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Composite Brake Rotor Assembly by Utilizing Replaceable Friction Surfaces

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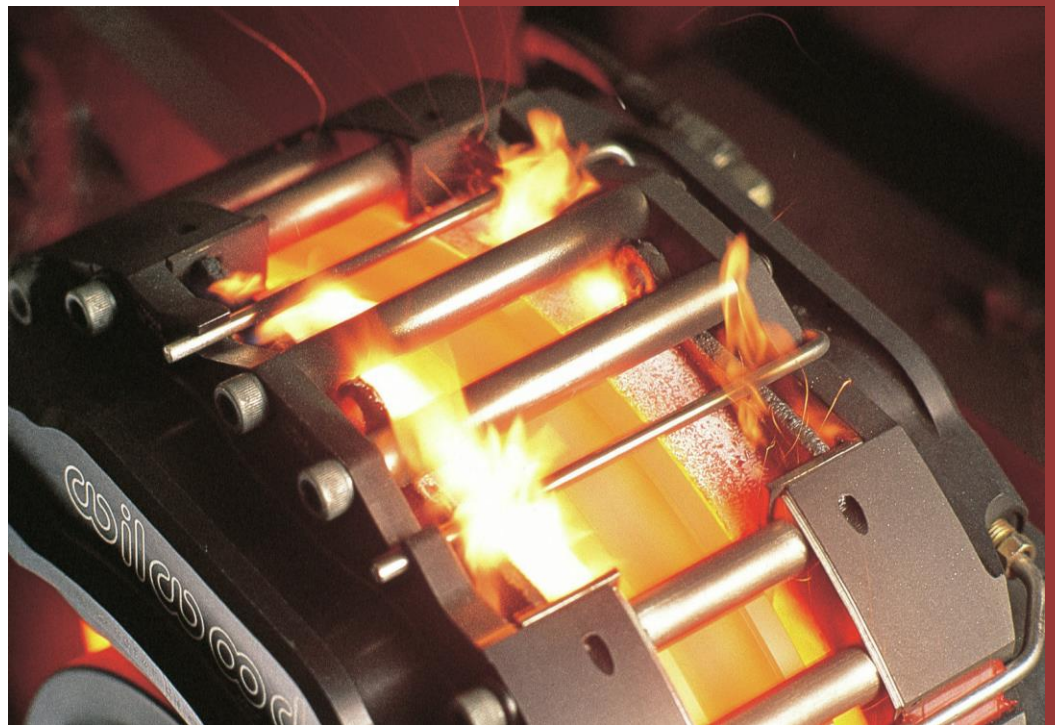
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COMPOSITE BRAKE ROTOR ASSEMBLY BY UTILIZING REPLACEABLE FRICTION SURFACES



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Mechanical Engineering Technology

6/2/2015

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ABSTRACT

This project investigated a proof of concept design involving a rotor fabricated from aluminum with replaceable friction surfaces with greater or equal performance characteristics in order to reduce cost and maintenance. The replaceable friction surfaces provide a means to mitigate cost to the end user. The structure is constrained by the dimensions, 11.75" diameter and 1.25" width and serves as a direct replacement rotor for a circle track racecar. Analyses provide a direct comparison in static mass, moments of inertia, and forced convection thermal calculations in order to determine if the concept was viable. Requirements for a successful design were a 22% reduction in total rotating mass, resist a linear deceleration rate of 8 meters per second, and the centripetal forces of an angular velocity of 315 radians per second. Off-car testing revealed a 4 pound reduction in static rotor mass and achieved a 34% reduction in the moment of inertia. On-vehicle testing involved data logging multiple laps at a local racetrack. The concept rotor assembly displayed a higher theoretical peak than the conventional design. In the composite structure the heat was rejected earlier in the cool down phase of the lap resulting in higher steady state of absorption/radiation characteristics. Means of monitoring the performance are by way of a GPS accelerometer and remote mounted infrared sensors mounted to each hub. This design offers the all the function of a conventional rotor with a 42% reduction in replacement cost and 18% reduction in replacement time.

