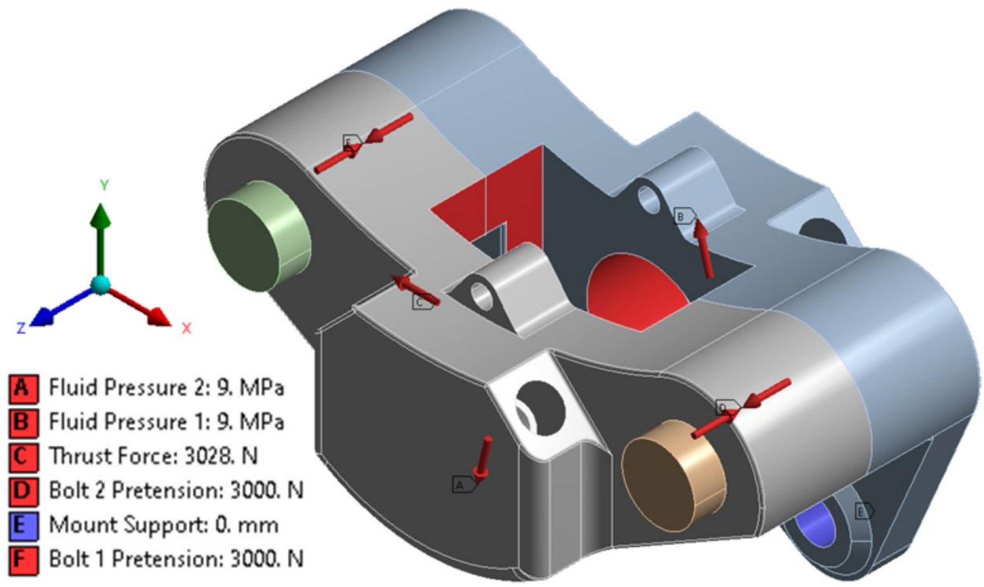


Figure 11 - Summary of loads applied in FEA model.

While a material was not final at this stage of analysis, an average Young's modulus of aluminium at 71 GPa was applied to the model and assumed to be a sufficient approximation for the final material [11]. This was done because the measure of deflection (and its effect on stress) was of interest in static analysis, thus the values were able to be considered a valid approximation of real-world conditions. A mesh convergence study resulted in a model that converged with the mesh parameters listed in table 4. It was found that mesh refinement was necessary at the point where the bolt and calliper halves touch and at the top edge of the calliper body where the two halves mate.

Table 4 - Calliper body FEA solution parameters.

Mesh / Solution Parameter	Value
Solve time	25 m 42 s
Global element size	0.7 mm
Number of nodes	548 609
Number of elements	323 247
Element skewness avg.	0.4524
Element quality avg.	0.6924

One stress concentration was found to be at the top of the calliper halves where compressive load results from the moment generated about the bolt axis as shown in figure 12. The maximum stress on the body was found to be 211 MPa on the outer half concentrated at the bolt hole edge location magnified in Figure 13. Without the presence of the fillet shown in figure 13, a high stress was seen here. This is due to the combined moment-tensile force on this bolt hole. The maximum directional displacement of the calliper was found to be 0.4 mm as shown in figure 14; a value comparable to that reported by multiple authors previously cited in the literature [12, 13]. A maximum stress value at ~ 500 MPa was detected on the dummy bolt; however, the bolts were not considered within the scope of this section of the analysis. Other stress areas of interest were the edge of the piston bore depth, the transition from the backing to the main body and the top edge of each calliper half. The magnitude of the peak stresses is relatively low and material selection was done based on this analysis before completing a fatigue simulation.

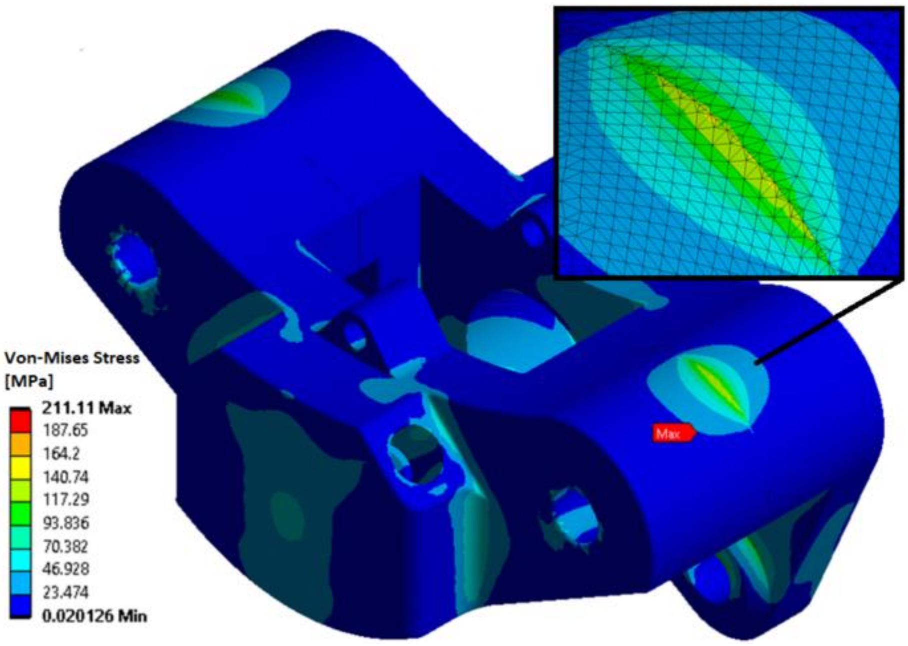


Figure 12 - Static stress contour plot of calliper body.

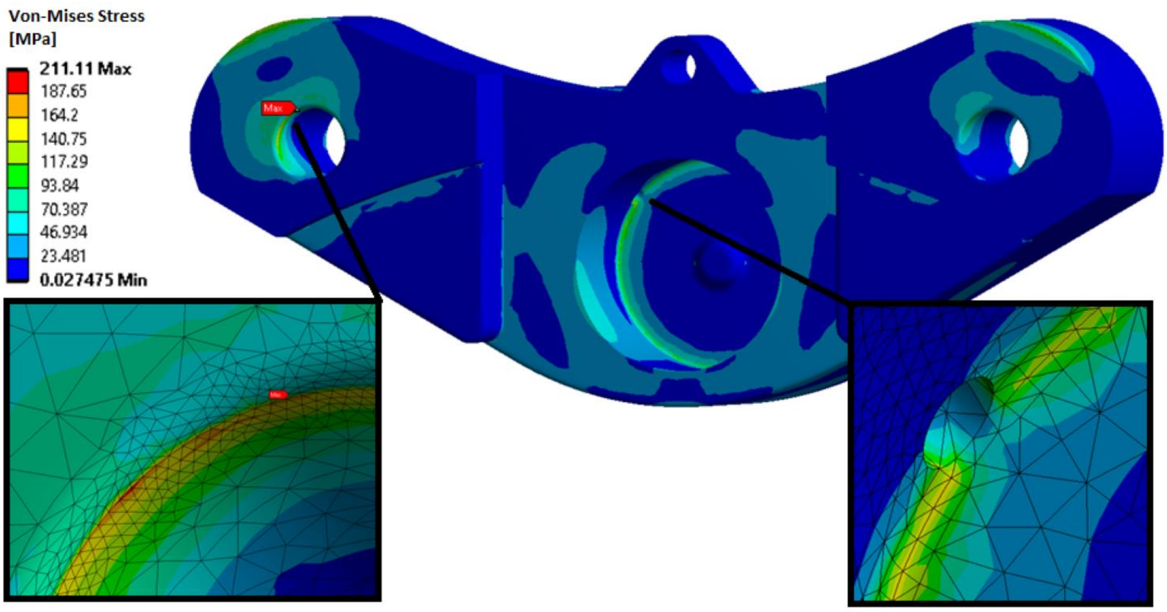


Figure 13 - Static stress contour plot of calliper body.

From the results it was concluded that the geometry of the calliper body was sufficient to withstand the static load cases it would be subjected to. This conclusion is subject to the outcomes from the fatigue study. At a maximum of 0.41 mm on the outboard edge of the calliper, the maximum static deflection is comparable to values reported by FSAE calliper designs reported in the literature which range from 0.17-0.75 mm [13, 14]. It was, however, identified that stiffness of the calliper may be improved if additional material is added to the transition between the extruded piston bore housing and the centre-chassis section. This may increase stiffness although additional weight would be the trade-off. An observation was made that a slight twisting motion can be detected due to the moment created about the mount point. However, the small deflection values led this to be concluded as having negligible effect. In summary, the level of observed stress was considered low enough for suitable material FOS against static failure.

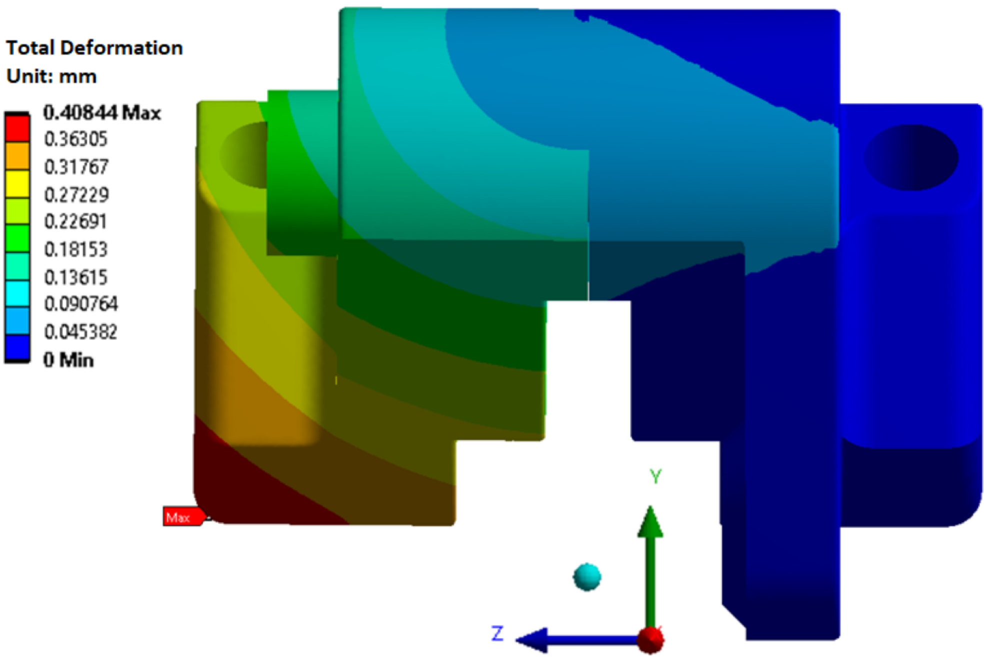


Figure 14 - Total deformation contour of calliper body simulation.

4.1.2 Material Selection

A candidate material for the brake calliper body should have an optimal strength-to-weight ratio, high machineability and maintain its strength at peak operation temperatures. In a material selection case study, Ashby [15] considered conductivity and relative mass to be key selection parameters. If both properties are equally weighted, such as in this case where light weight is desirable but cost is equally as important, then aluminium alloys are the best choice. The Ashby map in figure 15 illustrates the proximity of aluminium to other metals as stated by Ashby [15]. Additional requirement for the material to operate at temperatures up to 150°C was considered as it was assumed based on findings in the literature that this temperature could be reached in the calliper body under extreme conditions [16, 17]. Three candidate alloy grades were selected for comparison which is detailed in table 5. Primarily due to the provision of high temperature tensile strength, but also considering low mass and cost characteristics, 7075 series T6 aluminium was selected. It was therefore concluded that, using this material, the static analysis shows a factor of safety of 1.8

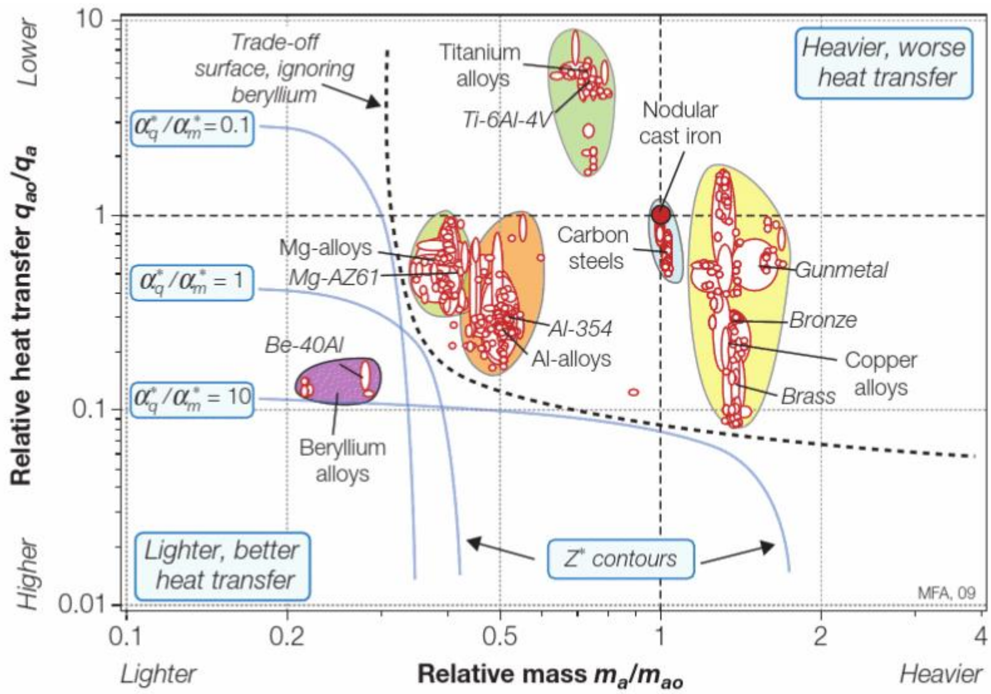


Figure 15 - Ashby chart showing material comparison for brake calliper case study [15].

Table 5 - Material selection summary for the calliper body.

Material Grade	YS at room temp	YS at 150°C	Mass/vol [kg/m^3]	Rank	Reference
7075 T6	570 MPa	380 MPa	2800	1	[18]
2014A T6	410 MPa	380 MPa	2780	2	[19]
A354 T6	294 MPa	234 MPa	2735	3	[20]

4.1.3 Fatigue Study

It was decided that the brake calliper body should be designed to survive fatigue loading up to 5×10^5 cycles given the peak braking conditions. As it is not a commercial brake calliper and the QEV3 car undergoes limited use, the fatigue life does not need to exceed a moderate/high value as designing for a higher fatigue life would only decrease short-term gain and make the system unnecessarily heavy. The load case for fatigue was zero-based (R = 0) as the force load was assumed zero/negligible when the brake is not engaged by the driver. The maximum thrust force was kept at 3028 N while the maximum pressure load was reduced to a practical value of 5 MPa corresponding to the required pressure to achieve wheel lockup, with a factor of safety of 2 applied to the load value. Under this regime, it was assumed that any FOS above 1.2 would be sufficient to give the design adequate fatigue life. An S-N curve with R = 0 from Kumar et al. [21] was used for the FEA fatigue analysis and its values are shown in figure 16.

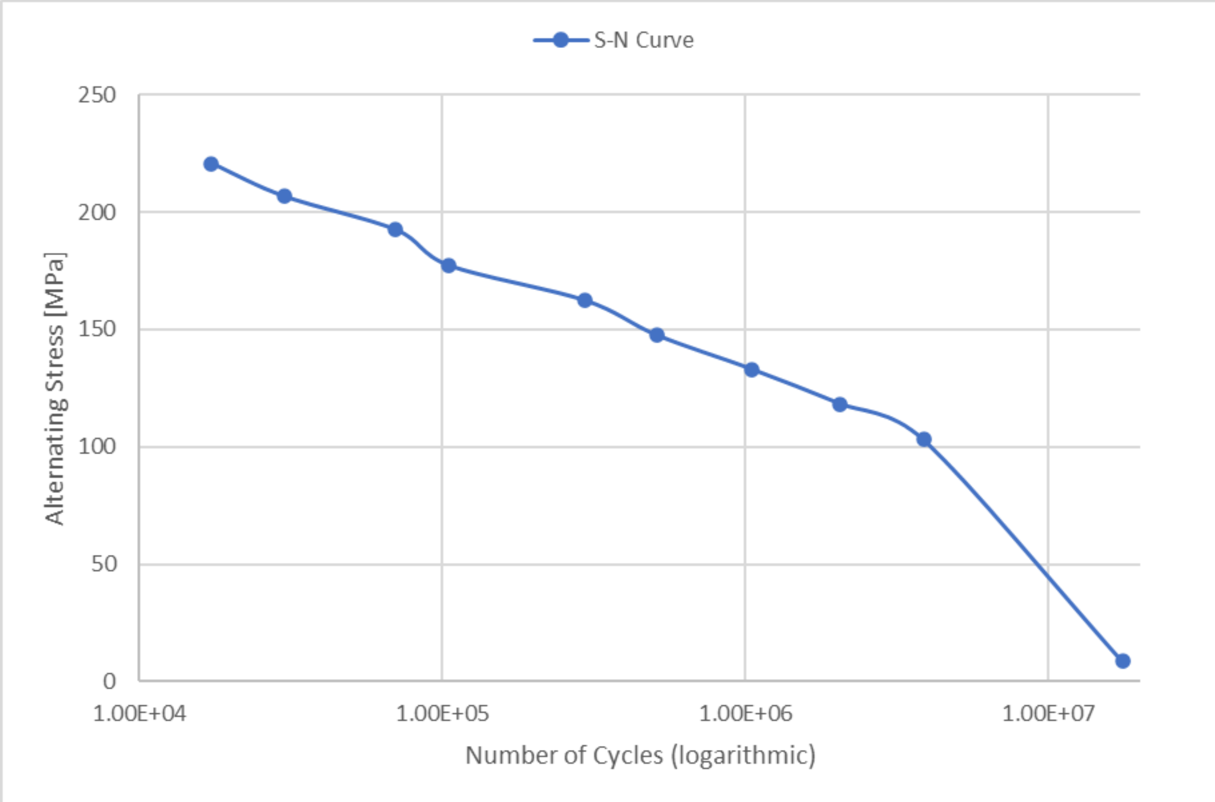


Figure 16 - S-N data for 7075 T6 alloy from Kumar et al [21].

Results from the fatigue simulation show a minimum safety factor of 1.7 to be present at the surface interface between the bolt and calliper half as indicated in figure 17. This FOS was considered acceptable for the purpose of this analysis and provides a high confidence that the service life of the calliper body would exceed the requirements of the car and the QUTMS team. It should be noted that most reported FSAE specific calliper designs were not simulated for fatigue loading, thus, this analysis step was above what is generally accepted for such a system [13, 16]. The validity of the fatigue model was supported by expectations extrapolated from the static model in terms of the FOS; however, it was interesting to see the lowest FOS was at a different location to the maximum static stress. It was thought that the lower pressure load was responsible for this as the thrust force from the pads forms a higher proportion of the overall load. Another observation made with respect to the fatigue results was the singularity FOS contours shown on the outboard half in figure 17. The presence of these apparent stress concentrations was not detected in the static simulation. It was concluded that the most likely source of this result was the bonded contact between the bolt head and calliper body producing non-ideal deformation. As such, this contour feature was ignored in the critical results.

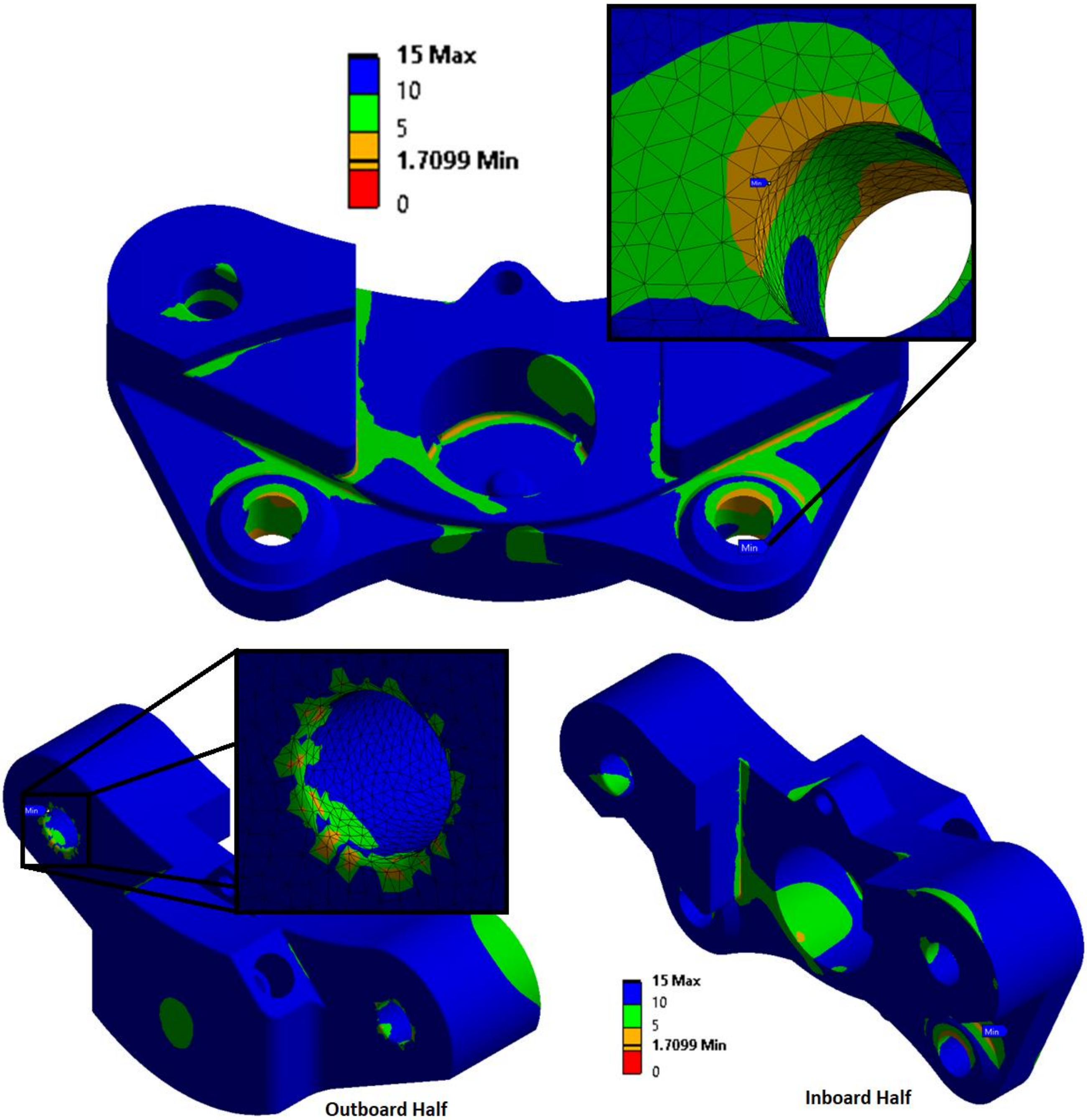


Figure 17 - Fatigue FOS contour plots for 5x10^5 cycles.

The static and fatigue analysis done here was concluded to be valid evidence to suggest the calliper body suitably meets the goals of the design. Although it could be argued that the calliper is over-built, in terms of structural strength and longevity, it was decided that additional removal of material to save weight would lead to increased production cost for diminishing return. Since all design criteria were deemed adequately met by the design, no further refinement was required and further changes were considered out of the scope of future work.

4.1.4 Cross-over Tube Design

The fluid cross-over tube and its associated fittings were required to have a pressure operating maximum above the maximum design pressure of the brake system in QEV3. The tubing selected for this item was standard 3/16-inch (outer diameter) AISI 304 stainless steel fluid line with a wall thickness of 0.71 mm which is required by SAE standard J3129_2019 to have a material yield strength of at least 205 MPa and ultimate tensile strength of 515 MPa (at room temperature) [22]. This corresponds to a maximum operating fluid pressure of 62 MPa as calculated using the following equation. The tubing is commonplace in commercial brake routing for cars, adding to confidence in the ability of this component to effectively perform its function [22].

$$P = \frac{2tS_y}{D} = \frac{2 \times 0.71 \times 10^{-3} \times 205 \times 10^{6}}{4.7 \times 10^{-3}} = 62\ MPa$$

Tubing is received in long lengths which must be cut, bent to shape and then flared using a purpose-made flaring tool. Of the three standard types of flare that can be used with this type of hard brake line, as shown in figure 18, a SAE single “bubble” flare was chosen, as this style provides a reliable fluid seal with crush-fittings compatible with the calliper body size constraints. The fittings were designed in accordance with SAE standard J512_2017 for automotive tubing fittings [5]. The standard outlines key dimensions of the brass nuts including the thread length, set at a minimum of 7 mm. A nut torque of 13.6 Nm was specified for tightness as per J512 also. The brass crush-fittings are shown in figure 19, and an expected burst pressure in the vicinity of 100 MPa is expected of the cross-over tube seal with these fittings. As such, the selection of parts for the cross-over tube was validated with adequate confidence. This assumes the quality of finish on the calliper body and off-the-shelf components can meet the design performance.

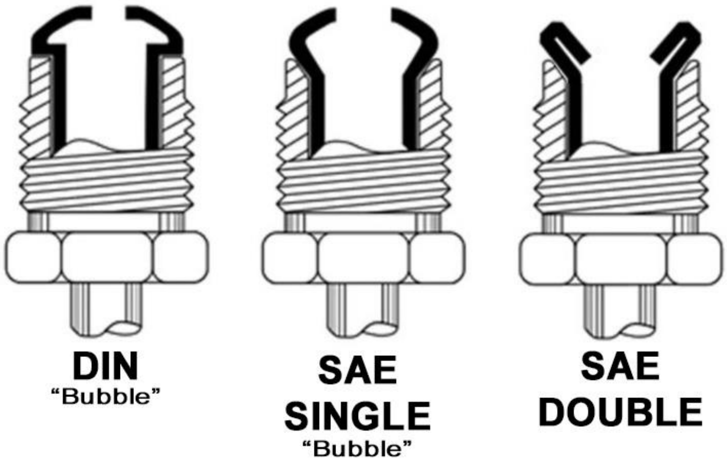


Figure 18 - Three standard flare types used for hard brake line fittings.

Figure 19 - Brass nuts for (left) SAE single bubble flare, and (right) DIN bubble flare.

4.2 Calliper Bolts, Mounting Bolts & Pad Retaining Pin (Jacob Garcia-Pavy, n10012478)

The calliper bolts have been considered the sole structural component which restrains the two-piece calliper halves together inducing significant loads. To ensure the function and safety of the brake calliper a thorough analysis must be conducted. The calliper bolts require manufacturing for proposed specificity and advantages to the system. This analysis will address bolt specifications regarding relevant standards and FSAE regulations. The analysis will include hand calculations to validate FEA simulations and facilitate selection of the most appropriate and favourable material. Which will be analysed under the same validated FEA simulation to ensure high confidence of the selected calliper bolt. Moreover, this section will analyse several other components such as the guiding pin and clip and mounting bolts. Shorthand calculations regarding FOS for the off-the-shelf products being the clip and mounting bolts will be performed to ensure safe operation. While an appropriate material will be selected for the manufactured guiding pin by determining the required material strength through hand calculations.

4.2.1 Calliper Bolts

The calliper bolt design uses a hexagonal head complying with the FSAE requirement outlined in section T.8.2.2 of FSAE rules 2021 [6]. While the required 8mm diameter utilises an M8 thread a preferred thread in ISO standards. The bolt's nominal length is 50mm which allows a minimum of two threads projecting as required in section T.8.2.4. The initial material used in the proceeding analysis is mild steel due to the quantity and quality of the respective standards. A fine thread pitch was chosen to increase strength and reduce the likelihood of loosening. Consequently, the bolt's nominal area is 39.2mm according to ISO 898-1 [23]. With a nominal diameter between 3mm and 39mm an 8.8 grade is allowable complying with a minimum of one of the following grades outlined in section T.8.2 of ISO 4014 [24]. Overall, a hexagon head M8 x 1.00mm x 50mm mild steel 8.8 grade was used to represent the initial calliper bolt which complies with all fastener regulations and is consistent with ISO and Australian standards. Moreover, due to the calliper bolts being self-manufactured having cut threads using a lathe the stress coefficient is 3.8 [25]. The following mechanical properties were specified in ISO 898-1 [23].

$$S_P = 600MPa, S_Y = 660MPa, S_{UT} = 830MPa$$

$$A_t = 39.2mm \,\&\, K_f = 3.8$$

The calliper bolt must maintain frictional support between the two calliper halves to prevent slippage and shear stress acting on the calliper bolts. The minimum required pre-tension was determined using a referenced textbook friction coefficient between the two machined aluminium faces of 1.05 [7]. In accordance with section 4.1 and Appendix B the support load calculated is approximately 380N in the x-axis. Thus, producing a Frictional support force of 400N.

$$F_{fSupport} = 380 \times 1.05 = 400\ N$$

In addition, the calliper undergoes significant axial loading during braking. Under maximum braking conditions it was concluded that the axial load produced by the moment induced by the brake pad was F_A, 2036N in the z-axis. The required pretension to maintain frictional support under this tensile load is calculated below.

$$F_A = 2036\ N$$

$$F_i \geq (F_{fSupport} + F_A)N$$

$$F_i \approx 3000N$$

4.2.1.1 HAND CALCULATIONS

Simple hand calculations were deduced to validate FEA simulation and to initialise bolt parameters. A static FOS was calculated with the maximum load being the specified pretension load $F_i \approx 3000N$ required by the bolt with no load deduction created by any braking. The calculations below display the static case rearranging the von mise's formula with previously discussed parameters calculating the FOS being equal/less than 2.27. Therefore, the 8.8 grade M8 x 1.25 x 50mm bolt is more than sufficient to handle the subjected loads.

$$\sigma' \leq \frac{S_y}{N}$$

$$\sigma' = \frac{F}{A_t} K_f = \frac{3000}{39.2} 3.8$$

$$\sigma' = 290.82 \ MPa$$

$$N \leq \frac{S_y}{\sigma'} \leq \frac{660}{290.82}$$

$$N \leq 2.27$$

Moreover, a fatigue FOS was calculated using both yield and goodman. The calculations are seen below in table 6. The required life cycle of the bolt should concur with the value suggested in section 4.1 being 5x10e+5 cycles. As the brake calliper has been designed for the QEV3 and will likely be replaced within the next design. However, the calliper bolt will be analytically modelled to last infinitely as the hand calculations will only guide FEA simulations. Which will thoroughly explore appropriate material for the calliper bolt. The yield equation produced the same FOS as the above static case of 2.27. While the goodman equation produced a critical FOS of 1.47 being quite low due to an inflated infinite lifecycle and significant alternating and mean stresses.

Table 6 - Endurance limit parameters of bolt [25].

Minimum Load	Maximum Load
$\mathbf{F_{min} = F_{PreLoad} - F_A}$ $\mathbf{F_{min} = 400N}$ $\boldsymbol{\sigma_{min} = \frac{400N}{39.2mm} 3.8}$ $\boldsymbol{\sigma_{min} = 38.776MPa}$	$F_{max} = 3000N$ $\sigma_{max} = \frac{3000N}{39.2mm} 3.8$ $\sigma_{max} = 290.816MPa$
$\boldsymbol{\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}}$ $\boldsymbol{\sigma_a = 126.02MPa}$	$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$ $\sigma_m = 164.796MPa$

$$S_n = S_n{}' C'$$

$\boldsymbol{S_n{}' = .5S_{UT}}$ $\boldsymbol{(infinite\ life\ cycle)}$ $\boldsymbol{S_n{}' = 415MPa}$	$\boldsymbol{C' = C_L C_S C_T C_G C_R}$ $\boldsymbol{C_L = 1\ (von\ mises)}$ $\boldsymbol{C_S = 1\ (assuming\ commercial\ surface\ finish)}$ $\boldsymbol{C_T = 1\ (assuming\ calliper\ bolts < 450\ celsius)}$ $\boldsymbol{C_G = .9\ (due\ to\ axial\ loading)}$ $\boldsymbol{C_R = .702\ (99.99\%\ reliability)}$

$$S_n = 262.197 \ MPa$$

Goodman Line	Yield Line
$\frac{\boldsymbol{\sigma_a}}{\boldsymbol{S_n}} + \frac{\boldsymbol{\sigma_m}}{\boldsymbol{S_{UT}}} = \frac{\mathbf{1}}{\boldsymbol{N}}$ $\left(\frac{\boldsymbol{\sigma_a}}{\boldsymbol{S_n}} + \frac{\boldsymbol{\sigma_m}}{\boldsymbol{S_{UT}}}\right)^{-\mathbf{1}} = \boldsymbol{N}$ $\boldsymbol{N} = \mathbf{1.47}$	$\frac{\sigma_a}{S_Y} + \frac{\sigma_m}{S_Y} = \frac{1}{N}$ $\left(\frac{\sigma_a}{S_Y} + \frac{\sigma_m}{S_Y}\right)^{-1} = N$ $N = 2.27$

Despite the low critical FOS calculated using goodman the hand calculations suggest the M8 x 1.25 x 50mm 8.8 mild steel bolt will outlast the calliper body being designed for 5x10e+5. However, more favourable materials exist. Mild steel with the calculations can be used as a reference value to facilitate material selection.

4.2.1.2 Material Selection

Appropriate material selection depends on several parameters utilised throughout the project. Firstly, weight has been a priority primarily within the design's developing stages. Weight removal reduces vehicle unsprung mass with performance advantages extending beyond acceleration of the car. Secondly, cost has been thoroughly considered throughout the design due to the limited funds available to the QUTM team. Thirdly, manufacturing has been carefully considered in later stages of the design's development being rigorously analysed with the DFM and DFA section in report 2. Material strength must be maintained or improved as mild steel noticeably produced a relatively low FOS under infinite lifecycles. To compare material candidates Ashby charts were used relating to the specified parameters. Figure 20 represent alloys and metals regarding their strength to density being weight. Several materials appear to be stronger than mild steel. Top tier Titanium has the most favourable density sharing the lowest weight while being considerably stronger. However, Figure 21 shows titanium as one of the most expensive metals being difficult to manufacture, while being extremely complex and energy intensive. Titanium alloy Ti-6Al-4V was selected as the other stronger materials also have increased costs and densities. This particular titanium alloy is readily available.

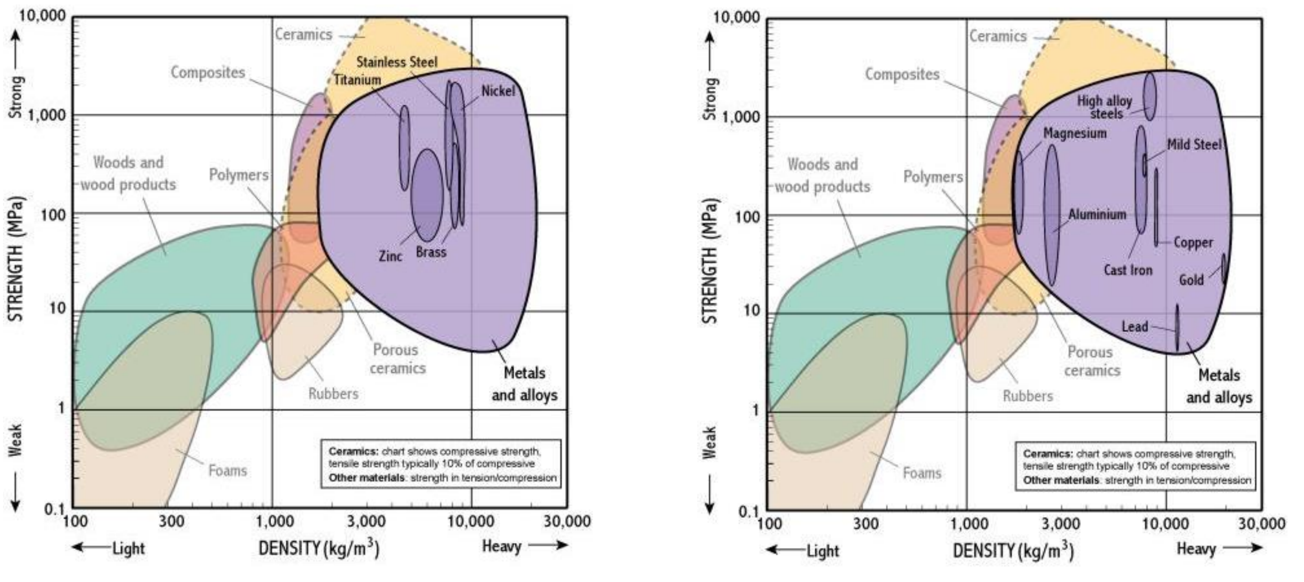


Figure 20 - Metal alloy strength vs. density Ashby chart [26].

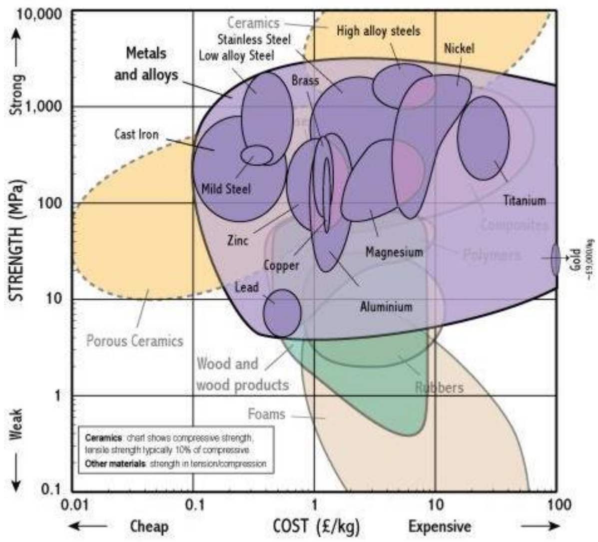


Figure 21 - Metal alloy and strength vs. cost, Ashby chart [26].

4.2.1.3 FEA

The hand calculations provide an appropriate foundation to validate the FEA simulations with the same selected hand calculation material. Which will allow a confident analyse of selected titanium bolt. The FEA utilises several simplifications and assumptions to model the system, however, conveys significant limitations.

