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Zips Racing Electric 2022 Brakes Subsystem Design

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Brakes Subsystem Design for Zips Racing Electric

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Honors Research Project

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Abstract

This project will include the design, manufacture, and assembly of the entire brakes subsystem for the Zips Racing Electric (ZRE) 2022 car. Goals of the brakes system include a lightweight, modular, machinable, and rigid design. The project will begin in the Fall semester with the design of the pedals and rotors as well as calculations to specify master cylinders, calipers, pads, etc. Software such as SolidWorks CAD, SolidWorks FEA, and Mastercam CAM will be employed. Next, the second semester will involve manufacturing or purchasing necessary components. Prototype pedals will be manufactured for physical testing.

The project will be completed individually. However, the project lead will work cooperatively with other undergraduate and graduate students on ZRE. Funding will be provided by company sponsors as well as funds budgeted by the University of Akron. The project lead will report to the project sponsor on a monthly basis and ZRE on a weekly basis.

Contents

1. Introduction

1.1 Principles of operation

In vehicle dynamics, the braking subsystem is designed to allow the driver to decelerate safely to a stop. Using traditional methods, a hydraulic system is commonly employed to achieve this task. The system is sealed and pressurized with brake fluid which is highly incompressible. Pressing the brake pedal is the input force which is multiplied by a piston called a master cylinder. This forces high pressure fluid into the calipers. The vehicle's momentum energy is converted into friction as the brake pads are clamped onto the rotor.

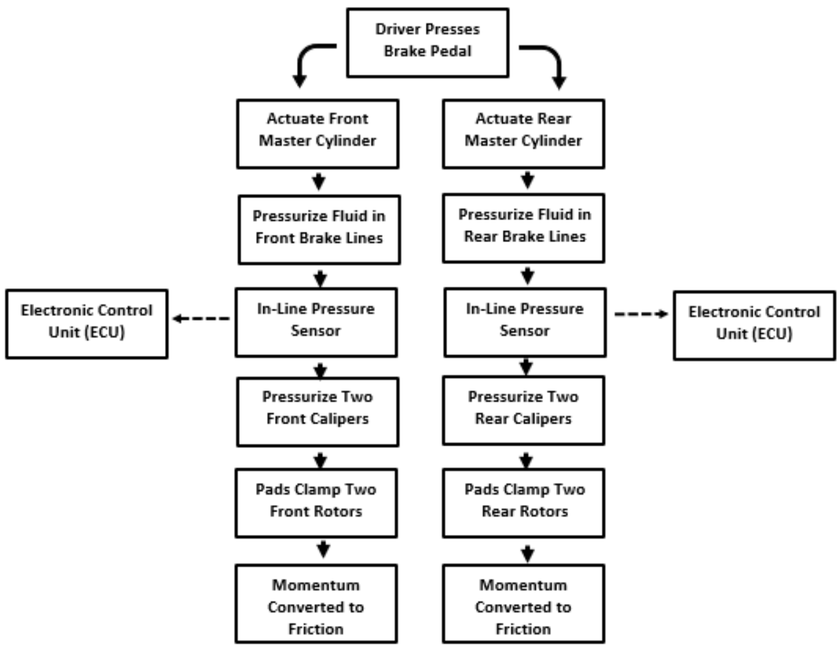

Figure 1: Function Structure Diagram for the brake system. Two systems are shown in parallel because the front and rear brake systems are independent circuits activated by pushing a single brake pedal.

1.2 Application

The goal of this honors design project is to design a braking system for use in the Zips Racing Electric (ZRE) 2022 car. The application for ZRE is to design and build a braking system that is optimized to perform on a formula style car under race conditions. Therefore, considerations such as weight, driver feel, and heat have increased importance. The design must carefully incorporate safety as well as performance to result in a system that will be successful for integration on the car. Finally, the system must be rules compliant per the FSAE handbook in order for ZRE to be eligible to attend competition.

The project will include the design of pedal box which includes both brake and accelerator pedals as well as their adjustable bases. Next, the rotors will be designed and optimized for cooling. Components to be specified and ordered include master cylinders, pads, and calipers. Finally, the model will be placed in the full car assembly where brake lines will be routed and measured. Previous iterations will be referenced with the objective being to improve upon past designs.

The goals of the 2022 design are as follows: light, rigid, machinable, modular. Building a system that is lightweight is imperative for the overall handling dynamics of the vehicle. To achieve this goal, material selection and FEA analysis will be used to reduce weight wherever possible. Next, rigidity is important as the brake system must not fail under any circumstance. In order to make the system realistic and marketable, it must be machinable. This means sacrifices in aesthetics and optimization must be made so the system is affordable and relatively easy to manufacture. Finally, a modular design will be employed for the pedal box to further decrease cost of manufacture and introduce part-interchangeability.

2. Design

This section will introduce the design and specification of all major components. These component groups are divided into pedal box, rotors/calipers, and master cylinders/fittings. Each group includes improvements upon the previous year's design. The design of each group involves specific design equations and selection criteria that are determined by the rules and goals of the system overall. While the pedal box was fully designed and manufactured in-house, other components were specified and ordered or sent out to be manufactured by sponsor companies with the required capabilities.

2.1 Pedal Box

The design of the pedal box incorporates two independent pedals (accelerator and brake) mounted to an adjustable base. The base must be easily adjustable by hand to accommodate the tallest and shortest driver. The accelerator pedal must actuate two independent position sensors to communicate a variable torque request to the ECU. The accelerator must also be spring loaded to return to the zero position when not pressed by the driver. On the other hand, the brake pedal must impinge on two independent master cylinders to pressurize the front and rear brake systems. A balance bar is commonly used to adjust bias between the two cylinders.

The design for the ZRE 22 pedal box began with a look at the previous design. The previous system had strong points as well as weaknesses. Use of a pedal machined out of a single piece of 7075 Aluminum proved to have a high strength to weight ratio, but it presented challenges to manufacture. The adjustment system utilized a dovetail, but the tolerances were incorrect, so it bound up and was difficult to use without tools. Finally, the throttle pedal return springs were too weak and the sensors showed oscillation as the vehicle coasted over bumpy surfaces

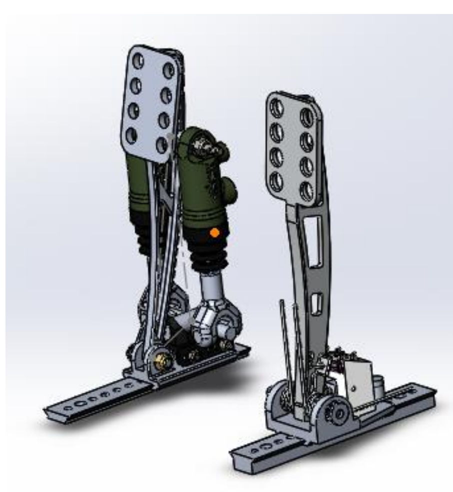

Figure 2: Front view of ZRE 2020 pedal box courtesy of Z. Demetriades, D. Seay, and J. Zogheib.

The new pedal box design involved a great deal of compromise. This was necessary to meet the overall goals of light, rigid, machinable and modular. First off, the brake and accelerator pedal as well as carriers were designed to be identical. This has the benefit of making the system cheaper to produce, however both pedals must be strong enough to meet the requirements of the brake pedal. Both pedals must be built to survive an input of 2 kN to the pedal face which is an FSAE rule requirement of the brake pedal only. The result is an optimized brake pedal, and an overbuild accelerator pedal.

Another major design choice was to make the pedals as an assembly as opposed to a single piece. This satisfies the desire to be machinable and modular as it is easier to manufacture and interchange/replace parts. The downside to this design is that the connection points introduce weaknesses and stress concentrations. Tolerancing of the components is very important as they must fit snugly together to ensure proper rigidity. Finally, the pedal face was designed to be 3D printed because of its complex geometry. Markforged Onyx was the selected material.

The adjustability of the pedal box was improved. Similar to the previous design, a dovetail was employed for front to back adjustment. However, the dovetail will be properly tolerance and contact area minimized. A qwik lok pin was used instead of a custom spring loaded taper pin to provide higher shear strength and easier adjustability. Two 3/16 inch qwik lok pins were sponsored by Jergens. Additionally, the pedal face is vertically adjustable to accommodate for the driver’s shoe size.

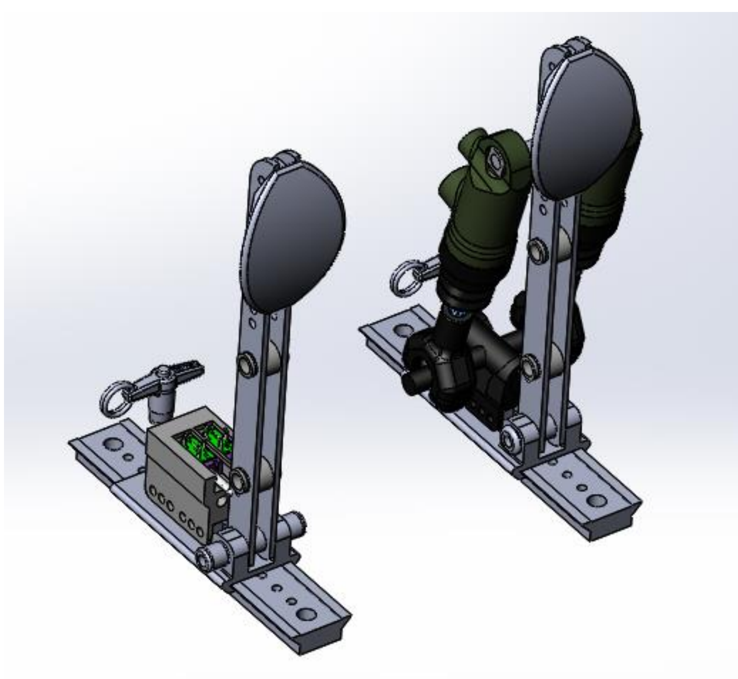

Figure 3: Final 2021-22 pedal box assembly

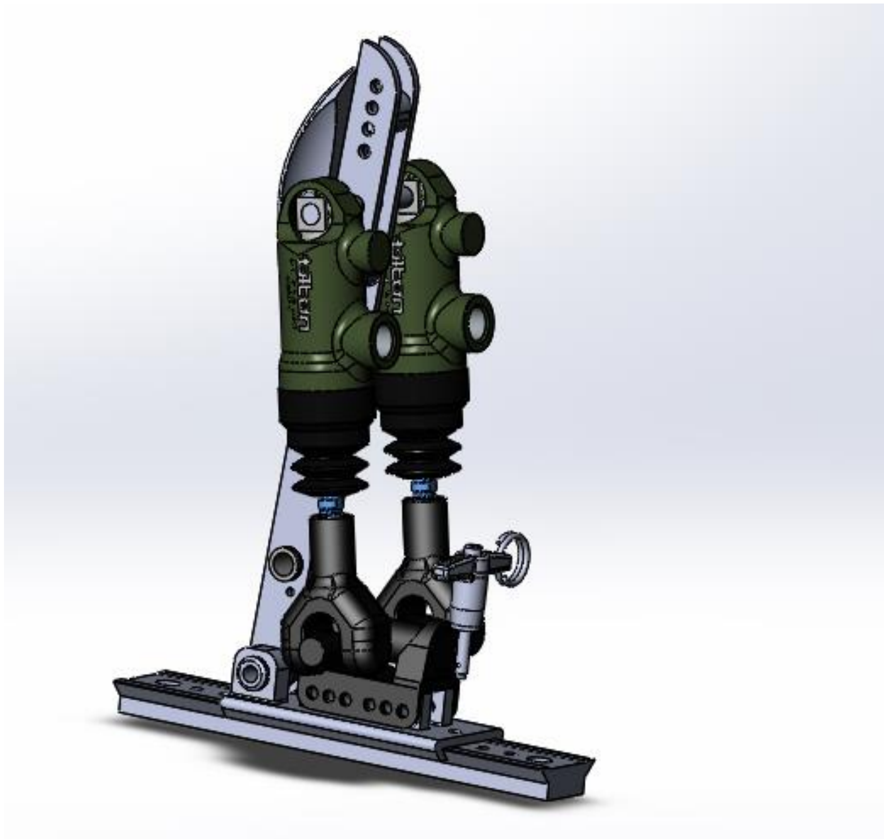

Figure 4: View of the brake pedal from the rear.