- Clamping force on the calliper was not investigated being outside the scope.
- No shear stress. Calliper bolt weight is negligible. The calculated pretension maintains frictional support preventing slippage and shear loads on the bolt.
- The calliper bolts do not experience significant temperatures which could alter properties and performance. The proceeding thermal analysis indicates significant temperatures at the brake pad and connecting components.
- The thread was not implemented within the geometry due to design and computation complexity.
- Instead, the bolt thread was represented through a bolt thread contact within the setup. With a pitch diameter, distance, and angle respectively of 7.5mm, 1mm and 60 degrees gathered from ISO 4014 [24].
- There is no separation of the threads which has been specified with the bolt thread contact. Which realistically is impossible due loosening. Which can be caused by improper tightening, initial loosen of around 10% preload, and through vibrations and temperature fluctuations.

The geometry consists of three separate bodies which include the bolt's head, body and the inboard calliper's thread seen in figure 22. However, as mentioned the threads were not implemented. The bolt's body contained several sharp stress concentrations and utilised a fillet of 0.4 mm and chamfer of 2 mm in compliance with ISO 4014 [24]. The inboard calliper was arbitrarily modelled principally sharing the same internal diameter. The bolt's head and body were separated to facilitate the meshing process. An automatic method was implemented on the head due to the simplicity of the shape. While a tetrahedral mesh with an element size of 1mm was utilised on the body due to the complexity of the shape. Overall, generating a quality mesh seen in figure 23 which displays the element quality. The element quality is quite high and uniformed across the geometry however is poor within the fillet of the calliper bolt which was concluded insignificant as the threads experienced the greatest stresses. The mesh was proven to converge at 22816 elements.

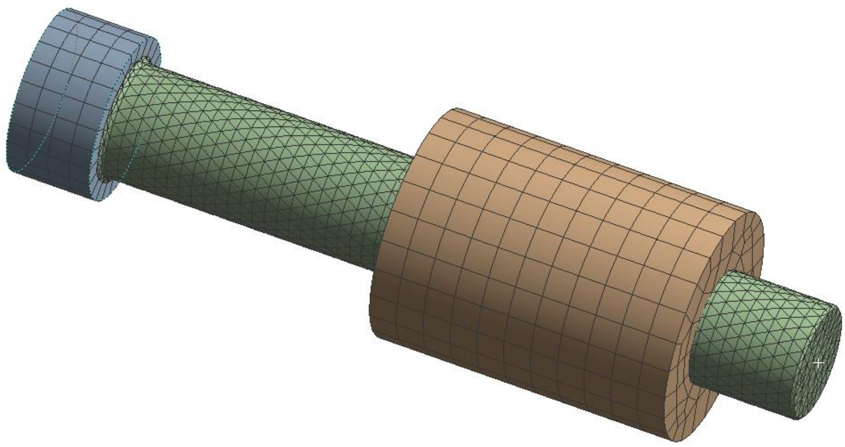


Figure 22 - Calliper bolt geometry and mesh.

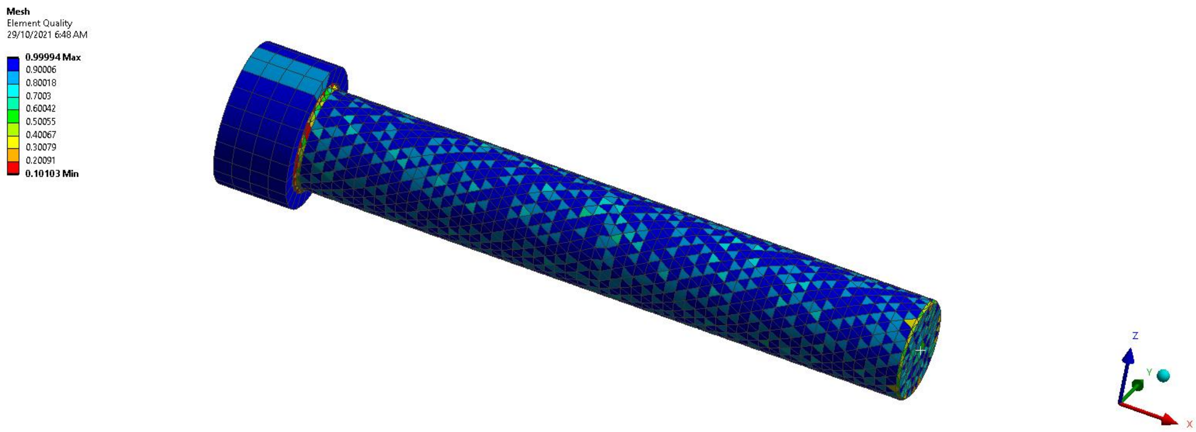


Figure 23 - Mesh element quality.

The following boundary conditions were applied and be seen in figure 24;

- A pretension load, A being 3000N was placed on the body which required a separate coordinate system for correct orientation.
- A tensile force, B being 2036N was placed on one end of the calliper bolt.
- A fixed support was placing on the ends of the calliper bolt.
- A Bond contact was specified at the calliper bolt’s head and body to connect into one assembly.
- A thread bond contact was used on the bolt’s body and inside of the calliper ‘thread’ to represent a generated thread.

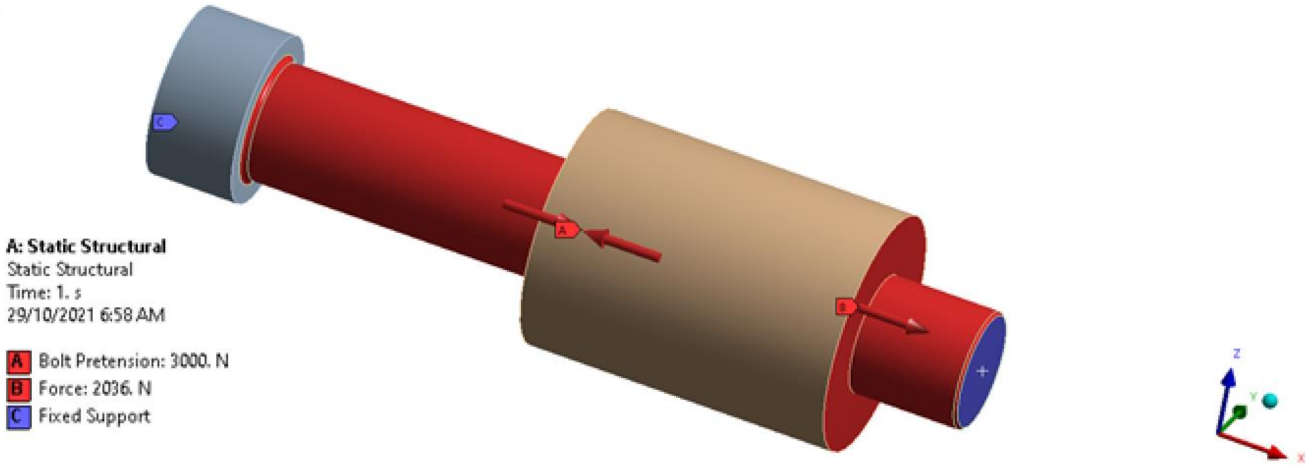


Figure 24 - Calliper bolt boundary conditions.

4.2.1.3.1 FEA Mild Steel

The FEA concluded a maximum stress of 290.76 MPa displayed in figure 25 which concurs with hand calculations with a difference of 0.056 MPa and an error percentage of 0.02%. However, figure 25 displays obvious simulation limitations. Firstly, there are significant portions of the bolt being insignificantly and unrealistically loaded this is likely due to the applied thread contact region. Secondly, the critical stress occurs within a small portion of the bolt most likely being the computational start of the thread. It was hypothesised that a stress pattern would emerge surrounding the thread. Overall, the simulation is not an accurate representation of experience forces on the bolt. However, accurately calculates stress experienced by the thread which many geometries can be reproduce.

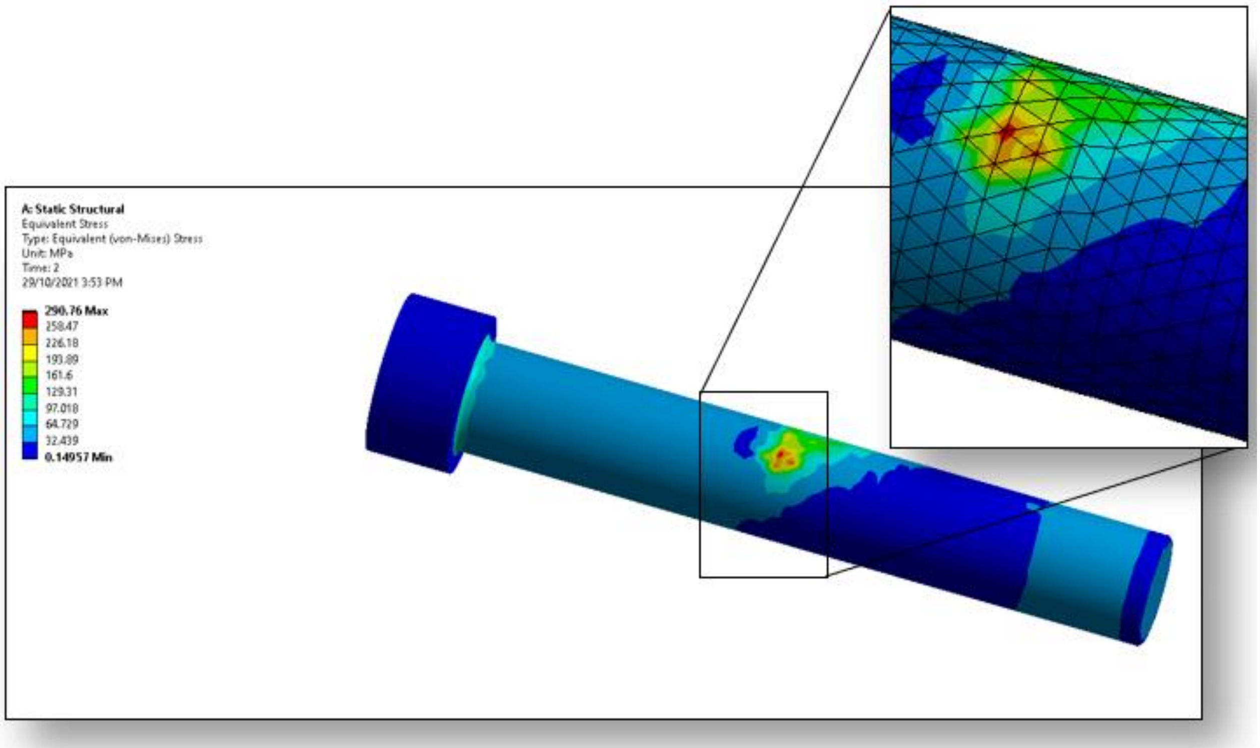


Figure 25 - Stress contour of calliper bolt Mild steel.

4.2.1.3.2 FEA Titanium Bolt Stress

The FEA simulation has been validated through hand calculations and produces quality results regarding initial thread engagement. However, has significant limitation regarding the entire calliper bolt which has been deemed unnecessary. Several similar studies establish the importance of the bolt's thread over the shank. A study was conduct on the load and stress distribution of thread pair which concluded the ultimate stresses on the bolt was experienced at the initial engagement [27]. However, the study by Lu et al. [27] showed significantly improved stress distribution throughout the bolt and particularly the thread. Despite this, the simulation shows the most significant point of interest, thus, it was used to analyse the bolt. The FEA simulation using Titanium Ti-6Al-4V found a maximum shear stress of 297.56 MPa at the predicted initial thread engagement seen in figure 26. Although not specified within the geometry the unseen computational thread contact applied most likely starts at this maximum shear stress. Figure 28 illustrates a critical FOS of 2.62 at the initial thread engagement while majority of the calliper bolt is significantly higher. Therefore, the FOS is more than required likely lasting beyond its operation life. Furthermore, figure 27 displays insignificant deformation with a maximum of deformation length of .0018 experience at the initial thread. Which is supported by the previous calculations and with the mentioned study. Overall, the Ti-6Al-4V alloy is an appropriate material for the M8 x 1.25 x 50mm calliper bolts to handle the prescribed loading conditions while offering favourable weight and performance advantages.

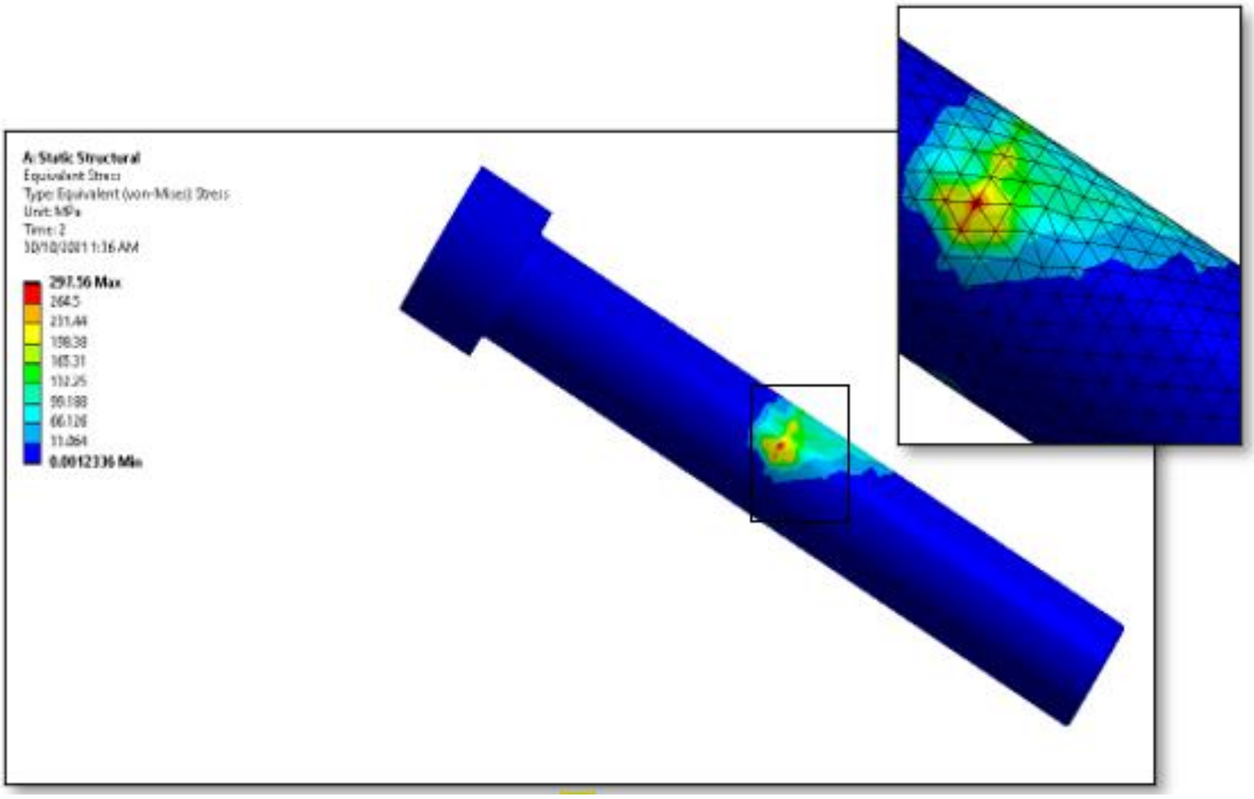


Figure 26 - Titanium bolt stress contour.

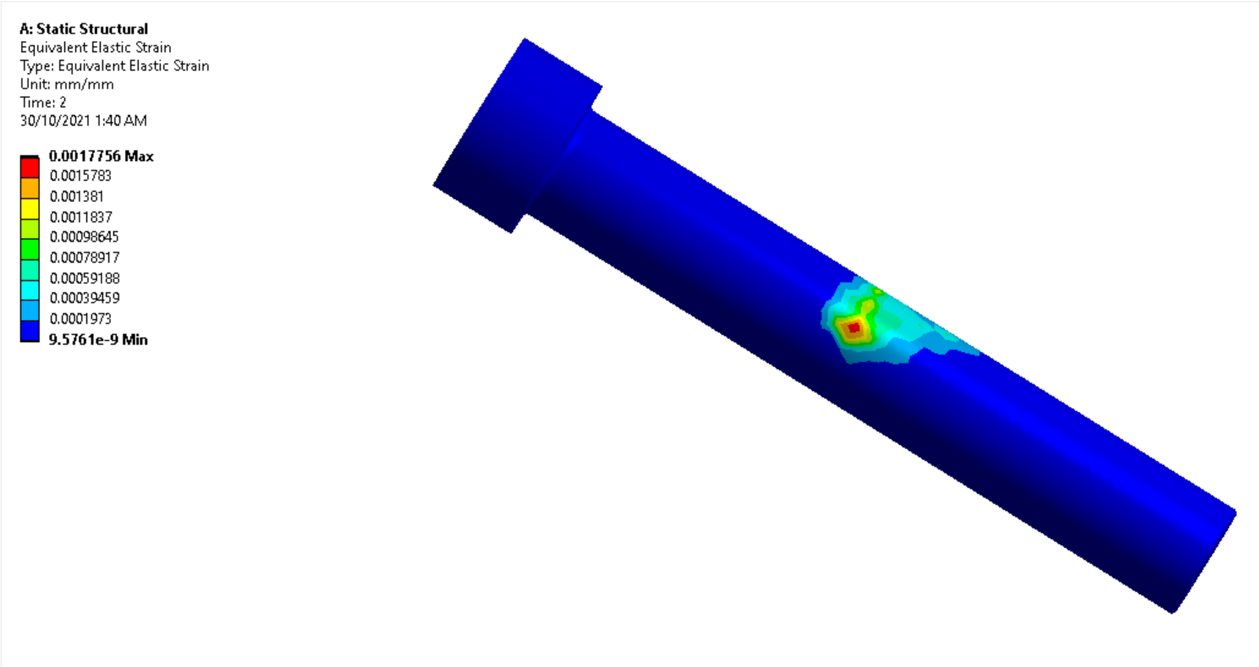


Figure 27 - Bolt deflection/strain contour (titanium).

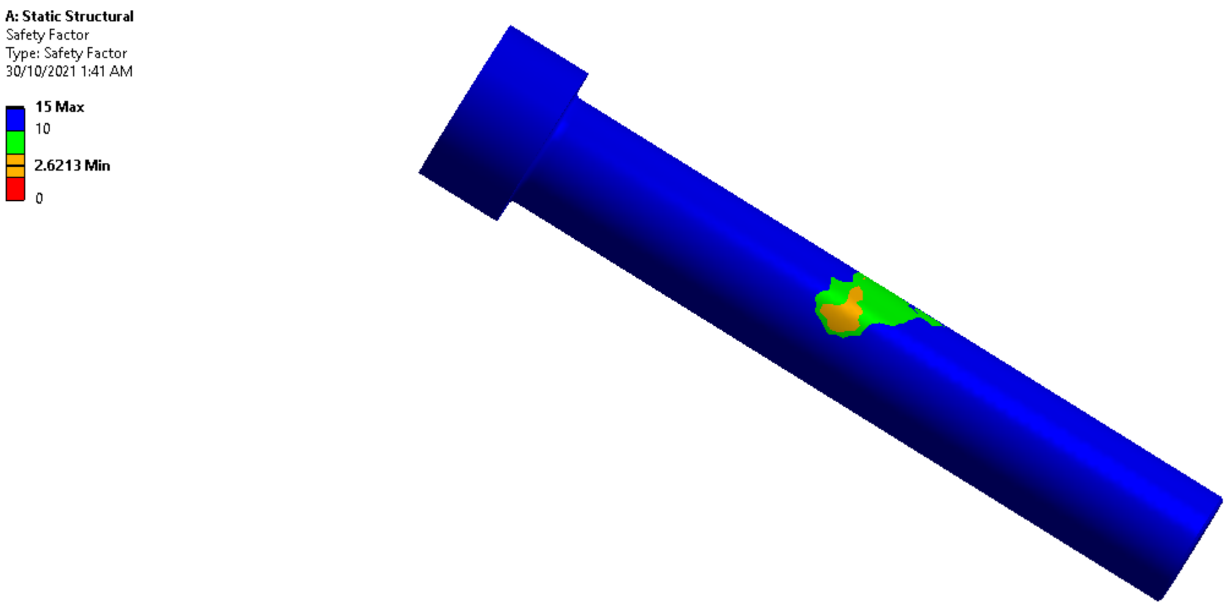


Figure 28 - Fatigue FOS contour titanium bolt.

4.2.2 Guiding Pin and Clip and Mounting Bolts

The brake pad retaining pin and clip allows easy accessibility to the brake pad and provides a structural guide and support for the brake pads. While the mounting bolts directly fix and transfer the brake calliper and loads to the rear wheel uprights. However, the components experience much smaller and insignificant loads and are considered relatively unimportant in terms of load-bearing capacity. Hand calculations were sufficient to analyse and determine the components required mechanical properties and FOS to maintain a functioning and safe brake calliper. Figure 29 displays the respective FBD of the guiding pin and clip with associated loads.

A few assumptions have been made to simplify the system including negligible self-weight. Sufficient clearance and lubrication between the guiding pin and the respective guiding hole preventing shear and moments along the pin. While Figure 30 displays the FBD of the mounting bolts with its associated pretension. It has been assumed the bolt's weight are negligible and the axial load generated during braking is solely experienced on the calliper bolts. Moreover, sufficient pretension will maintain frictional support between the bolt and calliper faces to prevent slippage, and shearing loads on the bolt.

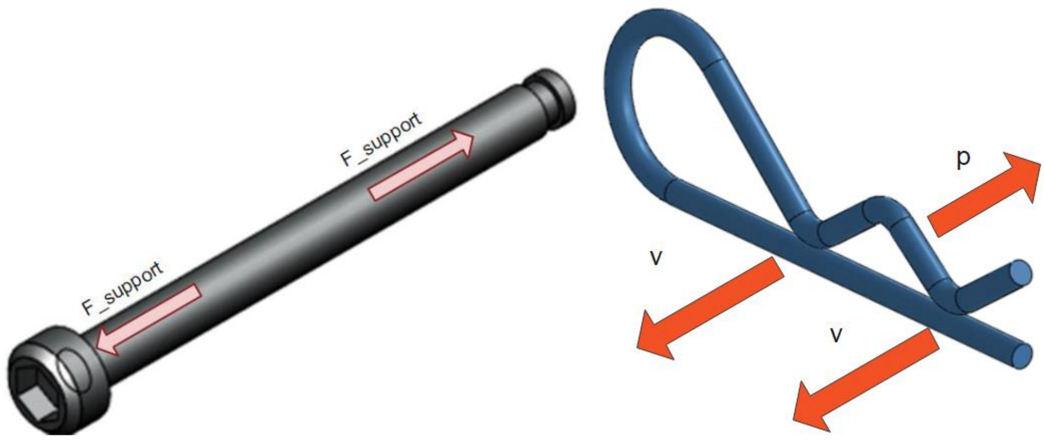


Figure 29 - FBD schematics of pad retaining/guiding pin and clip.

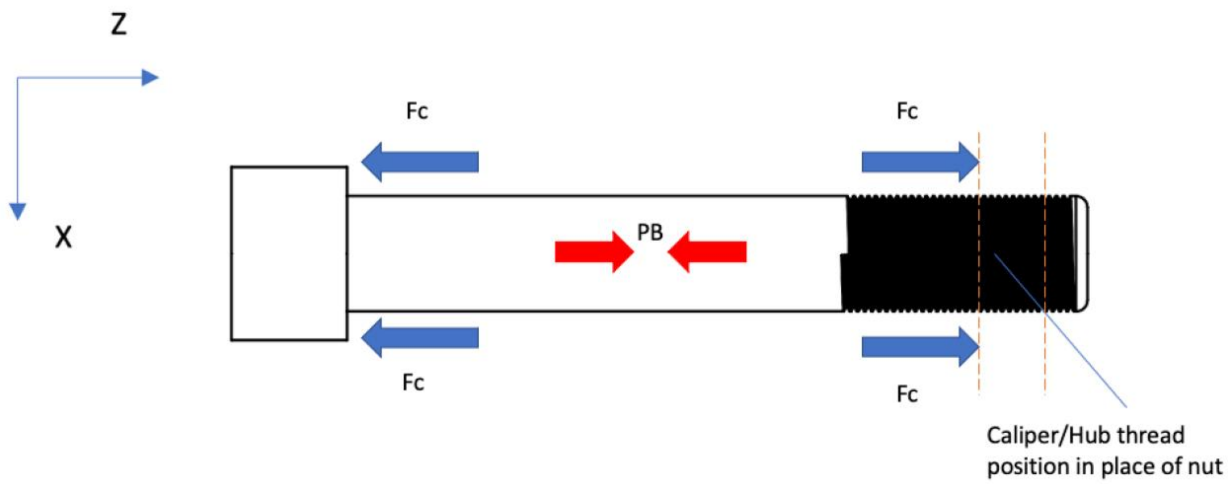


Figure 30 - Mounting bolts FBD.

4.2.2.1 Guiding Pin and Clip

Figure 31 displays the moment diagram used to equilibrate through the sum of the moments of the origin the guiding pin axial tension. The moment induced by braking is thought to be shared primarily through the two calliper bolts and partially through the guiding pin which has been assumed insignificant until this stage.

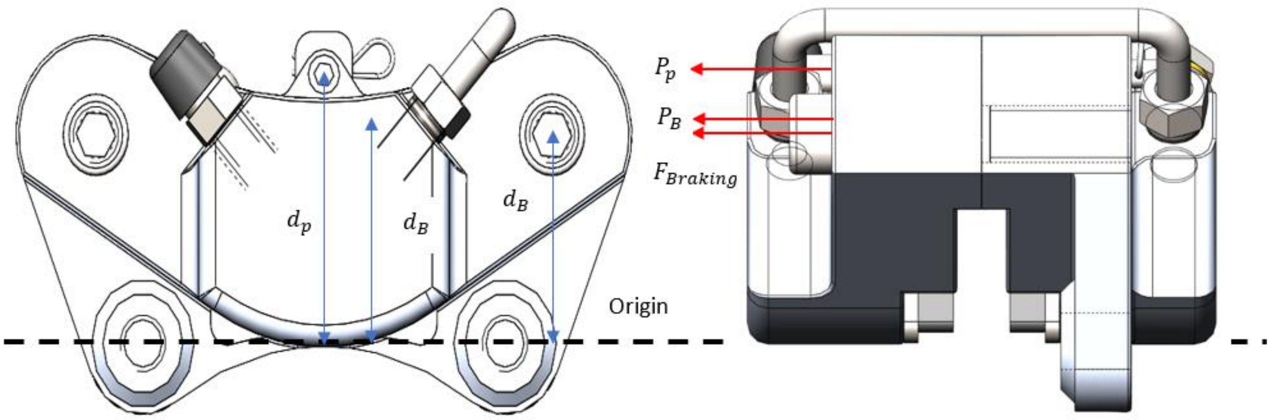


Figure 31 - Load diagram on calliper for loading of calliper mounting bolts.

The moment summation about the mounting bolts:

$$\sum M_o = F_{braking} \times d_{Braking} - P_P \times d_P - 2P_B \times d_B = 0$$

The following distances have been determined through Solidworks dimensions. While the $F_{Braking} = 1517N$ has been previously calculated.

$$d_{Braking} = .09023m, d_P = .093m \,\&\, d_B = .085m$$

$$136.879 = P_P \times d_P + 2P_B \times d_B$$

Deflection relationship:

$$\frac{\delta_P}{d_P} = \frac{\delta_B}{d_B} \rightarrow \frac{P_P}{d_P} = \frac{P_B}{d_B}$$

$$P_B = \frac{P_P\, d_B}{d_P}$$

$$136.879 = P_P \times d_P + \frac{2P_P d_B}{d_P} \times d_B$$

$$136.879 = .249P_P$$

$$P_P = 551.1N$$

$$v = P_P/2$$

$$v = 275.55N$$

Section 3 deemed the necessity to manufacture the guiding pin due to its specificity and added level of characterisation. The pin has several pre-existing design constraints having a diameter of 4mm while having a high enough tolerance within the brake pad to ensure smooth and uninterrupted travel to prevent accrual of shear and moment loads. The pin analysis will determine the minimum required material strength and property under fatigue conditions with FOS of 2. This FOS aims to account for potential shear and moment loads induced by improper guiding. The fatigue calculations are seen below and utilise a zero-based cycle of 1e+6.

4.2.2.1.1 Guiding Pin Hand Calculations

$$\sigma' = \sigma_P$$

$$\sigma_P = \frac{P_P}{A} = \frac{800}{\pi(.002)^2}$$

$$\sigma_P = 43.855\ MPa$$

$$\sigma_a = \sigma_m = \frac{\sigma_P}{2} = 21.93\ Mpa$$

$$S_n = S_n{}'C'$$

$\boldsymbol{S_n{}' = .5S_{UT}\ (infinite\ life\ cycle)}$	$\boldsymbol{C' = C_L C_S C_T C_G C_R}$ $\boldsymbol{C_L = 1\ (von\ mises)}$ $\boldsymbol{C_S = 1\ (assuming\ commercial\ surface\ finish)}$ $\boldsymbol{C_T = 1\ (assuming\ calliper\ bolts < 450\ celsius)}$ $\boldsymbol{C_G = .9\ (due\ to\ axial\ loading)}$ $\boldsymbol{C_R = .702\ (99.99\%\ reliability)}$

$$S_n = .316S_{UT}$$

Goodman

$$\frac{\sigma_a}{S_n} + \frac{\sigma_m}{S_{UT}} = \frac{1}{N}$$

$$\frac{21.93}{.316S_{UT}} + \frac{21.93}{S_{UT}} = \frac{1}{2}$$

$$\frac{91.33}{S_{UT}} = \frac{1}{2}$$

$$S_{UT} > 182.66\ MPa$$

Yield

$$\frac{\sigma_a}{S_Y} + \frac{\sigma_m}{S_Y} = \frac{1}{N}$$

$$\frac{\sigma_P}{S_Y} = \frac{1}{N}$$

$$S_Y > 87.71\ MPa$$

4.2.2.1.2 Material Selection

The calculated ultimate tensile and yield figures are likely obtained by most metals as shown in figure 32. Material selection will consist of the relation between three significant parameters. Firstly, weight illustrated in figure 32 - right. Removing pin weight decreases unsprung mass and adds performance advantages. The thermal analysis explored in section 4.4 determined the maximum temperature experienced at the brake pad after 2 seconds of operation was substantial. Which was demonstrated to largely dissipate and after 3 seconds the maximum temperature was seen at the top of the brake pad and guiding pin. To further reduce operational temperatures a more favourable heat transfer material can be selected. Secondly, costs represented in figure 32 left. Due to the limited funds the FSAE team will benefit from reduced costs within insignificant components. In conjunction with specified parameters and figures aluminium was chosen. Aluminium appears to be the most versatile metal being relatively light with favourable conductivity however is comparatively more expensive than a lot of other metals and alloys.

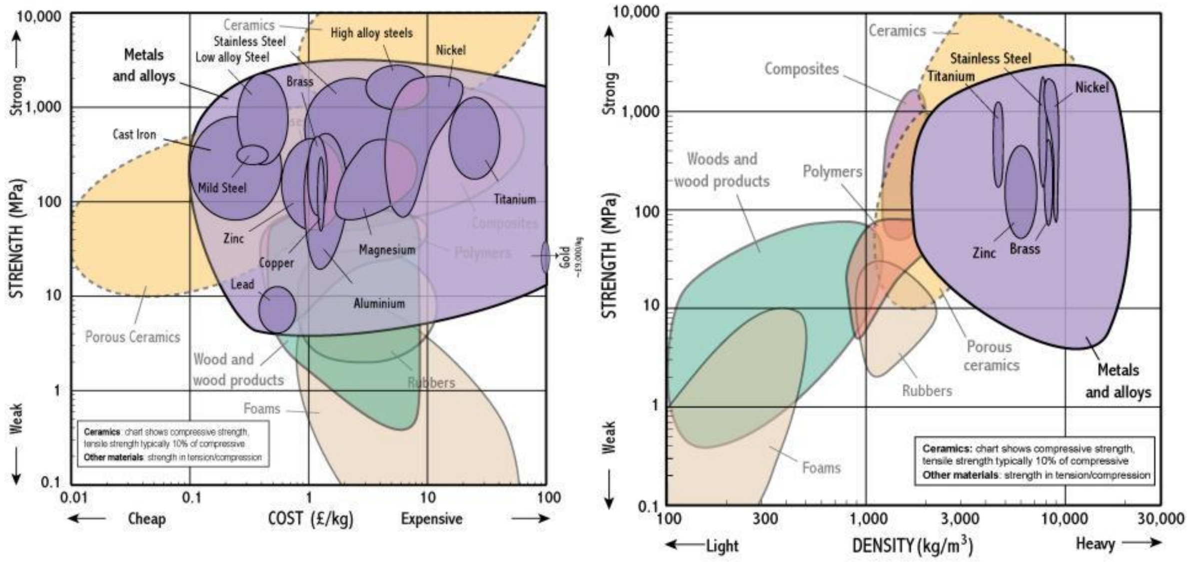


Figure 32 - (Left) Strength vs. cost Ashby material selection chart. (Right) Strength vs. density Ashby material selection chart [26].

4.2.2.2 Clip

While the Zinc Plate R-clip chosen in section 3 is an off-the-shelf product from Bunnings with a specified thickness of 5mm. The shear load was calculated in continuation with the previous axial load. Hand calculations regarding fatigue loading are required to ensure the contact between the clip and guiding pin to enable and support brake operation. The fatigue calculations seen below determines the clip's FOS. Which should be greater than 2 to account for and reduce the likelihood of failure due to vibrations and slips, and expansion induced by surrounding temperatures. Furthermore, the material has been described broadly as zinc finished steel with no suggestion of method. Thus, it was assumed maximum values for mild steel using having an ultimate tensile strength of 550 MPa. In addition, a zero-based infinite life cycle was assumed.

4.2.2.2.1 Clip Hand Calculations

$$v = 275.55N$$

$$\tau = \frac{v}{A} = \frac{275.55}{\pi(.0025)^2}$$

$$\tau = 14.03\ MPa$$

$$\sigma' = \sqrt{3\tau^2} = 24.307MPa$$

$$\sigma_a = \sigma_m = \frac{\sigma'}{2} = 12.15\ MPa$$

$$S_n = S_n{}'C'$$

$\boldsymbol{S_n{}' = .5S_{UT}\ (infinite\ life\ cycle)}$	$\boldsymbol{C' = C_L C_S C_T C_G C_R}$ $\boldsymbol{C_L = 1\ (von\ mises)}$ $\boldsymbol{C_S = 1\ (assuming\ commercial\ surface\ finish)}$ $\boldsymbol{C_T = 1\ (assuming\ calliper\ bolts < 450\ celsius)}$ $\boldsymbol{C_G = .9\ (due\ to\ axial\ loading)}$ $\boldsymbol{C_R = .702\ (99.99\%\ reliability)}$

$$S_n = 193.05\ MPa$$

$$\frac{\sigma_a}{S_n} + \frac{\sigma_m}{S_{UT}} = \frac{1}{N}$$

$$\frac{12.15}{193.05} + \frac{12.15}{550} = \frac{1}{N}$$

$$N = 11.67$$

Therefore, the zinc finished steel will likely outlast the system under ideal conditions being more realistic to fail during service and maintenance.

4.2.3 Mounting Bolts

Section 3 specifies the mounting bolt as an off-the-shelf product sourced from Titan Classics. The titanium Nissin calliper M8 x 1.25mm x 45mm bolt set has been chosen due to Titan's favourable strength to weight ratio and exceptional corrosion resistance. The bolts meet section T1 outlined in FSAE rules being made from grade 5 (Ti-6Al-4V). The following are the required parameters obtained from BS A 101 [28] on the design of titanium bolts in aerospace applications:

$$S_Y = 1100\ MPa, S_{UT} = 1170\ MPa, A_t = 39.2mm\ \&\ K_f = 3$$

The required pretension in the bolt is the frictional support between the inboard calliper surface and rear uprights to withstand the braking torque. The frictional coefficient use was 0.61 being aluminium on steel [29]. The pretension required by the bolt is:

$$F_{fSupport} = 380 \times .61 = 622.95\ N$$

$$F_i = 400 \times 1.2 = 747.54\ N$$

It is assumed that the mounting bolts only experience constant pretension with no cyclic conditions. This is due to the frictional support and lack of moment imparted to the bolts. Thus, static loading was considered with calculations seen below.

$$\sigma' \leq \frac{S_y}{N}$$

$$\sigma' = \frac{F}{A_t} K_f$$

$$N \leq 19.22$$

Thus, the titanium bolts will never fail under the proposed conditions and likely will not fail under serious incidents with a required breaking Force of $14373.3N$ which includes initial tightening.

$$\sigma' \leq 1100\ MPa$$

$$F = 14373.3N$$

4.3 Piston & Brake Pad (Todd Dalgliesh, n6004075)

The brake piston and pad were analysed and contacts between the two parts were undertaken together. The loads to be considered were the fluid pressure on the piston and the thrust force which is transferred to the pad on each application of the brake system. As the use case for the FSAE vehicle is below that of an automobile in daily use the applications per installation on vehicle would be below 100 times as mentioned in section 4.1. However, as the vehicle will be used by varying operators in many conditions the maximum pressure will be 9 MPa as was calculated in Appendix B. The thrust transferred from the rotor to pad will be set at 757N as outlined in Appendix B. The pad will experience an equal force from the opposite side of the calliper as the clamping force occurs onto the rotor in an idealised manner. The loads mentioned are shown below in 33.

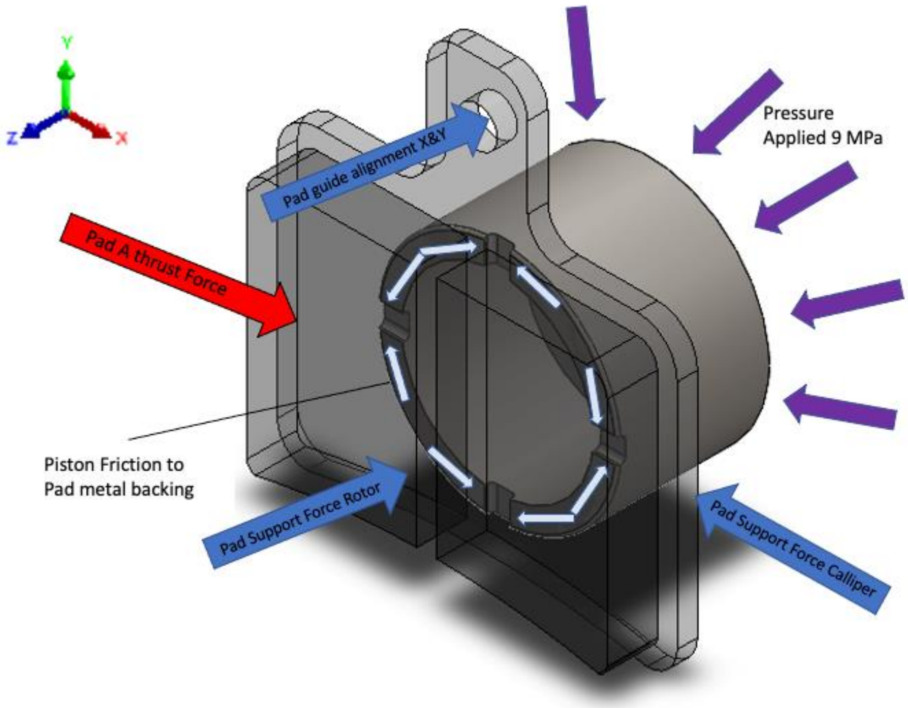


Figure 33 - Load diagram FBD for piston/pad structural analysis.

As the Brake pad is an OEM part the hand calculations will only be performed on the piston with contact stress to pad and Hoop stress, radial stress and radial displacement as these values can be checked via ANSYS structural. The maximum stress will be calculated for the contact area between the pad and piston. This contact area is a friction contact but is assumed that the piston does not rotate under application.

The following assumptions were implemented into the simulation which was run in ANSYS Static structural finite element analysis (FEM) for the pad and piston and are listed below:

- The pad has equal force applied from either brake pad and thus equal force from the piston
- Brake pad guide does not support any force and for service use only but was used to prevent the pad from moving or sliding in the Y-axis
- The material brake pad matched the materials which were available in the off the shelf version
- Piston analysis used stainless as the material as this is frequently used in many off the shelf brake systems and is corrosion resistant
- A frictionless support was used to constrain the system, during use the pad and piston move freely in the z-axis but was required for the system to calculate the maximum load cases

- The changes to the piston were minor and in line with machining practices that would be needed to ensure stress concentrations were minimised
- Weight of the pad and piston are negligible and thus effects of gravity were not input into this simulation
- As the seal is lubricated and piston moves freely the frictional force is negligible
- The support force for the pad was applied over the whole pad material surface
- Real world applications would have the pad operate at slight angles due to rotational effects however as these angles would be close to zero were negligible and not included
- The drop in friction coefficients due to heat was negligible as this could be covered in further work such as a full Fluent Thermal analysis
- The piston does not rotate and frictional contact between the pad and piston is stable

4.3.1 Hand calculations

The following hand calculations were used as a basis to check the FEA aligned with how the model would behave on a vehicle. As the shape of the components and parts had varying dimensional changes these calculations were used as a starting point but would need further refinement once all parts in the system were analysed and the final parts for production satisfied the criteria set by the team.

Contact Stress Piston to Pad

Dimension to measure piston contact area to pad:

Cut out area of piston face: 2.5mm*2.51mm

$Dia_{in} = 19mm \qquad Dia_{out} = 24mm \qquad \left(\frac{\pi Do^2}{4} - \frac{\pi Di^2}{4}\right) - 4 * \ (cut\ out\ area)$

$$(\left(\frac{\pi.*024^2}{4} - \frac{\pi*0.019^2}{4}\right) - 4 * \ (.0025 * .00251) = 0.000144m^2$$

$$\frac{2300}{0.000144} = 15.97\ MPa$$

A check was performed to verify in ANSYS that this force experienced on the contact surface was within specifications shown in figure 34.

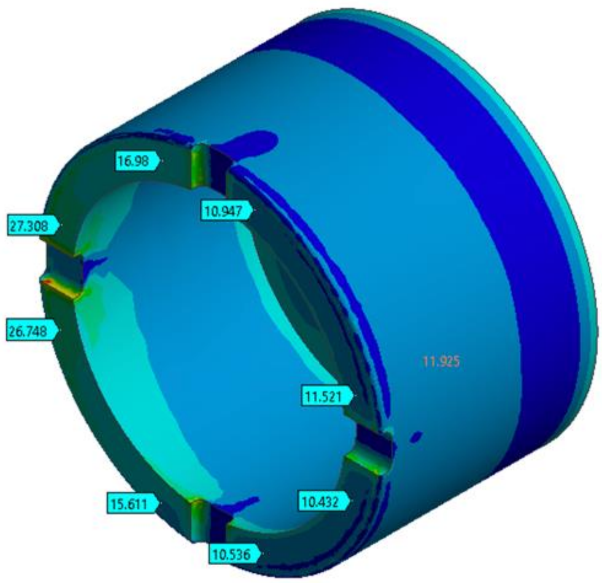


Figure 34 - Probe stress values on piston contact face.

Probe values on piston to pad face show below in Table 7:

Table 7 - Summary of probe stress values on piston.

Probe position	Value MPa
1	10.947
2	11.521
3	10.432
4	10.526
5	15.611
6	26.748
7	27.308
8	16.98
Average	16.2591 MPa

As the hand calculation did not include the force accounted for by the thrust force the average of these results is within 1.723% difference of the hand calculation therefore deemed accurate/valid.

Deflection of piston

Radial displacement & Hoop and radial stress using lame's equations for external pressure:

$$\sigma_r = -\frac{b^2 p_o}{b^2 - a^2}\left(1 - \frac{a^2}{r^2}\right)$$
$$\sigma_\theta = -\frac{b^2 p_o}{b^2 - a^2}\left(1 + \frac{a^2}{r^2}\right)$$
$$u = -\frac{b^2 p_o r}{E(b^2 - a^2)}\left[(1 - v) + (1 + v)\frac{a^2}{r^2}\right]$$

B = external radius = 24mm

A = internal radius= 19mm

$$\theta_r = \frac{(0.024^2 * 9 * 10^6)}{0.024^2 - 0.019^2}\left(1 - \frac{0.019^2}{0.024^2}\right) = 9\ MPa$$

$$\theta_h = \frac{(0.024^2 ** 10^6)}{0.024^2 - 0.019^2}\left(1 + \frac{0.019^2}{0.024^2}\right) = 39\ MPa$$

E = 200 GPA, v = 0.265 (Poisson ratio of stainless steel)

$$u = -\frac{(0.024^2 * 9 * 10^6)}{200 * 10^9 * (0.024^2 - 0.019^2)}\left((1 - 0.28) + (1 + 0.28)\frac{0.019^2}{0.024^2}\right) = 0.000171mm$$

This value was compared to minimum displacement in FEA and the value was 0.000224mm, as the FEA had additional forces increasing the total deformation. As this value is smaller compared to the tolerance required for high performance piston to bore clearance the deflection is within the scope of the project [30].

Therefore, using the fluid pressure gained from APPENDIX B, given

$$P_{system} = \sigma_{fluid} = \frac{F}{A} = \frac{2300}{24 * \pi} = 3.05MPa = 30.5\ bar$$

The value of seal protrusion will be **0.5mm** which will allow the seal to remain effective and resist leakage throughout the temperature range which is seen in figure 35. This seal dimension also aligns with **BS ISO 4928-2006** [31] seen in table 8 for wheel cylinder seal change after temperature is increased as the value for seals between 19.05mm and 25.4mm is **0.5mm.** The extrusion gap can be seen in figure 36 where the high-pressure fluid and motion of the piston creates the seal to deform slightly which is addressed with the seal protrusion calculation.

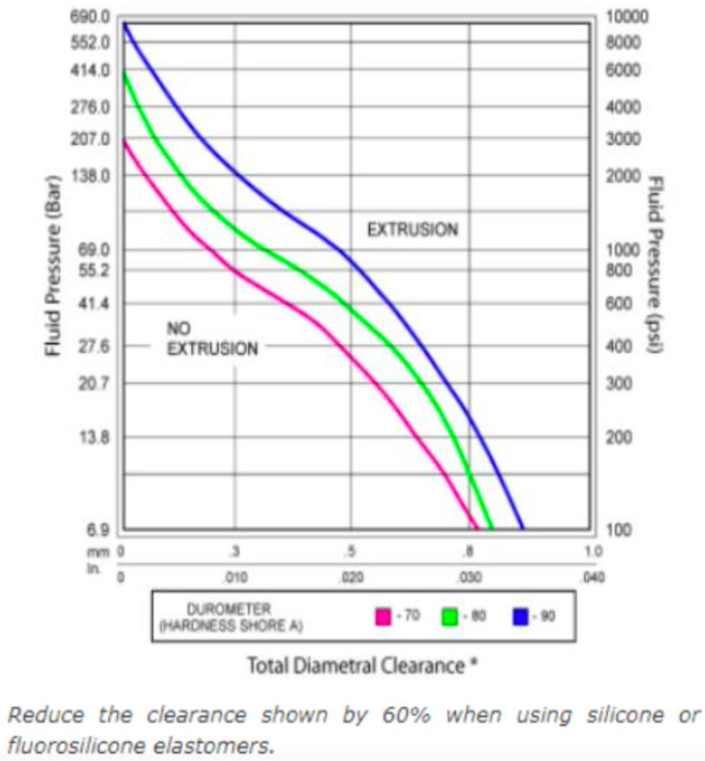


Figure 35 - Fluid pressure and associated clearance for piston seal [30].

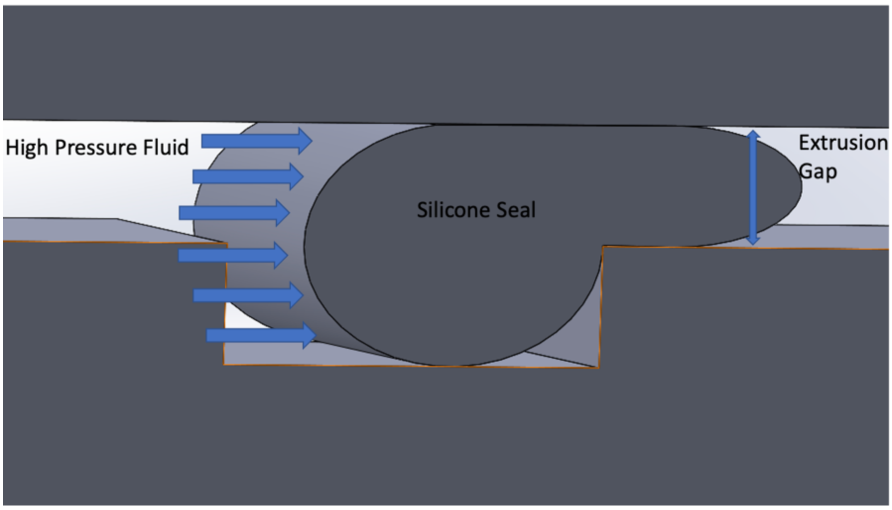


Figure 36 - Seal deformation and extrusion gap [30].

Table 8 - Lip diameter change of wheel cylinder seal per BS ISO 4928:2006 [31].

Dimensions in millimetres

Diameter of wheel cylinder bore	Minimum excess over bore
⩽ 19,05	0,40
> 19,05; ⩽ 25,4	0,50
> 25,4; ⩽ 38,1	0,65
> 38,1; ⩽ 60	0,75

4.3.2 FEA Model Development

The decision to use the maximum pressure experienced on the piston as this is a critical component the need for these parts to perform under any circumstance deemed this the correct decision for this analysis. Additionally, blend radii of 0.5mm were put on the 8 corners of the piston to ensure the stress concentrations were mitigated whilst reducing the changes that were required to the geometry after the mesh was set up shown in figure 37. The model was also split up so that the pad could have two separate materials and have a bonded surface between the metal backing and the organic pad material. These loads and contacts can be seen in figure 38.

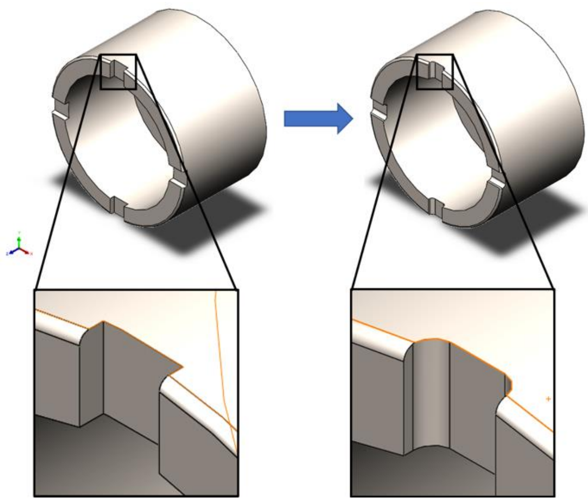


Figure 37 - Blend radii imparted to piston to minimise stress concentration at critical zone.

Following contacts and loads were input to the simulation to mirror operating conditions aggressive application of the brake system.

- The friction on from the pad to the piston was input as 0.2 as this was given value from steel to steel [32].
- A bonded contact was present for the pad material to pad steel lining
- As the pad would move slightly the RH side of the pad was used as a support to contact the calliper
- The pad guide hole was used a frictionless support and it carries no force
- The thrust force was applied to the surface of both pad materials to spread the load.
- The piston was supported using a frictionless support as this stabilised the location of the piston under compression.
- The pad Ultimate tensile strength was averaged from combinations of hybrid and composite materials obtained through studies by [33, 34] and input as the value of 33 MPa.

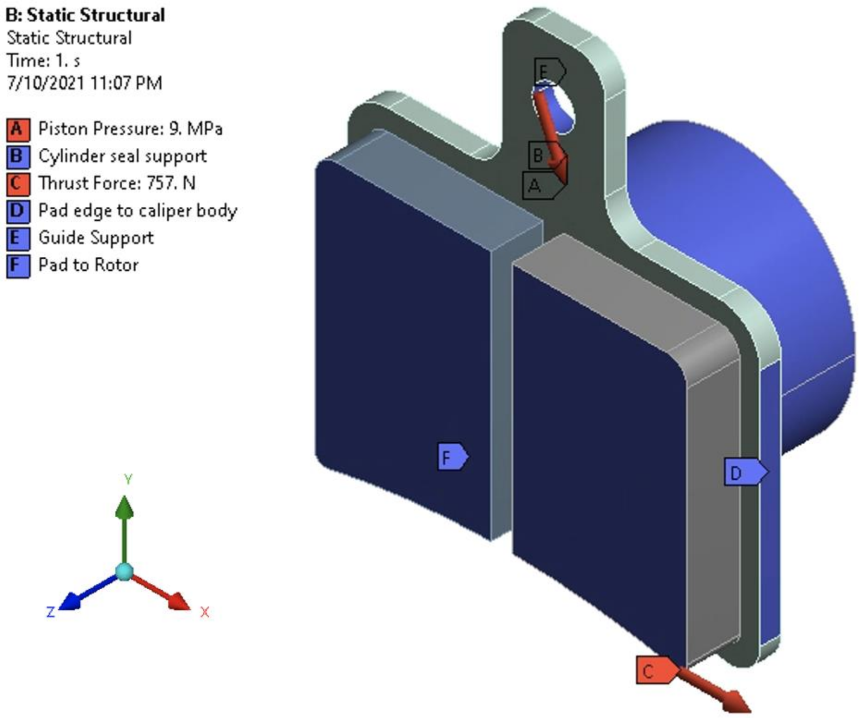


Figure 38 - Summary of loads and boundary conditions applied to FEA model.

Running the model simulation until a convergence occurred yielded the following values in table 9 below. As the main areas of concern were both ends of the piston and how the pad material was affected increased refinement was used in these areas.

Table 9 - Model statistics from converged mesh.

Mesh / Solution Parameter	Value
Solve time	177.59 (seconds)
Element sizes used	0.5mm (Piston body) 0.7mm (brake pad lining) 0.3mm (piston pad faces) 0.3mm (pad to rotor) 0.3mm (piston cut outs)
Number of nodes	281326
Number of elements	169972
Element skewness avg.	0.37654
Element quality avg.	0.748

As the original design had sharp edges where material was removed on the pad side of the piston this was a critical zone that required remedy to ensure the simulation ran smoothly. These were modified to be a 0.5mm fillet and allowed the model to pass all criteria. The maximum stress occurred at the edges contacting the pad and was measured to be 158.43 MPa shown in figure 39 below.

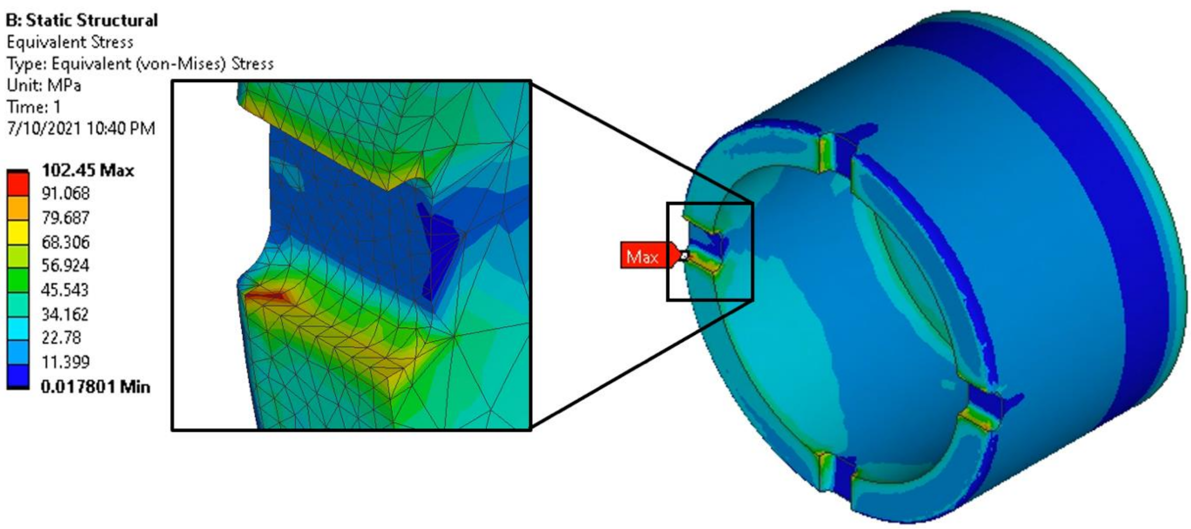


Figure 39 - Von mises stress contour of piston static analysis.

Attention was focused to the deformation that the piston would have on the pad material. Furthermore, the support contact that is experienced by the side of the pad is seen in figure 40. Due to the large area the force is spread across on the friction side, this stress is dispersed slightly compared to the pad/piston contact. The piston stress is shown in figure 41 with clear indications of where the force is applied to the pad**.** The large piston diameter is effective in covering a large area of the back of the pad while maintaining minimal contact area to limit heat transfer.

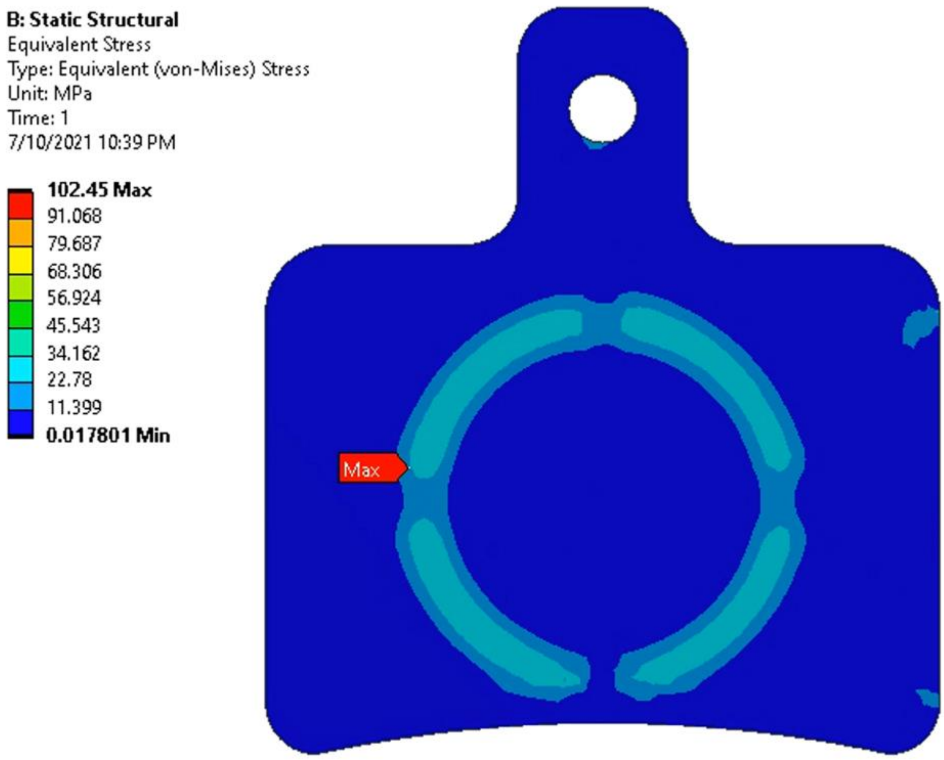


Figure 40 - Equivalent stress contour on pad resulting from piston contact.

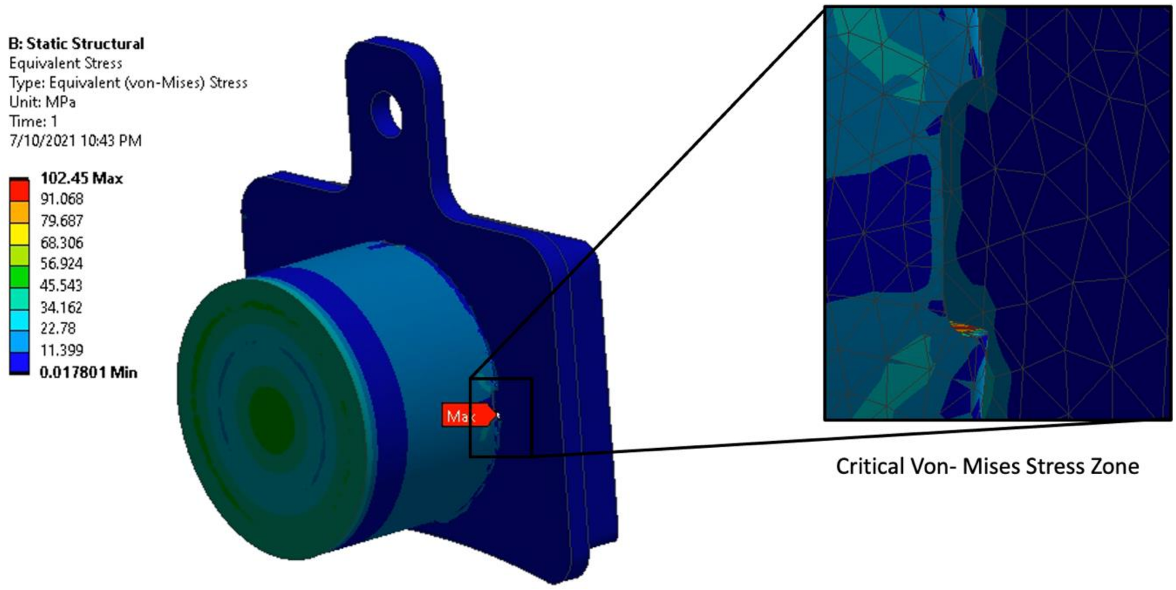


Figure 41 - Stress contour showing critical stress zone on piston.

As expected, the pad lining experienced deformation, however this occurred with only 0.00084mm on the rear lining side as shown in figure 42. This value was deemed within tolerances as this was not transferred to the pad material causing a misalignment and would only alter the pad wear slightly until the pad had bed in under continuous use.

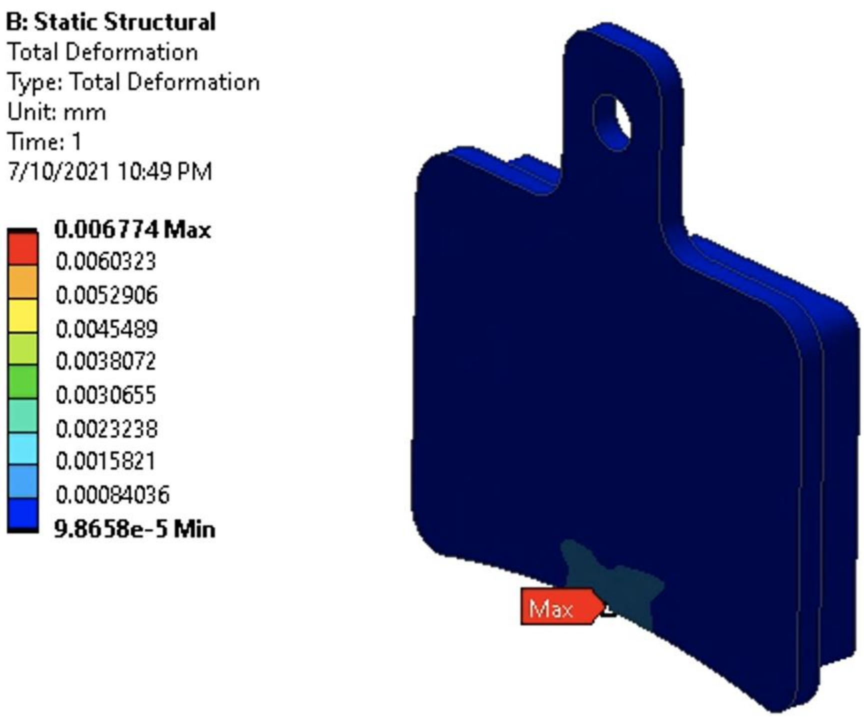


Figure 42 - Deformation contour plot of pad metal backing material.

Deformation was measured at a maximum at the rear of the piston in contact with the brake fluid seen in figure 43. This was measured at 0.01018mm which was within tolerances for the BS ISO 4928-2006 [31] which has permitted change of between 0% to 5.75% shown in table 10 and the calculated change of the piston design is 0.024% therefore is within the standard analysed in this section.

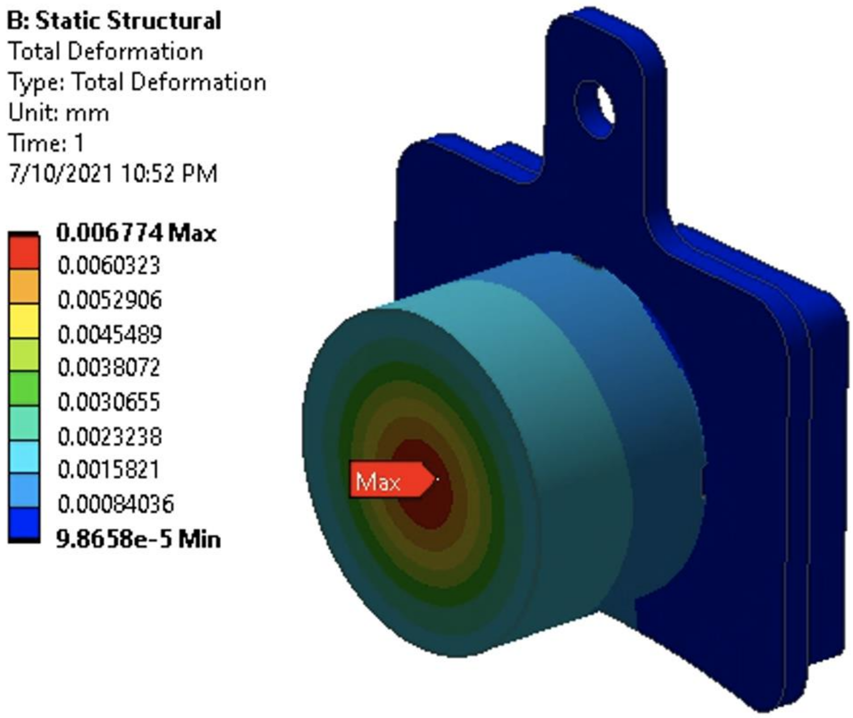


Figure 43 - Total directional deformation on piston from fluid pressure.

Table 10 - Piston deformation allowable from BS ISO 4928:2006 [31].

Characteristics	Permitted change
Volume	From 0,0 % to + 20,0 %
Outside diameter, lip	From 0,0 % to + 5,75 %
Outside diameter, base	From 0,0 % to + 5,75 %
Hardness	From – 15 IRHD to 0 IRHD

4.3.3 Material Selection

Due to the pad being manufactured outside of the team, material selection for the piston will only be under consideration. As the piston would be under a changing set of parameters throughout the operation, several of Ashby's material selection criteria will be used to determine a suitable material [26]. The key properties needed would then be corrosion resistant, high surface hardness, low thermal conductivity, thermal diffusivity, thermal expansion, high maximum service temperature, relative cost per unit volume and production energy per cubic volume which are shown in the table 11 below. These criteria would allow the piston to operate without inducing brake fade into the system through poor thermal qualities and resist wear of the piston [32]. By weighing all these equal and considering candidate materials which align with all the key properties mentioned previously the selection table 12 below shows that 316 stainless should be the candidate material of choice with a significantly better score than both aluminium and titanium.

Table 11 - Collection of applied Ashby material comparison maps used to select piston material [26].

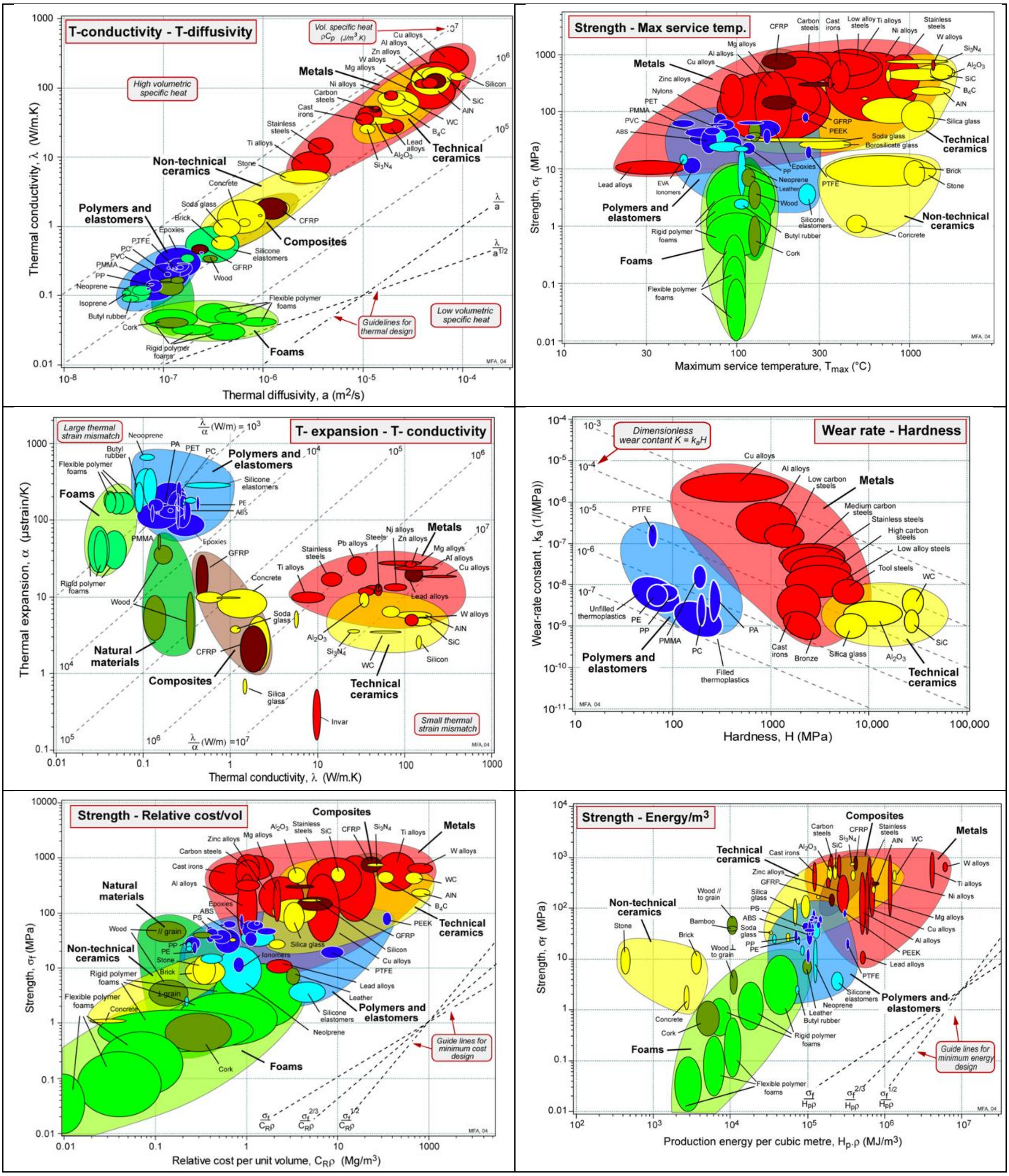


Table 12 - Material comparison and selection results for piston design.

Corrosion resistant Material	YS at room temp	Rank	Harness Rockwell B	Rank	Density [kg/m^3][7]	Rank	Max service temp.	Rank	Thermal conductivity (W/m.K)	Rank	Thermal diffusivity (m^2/s)	Rank	Relative cost per unit volume (mg/m^3)	Rank	Thermal expansion (strain/K)	Rank	Total score (lowest is selected)
Aluminium Alloy	240 MPa	3	34	3	2705	3	350 ℃	3[18]	251	3	5*10^-4	3	1	1	30	3	22
Titanium Alloy	241 MPa	2	70	2	4520	2	600 ℃	2	10	1	9*10^-5	1	300	3	10	1	14
Stainless Steel 316	290 MPa	1	79	1	7980	1	926 ℃	1	12	2	8*10^-5	2	10	2	20	2	12

As mentioned by [7] Neal, a good surface finish with a high corrosion resistance is necessary when fluid that is hydroscopic will be used as this can induce water accumulation into the system and for this reason along with the material selection table results from above 316 stainless steel should be selected for the piston. Aluminium has been used in some cases mentioned by Limpert [32], however this is not ideal as the poor thermal properties would raise the brake fluid heat vaporizing the fluid and creating brake fade which is not optimal in a performance vehicle such as the FSAE QEV3.

4.3.4 Fatigue study

Continuing from the fatigue study performed in 4.2.3, the fatigue loading was set at $5 * 10^5$ cycles as the FSAE vehicle has infrequent use much lower than an automobile in daily use. The loading scenario for the brake piston and pad was zero based as there is only the spring back occurring from the piston seal and fluid [3] and this is the release pressure acting on the system. The thrust force was used from 4.2.3 but set at 757N as this force was set for one pad. But the pressure on the piston of 9 MPa was set as the system maximum as this was the aim of the fatigue simulation to ensure the components would exceed the safety requirements in any brake use case.

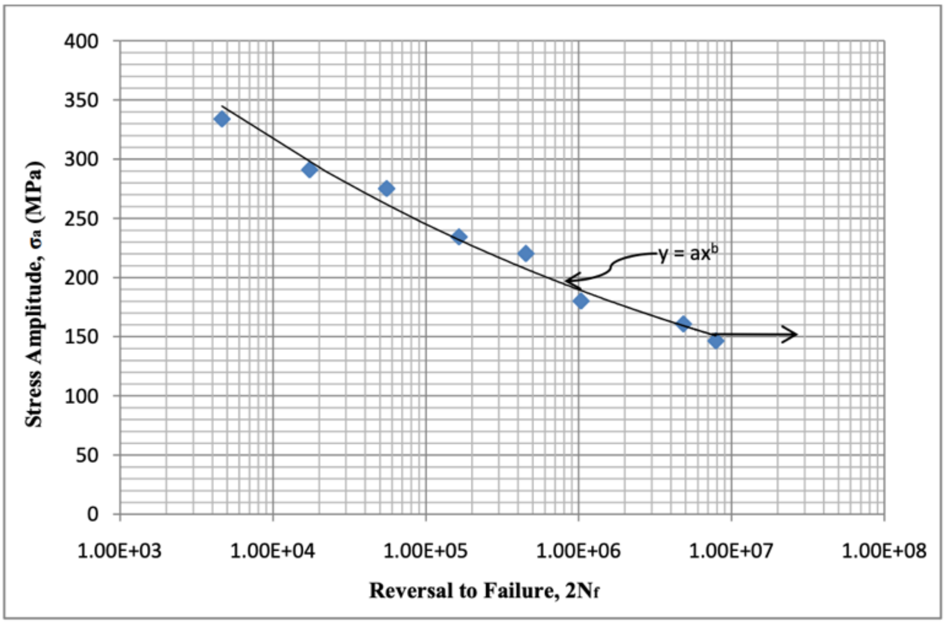


Figure 44 - Fatigue SN curve for 316 SS used for FEA analysis [35].

As the piston was the component which was being manufactured by the team the fatigue analysis for this was the main interest of the fatigue simulation. The minimum Factor of Safety was 2.2274 and occurred at the radius of the piston to pad machined section shown in figure 45**.** The pad and piston were simulated in conjunction and the pad had a FOS of around 1.5 on the pad material shown in figure 46, as the pad material is a combination of materials the lowest ultimate tensile strength of 37 MPa for the organic material was used which resulted in a minimum FOS location on the pad rear surface to metal pad lining. In line with the thrust added to the system by the rotor the key considerations for pad wear would be the lower edges and the pad wear would not be symmetrical as the strain on the pad is offset slightly due to this movement in the system seen in figure 47.