An important measure of the pedal design is the motion ratio. A greater motion ratio equates to greater mechanical advantage on the master cylinders. This parameter controls the pedal feel and driver input. Motion ratio is calculated as change in the pedal face position (pedal travel) divided by the change in length of the master cylinder from zero to fully compressed states (compression). The goal was to meet or exceed the previous year's figure of 3.49. The ZRE 22 design reached a motion ratio of 4.19. Coordinates and lengths for the calculation were taken from the SolidWorks model.

$$MR = \frac{\sqrt{1.714^2 + 0.141^2}}{MC_0 - MC_1} = \frac{1.720}{6.63 - 6.22} = 4.19$$

The morphological chart provides a roadmap through the design decisions for each of the sub-components. This is accurate to how the design process flowed through iterations of each assembly revision. A final design was reached by testing assembly fit and function as well as intuition and input from faculty advisors.

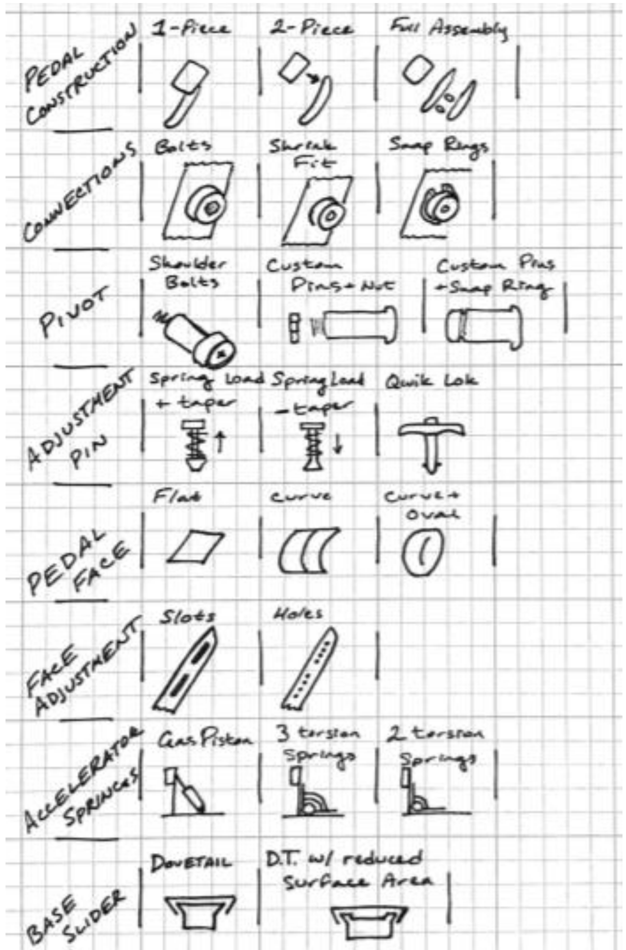

Figure 5: Pedal Box Morphological Chart

2.2 Rotors and Calipers

First and foremost, the selection of the brake pad plays a huge role in the overall feel and performance of the braking system overall. The chosen pad for the design is the SBS 655DC dual carbon brake pad. Benefits of the pad are provided on the SBS website,

- The upgrade choice for Superbike, Supersport and Superstock racers in National Championship as well as for Track Day riders
- High-tech carbon compound developed for racing and standard brake systems used for race and sport bikes
- Low heat transfer rate protects brake system and brake fluid against extreme temperatures
- Smooth initial bite, progressive in-stop performance with excellent brake lever feel and modulation
- DEST technology used for pre-bedding of the compound to eliminate fade and secure consistent performance

- NUCAP NRS technology secures a mechanical and indestructible bonding of the compound (SBS Friction A/S)

The first points advertise dual carbon pads as being the choice for aftermarket upgrades. This is true as the compound is an easy swap for stock pads because it shares many of the benefits of stock pads with an increase in stopping power. Dual carbon is able to conduct heat to the rotors. DC compound pads are also less aggressive than other compounds. This means their service life is similar to a stock pad. Technical datasheet for the compound is provided in Appendix B.

Bedding in brake pads is the process of breaking them in through actual use. Bedding is important because pads clamping on a perfectly clean rotor is actually less effective. In addition, bedding scrapes away the first few layers of the pad, exposing the higher friction compound beneath. Think of this process as heating up your tires before a drag race. Bedding warms up the compound and forms the pad to the rotor, increasing the coefficient of friction. Pre-bedding the pads is a must in the competition scenario. If the decision is made to swap out pads in competition, it is best to enter the race with a set of pads that is already bedded in. The same rule applies with tires.

Finally, the dual carbon pads have a few weak points. Primarily, the pads brake most effectively when hot. For this reason, the rotor design should not include excessive cooling. Rather, the rotor should build up to an operating temperature and then dissipate the rest.

Transitioning to the design of the rotors, an extensive literature and market search was conducted. With a multitude of designs on the market, it is difficult to discern which is best. Keeping in mind the application at hand, it was difficult to choose the best pattern. Blank (no pattern), drilled, dimpled, slotted, and hybrid designs were created in SolidWorks. Each pattern has its own list of pros and cons. Drilled rotors are lightweight and good at diverting water, but each hold introduces a stress concentration. Dimpled are similar to drilled but retain more material so they are stronger and heavier. Slotted provide a good balance as they are strong, divert water and debris, and wear the pad evenly.

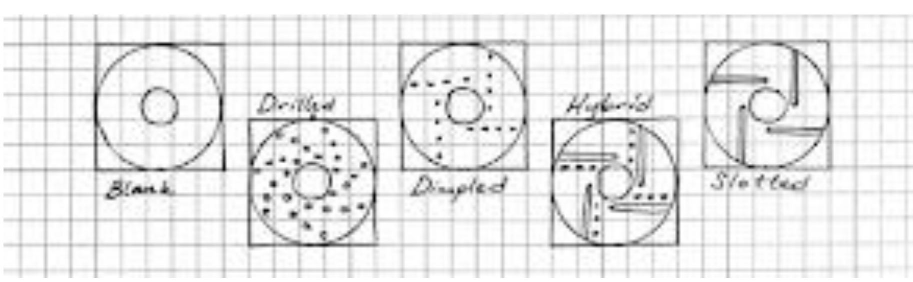

Figure 6: Rotor design morphological chart

The decision was made to go with slotted rotors with straight slots. The slot design helps to accelerate water and other debris away from the center of the hub. In addition, the slots wear the pad evenly to expose fresh material. On a blank rotor, pads can become ‘glazed over’ which limits performance. By uniformly scraping away this top layer, the pad surface remains effective. Finally, the rotor remains massive enough to retain more heat. This draws heat out of the pad and caliper to dissipate it and reduce brake fade.

The rotor will be fastened using a ‘floating’ design. This is important to account for unbalance in the part as well as expansion due to thermal load. Cylindrical aluminum bobbins will interface between the rotor and hub to allow for deformation. These parts will be hand fitted to control the play in the assembly.

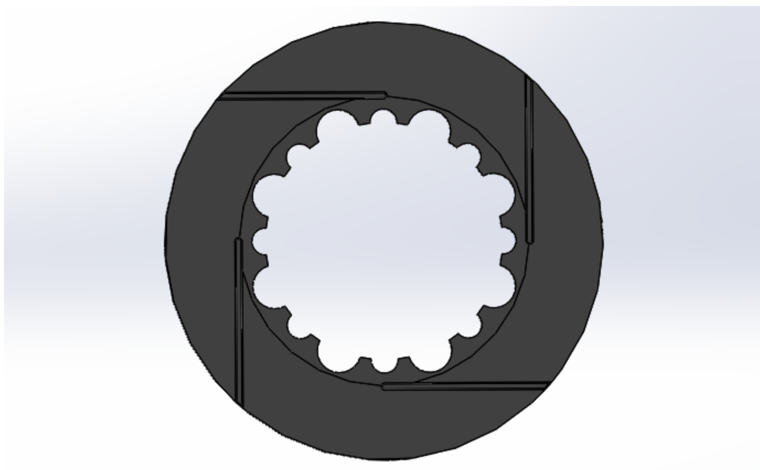

Figure 7: Final slotted rotor design. The disk will attach to the hub via aluminum bobbins to maintain a floating rotor design.

2.3 Master Cylinders and Fittings

In order to route the lines through the chassis, the subsystem was modeled in the frame assembly. This is important to communicate to the other subsystems where the lines will be located. With hand tools, the hard lines may be bent to any angle or radius of curve. However, the lines are preferred to be level and straight for the reduction of friction and losses in the system.

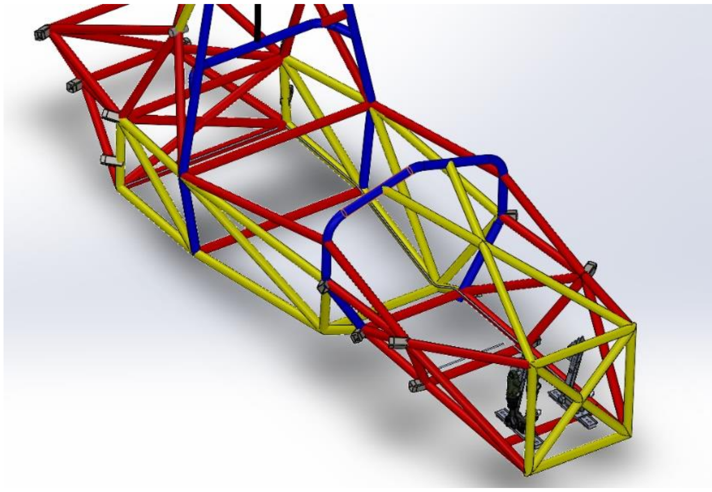

Figure 8: Pedal box and hard lines modeled in the frame. Rotors and calipers will be modeled with the upright assembly.

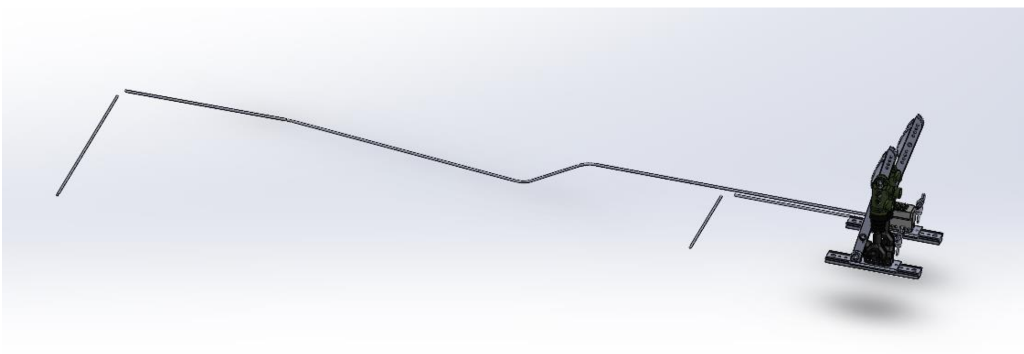

Figure 9: Model of the hard line geometry. This model is useful to measure, cut, bend, and fit the hard lines that physically go in the car.

For the same reason, the number of fittings and length of soft lines should be minimized. Each fitting introduces a possible leak point in the system while increasing friction and adding weight. For the 2022 design, the number of fittings was reduced by choosing different in-line pressure sensors and switching to banjo bolts on the master cylinders. Past designs stacked two or three fittings on the master cylinders, but the use of banjo bolts reduces that to a single fitting. Finally, appropriate lengths for the soft lines are measured from the frame model.

2.3.1 Master Cylinder Equations

In specifying the size of master cylinder required for the system, a string of hand calculations is required. It was decided to choose from the Tilton 78 series line of products as they are designed for the application and mate to a balance bar also provided by Tilton. A range of master cylinder piston sizes is offered, so the closest size to the design value is specified.

First, the static weight transfer of the vehicle must be established. This is important because the front and rear systems are independent, each requiring a dedicated master cylinder. Weight (W), wheelbase (L), and weight transfer (F_{WR}, R_{WR}) are determined by the frame team.

$$W = 525\ lb$$

$$L = 60.5\ in$$

$$F_{WR} = 48\%$$

$$R_{WR} = 52\%$$

$$H_{CG} = 11.5\ in$$

$$W_F = W * F_{WR} = 252\ lb$$

$$W_R = W * R_{WR} = 273\ lb$$

$$CG_F = W_R * \frac{L}{W} = 31.46\ in$$

$$CG_R = W_F * \frac{L}{W} = 29.04\ in$$

With the static weight transfer of the vehicle, it is possible to now calculate the dynamic weight transfer under maximum deceleration (a_{max}). This is defined as fully locking up the brakes at a speed of 30 mph. Under load, we can calculate the weight on front brakes (W_{FB}) and rear brakes (W_{RB}) including the downforce provided by the aerodynamic features at the front and rear axles.

$$F_{ax\ DF} = 80\ lbf$$

$$R_{ax\ DF} = 65\ lbf$$

$$W_{FB} = \frac{W}{L} * CG_F + \left(\frac{a_{max} * H_{CG}}{g}\right) + F_{ax\ DF} = 529.85\ lbf = 79.08\%$$

$$W_{RB} = \frac{W}{L} * CG_R + \left(\frac{a_{max} * H_{CG}}{g}\right) + R_{ax\ DF} = 140.15\ lbf = 20.92\%$$

Therefore, under maximum deceleration, 197.85 lb is transferred from the rear to the front. The suspension and brakes system must be able to dynamic force to the front. For this reason, the front brakes must produce a higher pressure.

The next figure to be calculated is the motion ratio of the pedal. The motion ratio (M) is given in section 2.1 as 4.19. The factors of friction for the brake pads and tires are also given below. These dynamic μ values are averaged and equal in both front and rear. These are used to calculate the friction forces on the tires.

$$M = 4.19$$

$$\mu_{tire} = 1.5$$

$$\mu_{pad} = 0.55$$

$$F_{fr\,f} = \mu_{tire} * W_{FB} = 794.776\ lbf$$

$$F_{fr\,r} = \mu_{tire} * W_{RB} = 210.224\ lbf$$

Multiplying the friction force by the radius of the tire yields brake torque.

$$R_{tire} = 10.05\ in$$

$$\tau_f = R_{tire} * F_{fr\,f} = 7987.494\ in\ lb$$

$$\tau_r = R_{tire} * F_{fr\,r} = 2112.756\ in\ lb$$

The last step before calculating the master cylinder diameter is solving for the maximum hydraulic pressure in the front and rear systems. A force on the pedal of 150 lbf (F_{pedal}) is assumed to be exerted by the driver. Other factors such as caliper dimensions are provided by ISR. The areas differ as four-piston calipers are used in the front and two-piston calipers in the rear.

$$A_{piston\,f} = 6.08\ in^2$$

$$A_{piston\,r} = 3.04\ in^2$$

$$P_{\max f} = \frac{F_{caliper\,f}}{A_{piston\,f} * \mu_{pad}} = 795.713\ psi$$

$$P_{\max r} = \frac{F_{caliper\,r}}{A_{piston\,r} * \mu_{pad}} = 420.945\ psi$$

Finally, the master cylinder diameter (bore) can be specified.

$$A_{piston\,f} = \frac{F_{pedal} * \frac{M}{2}}{P_{\max f}} = 0.395\ in^2$$

$$A_{piston\,r} = \frac{F_{pedal} * \frac{M}{2}}{P_{\max r}} = 0.747\ in^2$$

$$D_{piston\ f} = \sqrt{\frac{4}{\pi} * A_{piston\ f}} = 0.709\ in$$

$$D_{piston\ r} = \sqrt{\frac{4}{\pi} * A_{piston\ r}} = 0.975\ in$$

From here, it is necessary to choose from the list of available master cylinder bore sizes offered by Tilton Racing in the 78 series line. A 0.700 inch master cylinder is specified for the front and a 1.0 inch for the rear.

2.4 FMEA

The Failure Mode and Effects Analysis (FMEA) is a common tool used by the auto industry in the design phase of a vehicle. Teams from each subsystem generate an FMEA for their own system to analyze the risk of certain failure modes. The design is scrutinized and revised as many times as it takes to reach a goal. This method is very effective as it challenges designers to strengthen the weak links in their proposed design.

The metrics measured in an FMEA are severity, occurrence, and detection. Severity measures the outcome of the failure from small inconvenience to possible injury or death. Occurrence equates to the likelihood a failure will occur based on the life expectancy of the component. Finally, detection is calculated based on the ease of which the operator can measure component wear or requirement for service. Modern cars with Tire Pressure Monitoring Systems (TPMS) would score well in detection because incorrect tire pressure triggers a warning to the driver in the gage cluster.

After all of the metrics are determined, the Risk Priority Number (RPN) is calculated. The RPN is merely the product of the severity, occurrence, and detection. The higher RPN failure modes are revisited by the designer to mitigate risk in an improved design. The FMEA presented for the ZRE 22 brakes subsystem includes recommended actions to decrease RPN. The highest RPN results if the installer forgets to replace the quick pin. This failure mode is very severe and has a moderate level of occurrence because the installer may be in a hurry to adjust the pedal box in a race situation. A suggestion to mitigate risk is to include a lanyard to tether the pin to the frame so it is not forgotten.

FAILURE MODE AND EFFECTS ANALYSIS				
Item:	ZRE Brakes Subsystem	Responsibility:	N. Thomas	
Model:	2021-22	Prepared by:	N. Thomas	
Core Team:	N. Thomas, B. Wyler, Z. Koneval			

Process Function	Potential Failure Mode	Potential Cause(s)/ Mechanism(s) of Failure	Sev	Potential Effect(s) of Failure	Occur	Current Process Controls	Detect	RPN	Recommended Action(s)
Press brake pedal	Carrier moves on rail	Quick pin not installed	5	Car does not brake	2	Installation training	2	20	Install lanyard on quick pin
	Pedal disassembles	Improper assembly	5	Car does not brake	1	Operator training and instructions	2	10	Clear callout of assembly procedure in drawing
		Improper tolerance on machined components	5	Car does not brake	1	Operator training and instructions	2	10	Clear callout of tolerance in drawing
	Brake pedal does not support input force	Improper manufacturing	4	Car does not brake	1	Pedal FEA and physical test	2	8	Inspect all manufactured parts
	Braking is ineffective	Not enough brake fluid	3	Car does not brake	2	Check level in reservoirs	1	6	Practice routine inspection
		Brake fluid leak in lines	3	Car does not brake	2	Inspect lines, fittings for leaks	2	12	Practice routine inspection
	Foot slips off of pedal	Pedal face not adjusted properly, low friction	4	Car does not brake	1	Adjustable pedal face, grip tape	1	4	
Press throttle pedal	Carrier moves on rail	Quick pin not installed	3	Car does not accelerate	2	Installation training	2	12	Install lanyard on quick pin
	Pedal disassembles	Improper assembly	2	Car does not accelerate	1	Operator training and instructions	2	4	Clear callout of assembly procedure in drawing
		Improper tolerance on machined components	2	Car does not accelerate	1	Operator training and instructions	2	4	Clear callout of tolerance in drawing
	Foot slips off of pedal	Pedal face not adjusted properly, low friction	2	Car does not accelerate	1	Adjustable pedal face, grip tape	1	2	
	Car does not respond to request	Linear sensors not spaced correctly	2	Car does not accelerate	1	3D printed mount secured to carrier	2	4	Shims for 3D printed mount
		Hard-stop not adjusted properly	1	Car does not accelerate	2	Adjustable hard stop	2	4	Have replacements
		Linear sensors do not agree	1	Car does not accelerate	1	3D printed mount secured to carrier	3	3	Shims for 3D printed mount
Calipers engage rotors	Braking is ineffective	Brake pads are not bedded	2	Car does not brake	1	Pre-bedded pads	1	2	Follow bedding procedure
		Brake pads are worn out	3	Car does not brake	2	Inspect pads	3	18	Listen for chirping
		Debris, water on rotor	3	Car does not brake	2	Slots in rotor eject debris, water	1	6	Incorporate more modes of ejecting debris, water in design
		Fluid leak from caliper	4	Car does not brake	1	Inspect lines, fittings for leaks	2	8	Practice routine inspection

Figure 10: FMEA for the ZRE 22 brakes subsystem design.

3. Design Verification

With the design of any system, it is important to prove the design will meet requirements before entering mass production. In the case of automobiles, this is incredibly serious as failure of a component during operation can cause injury or death. In the iterative design phase, Finite Element Analysis (FEA) is employed to test different revisions. FEA is a software that uses boundary equations and applied forces to simulate the physical response of a model geometry. FEA is a very powerful tool and greatly simplifies the design process

However, FEA can have downsides. With any simulation, certain assumptions are made in terms of the boundary conditions. In addition, the model geometry is often a simplified version of the actual component. These shortcuts can lead to error in the results. For this reason, physical testing is necessary to further validate the results of the FEA simulation. This section will discuss the verification of the design of the pedals and the rotors.