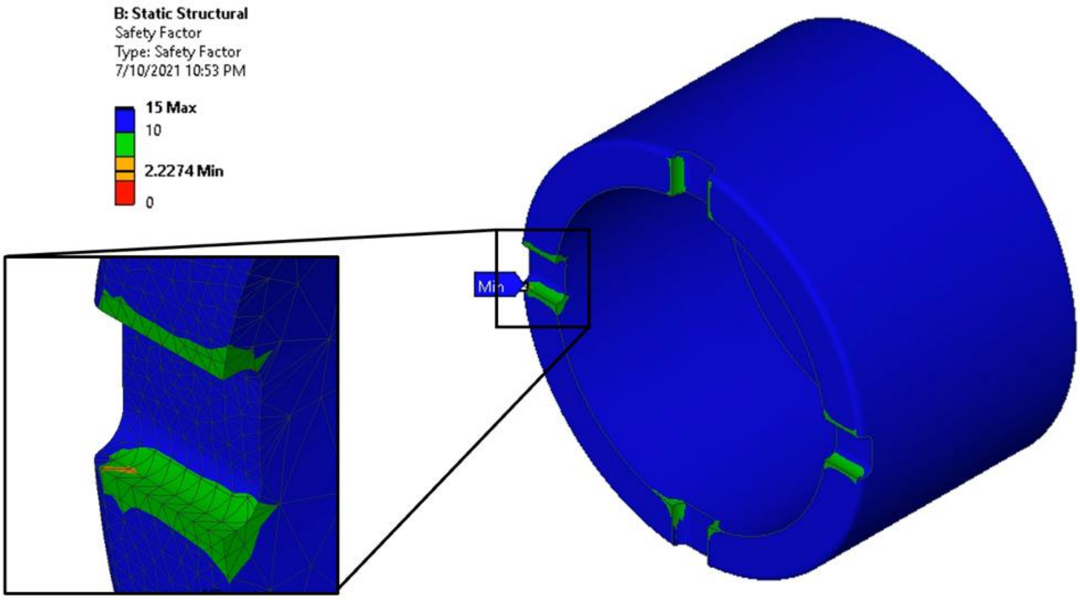


Figure 45 - Piston FOS contour against fatigue failure.

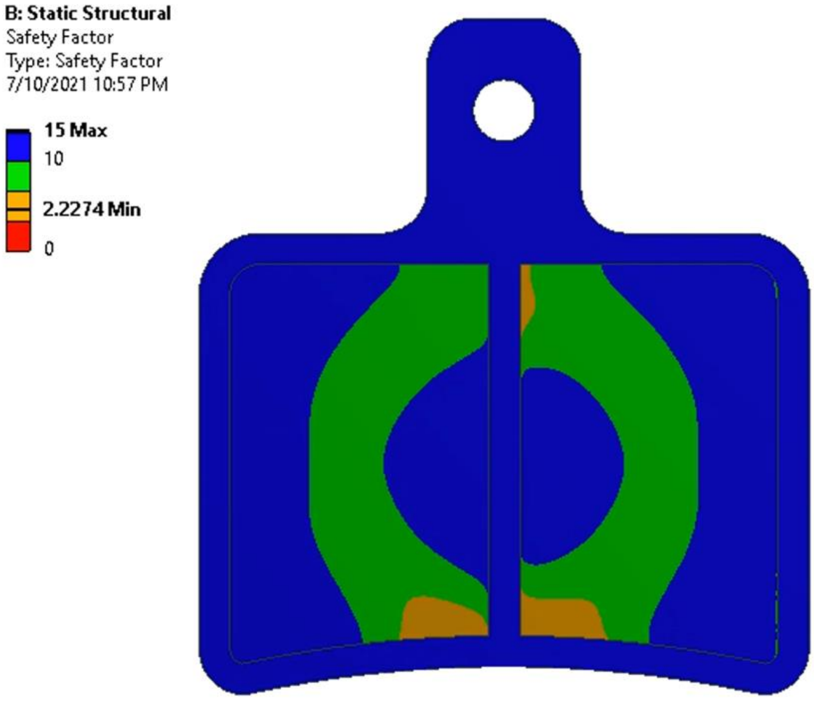


Figure 46 - Factor of safety of brake pad material.

4.3.5 Pad wear consideration

In line with the thrust added to the system by the rotor the key considerations for pad wear would be the lower edges and the pad wear would not be symmetrical as the strain on the pad is offset slightly due to this movement in the system seen in figure 47**.** As mentioned by Mullaikodi et al. [33] if the flexural resistance is improved the wear resistance can be improved. Therefore, for further work eliminating a higher degree of deformation from the pad lining would improve wear and reduce the chance pad drag would be induced into the system. The brake pads selected were considered adequate for this application.

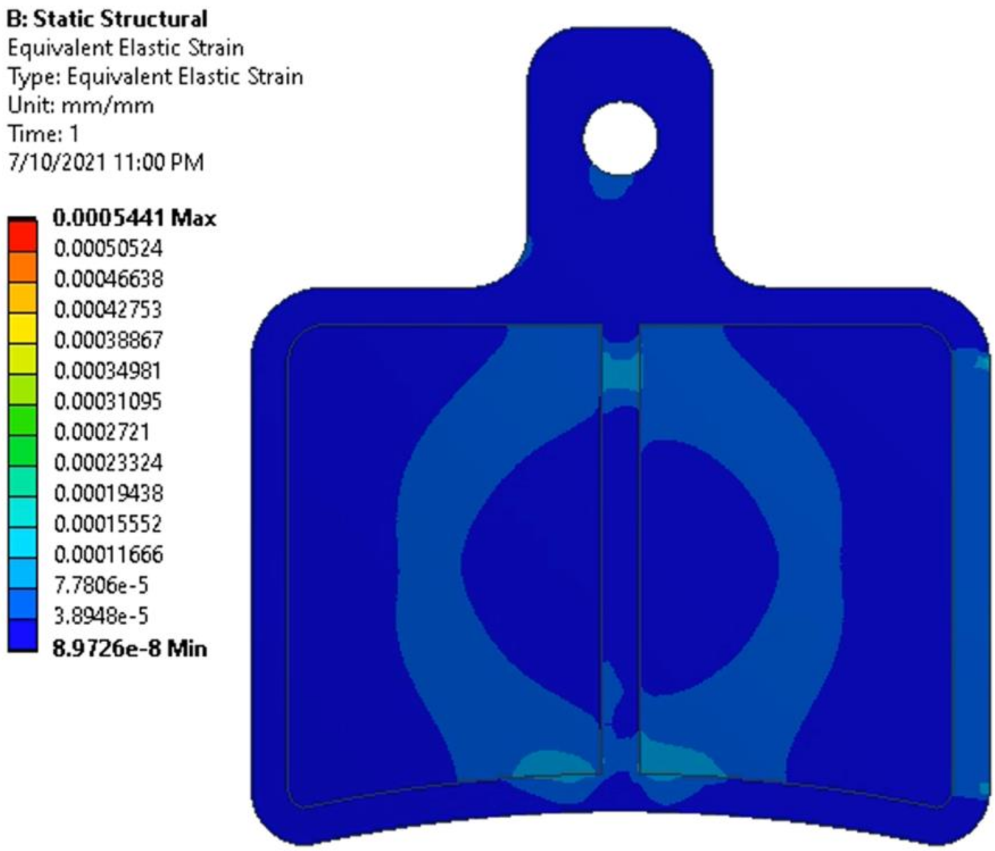


Figure 47 - Strain deformation of brake pad under braking load.

4.4 Rotor, Pad & Piston Thermal Analysis (Fletcher Johnson, n9896791)

4.4.1 Hand Calculations

A braking system works by converting kinetic energy (K.E) to thermal energy (T.E) through dynamic friction between the brake rotor and brake pads [2]. Most of the thermal energy is transferred through the calliper system as it is predominantly made from metallic materials which naturally have a high thermal conductivity [26]. Some of this thermal energy is radiated to the atmosphere before it connects through the calliper system as high velocity air passes over the brake pads and rotor, although this is largely dependent on how much air passes over the calliper [2]. For the QEV3 race car the calliper and rotor are housed deep within the wheel where is it hypothesised it will not receive much airflow, and hence will be reflected in the simulations.

The significance of performing a thermal analysis is to determine if the materials selected for the major components can safely operate during a worst-case loading scenario. During a FSAE competition, the cars compete in a series dynamic test, where one of these tests is measuring how long it takes to stop the race car from 100km/h to 0km/h in a straight line as quickly as possible. This event will be used as the worst-case loading scenario and the parameters are listed in table 13.

The thermal analysis is time dependant, meaning the temperatures read on the parts depends on how much time has passed. The thermal analysis will be broken down into two time-intervals, one immediately after the braking period has finished, and another 5 seconds after. The purpose of measuring the heat immediately after breaking is to find the maximum temperature possible on the brake pad, as it is expected the brake pad which reach the highest temperature in the system. Firstly, the amount of time spent breaking must be calculated:

Table 13 - Thermal analysis operational conditions.

Initial Conditions	Value
Initial velocity	$27.7\ \mathrm{m/s}$
Final velocity	$0\ \mathrm{m/s}$
Time	unknown
Deceleration	unknown
Mass	305kg
$\mathbf{F_{Breaking\ total}}$ (Appendix B)	4188.87 N

$$F = ma$$

$$-4188.87 = 305 \times a$$

$$a = -13.734\ m/s^2$$

$$a = \frac{v-u}{t}$$

$$-13.734 = \frac{0 - 27.7}{t}$$

$$t = 2.02\ seconds$$

Therefore, it will take 2.02 seconds to stop the car from 100km/h to 0km/h. This value will be used as the first time-interval to measure the maximum temperature on the brake pad.

ANSYS was used to simulate the thermal load on the calliper assembly and rotor. This will be achieved by putting a heat flux load on the faces of the brake pads, and on the rotor. However, this heat flux value must be calculated by hand:

$$Kinetic\ Energy = \left(\frac{mass\ on\ rear\ wheel}{2}\right) \times \left(\frac{initial\ velocity}{2}\right)$$

$$K.E = \left(\frac{41.45\ kg}{2}\right) \times \left(\frac{27.2^2}{2}\right)$$

$$K.E = 7951.04\ [J]$$

$$Power = \frac{K.E}{time}$$

$$P = \frac{7951.04}{2.02}$$

$$P = 3936.15\ [W]$$

$$Heat\ Flux\ (Brake\ pad) = \frac{Power}{Area\ of\ brake\ pad}$$

$$Heat\ Flux = \frac{3936.15}{0.001812}$$

$$Heat\ Flux = 2172272.856 \left[\frac{W}{m^2}\right] per\ rear\ wheel$$

$$Heat\ Flux = 1086136.428 \left[\frac{W}{m^2}\right] per\ brake\ pad$$

$$Heat\ Flux\ (Brake\ rotor) = \frac{3936.15}{Area\ of\ brake\ rotor}$$

$$Heat\ Flux = \frac{3936.15}{0.01119}$$

$$Heat\ Flux = 351756.78\ \left[\frac{W}{m^2}\right]\ per\ rear\ brake\ disc$$

Now the results were analysed:

Note: The ambient air temperature was set to 22 degrees Celsius.

4.4.2 Calliper Thermal Analysis

Firstly, the brake calliper just after the 2 second braking period will be analysed:

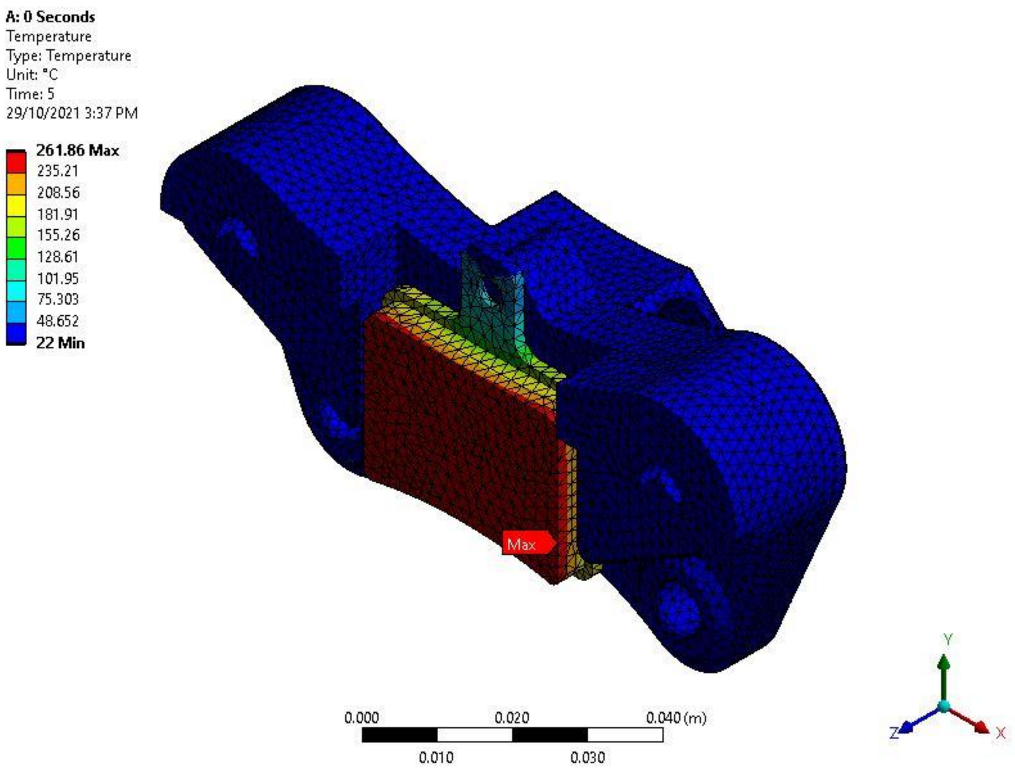


Figure 48: Brake Calliper Thermal at T=0

As it can be seen above in Figure 48, the maximum temperature achieved is on the brake pad at 261.86 degrees Celsius.

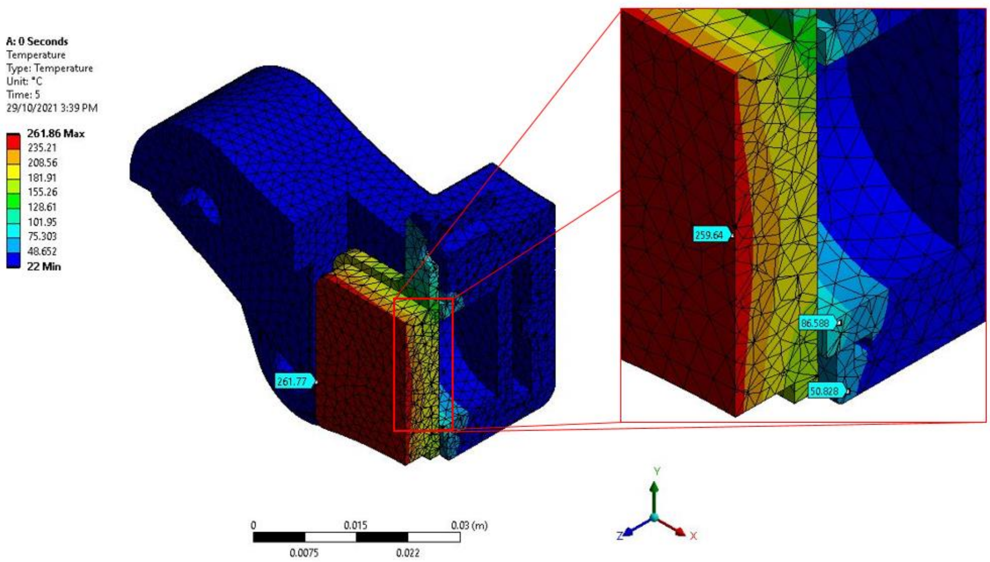


Figure 49: Brake Calliper Thermal at T=0 Section 1.

As seen above in figure 49 and in figure 50, during the worst-case scenario the piston had a maximum temperature of 86.6 degrees Celsius and the calliper body had a maximum temperature of 50.8 degrees Celsius.

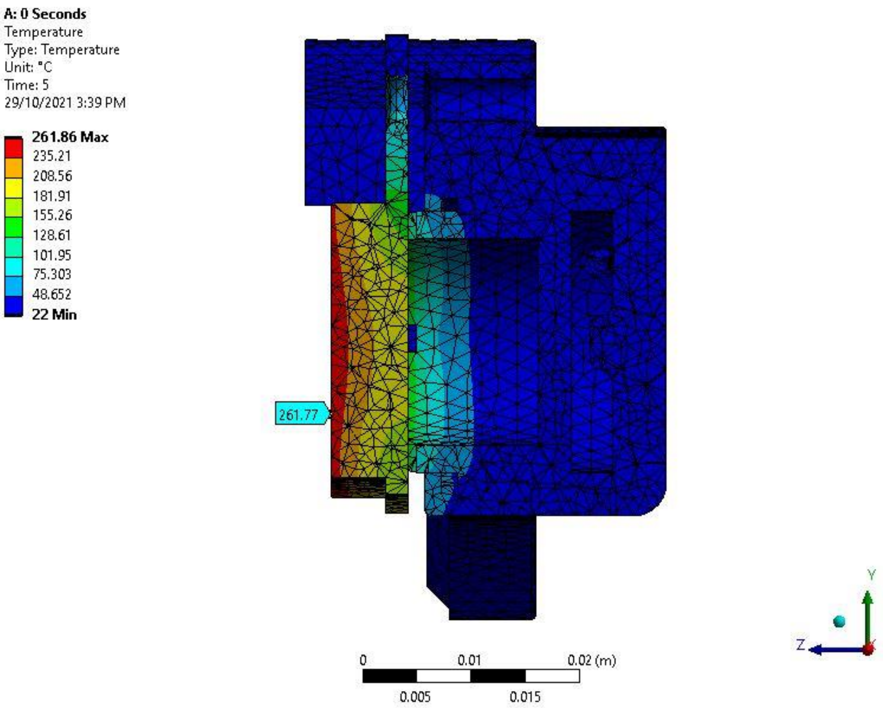


Figure 50: Brake Calliper Thermal at T=0 Section 2.

Now the brake calliper system will be analysed after 5 seconds of resting with no thermal load applied:

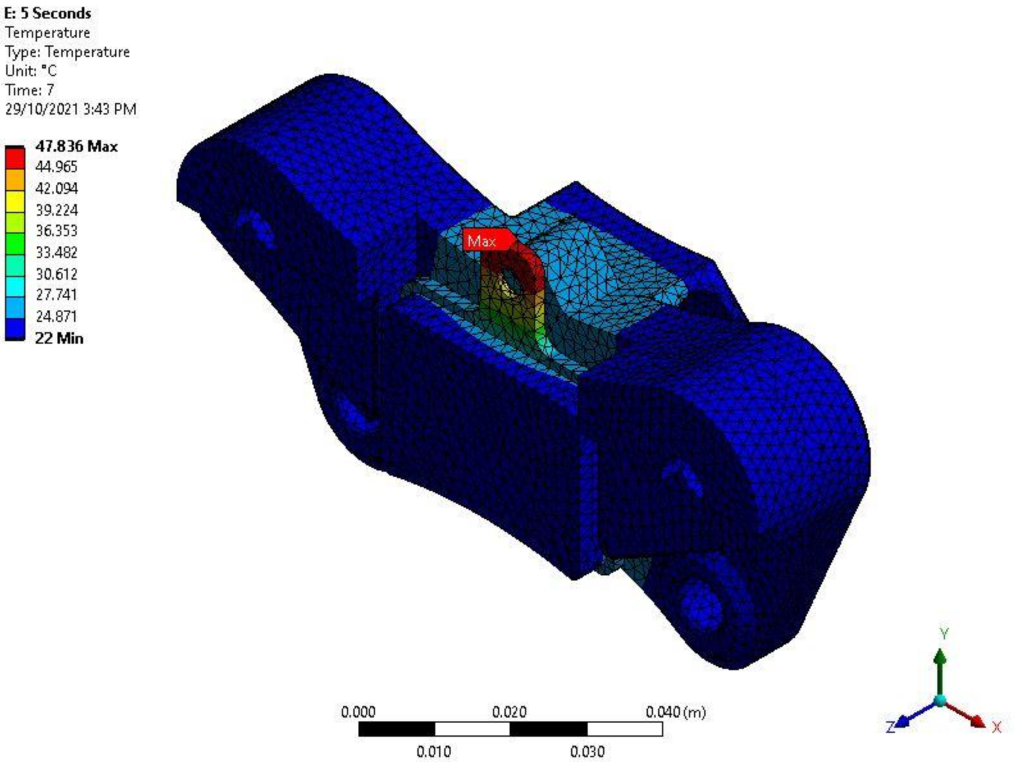


Figure 51: Brake Calliper Thermal at T = 5

As it can be seen above in Figure 51, the maximum temperature on the brake pad at 47.836 degrees Celsius.

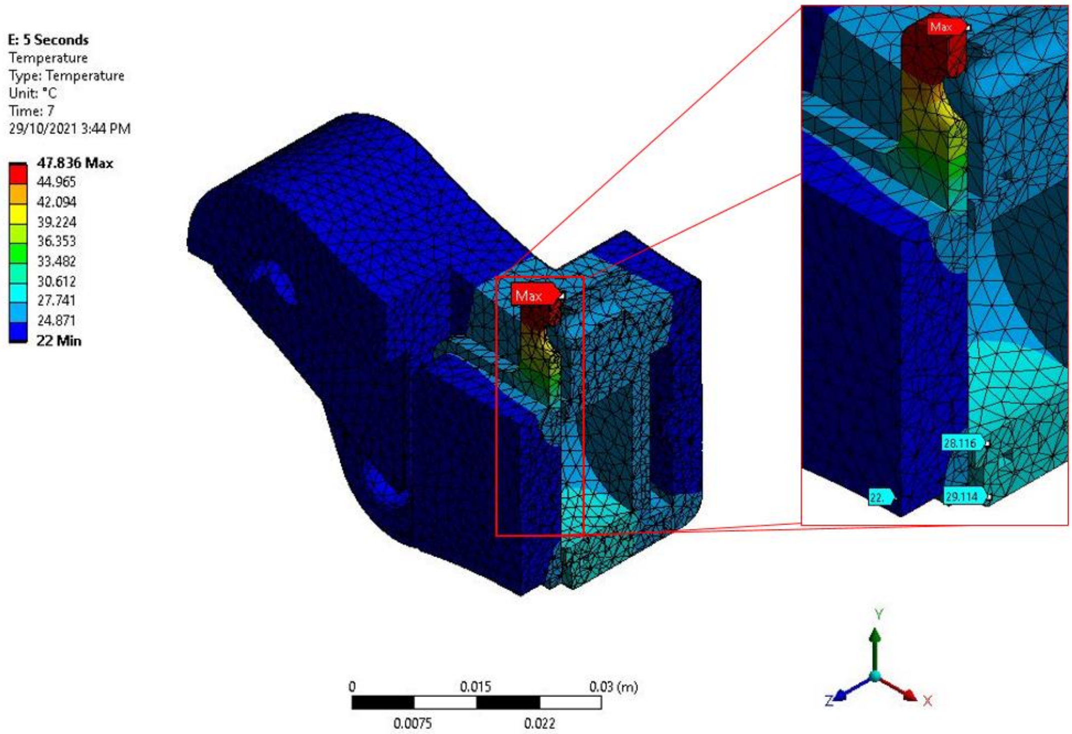


Figure 52: Brake Calliper Thermal at T=5 Section.

As seen above in figure 52, after 5 seconds of resting the piston had a maximum temperature of 28.116 degrees Celsius and the calliper body had a maximum temperature of 29.114 degrees Celsius. This is reflected in figure 53.

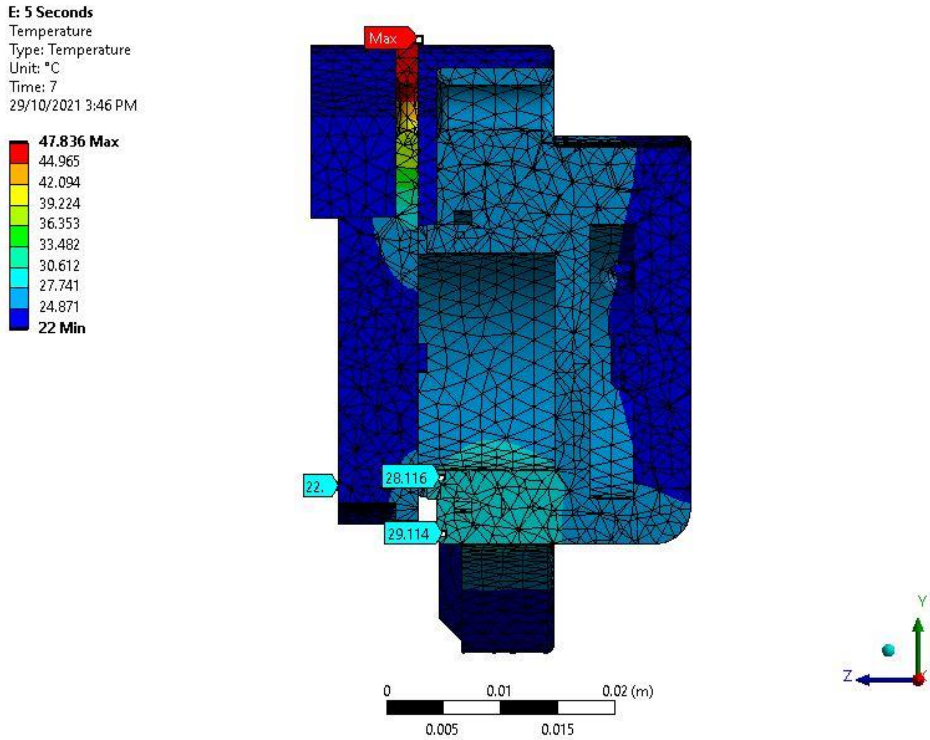


Figure 53: Brake Calliper Thermal at T=5 Section 2.

4.4.3 Rotor Thermal Analysis

Now the brake rotor will be analysed immediately after the braking has finished and it is allowed to cool and diffuse:

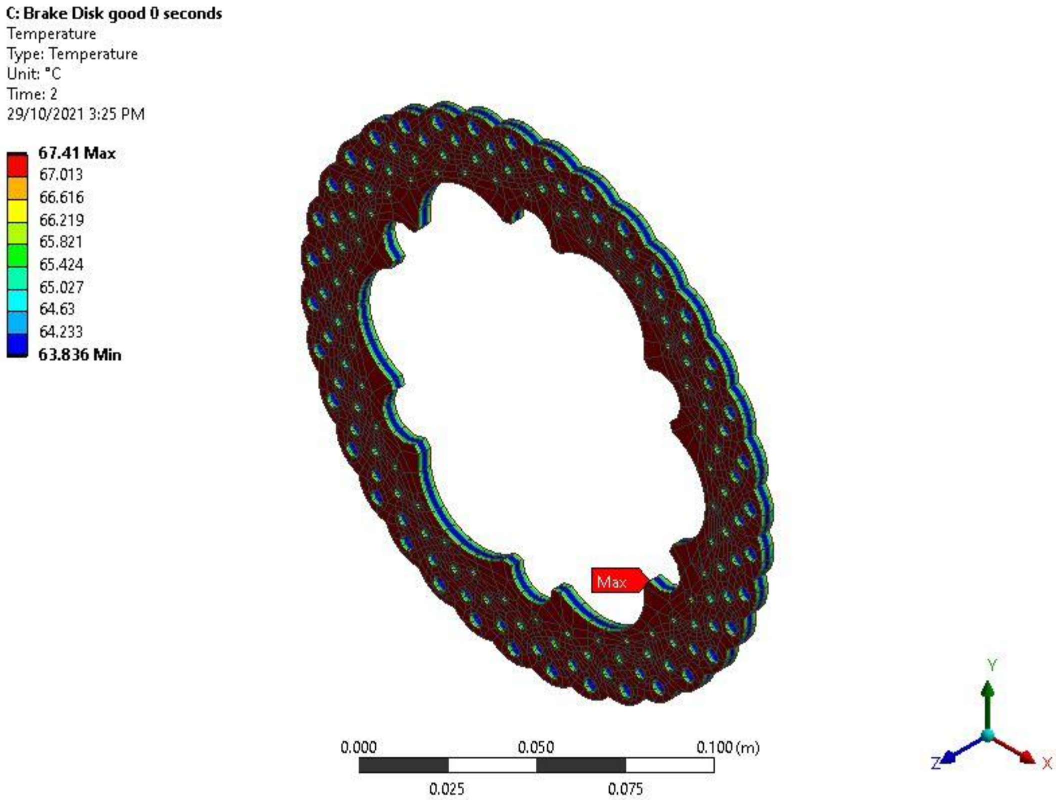


Figure 54: Brake rotor at T=0

As it can be seen above in Figure 54, the maximum temperature on the brake rotor is 67.41 degrees Celsius at the braking surface.

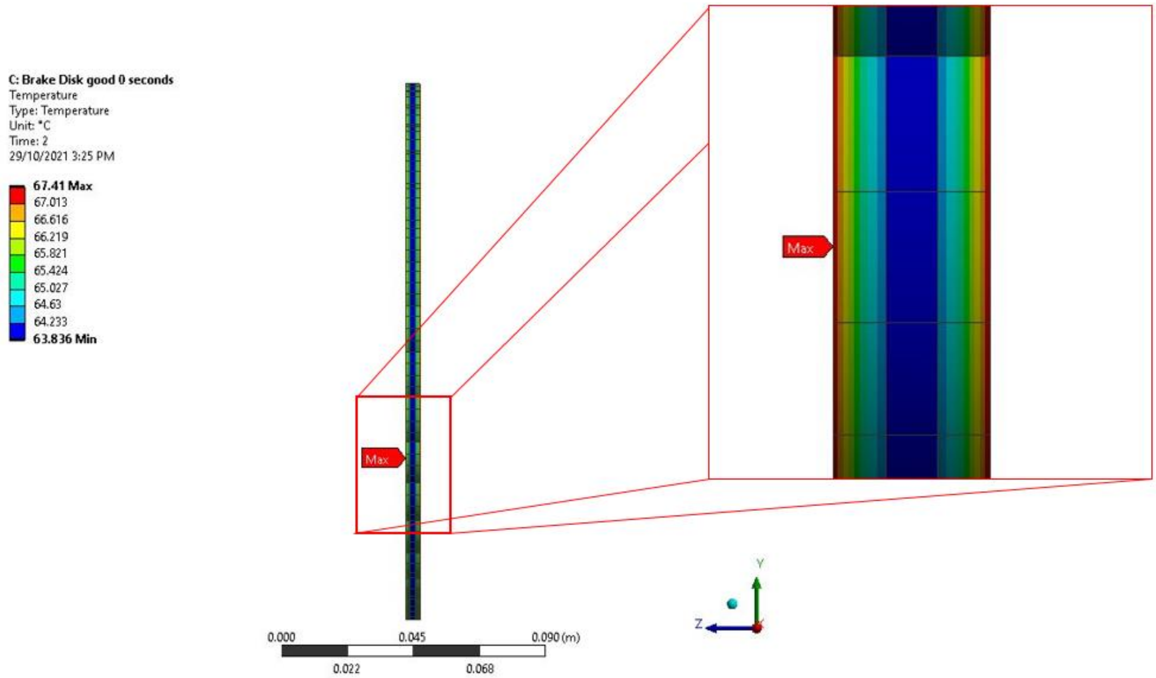


Figure 55: Brake rotor at T=0 detail

Now the brake rotor will be analysed 5 seconds after the braking has finished:

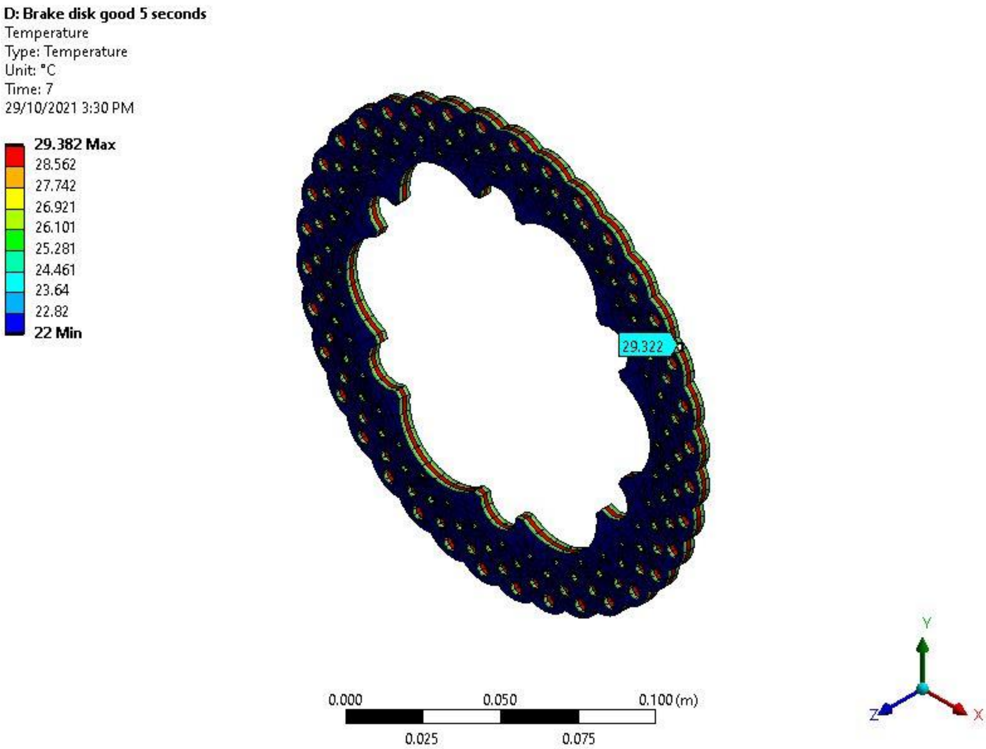


Figure 56: Brake rotor at T=5

As it can be seen above in Figure 56, the maximum temperature on the brake rotor is 29.382 degrees Celsius. This was reflected in figure 57 which shows a close-up view of the rotor lateral direction.

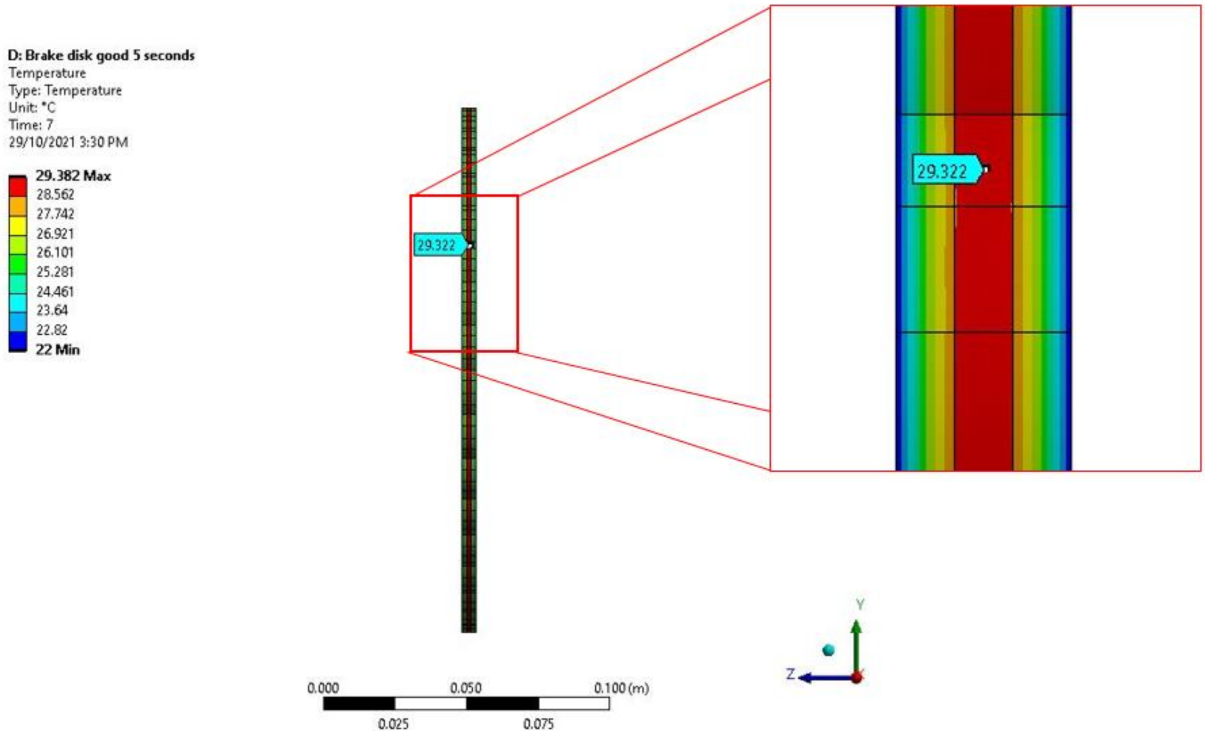


Figure 57: Brake rotor at T=5 detail.

4.4.4 Discussion and Material Evaluation

During the worst-case braking scenario, the hottest surface was on the brake pad, and it reached a maximum temperature of 261.86^oC. Although it is difficult to confirm these ANSYS results without a real-world test, a value around this range was expected as these are typical rear brake temperatures on FSAE cars [2]. After 5 seconds the temperature on the brake pad surface decreased to 22 degrees, as there is no thermal load, and the thermal energy is being dissipated through the calliper system. However, it must be noted the hottest surface of the calliper body after 5 seconds was the pad retaining pin holes on the brake pads at 47.83^oC. The most likely reason for this occurring is due to the pistons have a small contact area with the back of the brake pad, and may restrict conduction between these two parts, hence most of the thermal energy remains in the brake pads. This thermal analysis does not change the selection of brake pads for this calliper as the OTS brake pads are rated for temperatures up to 400°C.

During the worst-case braking scenario, the piston reached a maximum temperature of 86.6^oC, and after 5 seconds of no braking, the temperature decreased to 28.11^oC. This maximum temperature does not change the material selection of 316 stainless steel for the piston as it is well within safe operating temperatures, and it does not exceed the maximum service temperature of 916^oC for AISI 316 stainless steel. Additionally, the calliper body reached a maximum temperature of 50.8^oC, and after 5 seconds of no braking, the temperature decreased to 29.11^oC. This maximum temperature does not change the material selection of 7075 T6 Aluminium for the piston as it is well within safe operating temperatures, and it does not exceed the maximum service temperature of 477^oC.

The brake fluid which is used in the QEV3 race car has a boiling point of 270^oC. It is important this fluid does not approach its boiling point as braking performance is significantly reduced and may cause permanent damage to the entire braking system. Under the worst-case scenario, the piston and calliper body which house the braking fluid did not exceed 86.6^oC which is within the safe operating range for the fluid. It is unlikely the fluid will reach boiling point because the housing parts return to near ambient temperature after 5 seconds of cooling. Thus, it was concluded that the design was able to effectively manage heat.

Although the ANSYS simulation results are representative of real-world results, there are limitations which need to be discussed. These limitations may cause variances in real-world vs simulation results. Firstly, the thermal simulations do not account for brake fluid in the calliper. This is a limitation because brake fluid conducts heat from the pistons and calliper body. If the fluid could be modelled, it is hypothesised the temperature of the calliper body and pistons will be lower than the simulations seen above.

Another limitation of this model is it fails to account for the cooling effect of air passing over the calliper and rotor as the vehicle moves. Although the brakes will still reach the same temperature, they may cool down faster with flowing air. As mentioned before, it is unlikely much air will pass over this system however in reality the surface temperatures may be lower than the simulations predict.

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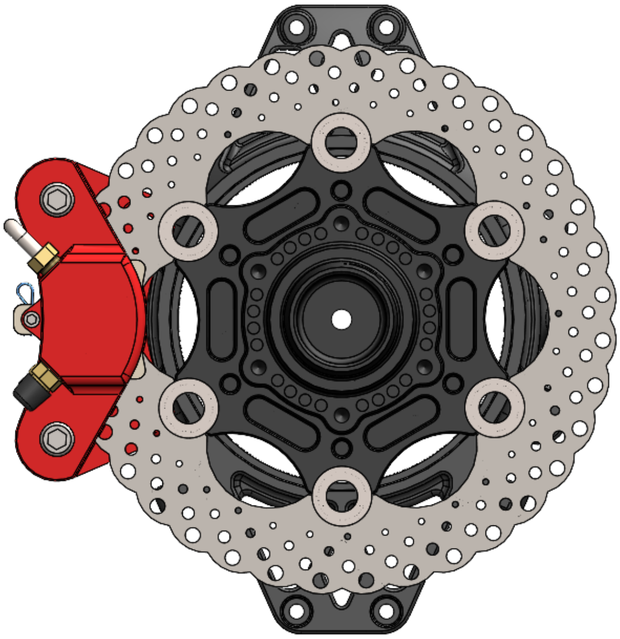
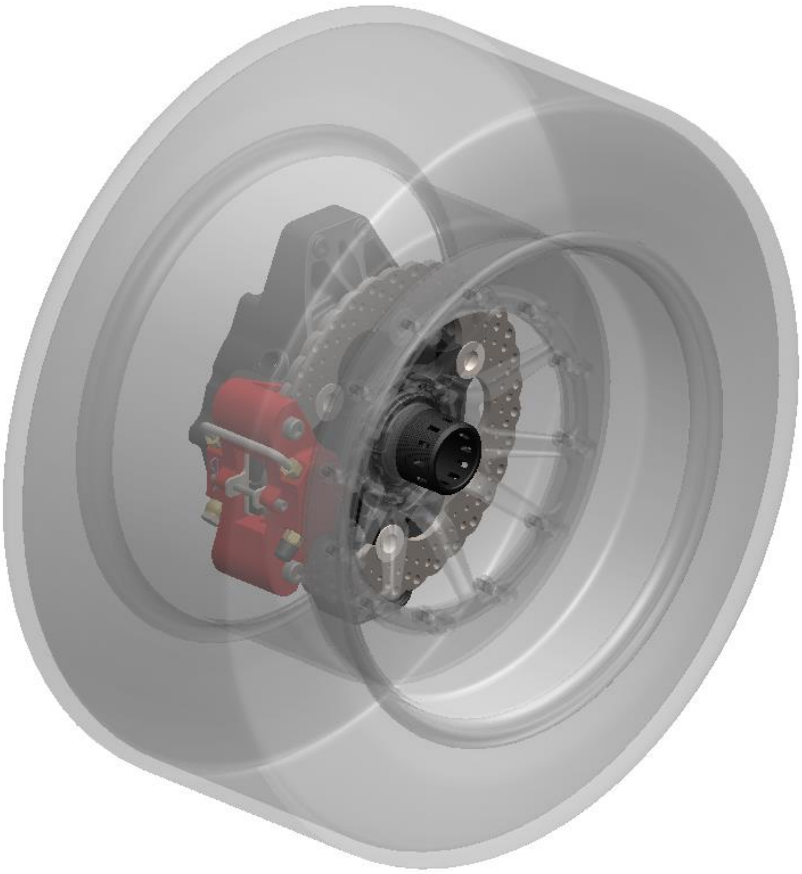
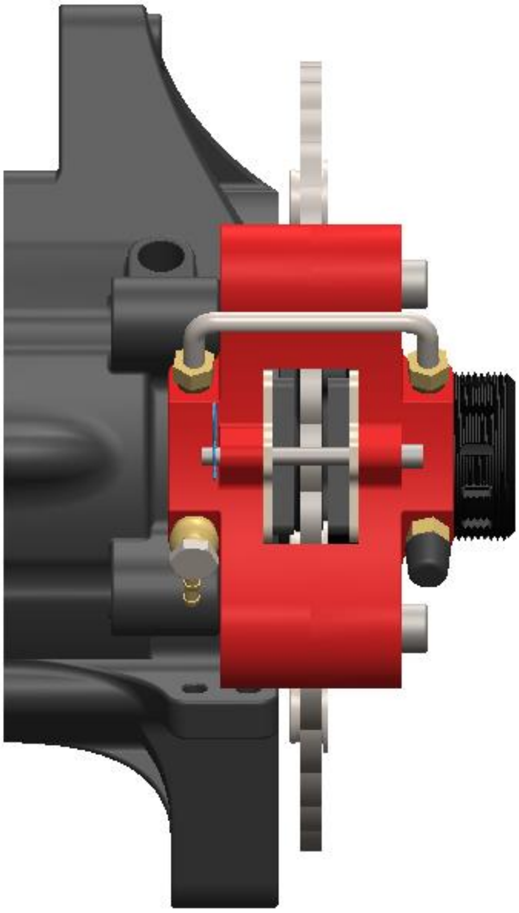
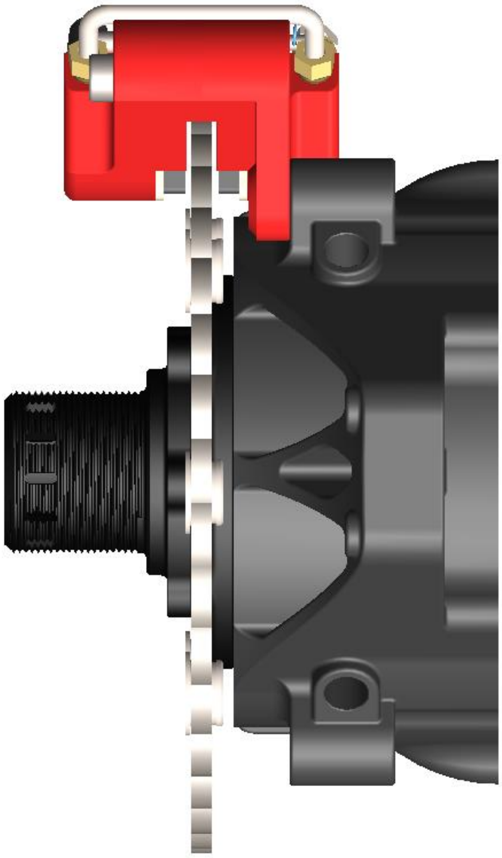
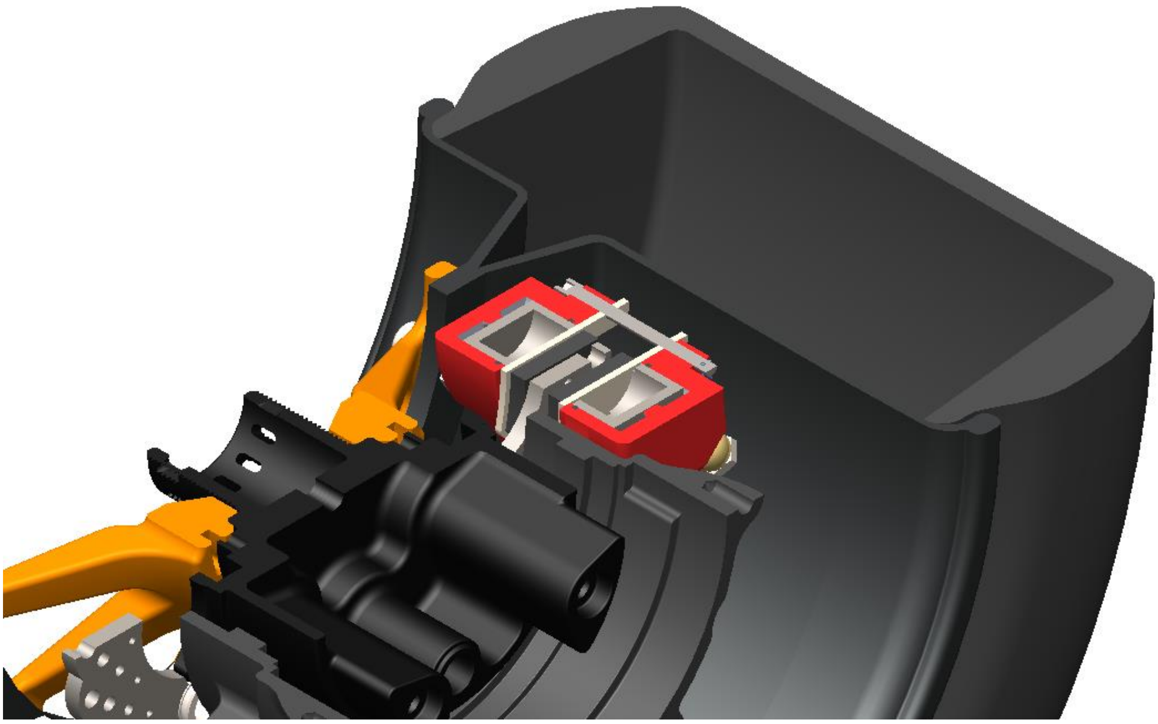
Appendix A – Design Summary Supplementary Images





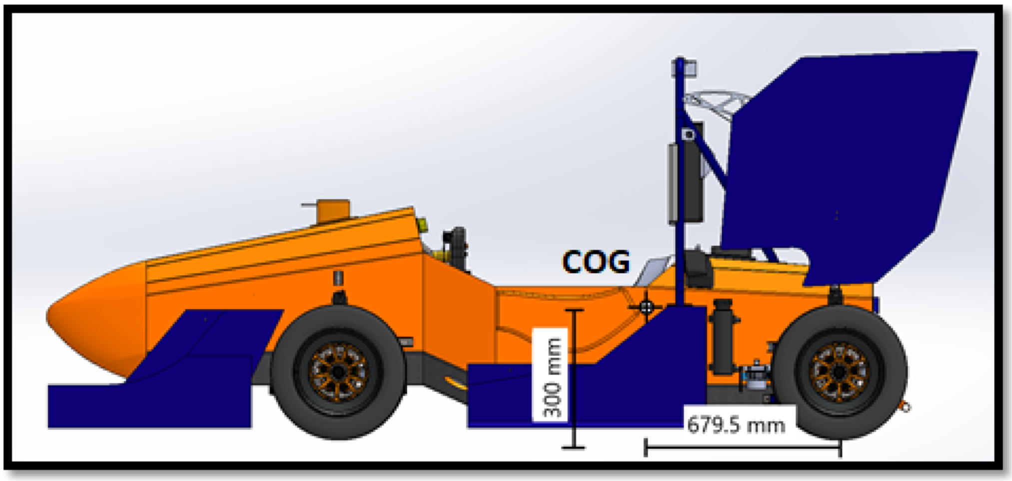






Appendix B – Systems Level Design Updated Calculation Summary

The mass of the car and an average driver is 305 kg. The wheelbase of the car is 1510 mm and the location of the COG in the QEV3 car is shown in the image below:



The parameter value inputs:

Parameter	Value	Unit
Tyre/Road friction coefficient (maximum), μ_s	1.4	-
Pad/Rotor friction coefficient (avg minimum), μ_k	0.32	-
Tyre rolling radius	210	mm
Rotor outer dia.	184	mm
Rotor contact minor dia.	141	mm
Mean rotor radius.	81.25	mm
Calliper piston dia.	24	mm
Number of pistons per rear calliper, n	2	-

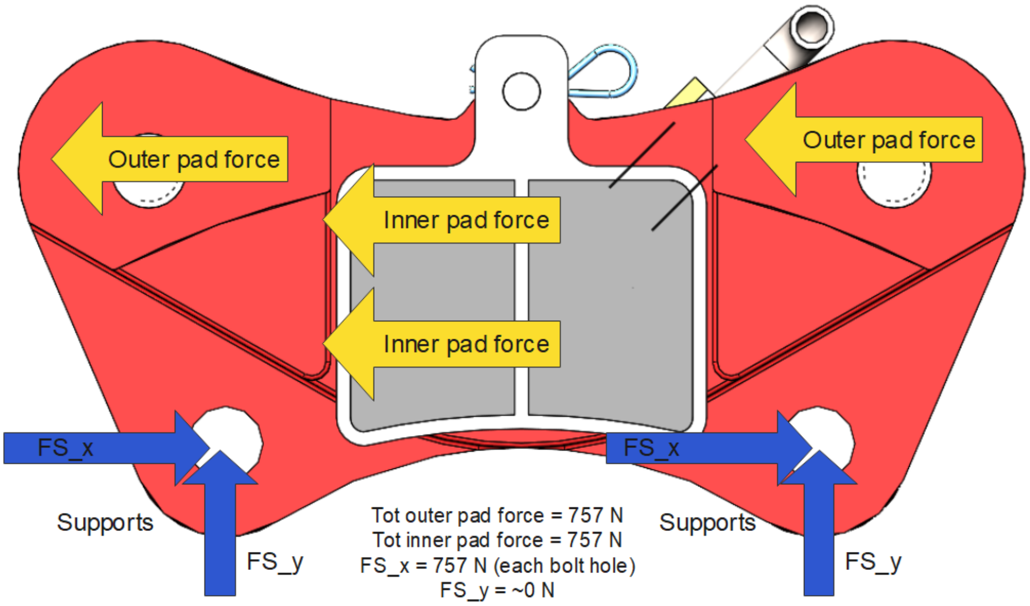
Calculations to determine the normal load applied to each rear wheel yielded the following values:

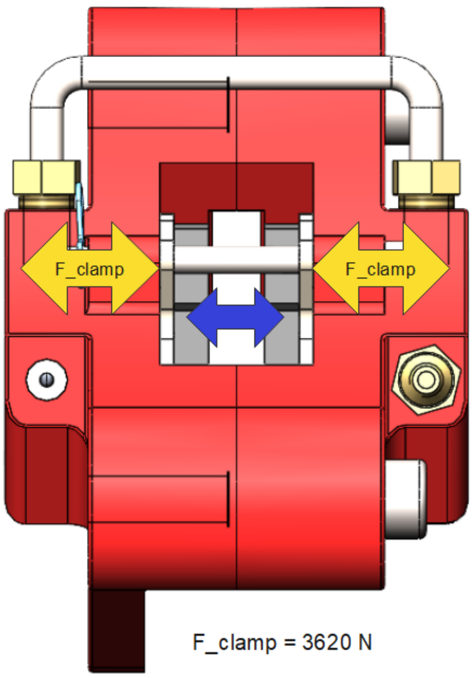
Parameter	Formula	Value	Unit
Weight force, W	$= 305 * 9.81$	2992.05	N
Static axle load front, FNF	$= W - FNR$	1346.42	N
Static axle load rear, FNR	$= 305 \times (1510 - 679.5)/1510$	1645.63	N
Total braking force, FBRAKE	$= W \times \mu_s$	4188.87	N
Longitudinal weight transfer, LW	$= FBRAKE \times \left(\frac{300}{1510}\right)$	832.23	N
Front wheel load (each), LF	$= FNF + LW$	1089.32	N
Rear wheel load (each), LR	$= FNR - LW$	406.70	N
Rear wheel brake force max, FBR	$= LR \times \mu_s$	569.4	N

Resulting torque required from brake: $T_{brake} = 569.4\ N \times 0.21\ m = 120\ Nm$

Required thrust support force from calliper: $F_{calliperLoad} = \frac{120}{0.08125} = 1513\ N$ (757 N per pad).

Thus, in the worst-case load, the driver gives maximum pressure input (9000 kPa) in response to wanting maximum brake power. This exceeds the required pressure (approximately 2500 kPa); however, as the tyre breaks traction and slips, the maximum tangential tractive force is limited by the normal force on the tyre and the friction coefficient between the tyre and the road. Thus, thrust force on the calliper becomes independent of input pressure. The following free body diagram captures these loads.


Outer pad force
Outer pad force
Inner pad force
Inner pad force
FS_x
FS_x
Supports
Supports
FS_y
FS_y
Tot outer pad force = 757 N
Tot inner pad force = 757 N
FS_x = 757 N (each bolt hole)
FS_y = ~0 N


F_clamp
F_clamp
F_clamp = 3620 N

Appendix C – Risk Assessment Matrix

(Over page)

RISK MANAGEMENT PLAN

This form consists of a Project Information page and Risk Register. You are required to complete both pages and submit for approval prior to commencing any project.

The form is linked to the QUT Assurance and Risk Management Services Charter (MOPP A/1.5), which should be read prior to completing this form and contains guidance notes, examples of common hazards, predictable risks and suggested controls.

Project Title:

Rear Brake Calliper Design for QEV3

Project Type (tick appropriate box / boxes)

UG Project (Individual or Group)	Y	**PG Project (Individual or Group)**		**Staff Research Project**		**Site Visit**		**Asset / Equip Procurement**			
								Corporate Level Project			
UG Class Exercise		**PG Class Project or Tutorial**		**Collaborative Project**		**Field Activity**	Y	**Commercial Project**			
UG Tutorial		**PG Research Project**		**Work Activity**		**Social Networking**		**Domestic**		**Overseas**	

Project Category (School / Portfolio / Discipline)

CPME (Specify Discipline Below)		**Design** (Specify Discipline Below)		**Urban Development** (Specify Discipline Below)		**Research** (Specify Discipline Below)	
		Mechanical					
Teaching & Learning		**Faculty Operations**		**External Relations**		**Other (detail below)**	

Other

Project/Work Details (Provide details of the exact nature of work – If space insufficient, add a page)
Attach copies of SOP, sketch, design, permit, authority or other relevant documents

Work entails assembly, installation, maintenance and operation of a new rear brake calliper on the QEV3 car of QUT Motorsport. The car is considered operating in FSAE guided race events and maintenance may be carried out at a race or track day or at QUT Motorsport lab.

Project Location: *QUT Gardens Point and various race venues in SEQ*	**Proposed commencement date**

Project Team Members names Contact phone/mobile numbers Student ID No. If applicable (Attach list if necessary)	**Names:** (Please Print)	**Student ID No.**	**Contact Phone / Mobile N0.**
	William Whigham	N10232869	
	Jacob Garcia-Pavy	N10012478	
	Fletcher Johnson	N9896791	
	Todd Dalgliesh	N6004075	

Project Supervisor/s (Name & Phone number)	**Name/s** (Please Print) *Veronica Gray*	**Phone No.**

Valid to / Review by (Insert DATE)	**Actual Review Date**	**Reviewed By** (Please Print)	**Signature**

Risk Assessment and Risk Register

No	Hazard, Activity, Task, or Process	Identified Risks	Risk Level Initial	Proposed Control Measures (All control measures must comply with legislative requirements and follow the Control Hierarchy below).	Residual Risk Level Final	Notes / Remarks 1) Who is responsible to implement controls? 2) Latest implementation date? 3) Other relevant detail.
1	*Sharp edges and sharp tools used for assembly and maintenance of the calliper. This includes sharp objects on the car that maintenance persons may come in contact with.*	*Cuts to hands and arms while assembling or handling calliper components. Particular components of concern are the calliper halves, suspension assembly and brake rotor (typically sharp).*	*Low -*	*QEV3 maintenance personnel to be appropriately trained in use of any tools needed for assembly/maintenance (Administrative).* *Burrs on calliper components to be removed during manufacture to reduce sharp edges (Elimination).* *Personnel to wear mechanic's gloves and long-sleeve shirts for protection (PPE).*	*Negligible*	*1) All persons involved in mechanical work on the vehicle are to adhere.* *2) Tooling subject to team assets*
2	*Removal of wheel from the car and associated hazards (i.e., car elevated) assuming car is not in the controlled workshop environment*	*Car could fall or move while wheel is off, injuring toes of persons.* *Lifting of wheel and car causing over-strain of person's back.*	*Low +*	*Chock car wheels to avoid it from rolling (Elimination).* *Use rated lifting equipment to support the car (Eng Controls).*	*Low -*	*1) QUTMS procedures to be followed here. Hazard stated for good measure.*
3	*Hot brake components and surrounding equipment such as motor units following operation.*	*Burns to maintenance personnel if hands or arms come in contact with calliper or brake rotor in particular.*	*Low -*	*Design and fitment of guard to brake rotor and calliper (Eng control).* *Personnel to wear long-sleeve shirt and gloves if working on hot brake (PPE).* *Wait for brake to cool before conducting maintenance (if viable) (Elimination).*	*Negligible*	*1) guard was identified as impractical considering this is a performance vehicle and additional weight associated would outweigh benefit in safety. Accept residual risk without.* *2) Team may need to work on hot brakes during an event.*
4	*Hazardous chemical (liquid) – Brake fluid DOT5 or 5.1*	*Risk of burns and irritation to skin if contact occurs.* *Burning and irritation to eyes, possibly causing loss of sight due to corrosive nature.* *Contamination of workspace floor and other parts of the car causing damage.* *Environmental contamination causing environmental degradation.*	*Moderate -*	*Personnel working on brake system to wear long-sleeve shirt and safety glasses at all times (PPE).* *Ensure operational safety shower or wash basin is near by to irrigate any contact with eyes or skin (Administrative).* *Dispose of waste fluids in accordance with QUTMS procedures. Spills to be captured or cleaned as they happen (Administrative).*	*Negligible*	*1) PPE is the main prevention method used.* *2) Assuming QUTMS has procedures in place as they already use other lubricants and coolant fluids in the car.*
5	*Disposal of worn brake pads, broken or damaged or worn calliper components.*	*Improper disposal could contaminate surrounding environments.* *Disturbance of brake dust particles and inhalation causing injury.*	*Low -*	*Personnel to wear particulate masks when disposing of soiled components and spent brake pads (PPE).*	*Negligible*	

6	*Incorrectly secured calliper on car (fasteners not tight enough)*	*Brake calliper may fail causing loss of braking to one or both rear wheels. Driver may lose control causing damage to vehicle and possible injury to driver or bystanders.*	*Moderate +*	*Bolt torques to be specified and verified as part of design specification. May also indicate torque spec on calliper itself (Administrative).* *Rear brakes are independent of front brakes; thus, driver should be capable of slowing down in this scenario (Eng control).*	*Low -*	*1) Assumed maintenance personnel are appropriately trained to interpret and follow assembly instruction.*
7	*Reduced or poor functionality of brakes due to improper manufacture or maintenance.*	*Risk of car being involved in an accident if brake fails to perform as driver expects. Injury to driver or bystander.*	*Low -*	*Specify inspection of all parts during maintenance processes and provide QUTMS with detailed step-by-step maintenance procedure (Administrative).*	*Negligible*	*1) Maintenance manual and inspection guidelines to be implemented in future work. Outside project scope.*
8	*Loss of brake fluid containment from the calliper during driving operation.*	*Risk of car being involved in an accident if brake fails to perform as driver expects. Injury to driver or bystander.* *Damage to surrounding components.*	*Low -*	*Specify inspection of all parts during maintenance processes and provide QUTMS with detailed step-by-step maintenance procedure (Administrative).* *Design calliper to minimise leak points and reduce the change of overpressure by designing with high FOS (Eng control)*	*Negligible*	*1) Major design change → replace internal fluid passages with external fluid pipe with high rated pressure. This is aimed at eliminating the risk of leakage between the calliper halves.*
9	*High torque required to tighten fasteners on calliper and calliper body.*	*Injury (pinching or bruising) to hands if tool is to slip.*	*Low -*	*Ensure fasteners are easily accessible on the car so that personnel don't require hands to be in the path of other components (Eng control)* *Persons to wear gloves (PPE)*	*Negligible*	*1) Calliper bolts to be orientated with bolt heads on the outboard side. This should facilitate the easy access of the bolts when the wheel is off the car.*
		Overtightening causing bolts to strip and calliper failure *Damage to tool if bolt head "rounds-out"*	*Low -*	*Torque wrench to be specified for use by personnel (Administrative)* *Selection of suitably hard material for fasteners so is to resist deformation at required torques (Eng control)*	*Negligible*	
10	*Corrosion of calliper body, fittings and/or other components.*	*Component failure causing loss of braking performance and possible crash causing injury to persons and monetary loss for the team.*	*Negligible*	*Select corrosion-stable materials suitable for the race car application and prospective environment (elevated temperature) (Eng control).*	*Negligible*	*1) Considered very low risk due to regular maintenance of the car and brakes. Low exposure to outside environment.*
11	*Damage or contamination to calliper from track debris*	*Performance degradation resulting in increased risk of crash, injury to persons and damage to car* *Lock-up or significant brake degradation enough to cause DNF in event. (Financial and reputation risk)*	*Low -*	*Design guard to protect vulnerable areas of the brake calliper from damage (Eng control).* *Visual checks of brake calliper and rotor on each wheel before and after each event (Administrative).*	*Negligible*	*1) Guard considered but may not be practical considering the weighting of performance over risk mitigation (considering low risk).* *2) Procedure to be implemented in future work as part of operations manual supplied to QUTMS.*

Risk Assessment Conducted By (Author)	Name (Print)	Appointment	Student No (If Applicable)	Signature	Date
	Jacob Garcia-Pavy & William Whigham	-	-	WW	25/10/21
I certify that I have consulted with appropriate personnel and have considered professional advice and/or relevant information in conducting this assessment and have obtained necessary approvals, permits, licences as applicable to the project.				Yes	

APPROVAL ***Note> Individuals approving this document accept responsibility for the appropriateness of controls and for the validity of the Risk Management Plan.***

Approved By	**Name** (PRINT)	Group leader	**Appointment / Title**	**Signature**	**Date**

Example Risk Assessment EGH420 Mechanical Systems Design

Distribution:

- **Original (Hard Copy) held by Author for duration of project and must be available to management, health and safety personnel, WH&S QLD Inspector or other authorised person on request.**
- **Electronic or Hard Copy held by Approver for the duration of the project.**

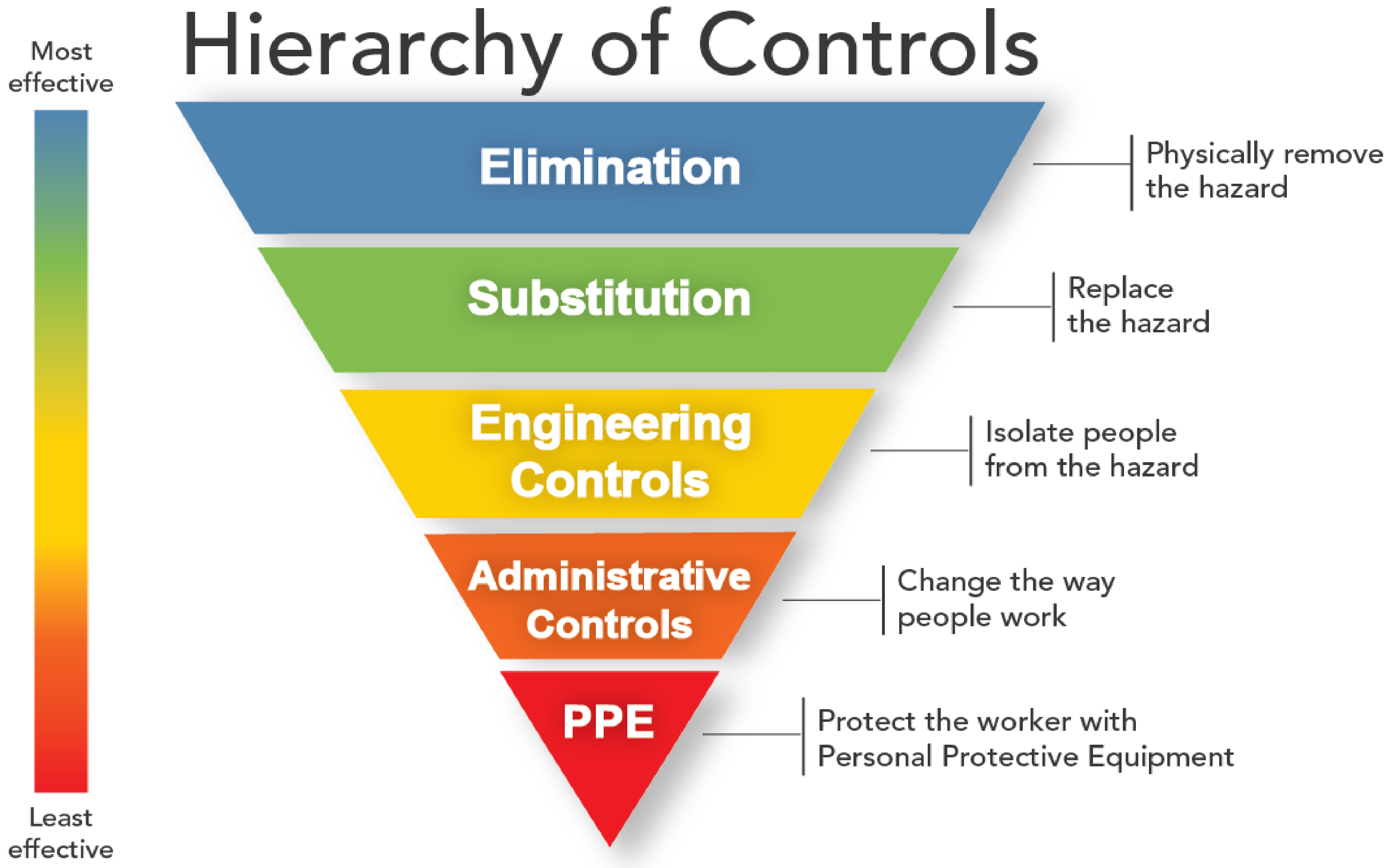
Risk Calculator

	Consequences				
Likelihood	**Insignificant**	**Minor**	**Moderate**	**Major**	**Catastrophic**
Almost Certain - Expected in most circumstances	Low +	Moderate +	High	Very High	Extreme
Likely - Will probably occur in most circumstances	Low -	Moderate -	Moderate +	High	Very High
Possible - Might occur at some time	Negligible	Low -	Moderate -	Moderate +	High
Unlikely - Could occur at some time	Negligible	Low -	Low +	Moderate -	Moderate +
Rare - May occur only in exceptional circumstances	Negligible	Negligible	Negligible	Low -	Low +

Consequences	How severely could it hurt someone/cause damage?
Catastrophic	Death or large number of serious injuries, environmental disaster, huge cost
Major	Serious injury, extensive injuries, severe environmental damage, major cost
Moderate	Medical treatment required, contained environmental impact, high cost
Minor	First aid treatment required, some environmental and/or financial impact
Insignificant	No injuries, low financial/environmental impact

Risk score	What should I do?
Extreme	Immediate action required
Very High	Senior management attention required
High	Action plan required, senior management attention needed
Moderate	Specific monitoring or procedures required, management responsibility must be specified
Low	Manage through routine procedures
Negligible	Accept the risk

<u>**Control Hierarchy**</u>



Appendix D – Cost Breakdown for All Components with Alternatives

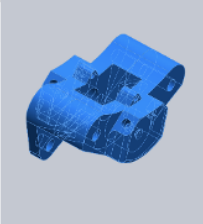
Part name	Manufacturer	Part Number	Part description	Supplier	Price	Length/ amount	QTY req	freight	Total price	selected
brake line	Aeroflow	AF66 - 4316	hardline for caliper crossover	Supercheap auto	69.99	7.5m		0	69.99	
	Aeroflow	AF66 - 4316	hardline for caliper crossover	outlawspeedshop	72	7.5m		0	72	
	Aeroflow	AF66 - 4316	hardline for caliper crossover	sparesbox	49.95	7.5m	0.3	9.9	59.85	
	Pirtek		cut per meter	Pirtek	6.66			0	6.66	6.66
									0	
flare nut	Aeroflow		S/S inverted flare nut tube Nut 3/16' hardline ti M12x1.0	Motorsport accessories	14.71	2	2	10.43	25.14	
	Aeroflow		S/S inverted flare nut tube Nut 3/16' hardline ti M10x1.0	sparesbox	13.95	2	2	15.21	29.16	
	Aeroflow	AF364	Stainless steel inverted Flare tube nut	Supercheap auto	19.99	2	2	0	19.99	
	Aeroflow	AF364	Stainless steel inverted Flare tube nut	RCE performance	14.16	2	2	0	14.16	
	Aeroflow	AF373	Stainless steel inverted Flare tube nut	RCE performance	13.13	2	2	0	26.26	26.26
			SAE inverted Flare brake fitting 3/16' kit	Ebay	42.44	24		0	0	
	Aeroflow	AF364	Stainless steel inverted Flare tube nut	Aeroflow	14.71	2		15.02	15.02	
									0	
R-clip	Pinnacle	I/N:2420762	3/8 x 1/2' Zinc plated R clip - 6 pack	Bunnings	4.82	6	1	0	4.82	4.82
	Pinnacle	I/N:2420759	3/16 - 7/16' Zinc plated Assorted r- clip	Bunnings	6.1	12		0	0	
	Brembo	5454218	Brembo caliper pad Pin r clip	Superbikesupply	3	1		5	5	
		PMBR6	retainig clip to suit brake pad disc pad small steel clip	Ebay	10	5.16		0	0	
	Brembo	5454218	Brembo caliper pad Pin r clip	Selexon Trading	0	100		10.45	10.45	
		CD4076P	retaining pin to hold pid CD4075 in place	Flying spares	1.07	1		0	0	
	Brembo	5454218	Brembo caliper pad Pin r clip	EasyR	2.49	1		5.43	5.43	
									0	
Pad Pin		CD4075P	Titanium Pins used to secure the pads from moving out of place	Flying spares	4.24	1	2	10.35	18.83	18.83
		D58-33-096	DRC KTM/HUSK brembo stainless front brake pin clip 56mm length	Mxstore	14.95	1		0	0	
	brembo	120280010	Brembo 05 caliper pin kit	ebay	20	1		0	0	
	Brembo		Husq/KTM brake pins	APEX import	17.96	2		0	0	
	Brembo	TIPINBP012	Titanium brake caliper pin 45mm length	PB Pro-bolt	20.99	1		0	0	
	Nissan	SKU:200-006	S14,s15,R33,r34, z32 brake pad pin front	EFI solutions	7.7	1		9.98	9.98	
	Nissan	SKU:200-007	R32,Z32 fron tbrake pad pin	EFI solutions	6.6	1		9.98	9.98	
	brembo	B00AP01YS8	titanium brake caliper pad pin	Alibaba	38.37	2		0	0	
									0	
Brake nipple		SSBN-2z1	Stainless steel brake nipple	PB pro bolt	14.5	2	1	0	14.5	
			M8 x 1.0 mm brake bleed screw	ebay	5.93	4	1	0	5.93	5.93
		BNM10x1.00	M10 x 1.0 mm brake bleed screw	Hel performance	5	2	1	10.45	15.45	
		H160-31BNC	M10x1.00 single banjo Bleed Nipple Bolt in stainless steel	Hel performance	15.75	1	1	10.45	26.2	
			Bleed bleeder caliper dent brand	Australian go kart spare	9.9	1	1	10.45	20.35	
		SKU:FL83_4PK	M6 x 1.00 mm 4 pack Brass bleed nipple	abtoolsonline	39.11	4	1	10.9	50.01	

Brake dust cap	Hel	BNDC	Blled nipple dust cap				Hel performance	1.5	1		5.45	5.45	
		SKU:A9726994	Blled screw cap 10 pk				repco	4.5	10		0	0	
	Hel	BNDC	Blled nipple dust cap				Hel performance	0.98	1		0	0	
	Keenso		4pc brake rubber dust cap				amazon	15.98	4		0	0	
			4 pieces brake dust cap				Aliexpress	3.2	4	1	0	3.2	3.2
												0	
Copper washer	righetti ridolfi	no 12 HAASE/corsa	brake banjo copper washer				Australian go kart spare	1.1	1		5	5	
	HEL		M12 10 pack	performance copper crush washer			ebay	9.5	10		0	0	
	Champion	BH069	copper washers				Supercheap auto	5.99	3	2	0	11.98	11.98
	brake quip	BQ13	ID 8mm OD 13mm thick 1.3mm weight 1g				wide bay brake hose service	1.55	1	4	6	12.2	
	Champion	BH073	copper washers				Supercheap auto	5.99	3		0	0	
												0	
brake pad	Ap racing	CP4226D27	39.7 x 44				Harris performance.com	41.6	1	2	14.35	97.55	
	Ap racing	CP4226D27	39.7 x 44				atomic	37.33	1	2	11.78	86.44	
	Ap racing	CP4226D27	39.7 x 44				mailordercarparts.co.uk	37.44	1	2	10.21	85.09	85.09
												0	
caliper bolt (body)	brembo	b00ATV4532	8mm dia x 42mm long		M8x1.25	titanium		13.66	4	2	35.98	63.3	
	Nissin	90108-mas-e01	m8x45mm	1 pack does 2 callipers		titanium		65	8	1	12.32	77.32	77.32
												0	
caliper bolt (to hub)	brembo		8 mm dia x 33.4mm		M8x1.25	titanium		13.66	2	4	23	77.64	
	Nissin	90108-mas-e01	m8x45mm		M8x45mm	titanium		65	8	1	12.32	77.32	
banjo fitting	RACEWORKS	RWF-720-06ABK	M8 to 6An Banjo fitting				ebay	31.67	1	2	8.66	72	72
	Supercheap auto	SPO3820313	Banjo to 6AN				supercheap auto	43.99	1	2	0	87.98	
												0	
braided brake lines rear	GKETCH		rear braided brake lines				Gktech	79	2	1	15.5	94.5	94.5
	Tora	H1717SSPAIR	set of 2 brake hose				Autoparts online	65.95	2	1	30.5	96.45	
Locktite for mounting bolt	Loctite	IN:1560362	Non-permanent adhesive threadlocker				Bunnings	15.58	1	1	0	15.58	15.58
	Loctite		Non-permanent adhesive threadlocker				ebay	17.5	1	1	0	17.5	
	Loctite		Non-permanent adhesive threadlocker				au-rs online	38.95	1	1	0	38.95	
Caliper Machine cost	CNC machining		CNC machining and finishing of both caliper body sides from 7075					416.73	2	1	0		416.73
Piston Manufacturing									2	1	0	47.06	94.12
Pin Manufacturing									2	1	0	3.45	6.9
									Total cost for rear caliper x 2				939.92