3.1 Pedal Box

For the design of the pedals, there is basically one parameter which determines their necessary strength. Per the 2022 FSAE Rulebook, the brake pedal must survive an input of 2kN to the face. This is tested at competition when a design judge sits in the vehicle and kicks the pedal with all of his/her power. To test conformance to this specification, SolidWorks FEA was used and the results were validated with a physical load test on the Instron machine.

In order to meet the 2kN spec, certain design choices become important. First, material selection is made. The material chosen for the pedals is 7075 aluminum. AA7075 is a commonly used alloy in the aircraft industry and is widely available in a variety of stock sizes. The alloy is known for good fatigue resistance, strength, and ductility. These are all critical for the design of a pedal as the design resembles a cantilevered beam under load. The material for the 3D printed face was selected to be Markforged Onyx. Onyx is a carbon fiber filled nylon material which is extremely rigid even with thin features. The layer adhesion is also very strong, so the face will not shear in the build plane.

Another critical design choice is geometry. Stress concentrations provide weak points under load and micro-cracks can lead to failure. The design should avoid thin features and sharp corners to minimize stress concentration points. Finally, the designer should consider the interface between each part. Force transmission is when input force is mapped through the components. Any extra force applied to the pedals should be directed to the frame. The force will effectively travel from the pedal to the carrier to the rail which is bolted to the frame. The qwik pin is rated for 23 kN in double shear; more than adequate for the application.

3.1.1 Pedal Box FEA

Solid mechanics FEA for the pedals was conducted in SolidWorks. The pedal was fixed at the pivot pin and master cylinder mount pin as 2kN was applied to the face. Assumptions for this simulation include programming the connection points as rigid. This means they all fit together perfectly as if the pedal were a single piece. In addition, the fixed condition at the master cylinder mount does not take into account the stiffness and damping ratio of the cylinders that mount to it. Therefore, the FEA serves only as an estimate of the system's response under load.

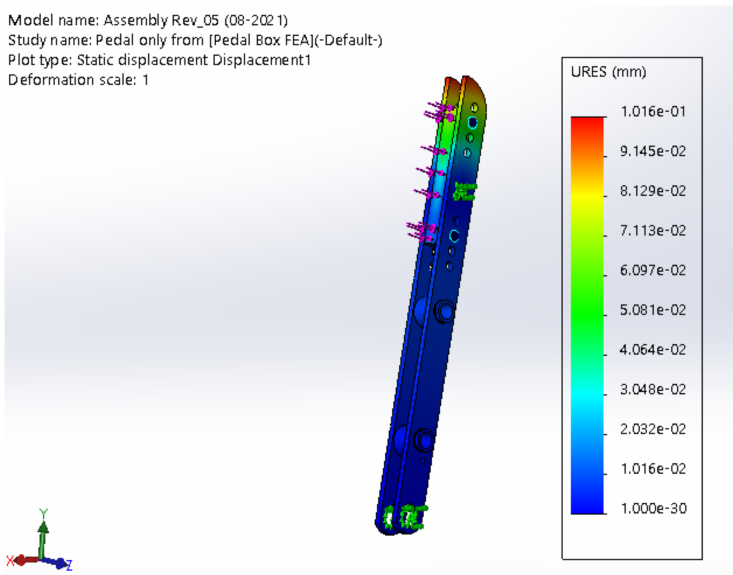

Figure 11: Displacement FEA of pedal under 2kN load. Displacement was observed to be acceptable.

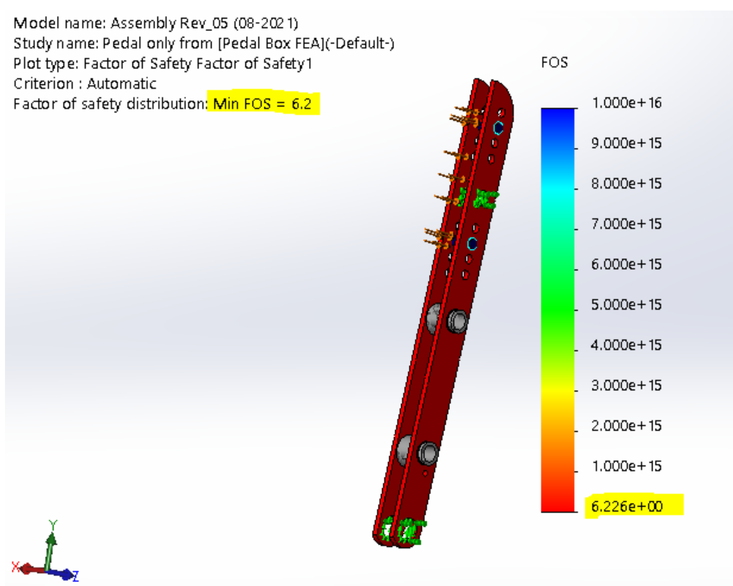

Figure 12: Factor of Safety FEA of pedal under 2kN load. Minimum FOS is 6.2 which is higher than desired.

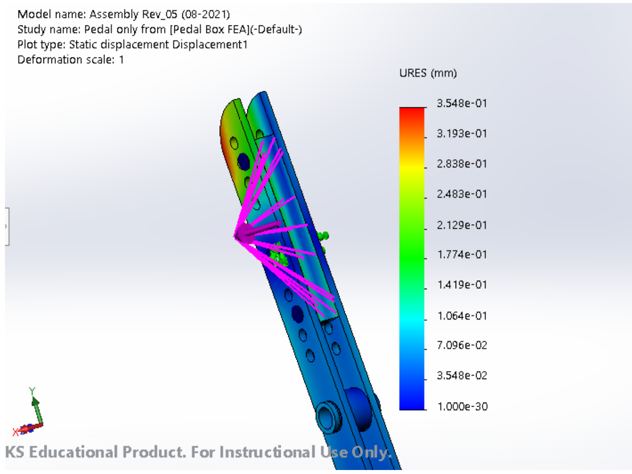

Figure 13: Displacement FEA of pedal under offset 2kN load. Displacement was observed to be acceptable.

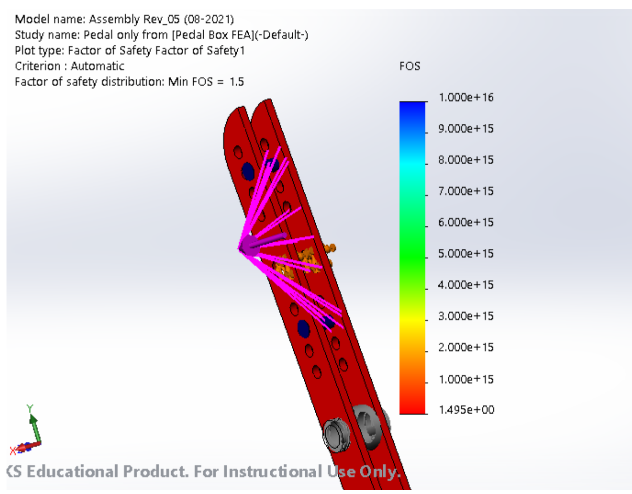

Figure 14: Factor of Safety FEA of pedal under offset 2kN load. Minimum FOS is 1.5 which is fully optimized.

3.1.2 Pedal Box Physical Test

The brake pedal FEA was further validated by a destructive physical test. A prototype pedal was manufactured to the exact design. The test was then set up on the Instron machine in the University's Materials Testing Lab. For measurement, the Gom Digital Image Correlation (DIC) system was used. The stem of the pedal was painted with a speckled pattern and the dots were monitored by the DIC cameras. The cameras track the position of the dots to plot displacement as force was applied. The pedal shifted in the fixture and failed around 3kN. This result confirms the FEA result.

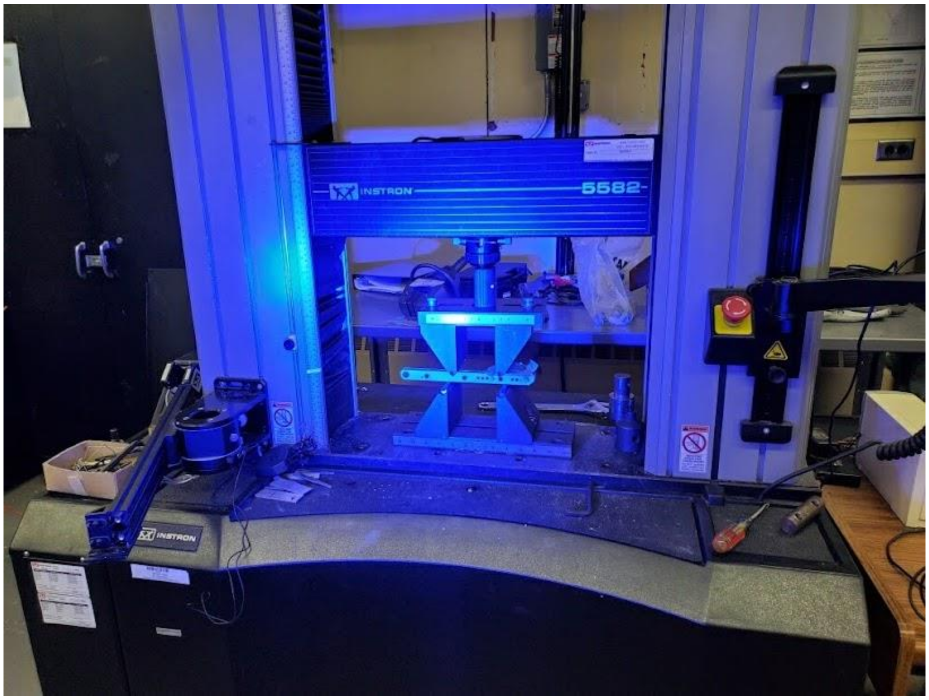

Figure 15: setup of the prototype pedal in the Instron machine.

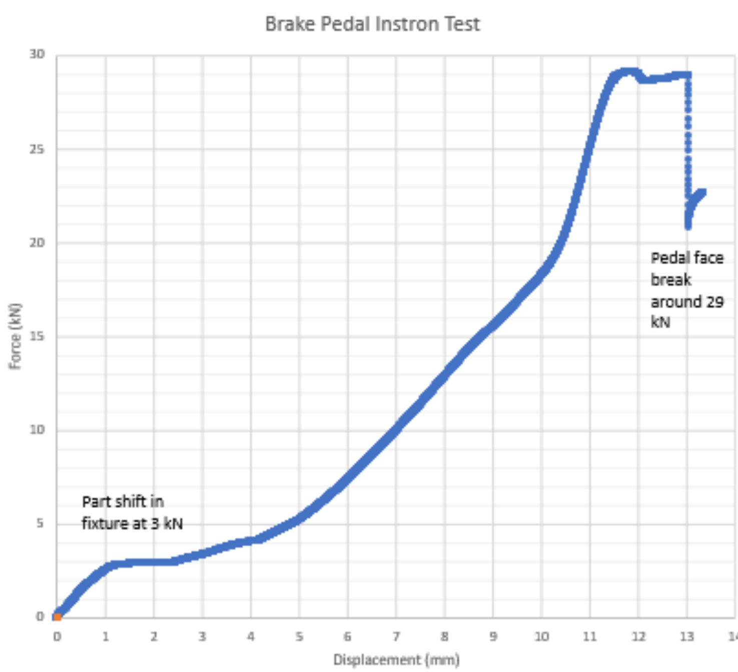

Figure 16: Plot of displacement vs applied force. Testing was conducted with Dr. Mani Kannan of the University of Akron.