Appendix E – Cost breakdown for final selected parts

Part name	Manufacturer	Part Number	Part description		Supplier	Price	Length/ No. of	QTY req	freight	Total price	selected
brake line	Pirtek		cut per meter		Pirtek	6.66			0	6.66	6.66
flare nut	Aeroflow	AF373	Stainless steel inverted Flare tube nut		RCE performance	13.13	2	2	0	26.26	26.26
R-clip	Pinnacle	I/N:2420762	3/8 x 1/2' Zinc plated R clip - 6 pack		Bunnings	4.82	6	1	0	4.82	4.82
Brake nipple			M8 x 1.0 mm brake bleed screw		ebay	5.93	4	1	0	5.93	5.93
Brake dust cap			4 pieces brake dust cap		Aliexpress	3.2	4	1	0	3.20	3.2
Copper washer	Champion	BH069	copper washers		Supercheap auto	5.99	3	2	0	11.98	11.98
brake pad	Ap racing	CP4226D27	39.7 x 44		mailordercarparts.co.uk	37.44	1	2	10.21	85.09	85.09
caliper bolt (body)	Nissin	90108-mas-e01	m8x45mm	can do both rear calipers with this pack		65	8	1	12.32	77.32	38.66
caliper bolt (to hub)	Nissin	90108-mas-e01	m8x45mm			65	8	1	12.32	77.32	38.66
banjo fitting	RACEWORKS	RWF-720-06ABK	M8 to 6An Banjo fitting		ebay	31.67	1	2	8.66	72.00	72
braided brake lines rear	GKETCH		rear braided brake lines		Gktech	79	2	1	15.5	94.50	94.5
Locktite for mounting bolt	Loctite	IN:1560362	Non-permanent adhesive threadlocker		Bunnings	15.58	1	1	0	15.58	15.58
Caliper Machine cost			Both caliper body sides - performed by CNC			208.37	2	1	0	416.73	416.73
Piston Manufacturing			Stainless steel - CNC				2	1	0	47.06	94.12
Pin Manufacturing			Steel - CNC				2	1	0	3.45	6.9
								Total cost for rear caliper x 2			939.9

Appendix F – Manufacturing cost summary parts – Calliper Body



SOLIDWORKS Costing Report for caliper body

Model Name:	Caliper machine cost
Material	Aluminium 6061
Total weight:	0.9163 kg
Total stock weight:	2.71 kg

Quantity to Produce	
Total number of parts:	2
Lot size:	1

Estimated machine cost per part:	293.22
Cost operation	120 AUD/ hour
Run time approximation:	2.44 hours

Cost of part ready to install:		
Machine cost:	293.22 AUD	100%
Material:	68.51 AUD	100%
Anodizing the caliper	25 AUD	100% - red or blue standard pricing surcharge for other colours
Environmental levy:	30 AUD	100%
Markup:	N/A	0%
TOTAL: (x2 rear calipers)	416.73	

Final cost of 2 rear calipers: $416.73

Custom Body Name	Custom Process	Cost (AUD / Part)	Costing Template
Split1[2]	Custom Cost per Volume Removed	157.57	c:\programdata\solidworks\solidworks 2020\lang\english\costing templates\machiningtemplate_default(englishstandard).sldctm
Split1[1]	Custom Cost per Volume Removed	136.46	c:\programdata\solidworks\solidworks 2020\lang\english\costing templates\machiningtemplate_default(englishstandard).sldctm

Appendix G - Manufacturing cost summary parts – Piston

SOLIDWORKS Costing Report for piston

Model Name:	Piston
Produced for QUT FSAE	Group 5
Manufacturing Method:	Machining
Material:	304 Stainless Steel
Stock weight:	0.068 kg
Stock Type	Block
Block Size:	23.876x23.876x14.986 mm
Material cost/weight:	9.955 AUD/kg
Shop Rate:	30.00 AUD

Quantity to Produce

Total number of parts:	2
Lot size:	1

Estimated cost per part: 47.06 AUD

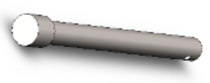
Cost Breakdown

Material:	1.49 AUD	3%
Manufacturing:	45.57 AUD	97%

Estimated time per part: 01:06:52

Setups:	01:05:00
Operations:	00:01:52

Appendix H - Manufacturing cost summary parts – Retaining Pin



SOLIDWORKS Costing Report Pad retaining Pin

Model Name:	Pad Retaining Pin
Produced for QUT FSAE	Group 5
Manufacturing Method:	Machining
Material:	Plain Carbon Steel
Stock weight:	0.0136 kg
Stock Type	Plate
Cylinder Thickness:	0.25 in
Material cost/weight:	0.20 AUD/kg
Shop Rate:	N/A

Quantity to Produce

Total number of parts:	2
Lot size:	1

Estimated cost per part: 3.45 AUD

Cost Breakdown

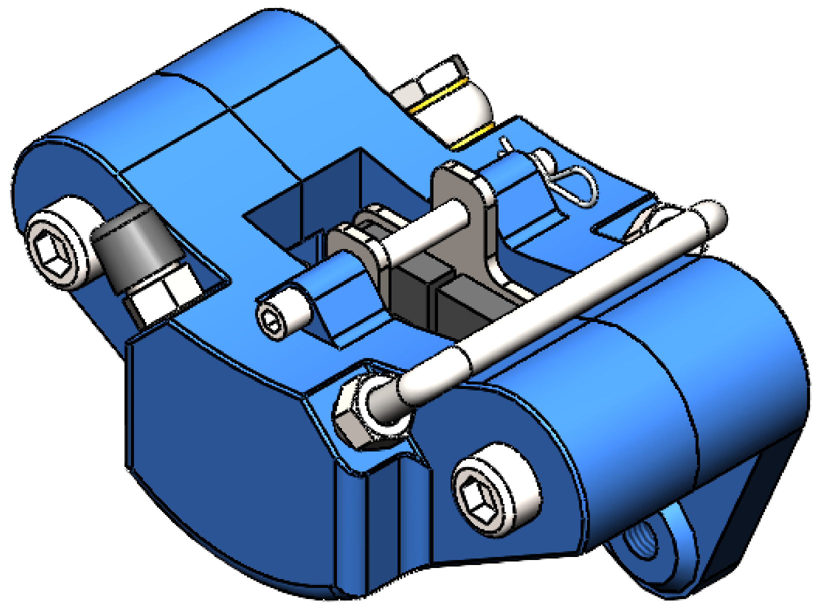
Manufacturing:	3.45 AUD	100%

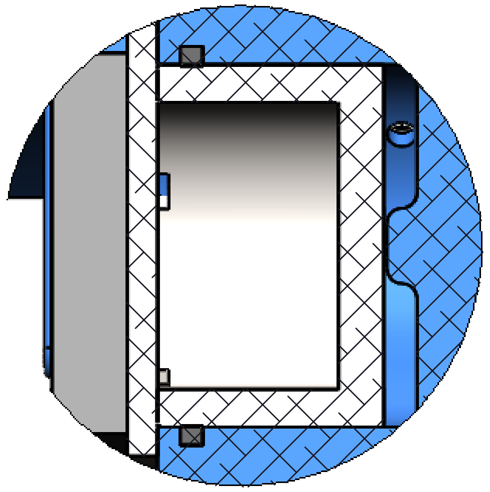
Estimated time per part: 00:05:03

Setups:	00:05:03
Operations:	00:00:00

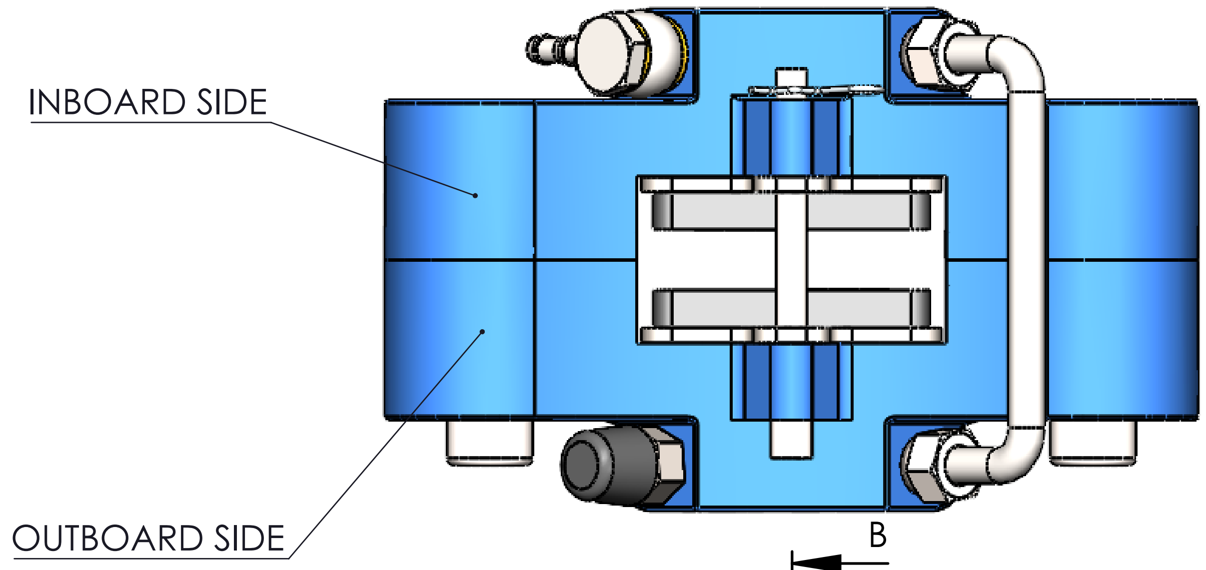
Appendix I – Manufacturing Drawing Set

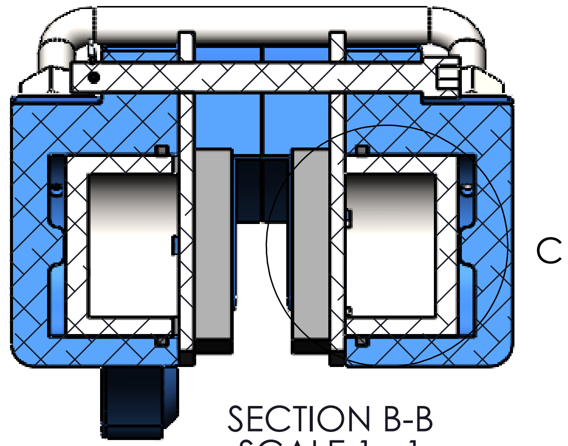
(Over page)



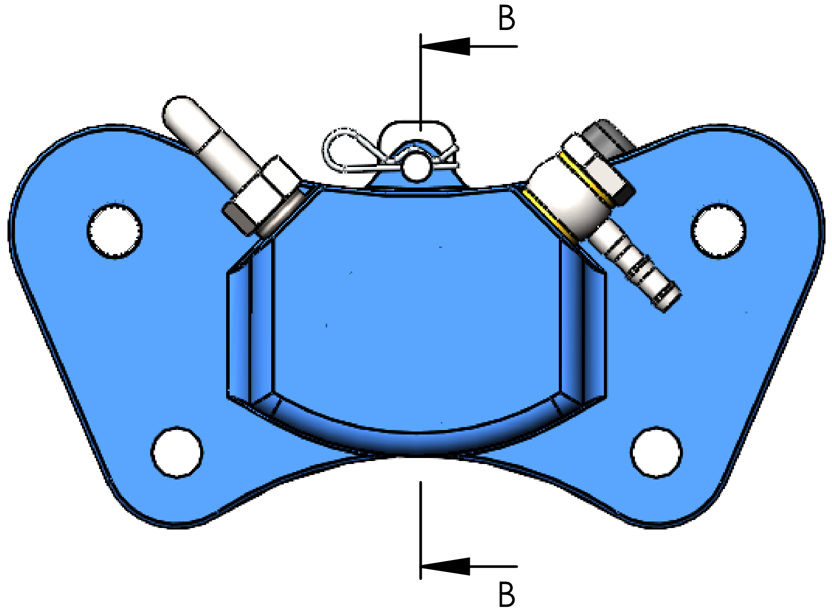


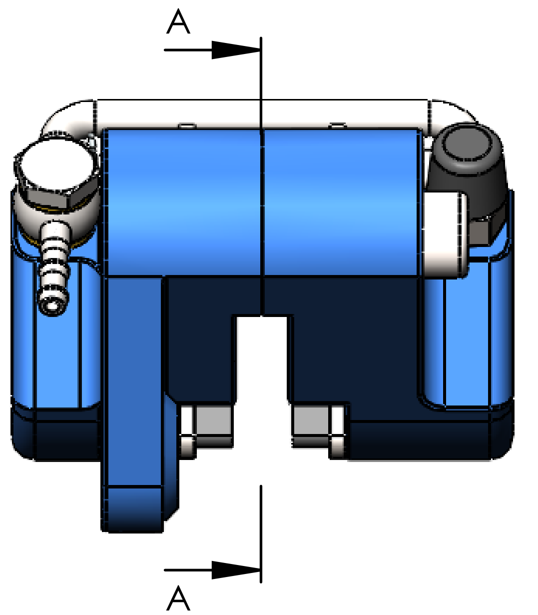
DETAIL C

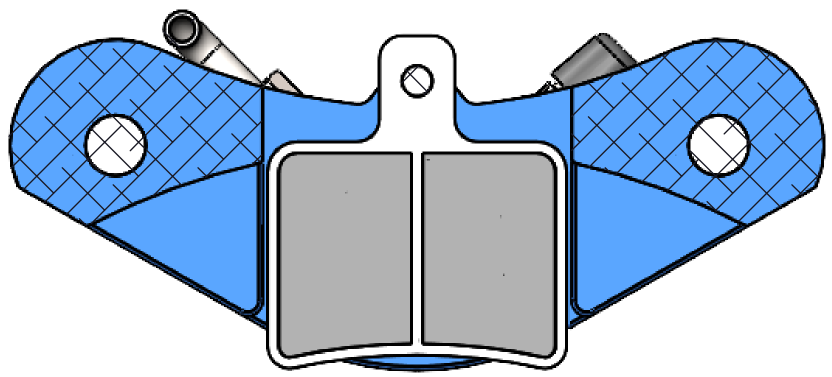




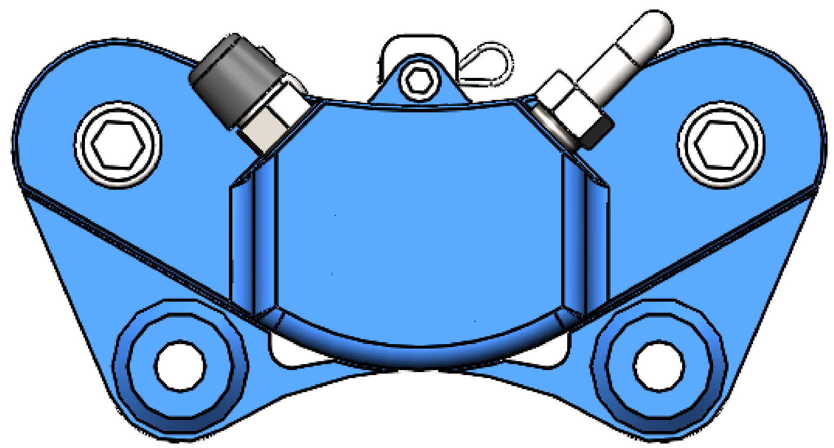
SECTION B-B
SCALE 1 : 1







SECTION A-A
SCALE 1 : 1



QUT

GEN. LIMITS UNLESS STATED

NO PLACES DEC.	± 0.25mm
1 PLACE DEC.	± 0.10mm
2 PLACES DEC.	± 0.05mm
ANGLE 1 PLACE DEC.	± 0.50°
ANGLE 2 PLACES DEC.	± 0.20°
SURFACE FINISH UNLESS STATED	1.6

ALL UNSPECIFIED DIMENSIONS ARE IN mm

NEXT HIGHEST ASSY	**N/A**	QUANTITY	**2**
HEAT TREATMENT	**N/A**	MASS	**560 g**
FINISH	**N/A**		
MATERIAL	**N/A**		
DESIGNED BY	FJ	DATE	10/10/2021
APPROVED BY	WW	DATE	10/10/2021

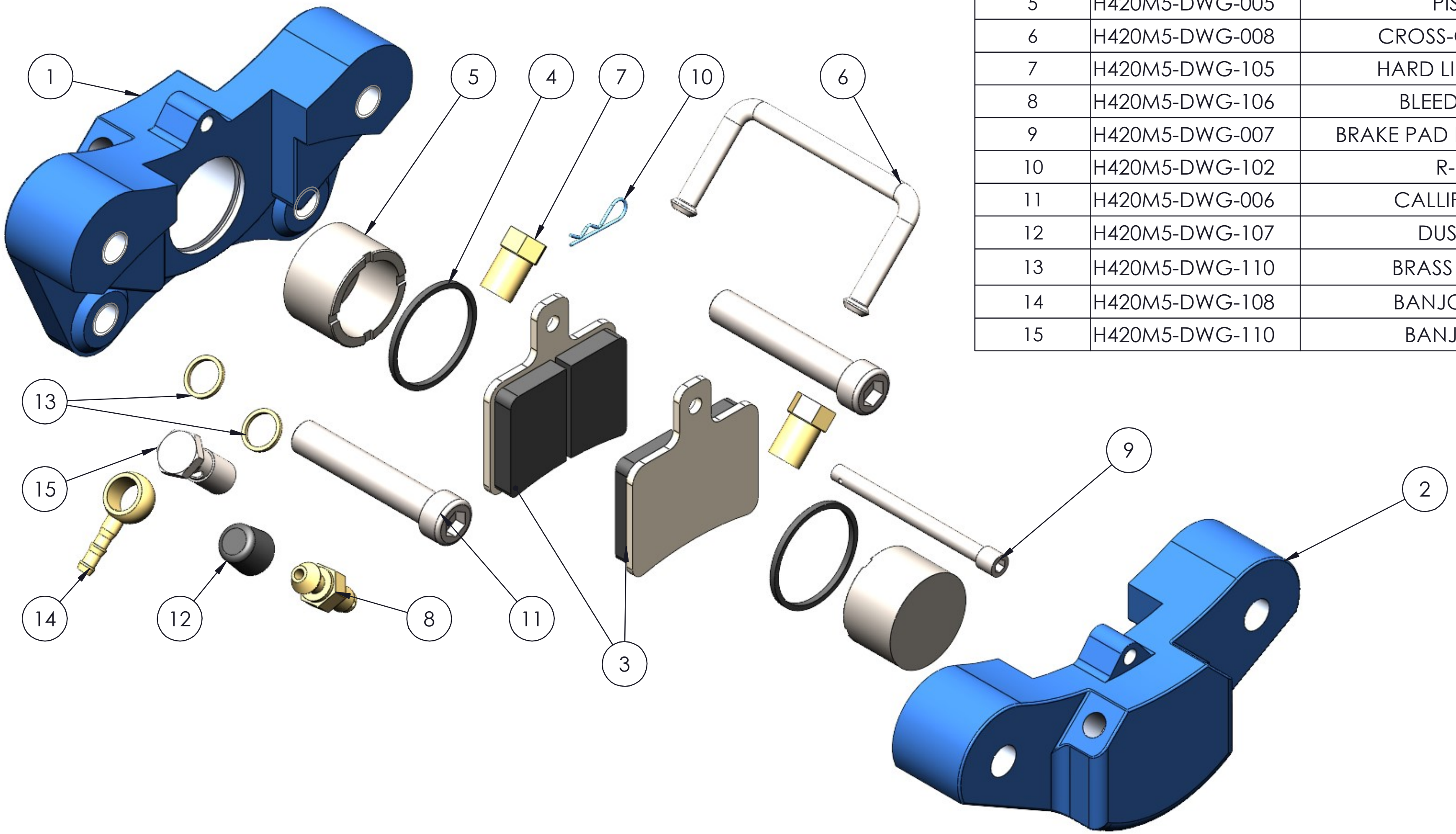
TITLE

BRAKE CALLIPER SYSTEM ASSEMBLY

THIRD ANGLE PROJECTION	SHEET SIZE A3	SCALE 1:1

DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-001	1.0	1	1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	H420M5-DWG-003	CALLIPER BODY - INBOARD	1
2	H420M5-DWG-004	CALLIPER BODY - OUTBOARD	1
3	H420M5-DWG-104	BRAKE PAD	2
4	H420M5-DWG-101	PISTON SEAL	2
5	H420M5-DWG-005	PISTON	2
6	H420M5-DWG-008	CROSS-OVER PIPE	1
7	H420M5-DWG-105	HARD LINE FITTING	2
8	H420M5-DWG-106	BLEED SCREW	1
9	H420M5-DWG-007	BRAKE PAD RETAINING PIN	1
10	H420M5-DWG-102	R-CLIP	1
11	H420M5-DWG-006	CALLIPER BOLT	2
12	H420M5-DWG-107	DUST CAP	1
13	H420M5-DWG-110	BRASS WASHER	2
14	H420M5-DWG-108	BANJO FITTING	1
15	H420M5-DWG-110	BANJO BOLT	1



QUT

GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH UNLESS STATED 1.6
ALL UNSPECIFIED DIMENSIONS ARE IN mm

NEXT HIGHEST ASSY	N/A	QUANTITY	1
HEAT TREATMENT	N/A	MASS	351 g
FINISH	N/A		
MATERIAL	N/A		
DESIGNED BY	FJ	DATE	10/10/21
APPROVED BY	WW	DATE	10/10/21

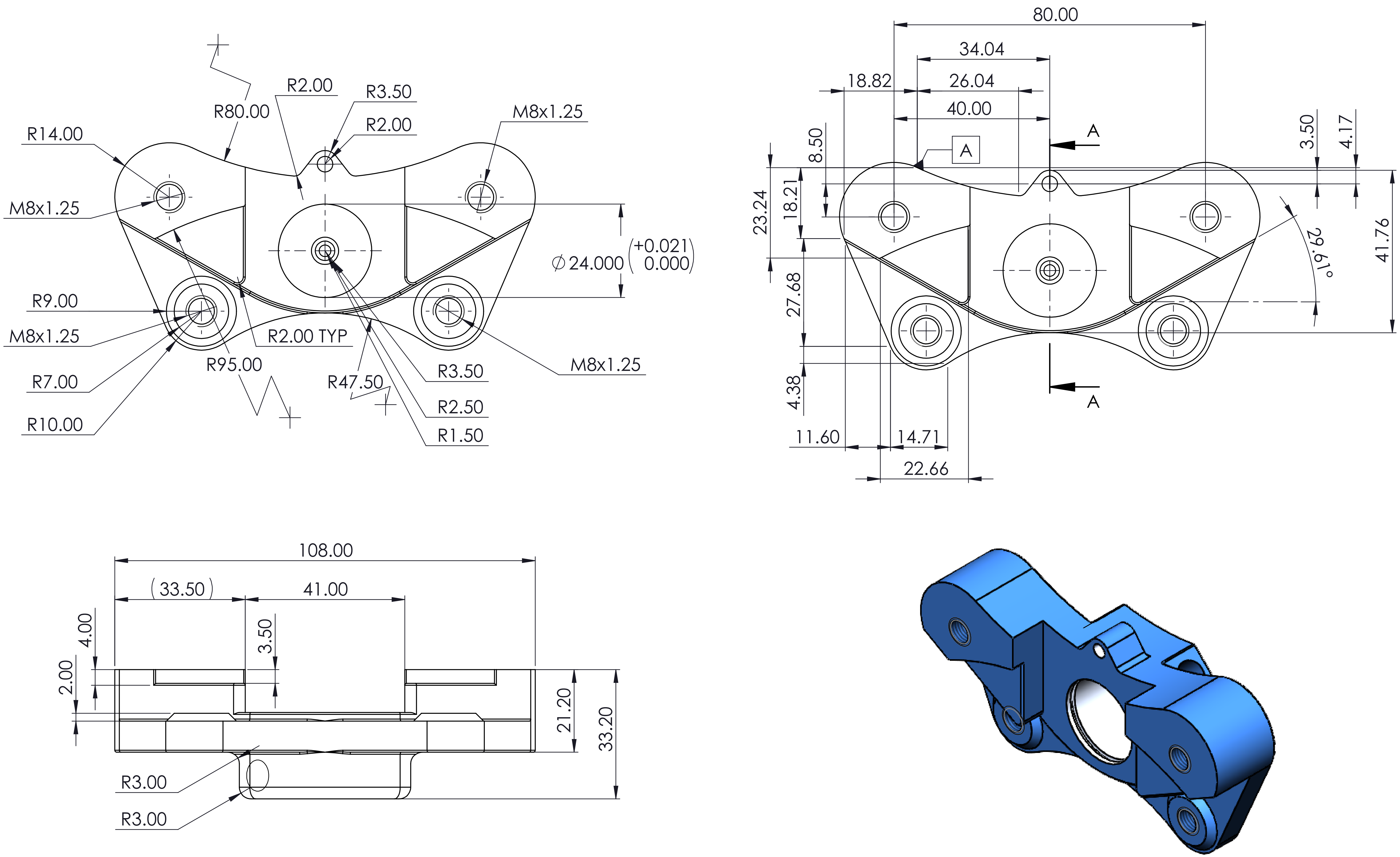
TITLE

BRAKE CALLIPER SYSTEM ASSEMBLY

THIRD ANGLE PROJECTION

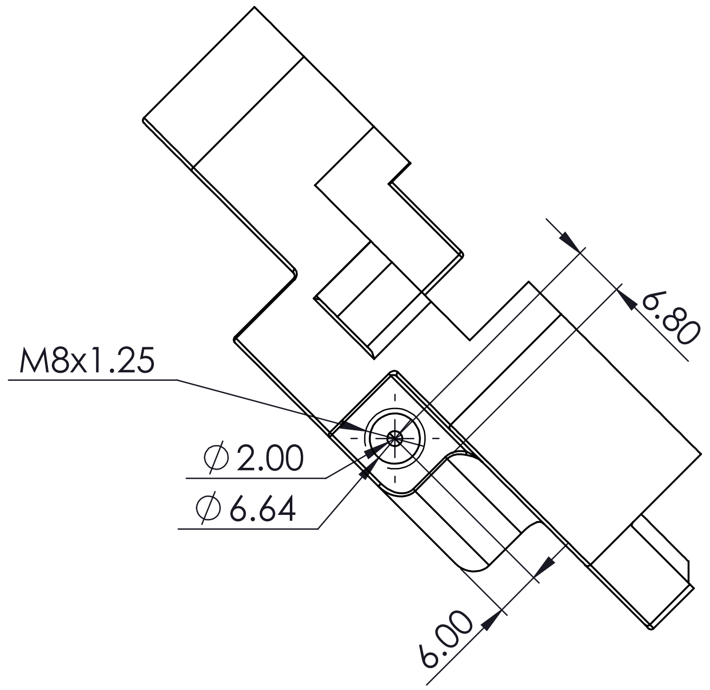
SHEET SIZE A3 | SCALE 1:5

DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-002	1	1	1

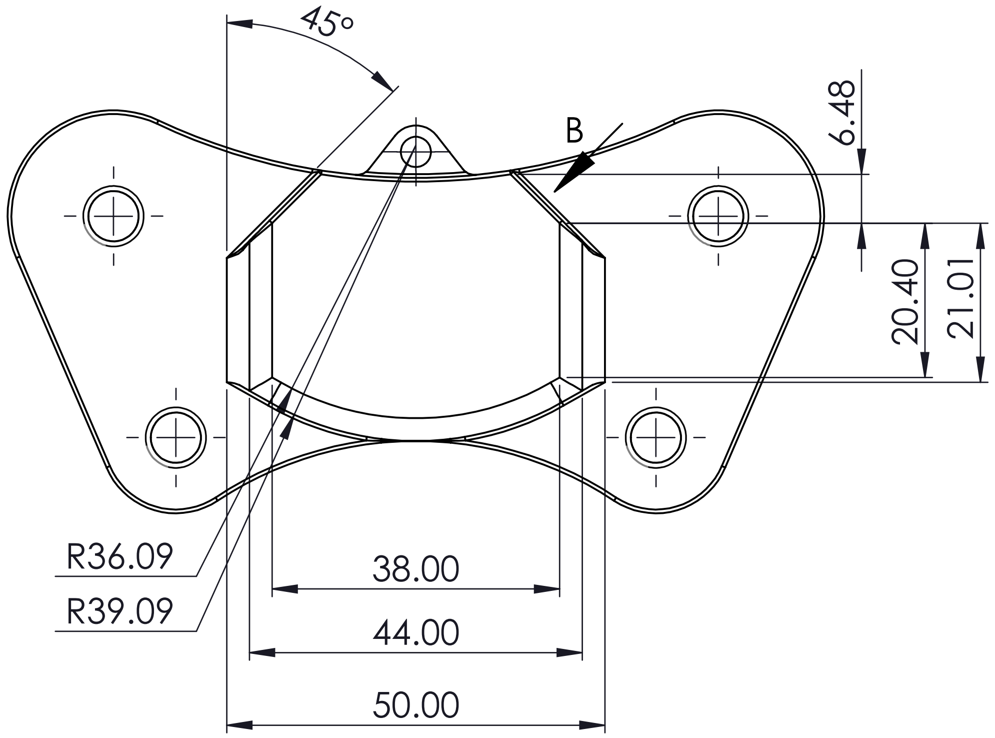


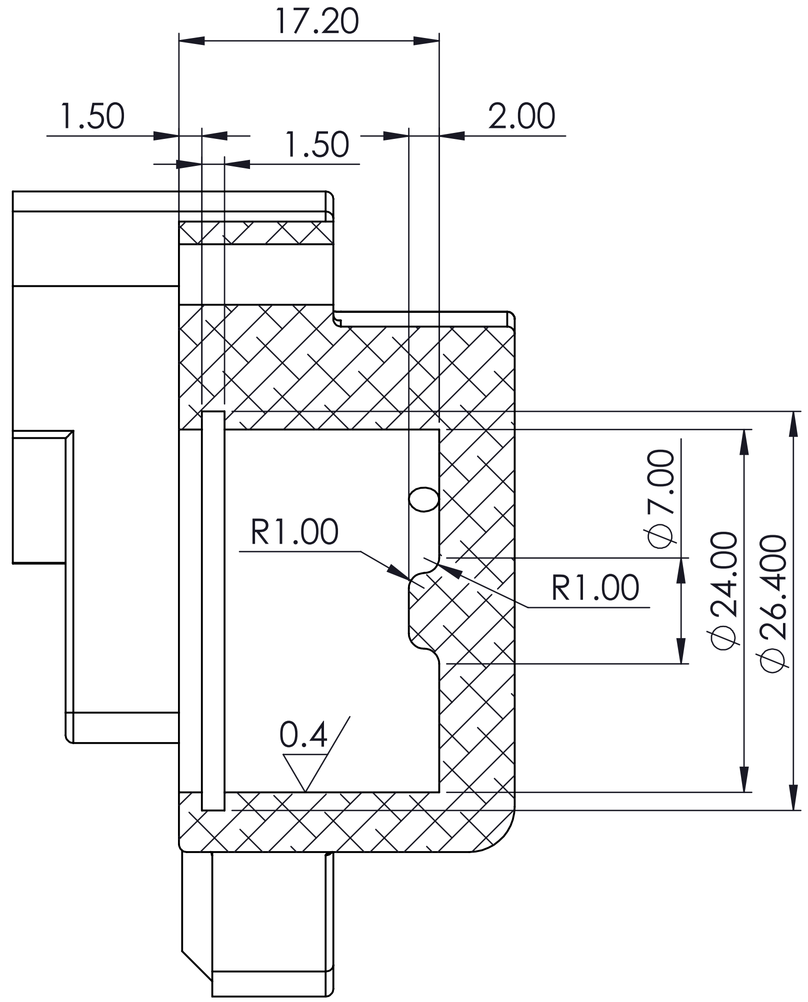
QUT	THIS DRAWING, THE COPYRIGHT AND THE DESIGN THEREIN, ARE ALL THE SOLE PROPERTY OF QUEENSLAND UNIVERSITY of TECHNOLOGY; AND MAY NOT BE COPIED OR REPRODUCED, WHOLLY OR IN PART, WITHOUT THE PRIOR WRITTEN PERMISSION OF THE SAID ORGANISATION.	GEN. LIMITS UNLESS STATED NO PLACES DEC. ± 0.25mm 1 PLACE DEC. ± 0.10mm 2 PLACES DEC. ± 0.05mm ANGLE 1 PLACE DEC. ± 0.50° ANGLE 2 PLACES DEC. ± 0.20° SURFACE FINISH UNLESS STATED 1.6 ALL UNSPECIFIED DIMENSIONS ARE IN mm	NEXT HIGHEST ASSY	CALLIPER	QUANTITY	1
			HEAT TREATMENT	N/A	MASS	181.52 g
			FINISH	**MACHINED**		
			MATERIAL	7075 T6 ALUMINIUM		
			DESIGNED BY	WW	DATE	10/10/2021
			APPROVED BY	FJ	DATE	10/10/2021

TITLE **CALLIPER BODY - INBOARD**			
THIRD ANGLE PROJECTION	SHEET SIZE A3		SCALE 1:1
DRAWING NUMBER **H420M5-DWG-003.1**	ISSUE 1	SHEET 1	OF 2



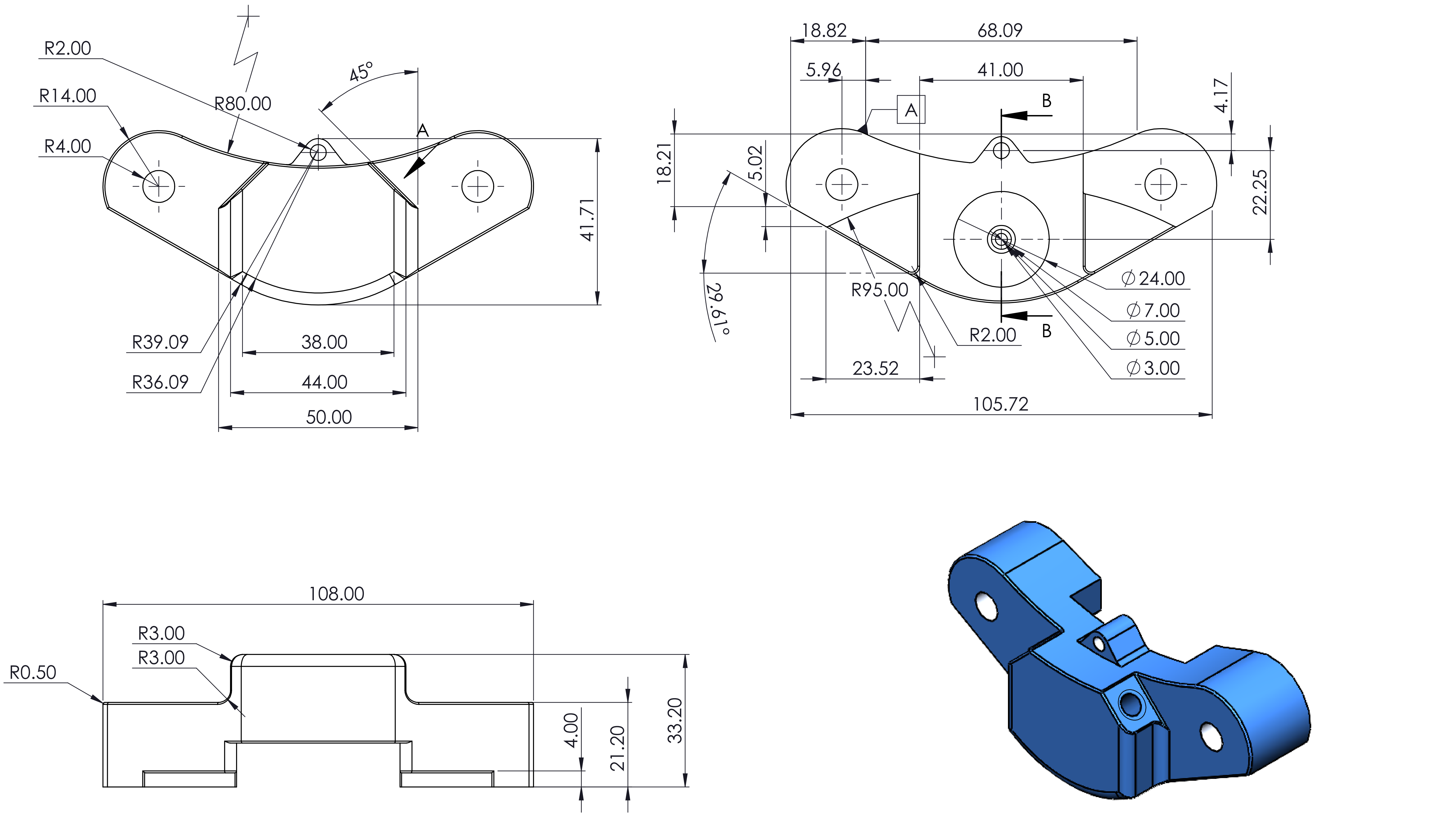
VIEW B
TYPICAL 2 PLACES





SECTION A-A
SCALE 2 : 1

QUT		GEN. LIMITS UNLESS STATED NO PLACES DEC. ± 0.25mm 1 PLACE DEC. ± 0.10mm 2 PLACES DEC. ± 0.05mm ANGLE 1 PLACE DEC. ± 0.50° ANGLE 2 PLACES DEC. ± 0.20° SURFACE FINISH UNLESS STATED 1.6 ALL UNSPECIFIED DIMENSIONS ARE IN mm	NEXT HIGHEST ASSY	CALLIPER	QUANTITY	1	TITLE
			HEAT TREATMENT	N/A	MASS	181.52 g	**CALLIPER BODY - INBOARD**
			FINISH	MACHINED			
			MATERIAL	7075 T6 ALUMINIUM			THIRD ANGLE PROJECTION · SHEET SIZE A3 · SCALE 1:1
			DESIGNED BY	WW	DATE		DRAWING NUMBER H420M5-DWG-003.2 · ISSUE 1.0 · SHEET 2 · OF 2
			APPROVED BY	FJ	DATE		