3.2 Rotors

The brake rotors are subject to both mechanical and thermal loads. The mechanical load is the braking torque which is defined as the braking force applied by the caliper multiplied by the radius of the tire which acts as the moment arm. Additionally, the thermal load is generated by the friction force of the pads contacting the disk. The heat must be dissipated correctly to avoid brake fade and damaging components. Grey cast iron was chosen for the material because its properties are well suited for the application. These properties include:

- Low thermal expansion coefficient so disc doesn't deform as much
- Low thermal conductivity so disc acts as a heat sink
- Good coefficient of friction
- Pearlitic structured material to improve strength and wear properties
- Natural damping properties to minimize vibration
- Cost-effective (Chatfield, Kucera 2018)

3.2.1 Solid Mechanics FEA

The solid mechanics FEA was conducted in SolidWorks. With the bobbin interfaces fixed, the maximum brake torque of 7987.494 in lb (see section 2.3.2) was applied to the surface on both sides of the disk. Assumptions of this simulation include the fixed condition of the bobbins. In reality, there will be a small amount of 'give' in the aluminum bobbin material as well as the tolerance allowed in the floating rotor design. However, the FEA is generally a good representation of the system's behavior. Physical validation will not be pursued.

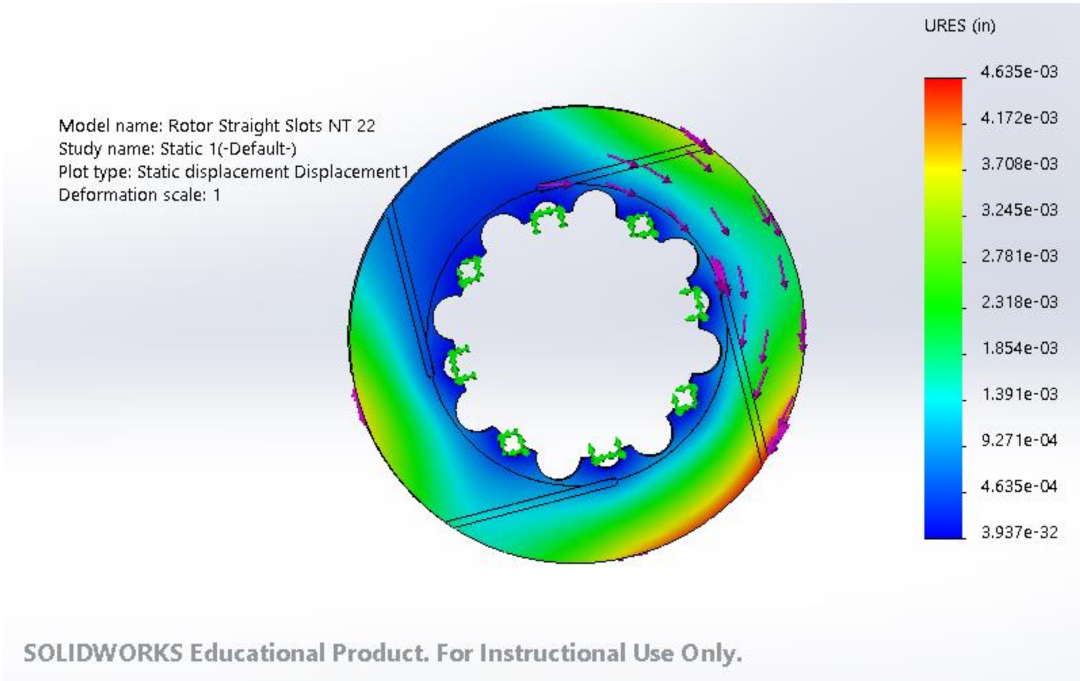

Figure 17: Displacement of the brake rotor under maximum load. The deformation is acceptable.

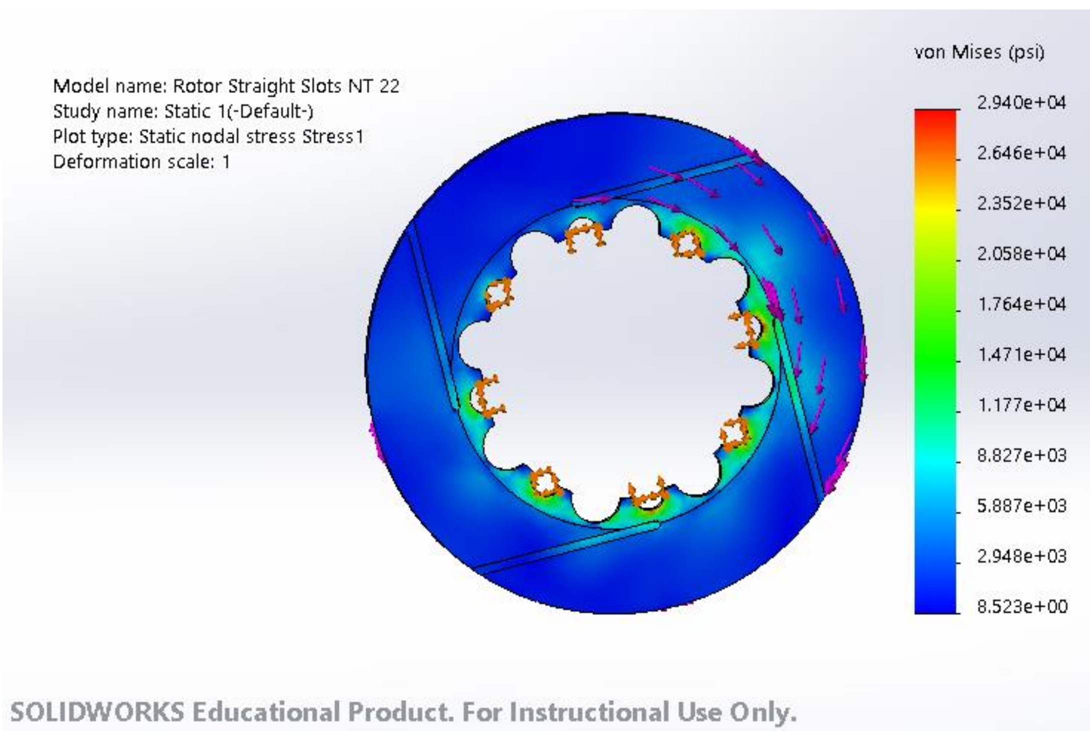

Figure 18: Stress plot of the rotor under maximum load. Stress concentration is seen at the interface with the bobbins.

3.2.2 Heat Transfer FEA

Once again, SolidWorks FEA was employed to simulate the rotor under maximum load. The study was set up with three values: initial temperature, convection, and heat power. Initial temperature of the rotor was set to standard condition of 25°C. Convection was set to simulate ambient air passing across the surfaces of the rotor with an approximated coefficient of 100 W/m^{2}K. Finally, heat power was calculated based on maximum braking over 3 seconds with 80% front braking bias.

$$KE = \frac{1}{2}mv^2 = 21367.64\frac{kg\ m^2}{s^2}$$

$$Heat\ Power_{total} = \frac{KE}{\Delta t} = 7122.55\ W$$

$$Heat\ Power_{front\ rotor} = \frac{0.8\ HP_t}{2} = 2849.02\ W$$

The heat power accounts for energy transferred through the brake pads to the rotor under maximum braking. This calculation is not the most accurate method to model the heat energy transferred, but it is enough to get an idea of the absolute maximum service temperature of the rotors.

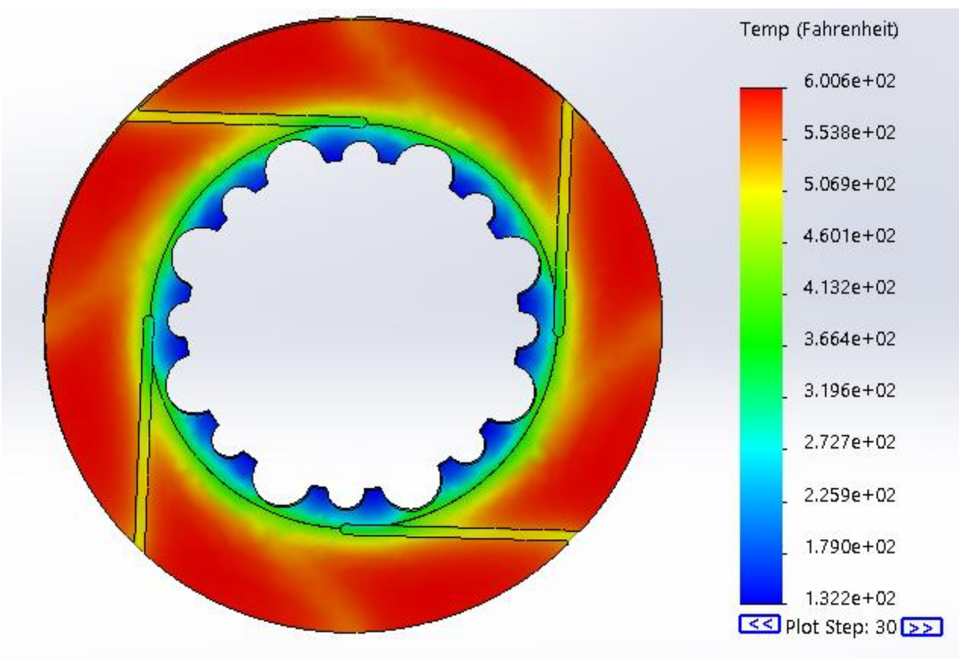

Figure 19: Heat transfer FEA of brake rotor conducted in SolidWorks.

Mesh independency analysis was conducted and very similar results were achieved at varying mesh size. Curvature based mesh was used for simulation. As shown in the figure, the maximum temperature of the rotor at the end of a 3-second full lockup is about 600°F. This is well below grey cast iron's maximum service temperature of 842°F (T_M = 2060-2200°F). The FEA results will be verified when a running car is completed and an infrared heat gage can be used to measure the rotor temperature after a hard braking cycle.

4. Cost

For budgeting reasons, the design teams at the University of Akron are required to present a projected cost for the year. Each subsystem completes an individual budget that is summed up to generate a predicted cost for a complete car. After sponsorship, the remainder of the budget is supplied by the College of Engineering and Polymer Science (CEPS).

An important comparison to note is the difference in cost between the previous and current pedal designs. This is the greatest difference among the two budgets. For the previous, one-piece pedal design, the pedals cost about $94 each including necessary hardware. The ZRE 22 pedal costs only about $25. In addition, the previous pedal required a CNC program with several setups while the ZRE 22 pedal components could be mass produced much more economically.

Description	Link/SKU	Supplier	Discount	Cost/Unit	Qty.	Cost	Expenditure
Fittings ---------------					-		
Permatex pneumatic/Hydraulic Sealant	54540	Summit Racing	gift card	$ 28.99	1	$ 28.99	$ 28.99
Goodridge 3AN Male x 1/8 inch NPT Female	GDR-MCHD-014BK	Summit Racing	gift card	$ 17.40	2	$ 34.80	$ 34.80
Summit Racing 3AN Tube Sleeves	SUM-220334-6B	Summit Racing	gift card	$ 4.99	3	$ 14.97	$ 14.97
Summit Racing 3AN Tube Nuts	561803ERL	Summit Racing	gift card	$ 4.63	8	$ 37.04	$ 37.04
Fragola Performance Systems Hose Barb to AN	484106-BL	Summit Racing	gift card	$ 9.99	2	$ 19.98	$ 19.98
Fragola Peformance Systems Banjo Bolts	980358-BL	Summit Racing	gift card	$ 16.99	2	$ 33.98	$ 33.98
Brake Rotors ----------------					-		
Brake rotor stock		Alro Steel	In Stock	$ 238.19	1	$ 238.19	$ -
Bobbin material - 7075 Round 1.0 dia		Clinton Aluminum	70%	$ 15.00	1	$ 15.00	$ 4.00
Snap Rings for rotors	97633A190	McMaster		$ 12.08	1	$ 12.08	$ 12.08
Brake Cleaner	SUM-941241	Summit Racing	gift card	$ 3.99	1	$ 3.99	$ -
Rotor machining/surfacing		Precision Componen	100%				
Calipers ----------------					-		
Front calipers	22-048	ISR		$ 223.32	2	$ 446.64	$ 446.64
Rear Calipers	22-049	ISR		$ 131.90	2	$ 263.80	$ 263.80
Shipping and VAT on Calipers		ISR		$ 16.30	1	$ 16.30	$ 16.30
Brake Pads		SBS	In Stock	$ 110.00	1	$ 110.00	$ -
Caliper Mount Bolts		Fastenal	In Stock	$ 18.00	1	$ 18.00	$ -
Pedal Box Material ----------------					-		
7075 plate - Pedal Stems	8885k862	McMaster		$ 30.65	1	$ 30.65	$ 30.65
6061 Block - Baseplates		Clinton Aluminum	In Stock	$ 23.33	2	$ 14.00	$ -
6061 Block - Rails		Clinton Aluminum	In Stock	$ 10.00	1	$ 3.00	$ -
MC Mount Material - Steel Round			In Stock	$ 10.00	1	$ 10.00	$ -
Crossmember /Pivot Pin Material - 6061 Round (:	8974k11	McMaster		$ 10.74	1	$ 10.74	$ 10.74
Jergens Qwik Pin	801000	Jergens	100%	$ 19.24	2	$ 38.48	$ -
Front Master Clylinder Rebuild Kit	78-700RK	Tilton	see form	$ 29.00	1	$ 29.00	$ 29.00
Rear master Clylinder Rebuild Kit	78-1000RK	Tilton	see form	$ 29.00	1	$ 29.00	$ 29.00
Brake Bias Adjustment (Balance Bar)	72-280	Tilton	see form	$ 625.00	1	$ 453.13	$ 453.13
BOTS Mount Material			In Stock	$ -	1	$ -	$ -
LH Rotary Spring for pedal box (+tx)	3HPN1	Grainger		$ 4.80	1	$ 4.80	$ 4.80
RH Rotary Spring for pedal box (+tx)	3HPK7	Grainger		$ 5.25	1	$ 5.25	$ 5.25
Pedal Face mount bolts	2123167	Fastenal	In Stock	$ 3.50	1	$ 3.50	$ -
Balance Bar mount bolts	2123170	Fastenal	In Stock	$ 3.50	1	$ 3.50	$ -
Balance Bar/Pedal Face mount nuts	37015	Fastenal	In Stock	$ 0.09	4	$ 0.36	$ -
Pivot pin/Barrel Connector snap rings	97633A200	McMaster		$ 12.67	1	$ 12.67	$ 12.67
MC Mount Snap Rings	97633A150	McMaster		$ 11.04	1	$ 11.04	$ 11.04
Lines -------------					-		
12" Flex Line		Speedway		$ 12.99	2	$ 25.98	$ 25.98
25" Flex Line		Speedway		$ 14.99	4	$ 59.96	$ 59.96
Summit Racing Copper/Nickel Tubing	SUM--220216-25	Summit Racing		$ 31.99	1	$ 31.99	$ 31.99
Brake Reservoirs		Amazon		$ 6.99	2	$ 13.98	$ 13.98
Brake Fluid		Summit Racing	gift card	$ 14.99	4	$ 60.00	$ 60.00
Flex Brake Line to reservoirs	https://www.summitracinc	Amazon		$ 9.86	1	$ 9.86	$ 9.86

Figure 20: ZRE 22 brakes subsystem budget totaling $1700.63

The manual machining for each pedal in the ZRE 22 design came with a cost of about 2 hours of machine time per pedal. This figure is including the fact that the material purchased was already roughly cut to final dimensions. One of the most time consuming components was machining the pedal stems. For mass production, these pieces would be cut out of a large sheet of 7075 aluminum with a CNC waterjet. This method would save time and preserve both the microstructure properties and the accuracy required of the component. Another suggestion would be to use an extruded material for the rails. This would save the time of cutting the dovetail.

5. Conclusion

The design and manufacture of the ZRE 22 brakes subsystem was a success that improved upon the previous year. The new design is lighter, simpler, and more cost effective. With all of the changes, the design is still rules compliant per FSAE. This system is expected to perform well during competition design and dynamic events.

Some future areas of improvement could include the design of the rotors and a few aspects of the pedal box. First, the rotors could be much lighter if the design included ventilation or through holes. Strict analysis would be required to ensure stress concentrations at the holes would not cause the rotor to crack. Moving on, the pedal box could be improved by including bearings at the base. This would increase the lifespan of the system and decrease the friction of pressing the pedal. Additionally, the packaging of the throttle pedal could be improved by employing rotary instead of linear sensors and gas pistons in place of torsion springs.

Figure 21: Final manufactured pedal box

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Appendix A

Rules Questions

Often, rules questions must be submitted to FSAE for review if it is unclear whether the proposed design meets specific requirements. A rule question was asked to confirm that employing a 3D printed pedal face would be allowed.

T.3.1.11 The brake pedal must be one of:

- Fabricated from steel or aluminum
- Machined from steel, aluminum or titanium

Figure: Rule T.3.1.11 from the 2022 FSAE rulebook.

17111

3D Print Brake Pedal Face

Rule Numbers:

2022:T.3.1.11

Opened: 2021-09-02 17:24:03 ET

State: Closed

Status: Closed By Team

Team Member 2021-09-02 17:25:42 ET EST (GMT-5)

Nicholas Thomas
Team Member for Univ of Akron, Zips Racing Electric since 2019

I would like to employ a brake pedal design where the stems are machined from aluminum and the face is 3D printed. Would this be allowed?

- 1-14 corner view.PNG

Approved Resolution 2021-12-15 00:15:37 ET EST (GMT-5)

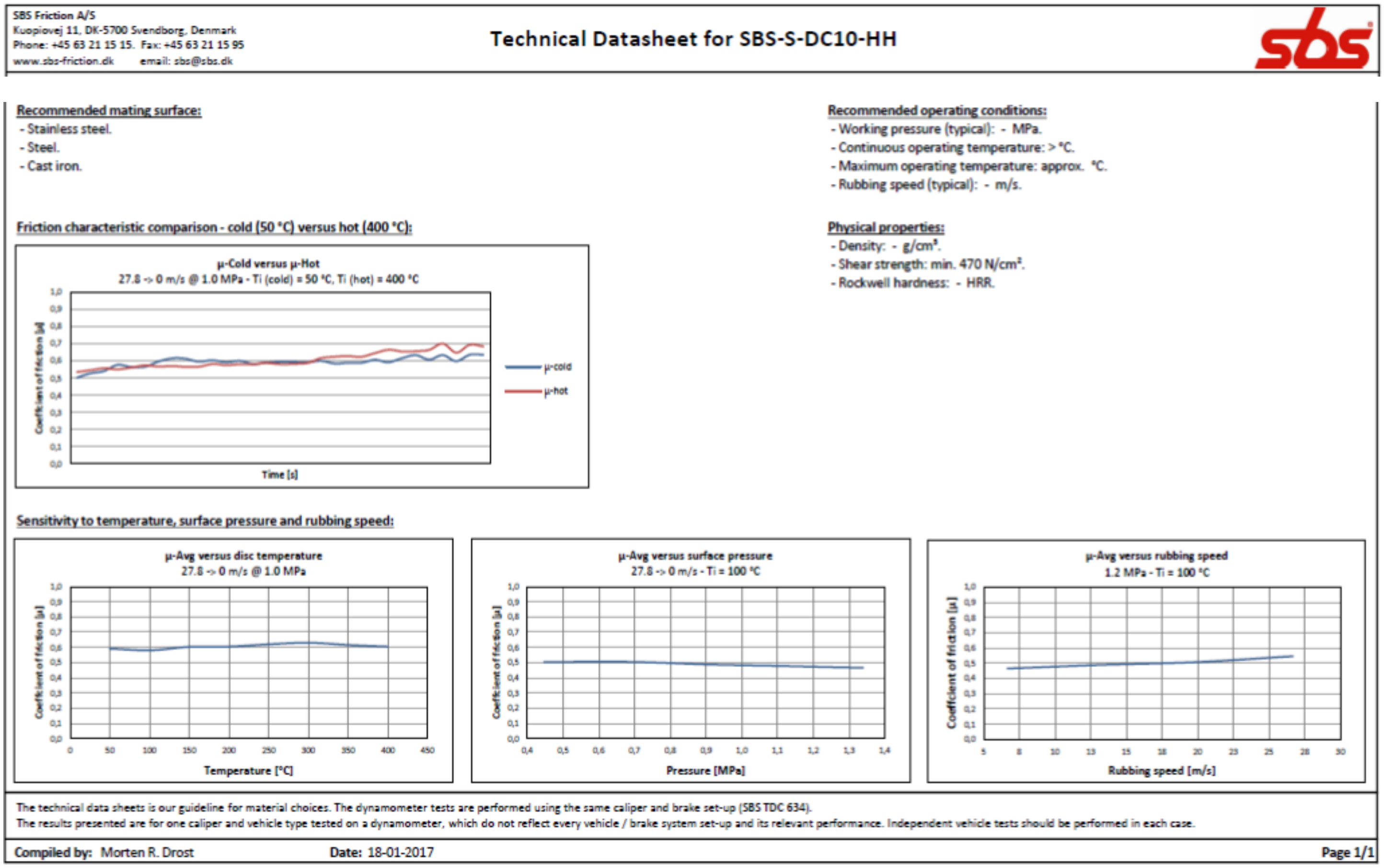
Mark Muddiman
Rules Rep since 2012

Nicholas,
In addition to meeting T.3.1.2 and T.3.1.13, any failure of the pedal face must not injure a driver's foot when applying 2000N.

Figure A-1: Inquiry number 17111 submitted 09/02/2021.