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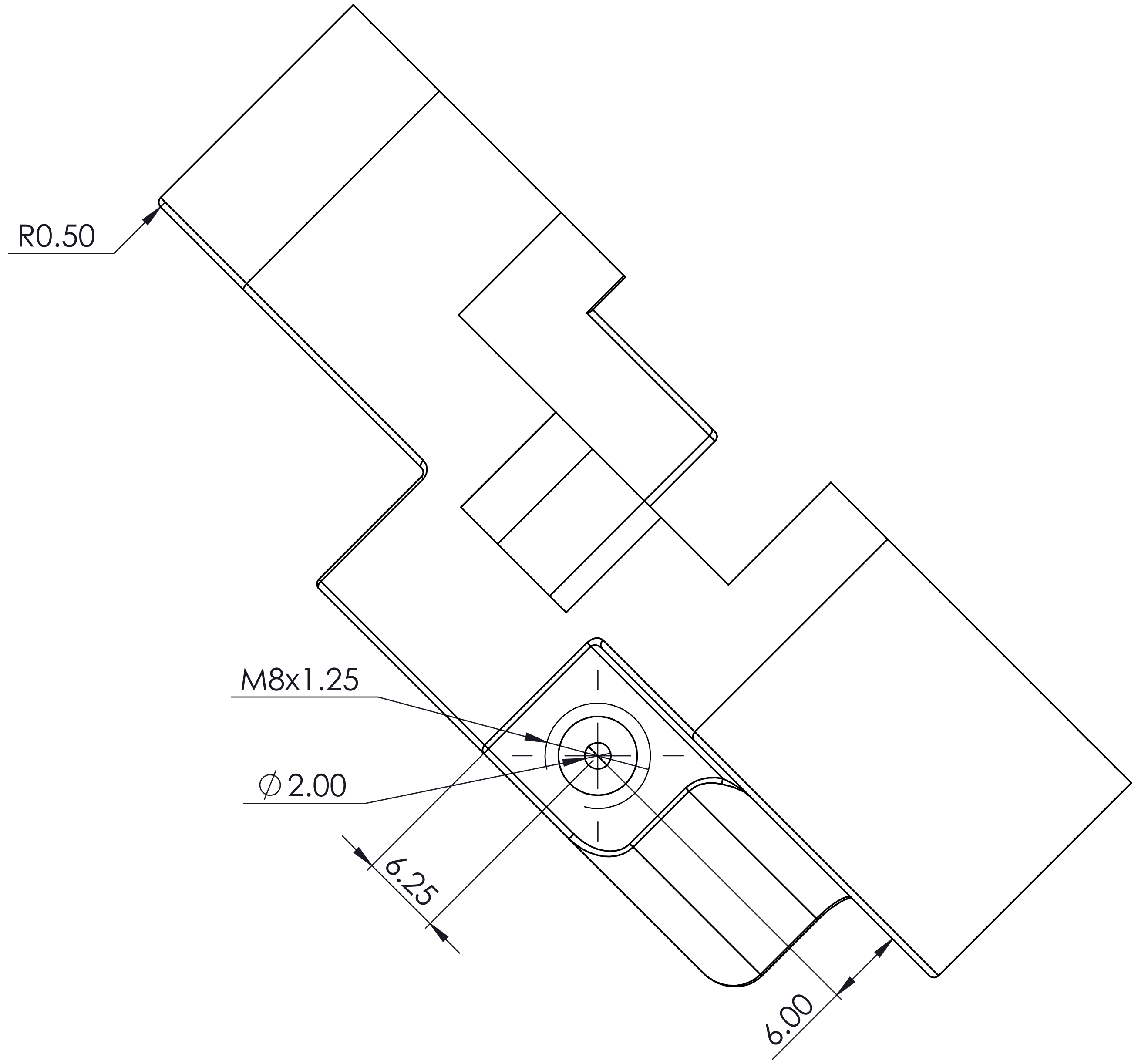
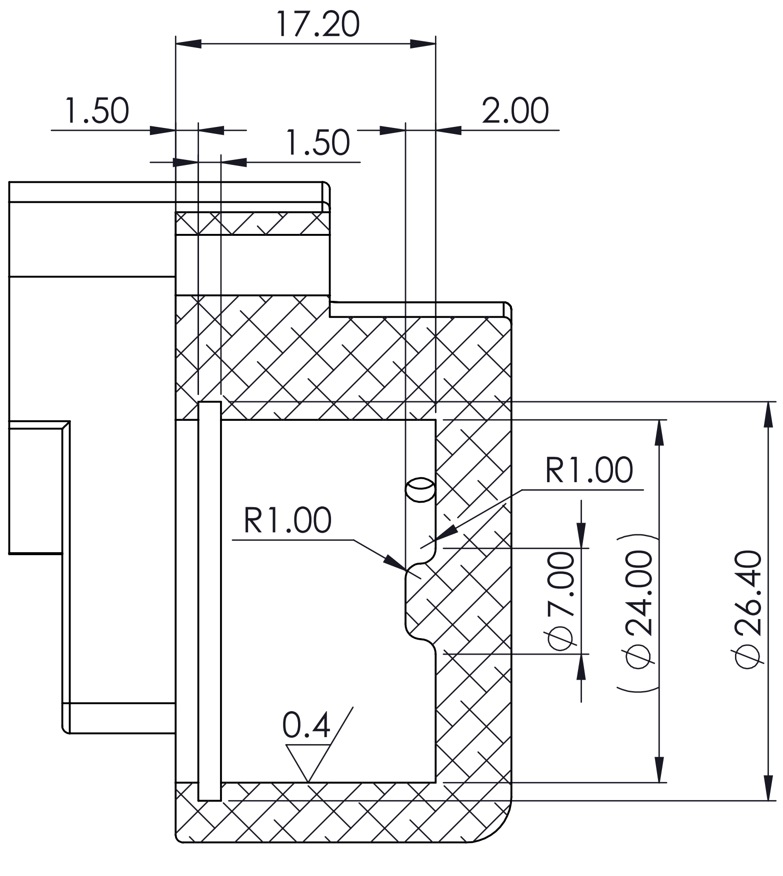
			HEAT TREATMENT	N/A	MASS	153.07 g	**CALLIPER BODY - OUTBOARD**	
			FINISH	N/A				
			MATERIAL	7075 T6 ALUMINIUM			THIRD ANGLE PROJECTION	SHEET SIZE A3 / SCALE 1:1
			DESIGNED BY	WW	DATE	10/10/2021	DRAWING NUMBER **H420M5-DWG-004.1**	ISSUE 1 / SHEET 1 / OF 2
			APPROVED BY	FJ	DATE	10/10/2021		



VIEW A
SCALE 2 : 1

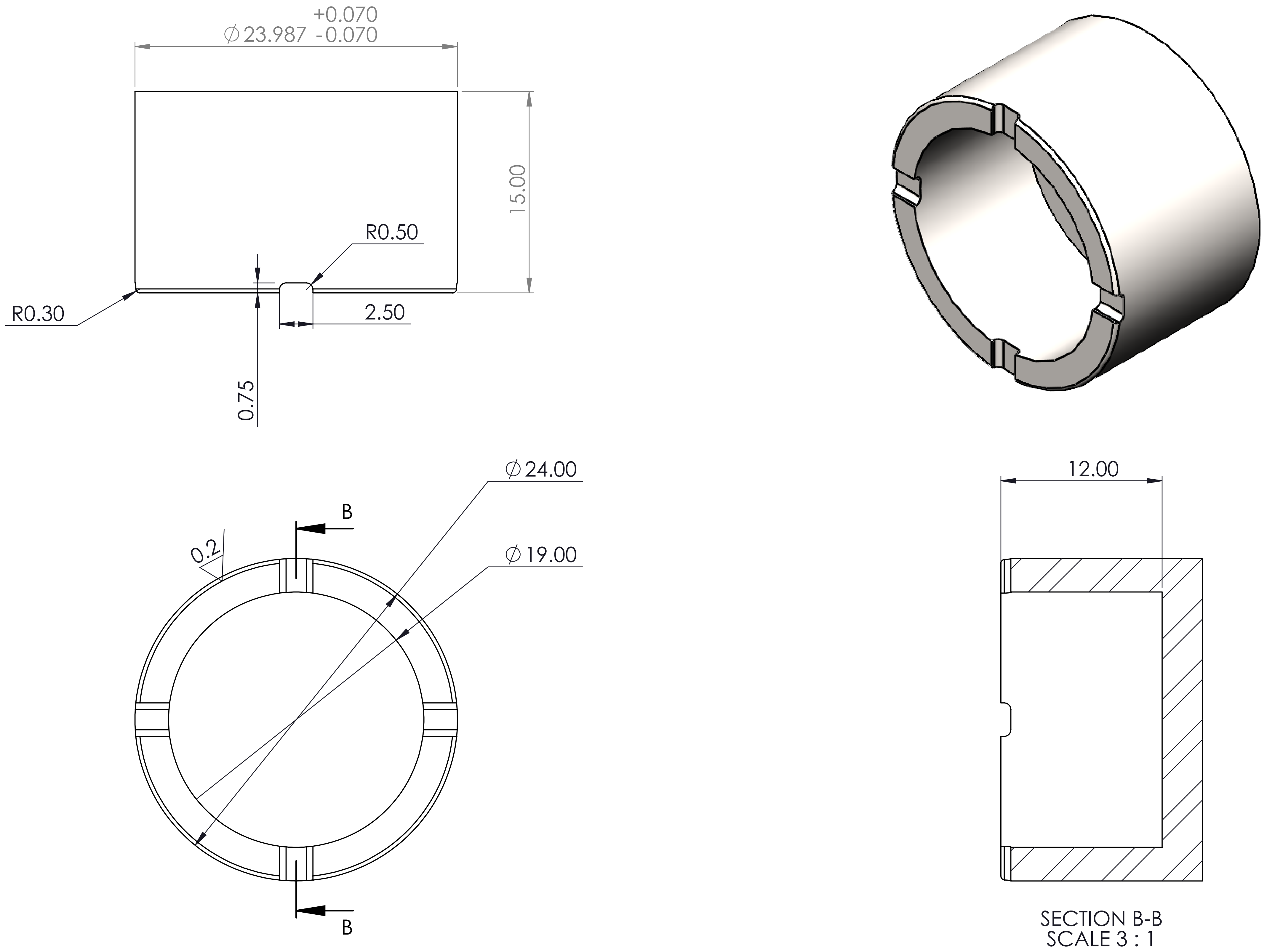


SECTION B-B
SCALE 2 : 1

NEXT HIGHEST ASSY	CALLIPER	QUANTITY	1	TITLE	
HEAT TREATMENT	N/A	MASS	153.07 g	CALLIPER BODY - OUTBOARD	
FINISH	MACHINED				
MATERIAL	7075 T6 ALUMINIUM			THIRD ANGLE PROJECTION	SHEET SIZE A3 / SCALE 1:1
DESIGNED BY	WW	DATE		DRAWING NUMBER H420M5-DWG-004.2	ISSUE 1.0 / SHEET 2 / OF 2
APPROVED BY	FJ	DATE			

QUT

GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH UNLESS STATED 1.6
ALL UNSPECIFIED DIMENSIONS ARE IN mm



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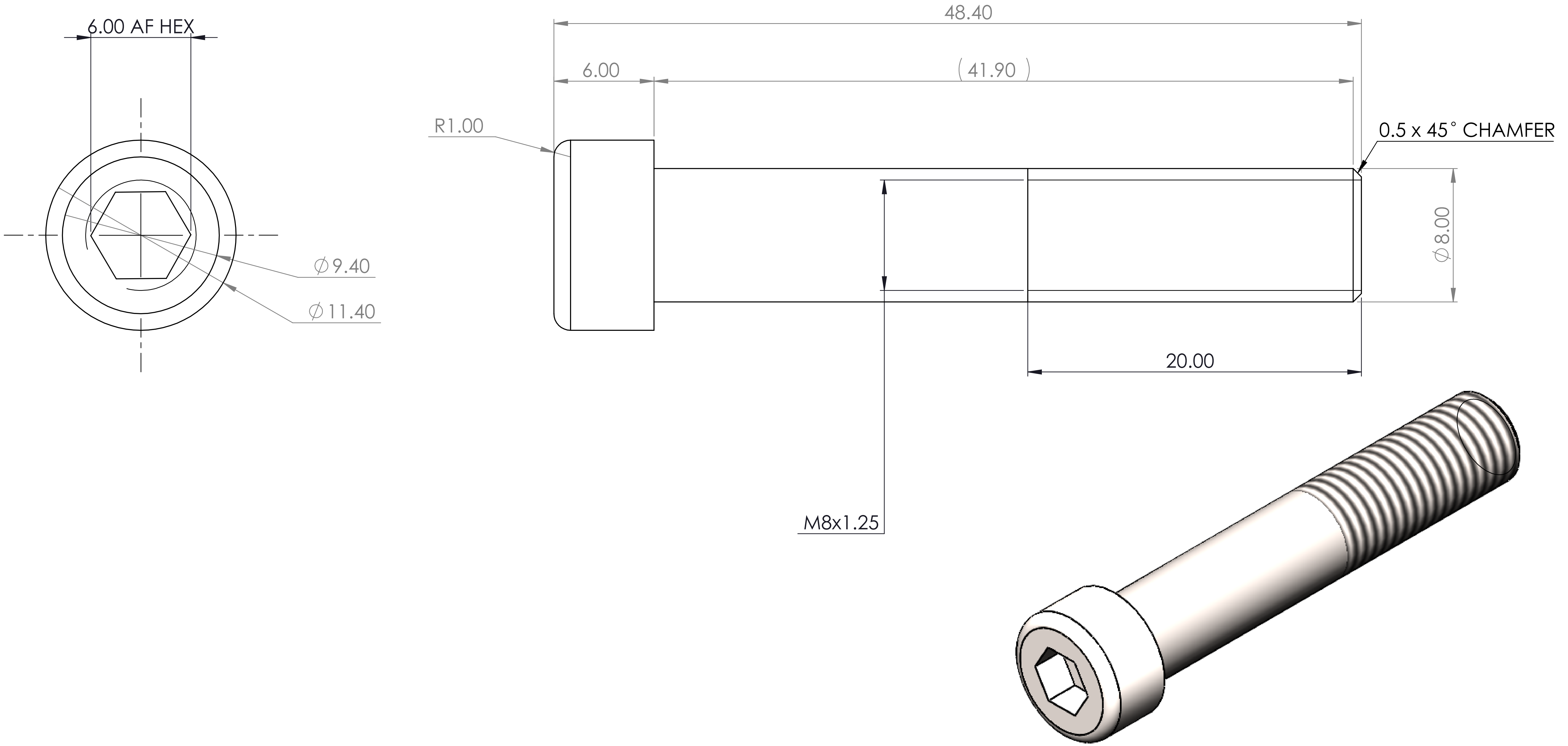
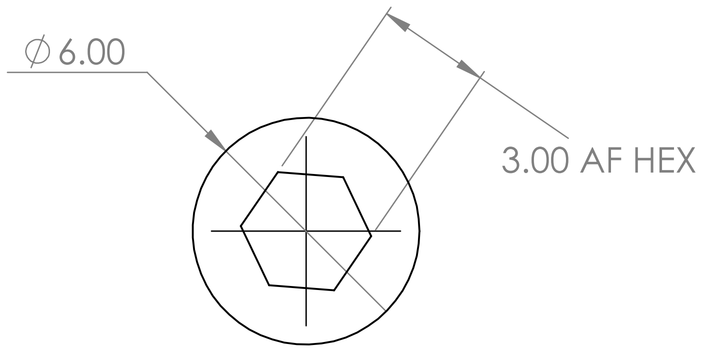
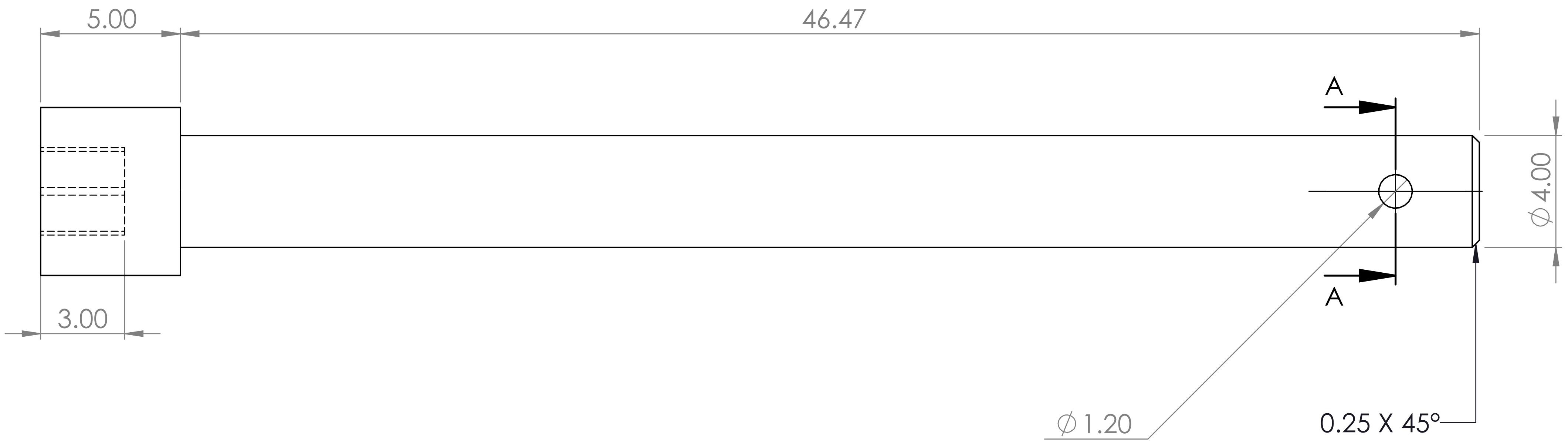
			HEAT TREATMENT	N/A	MASS	26.91 g
			FINISH	N/A		
			MATERIAL	316 STAINLESS STEEL		
			DESIGNED BY	WW	DATE	10/10/2021
			APPROVED BY	FJ	DATE	10/10/2021

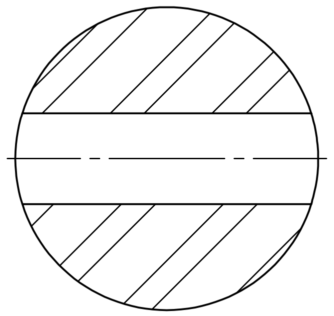
TITLE			
PISTON			
THIRD ANGLE PROJECTION	SHEET SIZE A3	SCALE 3:1	
DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-005	1	1	1



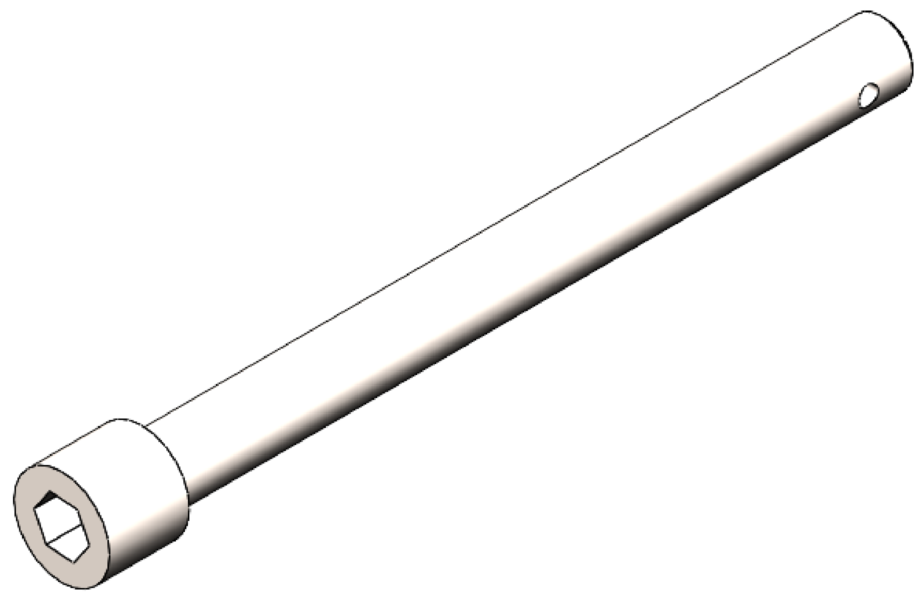
QUT		GEN. LIMITS UNLESS STATED NO PLACES DEC. ± 0.25mm 1 PLACE DEC. ± 0.10mm 2 PLACES DEC. ± 0.05mm ANGLE 1 PLACE DEC. ± 0.50° ANGLE 2 PLACES DEC. ± 0.20° SURFACE FINISH 1.6 UNLESS STATED ALL UNSPECIFIED DIMENSIONS ARE IN mm	NEXT HIGHEST ASSY	CALLIPER	QUANTITY 2	TITLE **CALLIPER BODY BOLT**
			HEAT TREATMENT	N/A	MASS 12.72 g	
			FINISH	N/A		
			MATERIAL	TITANIUM		THIRD ANGLE PROJECTION — SHEET SIZE A3 — SCALE 2:1
			DESIGNED BY	WW	DATE 10/10/2021	DRAWING NUMBER **H420M5-DWG-006** — ISSUE 1 — SHEET 1 — OF 1
			APPROVED BY	FJ	DATE 10/10/2021	







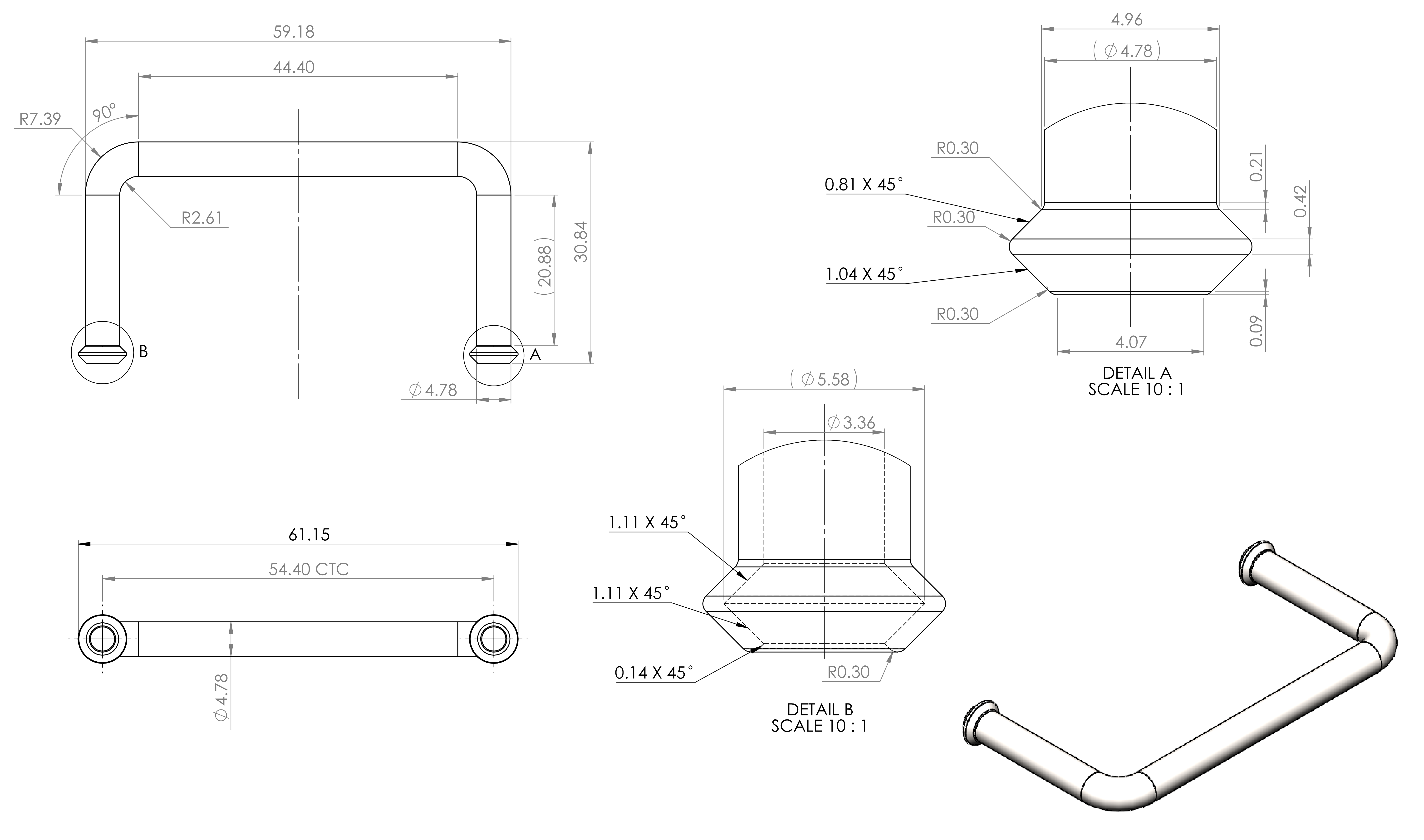
SECTION A-A
SCALE 10 : 1



NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	1	TITLE	**BRAKE PAD RETAINING PIN**
HEAT TREATMENT	**N/A**	MASS	**5.42 g**		
FINISH	**MACHINED**			THIRD ANGLE PROJECTION	SHEET SIZE A3 / SCALE 2:1
MATERIAL	**MILD STEEL**				
DESIGNED BY	**WW**	DATE	10/10/2021	DRAWING NUMBER **H420M5-DWG-007**	ISSUE 1 / SHEET 1 / OF 1
APPROVED BY	**FJ**	DATE	10/10/2021		

GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH 1.6
UNLESS STATED
ALL UNSPECIFIED DIMENSIONS ARE IN mm

QUT



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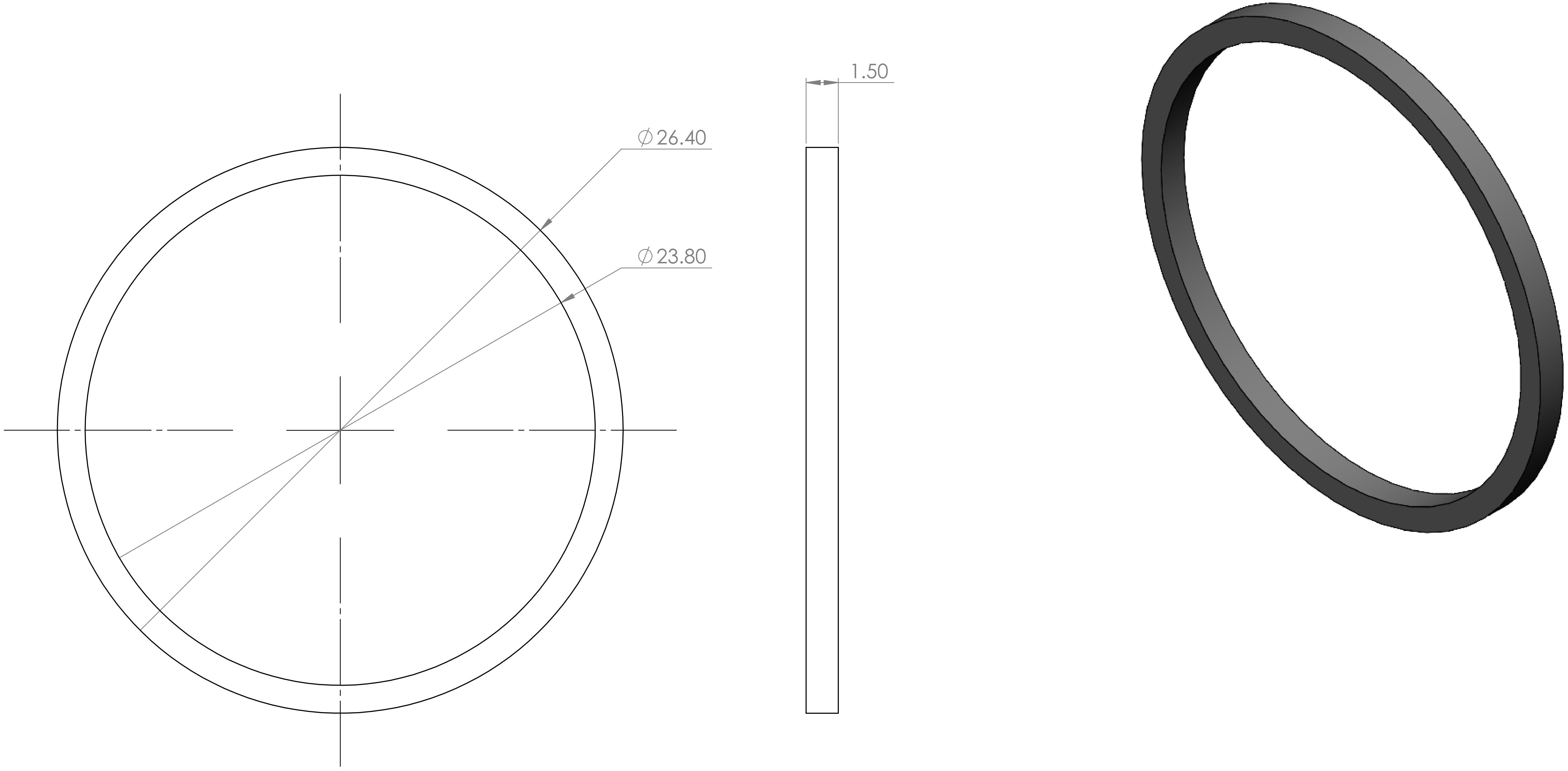
			HEAT TREATMENT	**ANNEALED**	MASS	**7.84 g**
			FINISH	**N/A**		
			MATERIAL	**AISI 304**		
			DESIGNED BY	WW	DATE	10/10/2021
			APPROVED BY	FJ	DATE	10/10/2021

TITLE	**CROSS-OVER TUBE**		
THIRD ANGLE PROJECTION	SHEET SIZE A3		SCALE 2:1
DRAWING NUMBER **H420M5-DWG-008**	ISSUE 1	SHEET 1	OF 1



NEXT HIGHEST ASSY	CALLIPER	QUANTITY	2	
HEAT TREATMENT	N/A	MASS	0.18 g	
FINISH	N/A			
MATERIAL	EPDM RUBBER			
DESIGNED BY	N/A	DATE		
APPROVED BY	N/A	DATE		

GEN. LIMITS UNLESS STATED

NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°

SURFACE FINISH UNLESS STATED 1.6

ALL UNSPECIFIED DIMENSIONS ARE IN mm

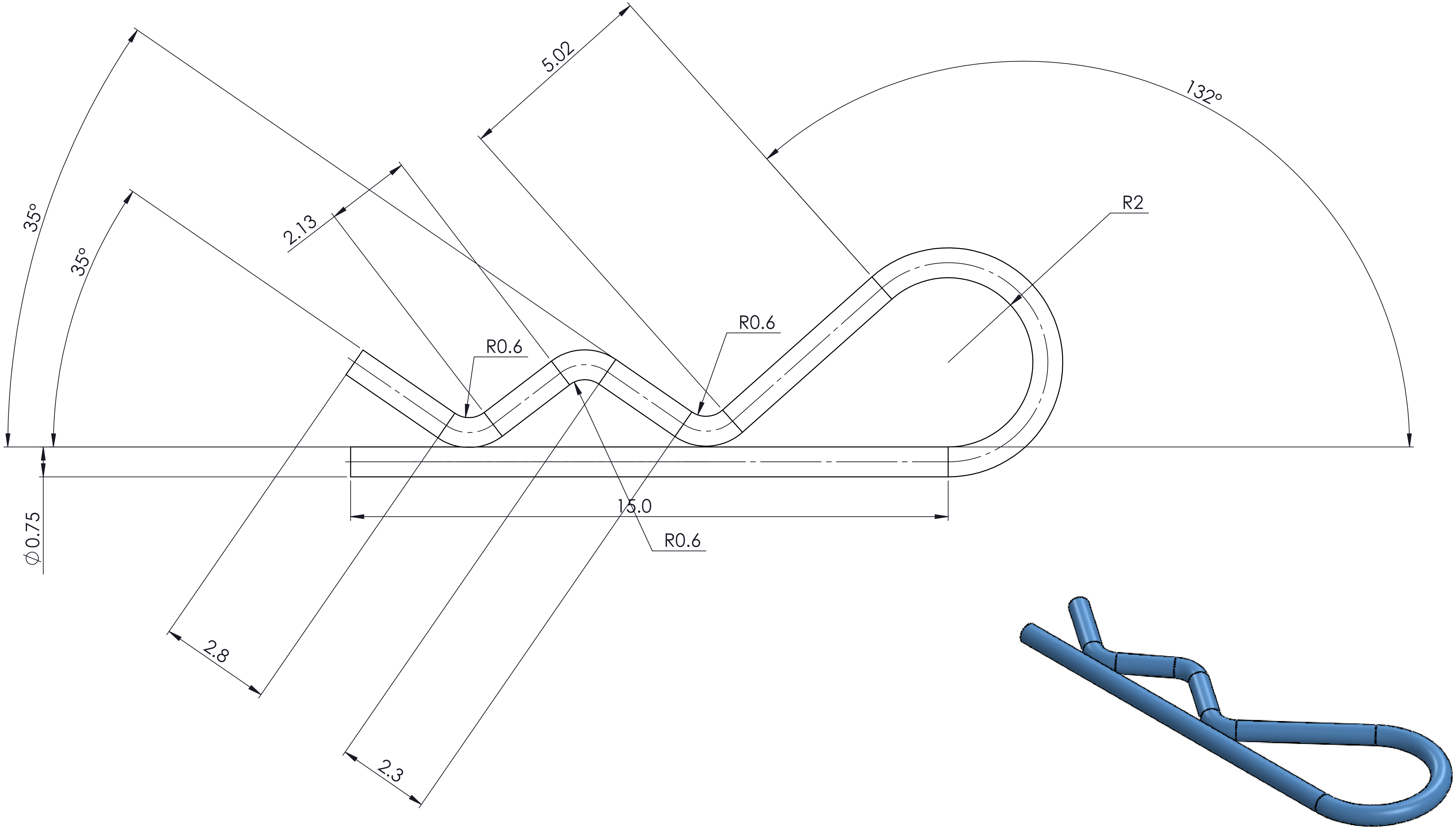
TITLE

PISTON SEAL

THIRD ANGLE PROJECTION

SHEET SIZE	A3	SCALE	5:1

DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-101	1.0	1	1



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GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH UNLESS STATED 1.6
ALL UNSPECIFIED DIMENSIONS ARE IN mm

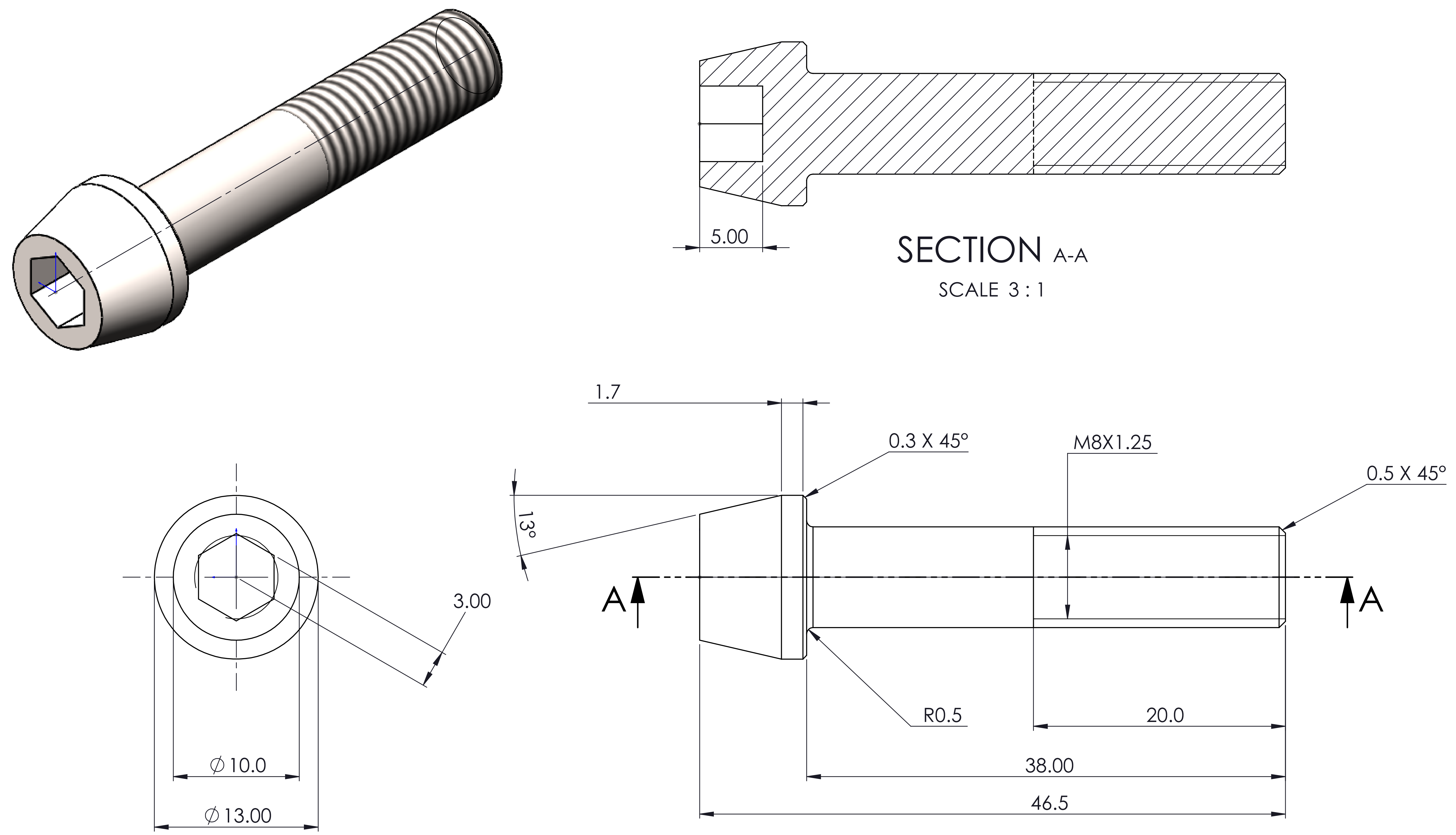
NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	1
HEAT TREATMENT	**Q&T**	MASS	**0.9 g**
FINISH	**N/A**		
MATERIAL	**AISI 304**		
DESIGNED BY		DATE	
APPROVED BY		DATE	