Appendix B

SBS Dual Carbon Brake Pads Technical Datasheet


SBS Friction A/S
Kuopiovej 11, DK-5700 Svendborg, Denmark
Phone: +45 63 21 15 15. Fax: +45 63 21 15 95
www.sbs-friction.dk email: sbs@sbs.dk

Technical Datasheet for SBS-S-DC10-HH

Recommended mating surface:
- Stainless steel.
- Steel.
- Cast iron.

Recommended operating conditions:
- Working pressure (typical): - MPa.
- Continuous operating temperature: > °C.
- Maximum operating temperature: approx. °C.
- Rubbing speed (typical): - m/s.

Friction characteristic comparison - cold (50 °C) versus hot (400 °C):

Physical properties:
- Density: - g/cm³.
- Shear strength: min. 470 N/cm².
- Rockwell hardness: - HRR.

Sensitivity to temperature, surface pressure and rubbing speed:

The technical data sheets is our guideline for material choices. The dynamometer tests are performed using the same caliper and brake set-up (SBS TDC 634).
The results presented are for one caliper and vehicle type tested on a dynamometer, which do not reflect every vehicle / brake system set-up and its relevant performance. Independent vehicle tests should be performed in each case.

Compiled by: Morten R. Drost **Date:** 18-01-2017 **Page 1/1**

Figure B-1: Technical datasheet for dual carbon brake pad material provided by SBS.

Appendix C

Purchased Components

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BRAKES - SWEDEN

Home | SiteMap | Contact

Products
Price list
Catalogues
Order info
Gallery
About ISR

You are here: Home / Products / Calipers / 22-049

22-049 2-piston caliper

<< Back

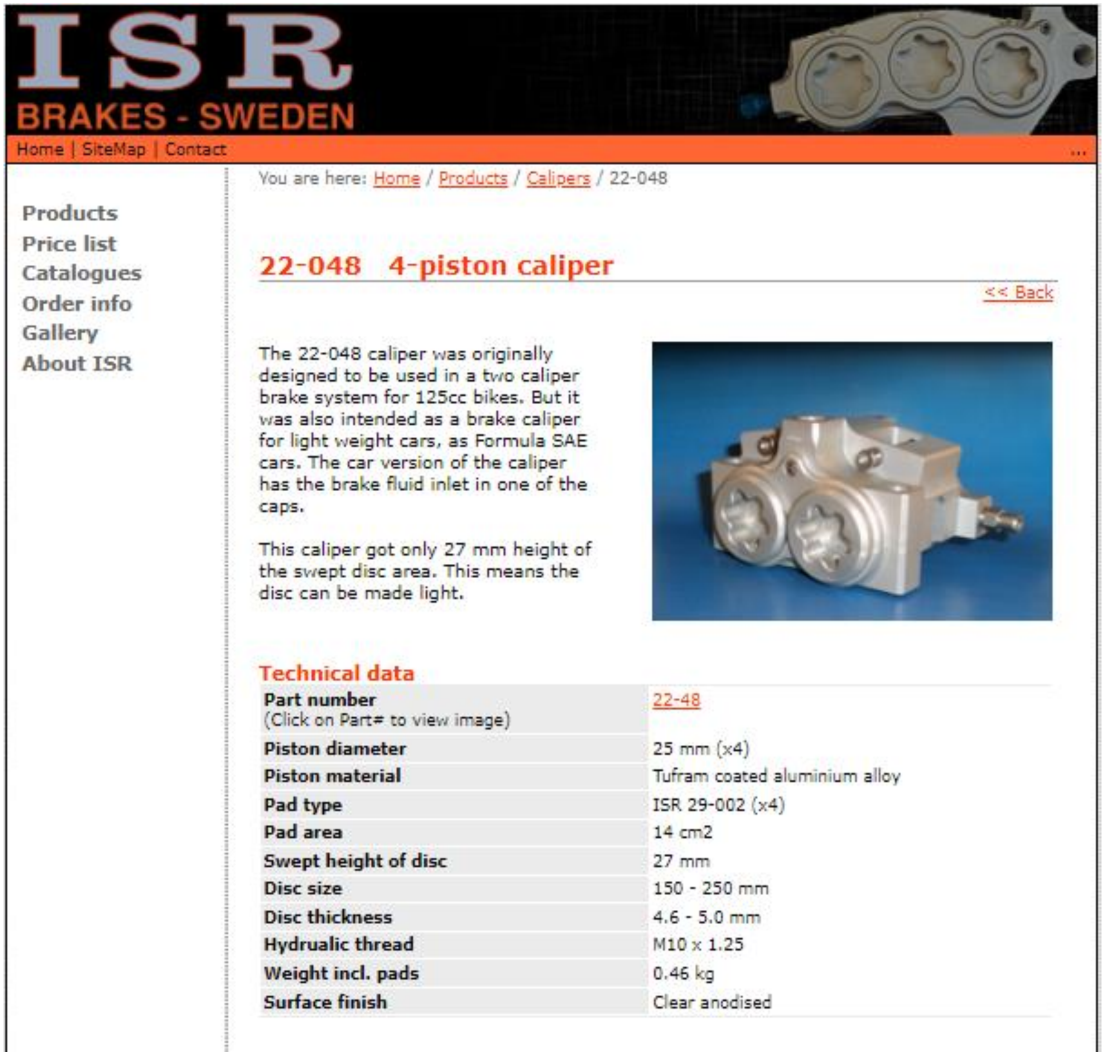
The 22-049 caliper was originally designed to be used for Motocross bikes but the caliper is also suitable for all kinds of lighter bikes and cars . It can be used in a one or two caliper brake system.

This caliper has only got a 27 mm height of the swept disc area. This means the disc can be made light. It also means that the brake disc gets a clean "low profile" design.

Technical data

Part number (Click on Part# to view image)	22-049-OA (RH) 22-049-OB (LH) 22-049-OC (RH car) 22-049-OD (LH car)
Piston diameter	25 mm (x2)
Piston material	Tufram coated aluminium alloy
Pad type	ISR 29-002 (x2)
Pad area	7 cm2
Swept height of disc	27 mm
Disc size	150 - 250 mm
Disc thickness	4.6 - 5.0 mm
Hydrualic threads	M10 x 1.25
Weight incl. pads	0.29kg
Surface finish	Clear anodised

Figure C-1: ISR 22-049 two-piston caliper used in the rear.



ISR
BRAKES - SWEDEN

Home | SiteMap | Contact

You are here: Home / Products / Calipers / 22-048

- Products
- Price list
- Catalogues
- Order info
- Gallery
- About ISR

22-048 4-piston caliper

<< Back

The 22-048 caliper was originally designed to be used in a two caliper brake system for 125cc bikes. But it was also intended as a brake caliper for light weight cars, as Formula SAE cars. The car version of the caliper has the brake fluid inlet in one of the caps.

This caliper got only 27 mm height of the swept disc area. This means the disc can be made light.

Technical data

Part number (Click on Part# to view image)	22-48
Piston diameter	25 mm (x4)
Piston material	Tufram coated aluminium alloy
Pad type	ISR 29-002 (x4)
Pad area	14 cm2
Swept height of disc	27 mm
Disc size	150 - 250 mm
Disc thickness	4.6 - 5.0 mm
Hydrualic thread	M10 x 1.25
Weight incl. pads	0.46 kg
Surface finish	Clear anodised

Figure C-2: ISR 22-048 four-piston caliper used in the front.

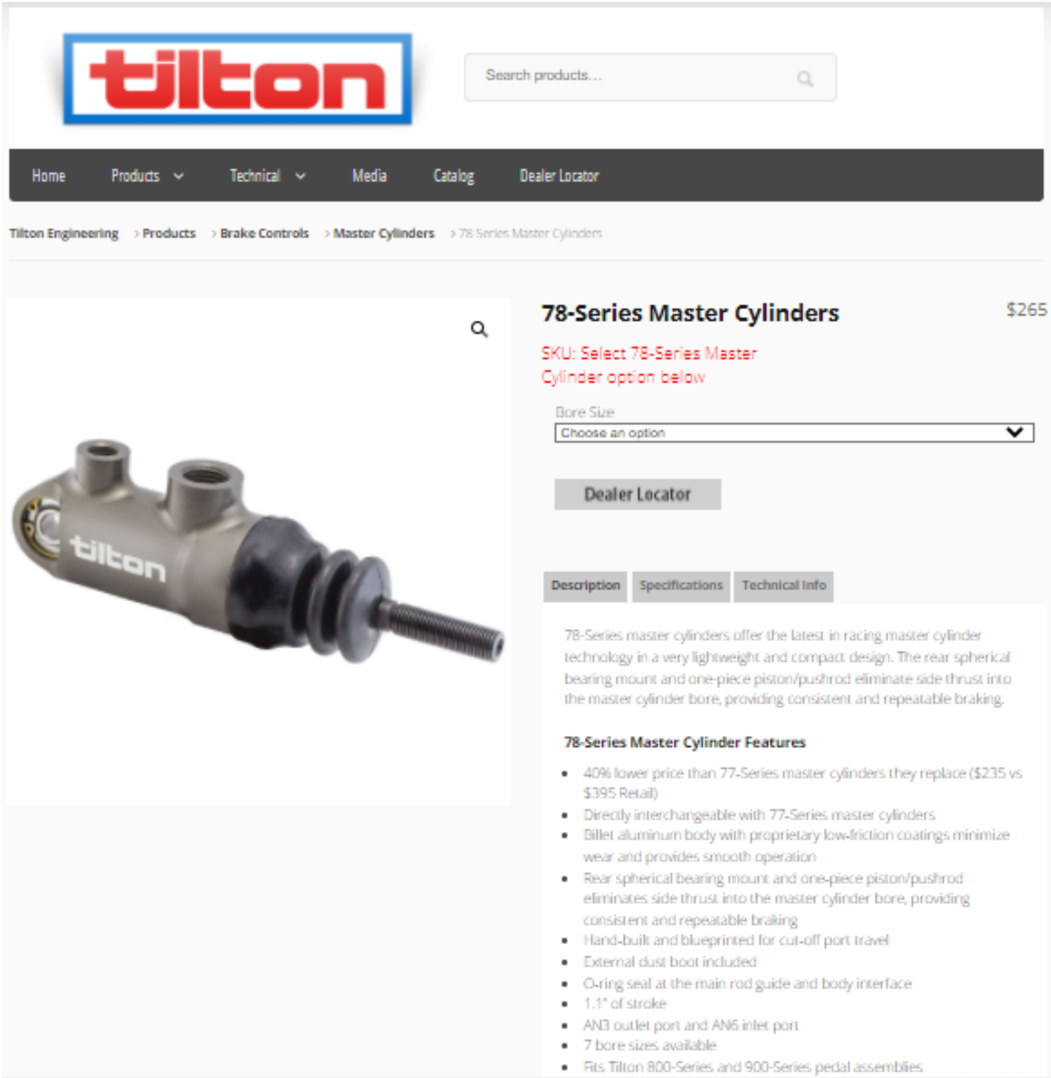


Figure C-3: Tilton 78 series master cylinders.