TITLE
R-CLIP

THIRD ANGLE PROJECTION

SHEET SIZE	A3	SCALE	5:1

DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-102	1.0	1	1



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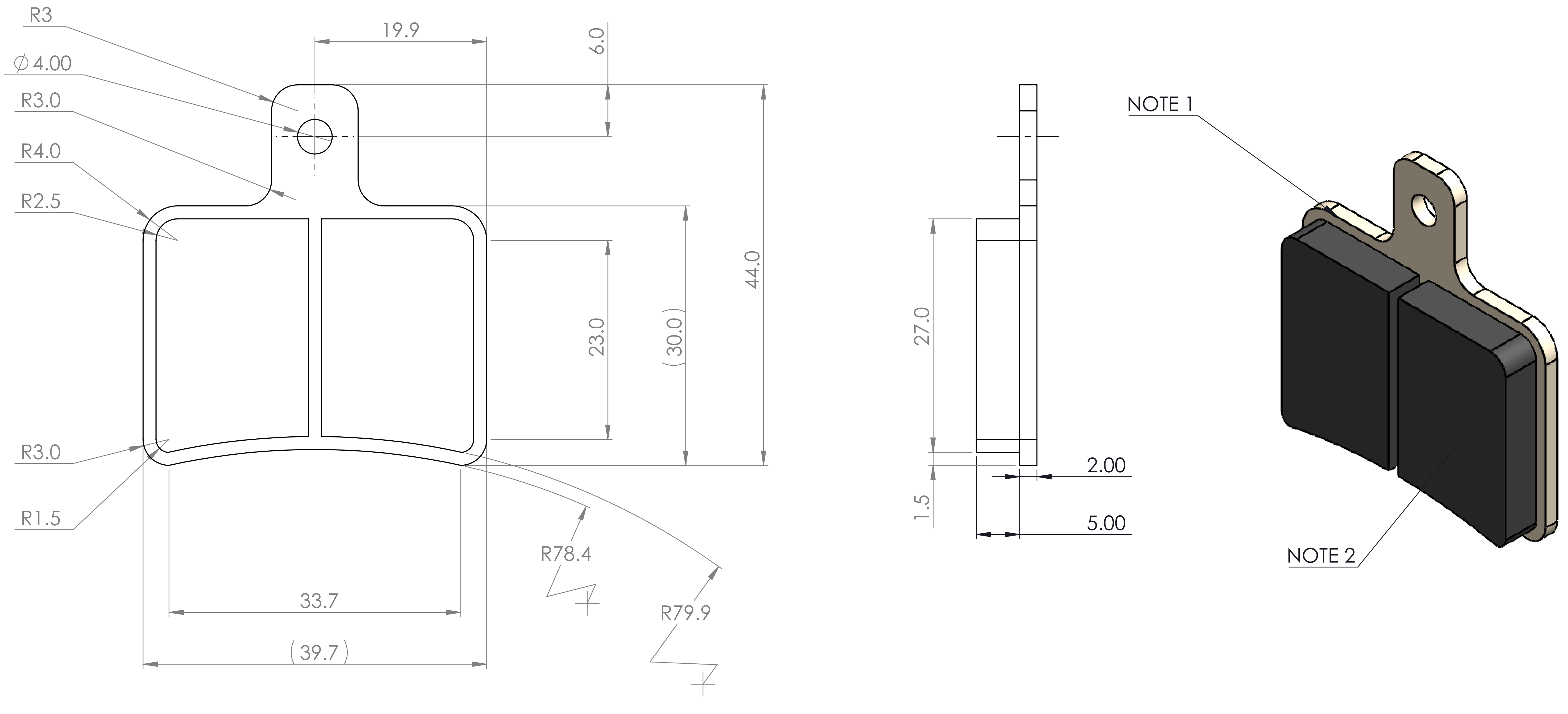
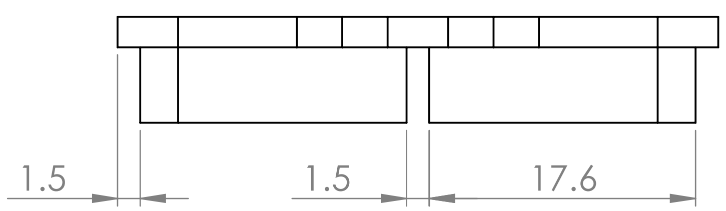




NOTES

1. BACKING MATERIAL IS STAINLESS STEEL
2. FRICTION MATERIAL IS SINTERED METALLIC COMPOUND
3. DIMENSIONS OF FRICTION MATERIAL ARE APPROXIMATE DUE TO LACK OF MANUFACTURER SPECIFICATION.

QUT

GEN. LIMITS UNLESS STATED

NO PLACES DEC.	± 0.25mm
1 PLACE DEC.	± 0.10mm
2 PLACES DEC.	± 0.05mm
ANGLE 1 PLACE DEC.	± 0.50°
ANGLE 2 PLACES DEC.	± 0.20°

SURFACE FINISH UNLESS STATED 1.6

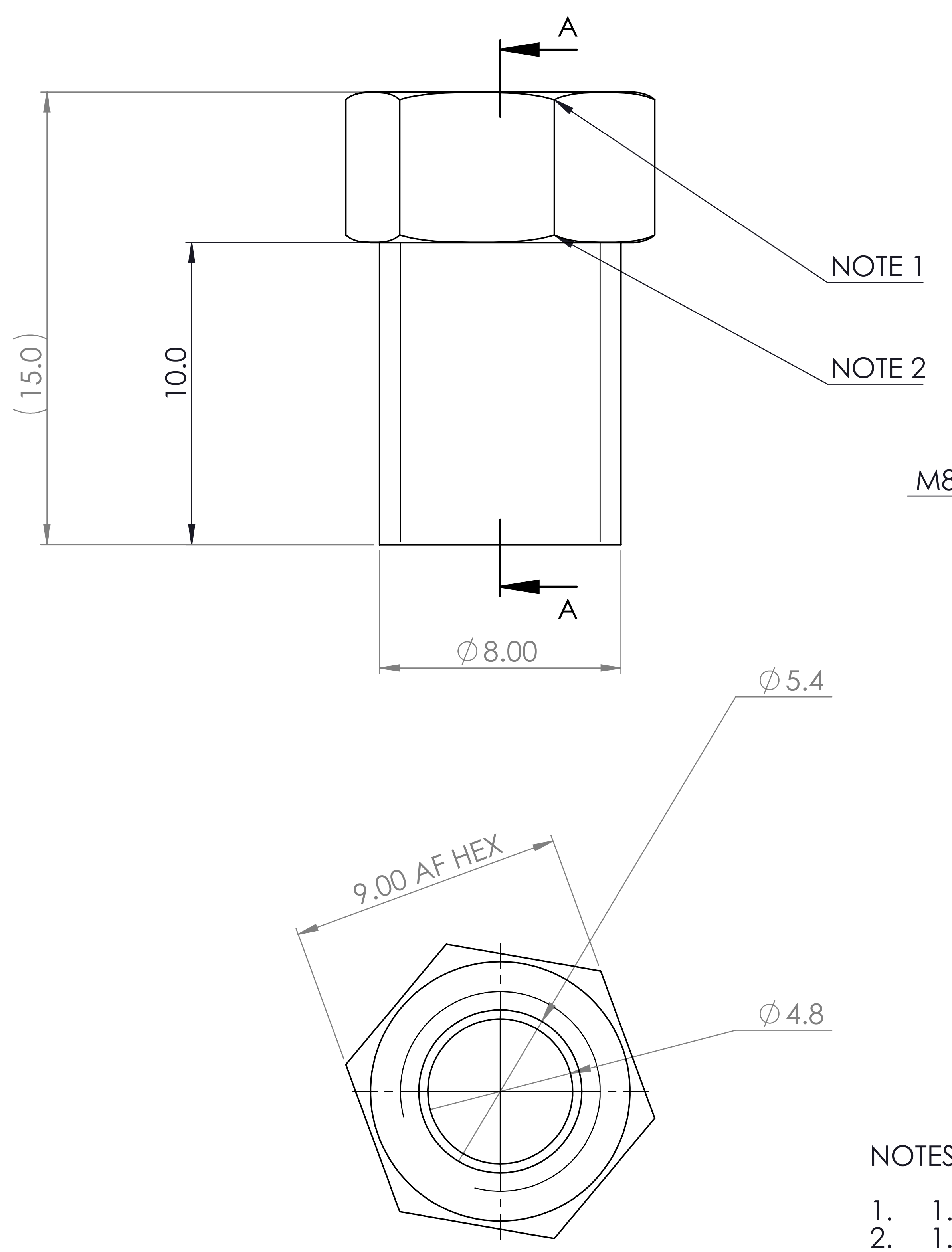
ALL UNSPECIFIED DIMENSIONS ARE IN mm

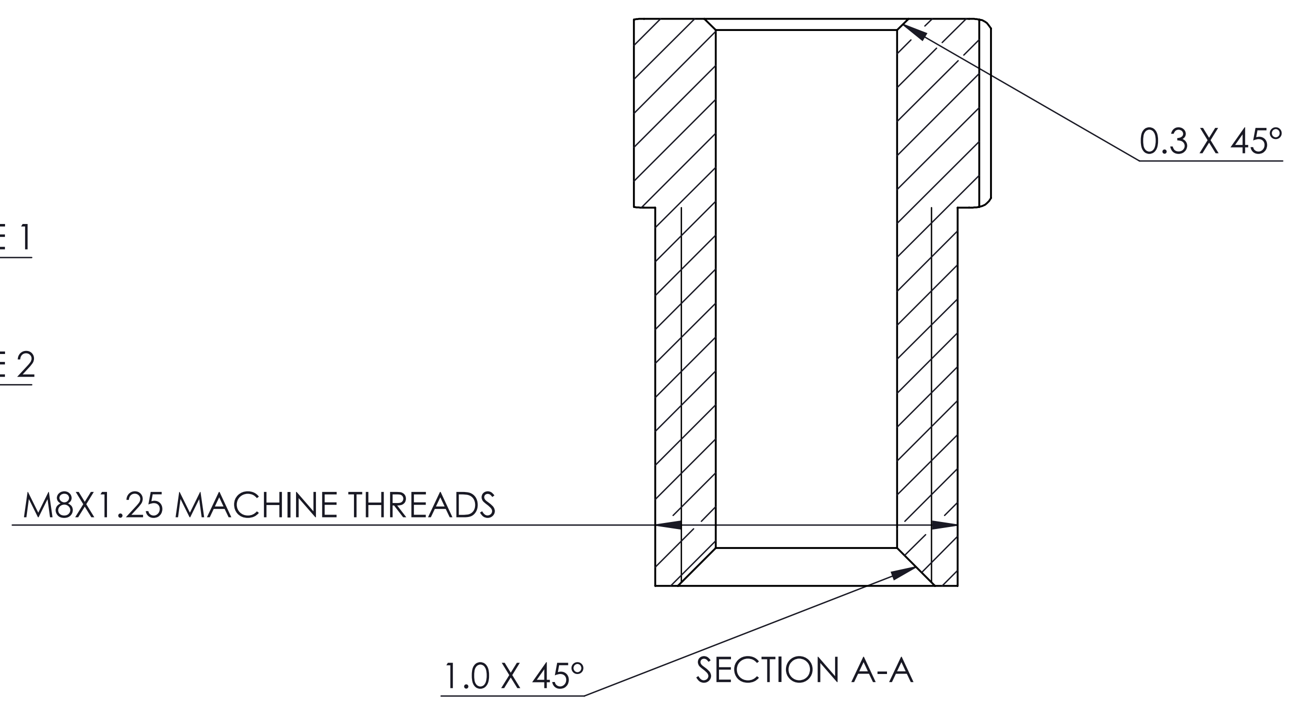
NEXT HIGHEST ASSY	CALLIPER	QUANTITY	2
HEAT TREATMENT	SINTERED	MASS	8 g
FINISH	N/A		
MATERIAL	STAINLESS STEEL		
DESIGNED BY	N/A	DATE	
APPROVED BY	N/A	DATE	

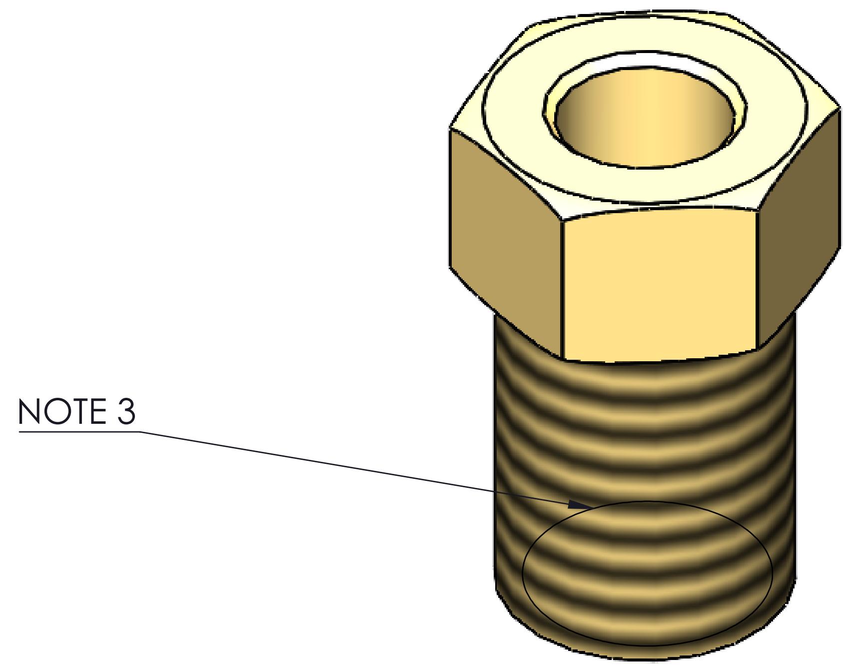
TITLE

CP4226D27 BRAKE PAD

THIRD ANGLE PROJECTION	SHEET SIZE A3		SCALE 2:1
DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-104	1.0	1	1







NOTES

1. 1.7 MM FILLET TO TOP OF HEX.
2. 1.0 MM FILLET TO BASE OF HEX
3. ROLLED THREADS ACCEPTABLE

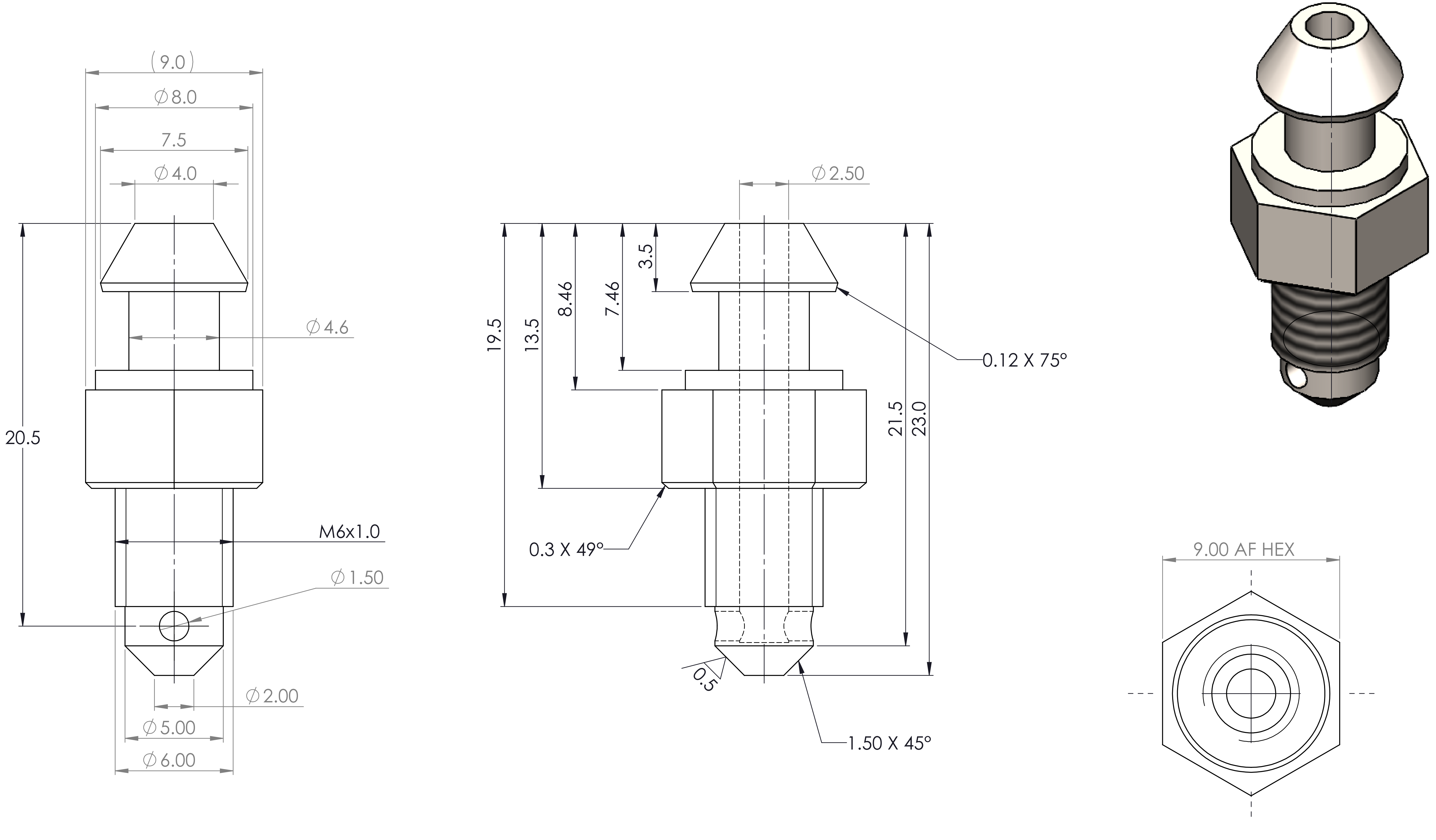
GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH UNLESS STATED 1.6
ALL UNSPECIFIED DIMENSIONS ARE IN mm

NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	**2**
HEAT TREATMENT	**N/A**	MASS	**1.0 g**
FINISH	**AS MACHINED**		
MATERIAL	**304 STAINLESS STEEL**		
DESIGNED BY		DATE	
APPROVED BY		DATE	

TITLE

FLARE TUBING NUT

THIRD ANGLE PROJECTION	SHEET SIZE A3		SCALE 5:1
DRAWING NUMBER **H420M5-DWG-105**	ISSUE 1.0	SHEET 1	OF 1



QUT

GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH UNLESS STATED 1.6
ALL UNSPECIFIED DIMENSIONS ARE IN mm

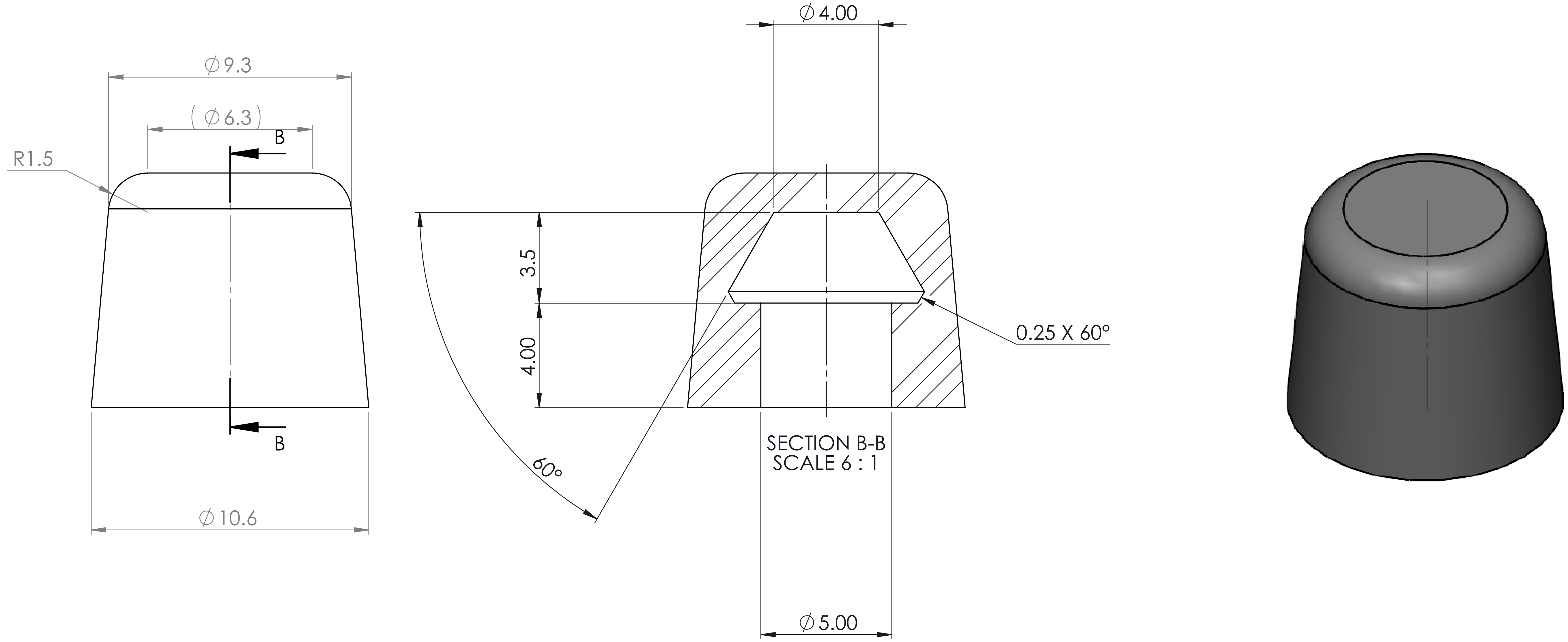
NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	**1**
HEAT TREATMENT	**N/A**	MASS	**5.24 g**
FINISH	**N/A**		
MATERIAL	**STAINLESS STEEL**		
DESIGNED BY	N/A	DATE	
APPROVED BY	N/A	DATE	

TITLE

BLEED SCREW

THIRD ANGLE PROJECTION

SHEET SIZE A3	SCALE 2:1	
DRAWING NUMBER H420M5-DWG-106		
ISSUE	SHEET	OF
1.0	1	1



QUT

GEN. LIMITS UNLESS STATED

NO PLACES DEC.	± 0.25mm
1 PLACE DEC.	± 0.10mm
2 PLACES DEC.	± 0.05mm
ANGLE 1 PLACE DEC.	± 0.50°
ANGLE 2 PLACES DEC.	± 0.20°
SURFACE FINISH UNLESS STATED	1.6

ALL UNSPECIFIED DIMENSIONS ARE IN mm

NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	1
HEAT TREATMENT	**N/A**	MASS	**0.50 g**
FINISH	**N/A**		
MATERIAL	**RUBBER**		
DESIGNED BY	**N/A**	DATE	
APPROVED BY	**W WHIGHAM**	DATE	10/10/2021

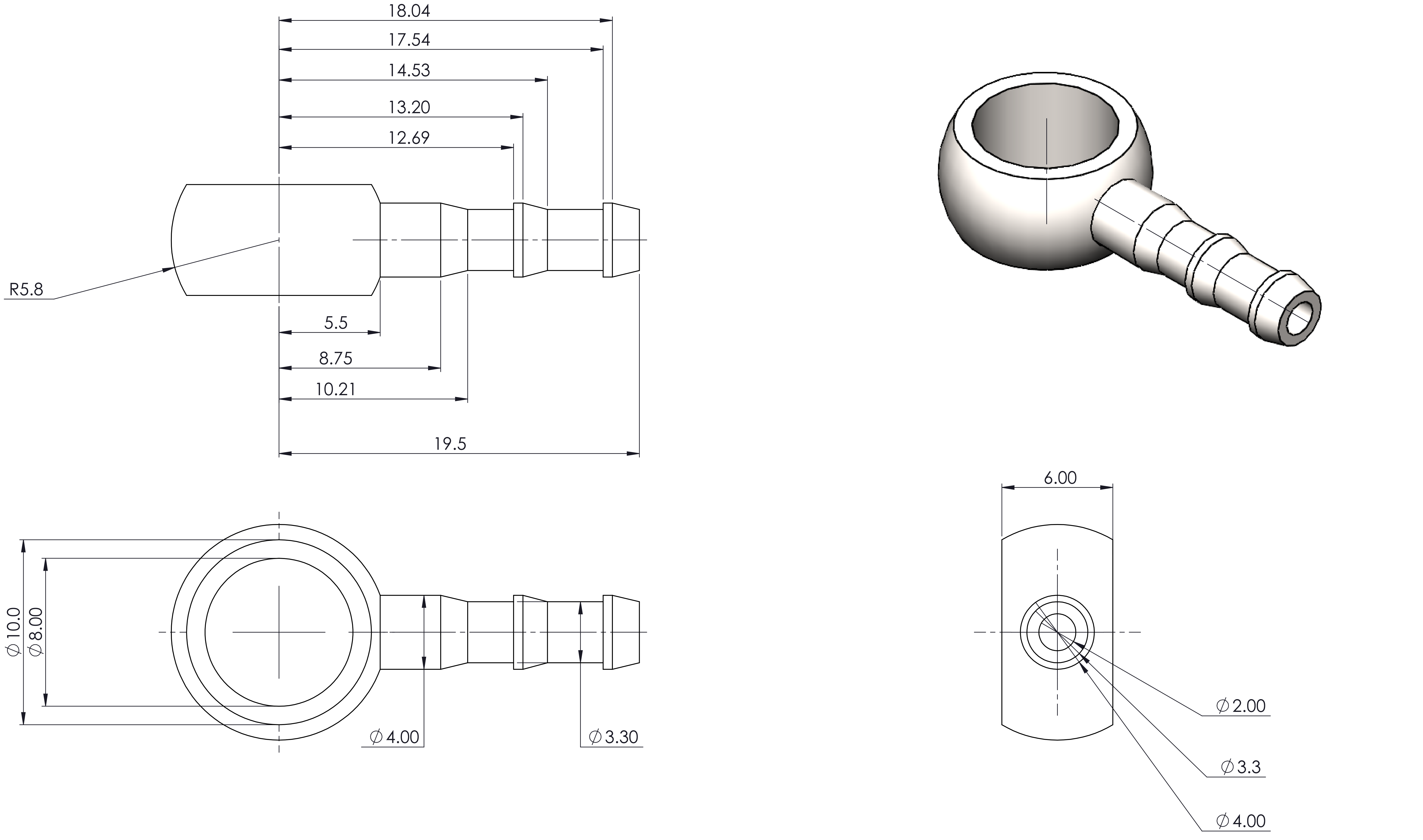
TITLE

BLEEDPORT DUST CAP

THIRD ANGLE PROJECTION

SHEET SIZE	SCALE
A3	6:1

DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-107	1.0	1	1



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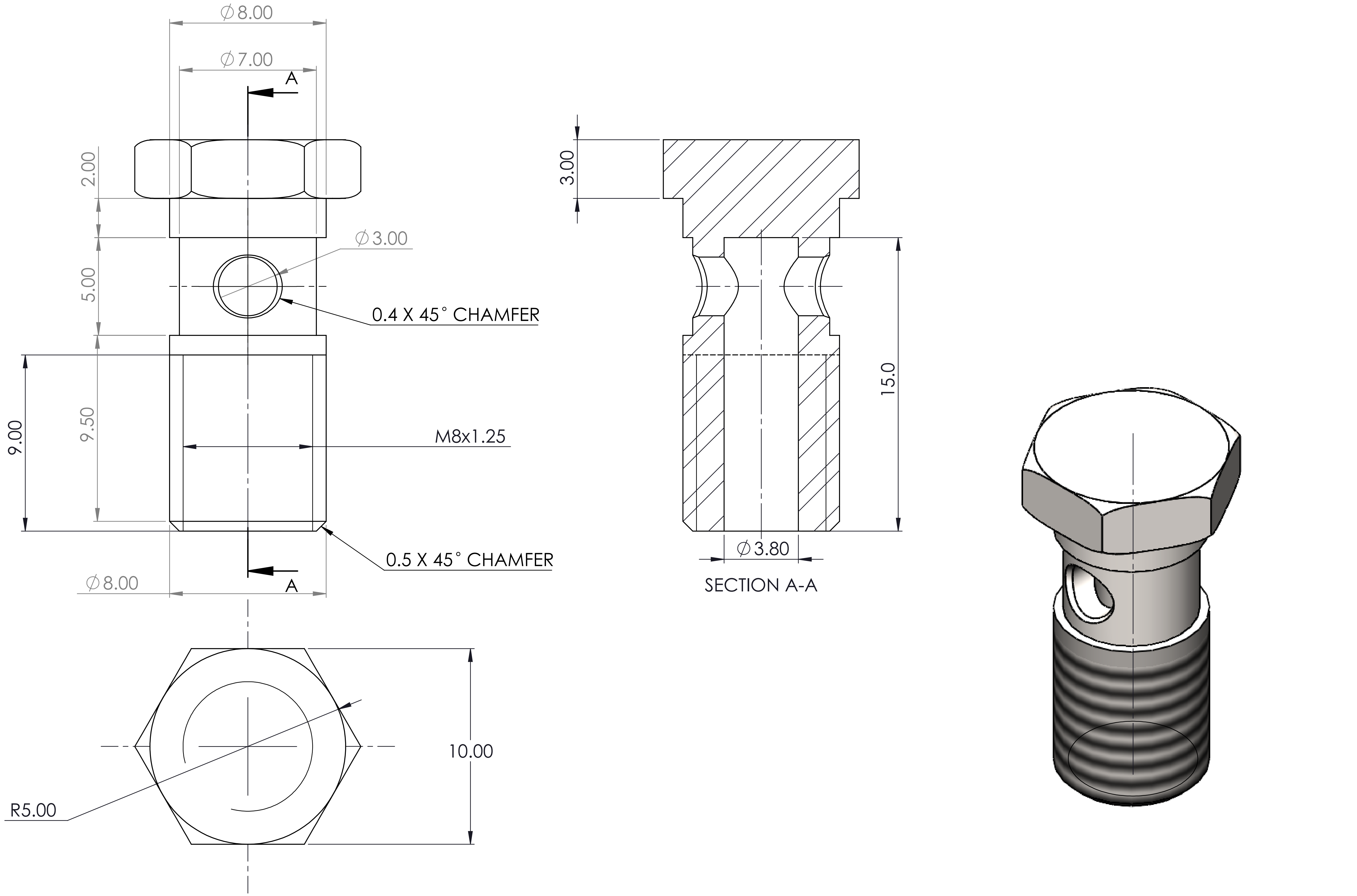
			HEAT TREATMENT	**N/A**	MASS **3.0 g**	
			FINISH	**N/A**		
			MATERIAL	**STAINLESS STEEL**		THIRD ANGLE PROJECTION; SHEET SIZE A3; SCALE 5:1
			DESIGNED BY		DATE	DRAWING NUMBER **H420M5-DWG-108**; ISSUE 1.0; SHEET 1 OF 1
			APPROVED BY	**F JOHNSON**	DATE	



QUT

GEN. LIMITS UNLESS STATED
NO PLACES DEC. ± 0.25mm
1 PLACE DEC. ± 0.10mm
2 PLACES DEC. ± 0.05mm
ANGLE 1 PLACE DEC. ± 0.50°
ANGLE 2 PLACES DEC. ± 0.20°
SURFACE FINISH UNLESS STATED 1.6
ALL UNSPECIFIED DIMENSIONS ARE IN mm

NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	1
HEAT TREATMENT	**N/A**	MASS	**6.9 g**
FINISH	**N/A**		
MATERIAL	**STAINLESS STEEL**		
DESIGNED BY	**N/A**	DATE	
APPROVED BY	**F JOHNSON**	DATE	

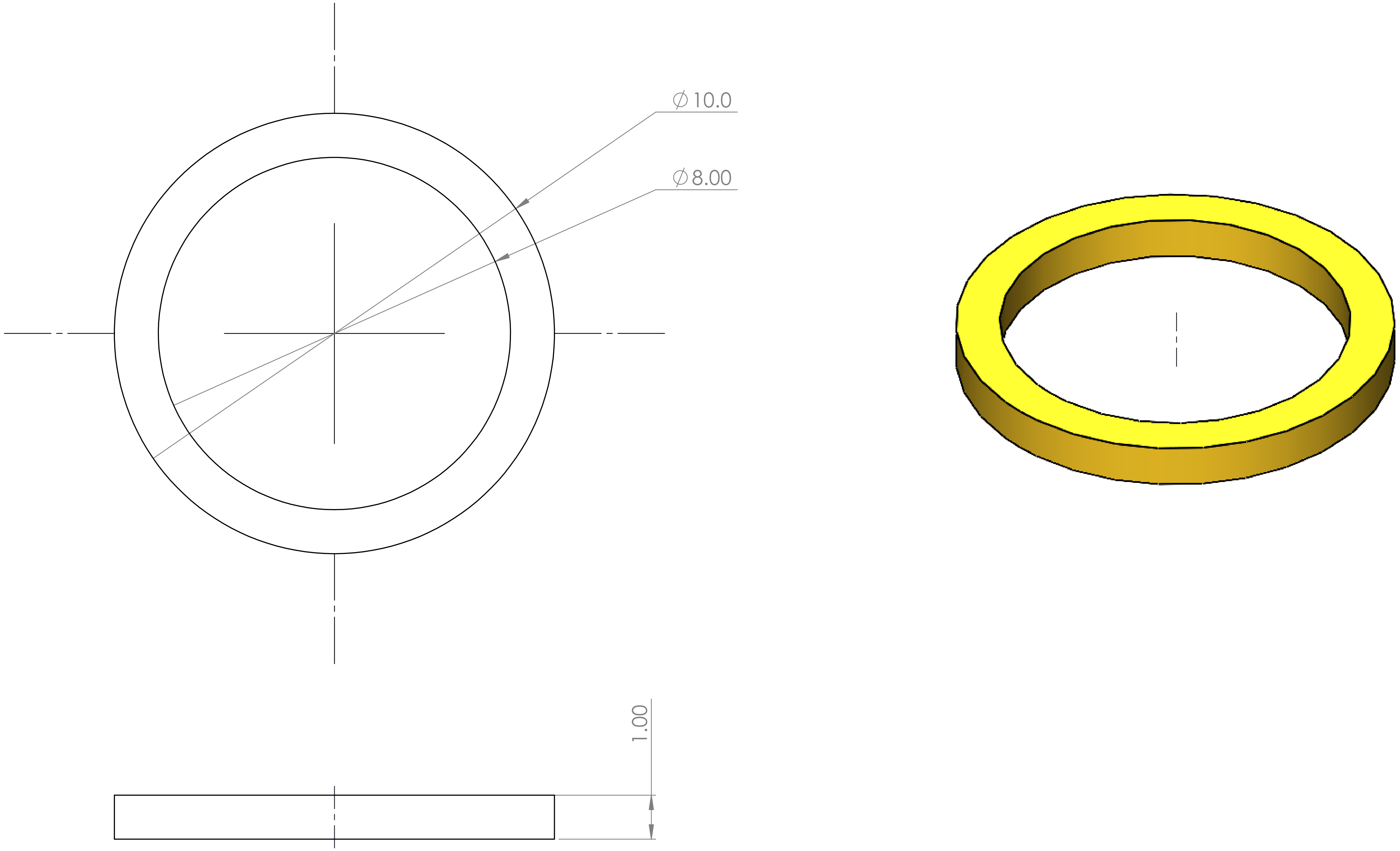
TITLE

BANJO BOLT

THIRD ANGLE PROJECTION

SHEET SIZE	SCALE
A3	5:1

DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-109	1.0	1	1



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GEN. LIMITS UNLESS STATED

NO PLACES DEC.	± 0.25mm
1 PLACE DEC.	± 0.10mm
2 PLACES DEC.	± 0.05mm
ANGLE 1 PLACE DEC.	± 0.50°
ANGLE 2 PLACES DEC.	± 0.20°

SURFACE FINISH UNLESS STATED 1.6

ALL UNSPECIFIED DIMENSIONS ARE IN mm

NEXT HIGHEST ASSY	**CALLIPER**	QUANTITY	**2**
HEAT TREATMENT	**ANNEALED**	MASS	**0.3 g**
FINISH	**N/A**		
MATERIAL	**COPPER / BRASS**		
DESIGNED BY	**N/A**	DATE	
APPROVED BY	**F JOHNSON**	DATE	**10/10/21**

TITLE

BRASS WASHER

THIRD ANGLE PROJECTION	SHEET SIZE A3		SCALE 10:1
DRAWING NUMBER	ISSUE	SHEET	OF
H420M5-DWG-110	1.0	1	1