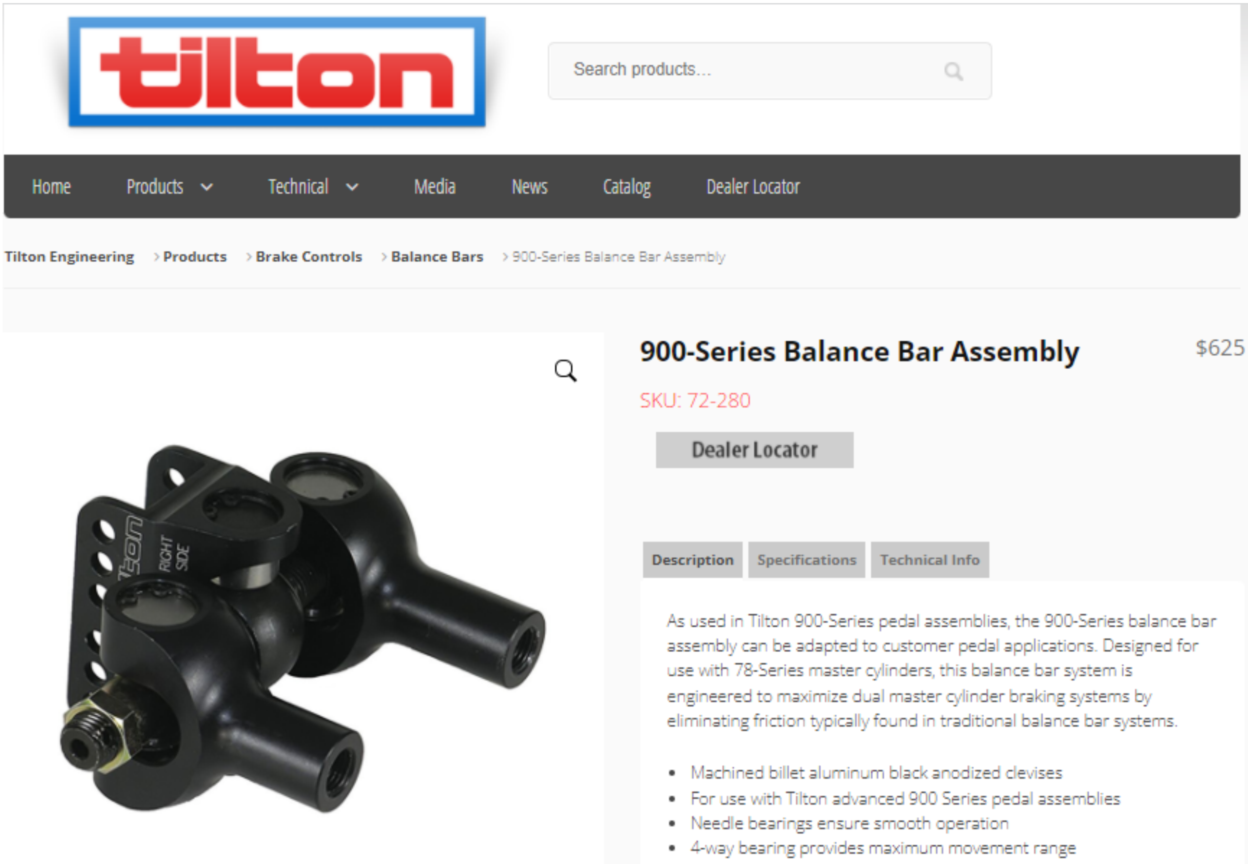


Figure C-4: 900 series balance bar from Tilton used to adjust master cylinder bias.

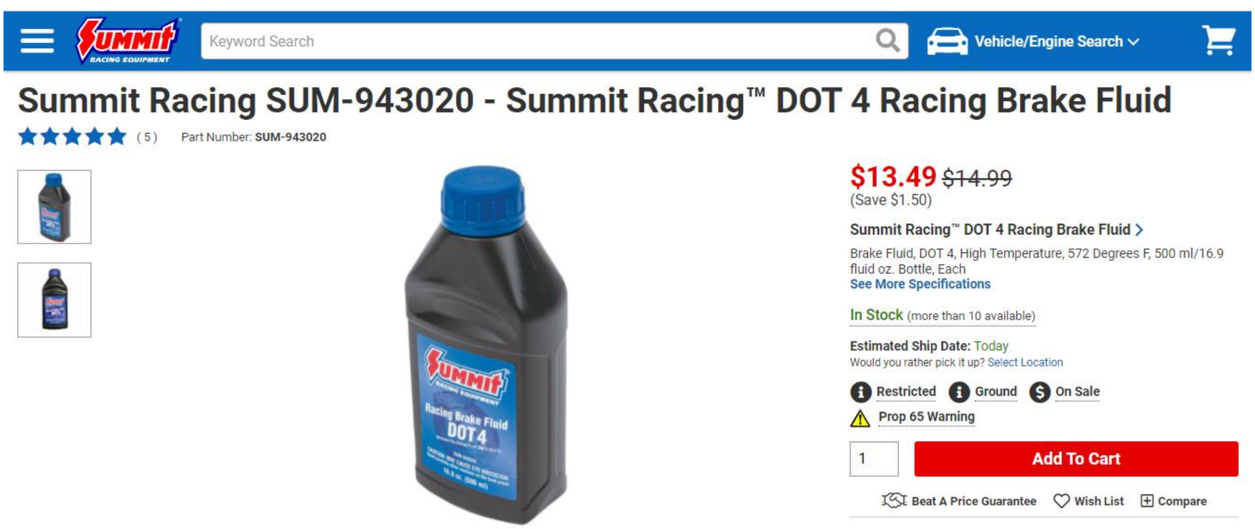


Figure C-5: DOT 4 brake fluid.

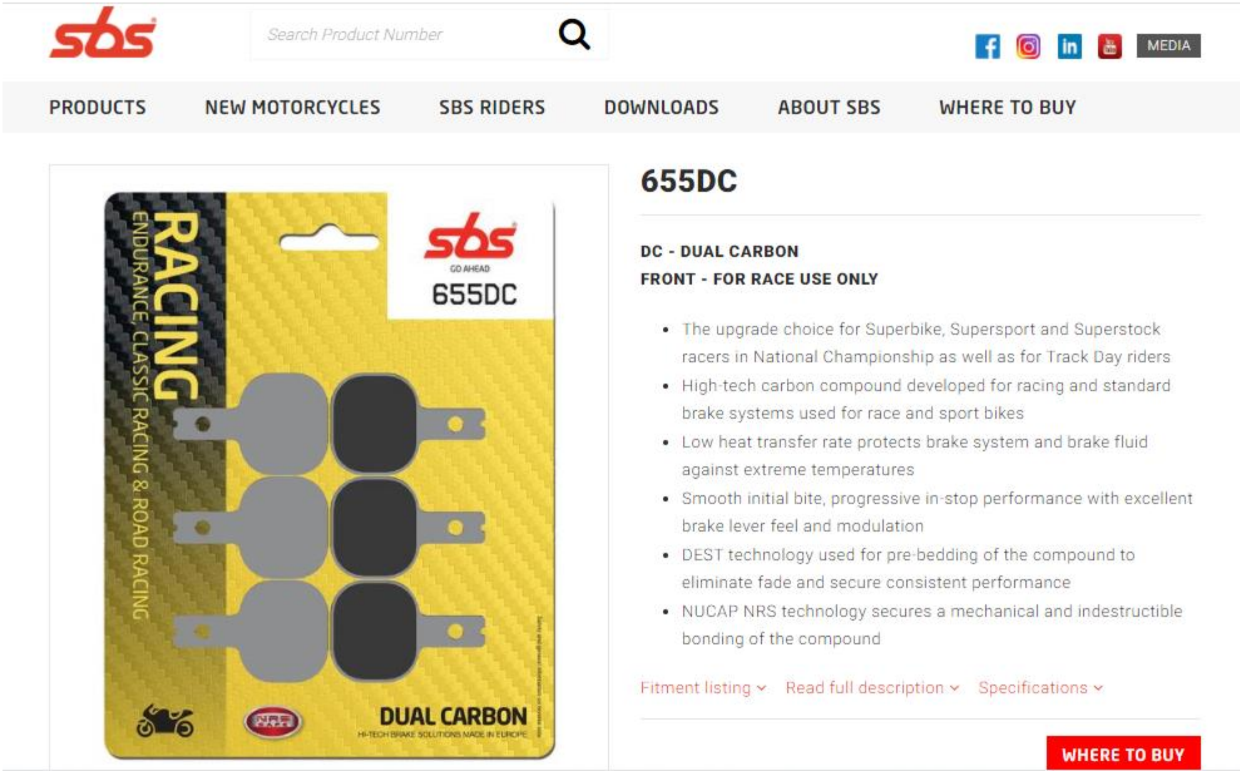


Figure C-6: SBS 655DC dual carbon brake pads.

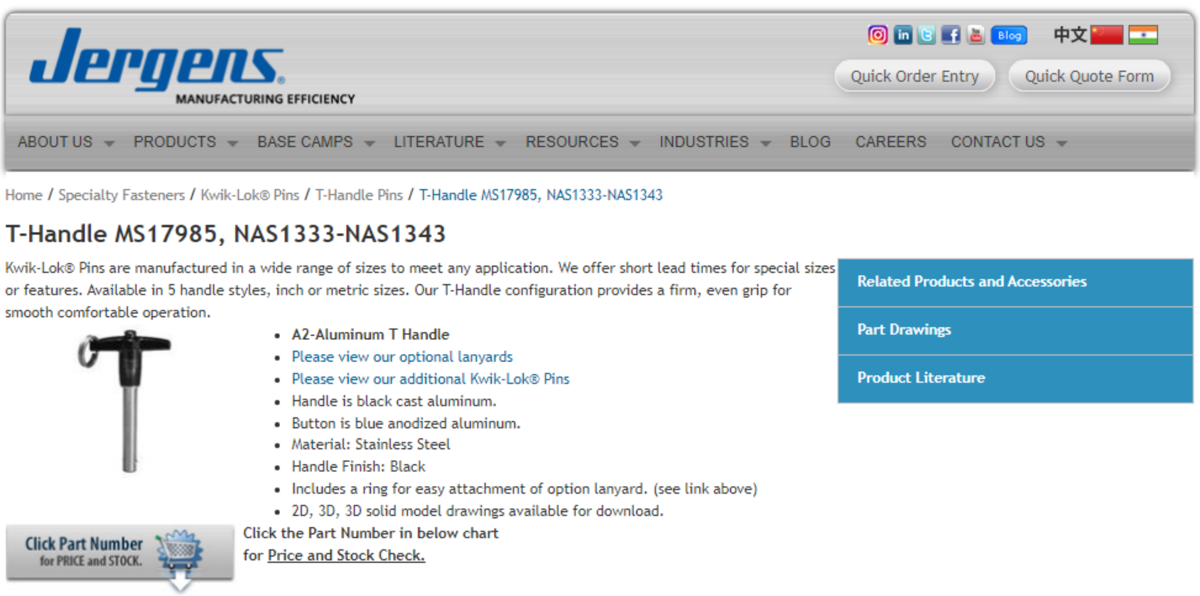


Figure C-7: Jergens T-Handle Kwik-Lok Pin.