INTRODUCTION

ENGINEERING PROBLEM

Current brake rotor metallurgy has only two paths; grey iron, carbon ceramic, and derivations of each respective material; designs consist of a one or two-piece rotor that once the friction surface has reached a minimum thickness threshold it is discarded and replaced with an all new casting. A similar relationship exists between the brake rotor and a pneumatic tire; once the tread has worn down, the tire is replaced with a new unit, leaving the “worn” unit with approximately 90% of the overall structure intact. This project was motivated by a need for an alternative system that consists of a rotor structure with replaceable friction surfaces that is inexpensive manufacture and maintain while both lighter in rotating mass as well as static mass.

Because of grey iron’s high material density, $\rho = 7196 \text{ kg/m}^3$, a typical brake rotor mass can be as much as 10-20 kg with most of that mass concentrated along the outside diameter furthest from the point of rotation. Potential energy, e.g. combustible fuel, is wasted overcoming the moment of inertia during acceleration and consequently extra brake pedal effort overcoming the flywheel effect during deceleration. Rust is also prone in areas that are not the friction surface. The internal venting channels proximal to the rotor faces degrade the effectiveness of heat dissipation through inhibiting centrifugal convection currents whereby the internal rotor temperature rises and reducing pad friction efficacy.

A proposed structure of 6061-T6 aluminum, $\rho = 2712 \text{ kg/m}^3$, or similar material, for the rotor body with replaceable faces of a high friction ferrous material shall serve as the composite rotor structure. It is by this means that a lighter mass unit with similar or greater braking characteristics can be quantified.

MOTIVATION

With skill in vehicle fabrication and a familiarity with oval, drag, and road course racing, this project is of great interest. If a lighter mass system is achievable while maintaining similar, or improved, braking characteristics there is more energy available to accelerate and therefore the amount of fuel consumed per lap is diminished. The car is less susceptible to reactionary and transient forces caused by steering input, wide-ranging track conditions and driver error. The ideal design is a perfect retrofit into existing braking technology, lightweight and inexpensive to replace since it is the surfaces and not the whole rotor that is replaced.

The rotor design has two paramount concerns; weight and strength. Multiple materials such as: high strength aluminum, low carbon steels, and as cast metal composites are under consideration. Should the project progress into an evolutionary development phase, carbon ceramics are a consideration as well. All of the materials can potentially work, but the question is will they work within the design requirements? The final design criterion is the cost to manufacture must not be so expensive that higher performing materials, i.e. carbon ceramics, are advantageous. Using the stated materials may require the part to be larger so that it can stand up to the design requirements which may result in the part being too heavy or not fit within the constraints set forth by the design criteria.

FUNCTION STATEMENT

Rotor design must be a sufficient reduction in rotating and static mass while providing similar braking characteristics to that of a conventional metallic assembly.

REQUIREMENTS

In order for the brake rotor to be successful it must be a single unit of complementary components that serve two purposes: provide the car a means of converting frictional force into deceleration, and be easily replaceable. Keeping the design simple and choosing proper materials will lead to a strong yet lightweight product that with a relatively inexpensive consumable. Design criteria fall into three major categories of interest:

Dimensionality

- Design is constrained so that it must function as replacement for a conventional circle track racing car design of 11.75" x 1.25" rotor sourced from an early 1970's Chevrolet Impala, 3rd generation Corvette or 2wd full-size pickup of that era.
- Rotor body must mount to a dedicated hub bearing carrier assembly with eight 5/16" fasteners on a 7-inch bolt circle. This hub is a components-off-the-shelf assembly utilizing 2.0-inch inside diameter tapered roller bearings and serves as the conventional means for which the rotor is mounted by design to modern racecar.

Off-car testing

- Removal of 20% of the static mass from the assembly.
- Confirm an existing manufacturer's claim of 34% reduction in the moment of inertia in a similar product.

On-car testing

- Rotor assembly must be able to resist the forces created by an angular velocity of 315 rad/sec.
- Rotor must be able to dissipate the heat generated from 125 kilowatts of kinetic energy if mounted on the front axle of the racing car.
- Rotor must be able to resist the torque provided by the linear deceleration rate of 8 m/s^2
- Provide an experimental means to confirm the assumed forced convection constant.

ENGINEERING MERIT

In order to design the brake rotor to fit the stated design requirements several equations will be used. Equilibrium equations are necessary to determine resultant forces and moments about the X and Y axes. Determinants of the thermodynamic properties such as coefficient of expansion, theoretical temperature increase, and theoretical rate of thermal dissipation are all necessary to provide a baseline in the selection of the proposed materials. The dimensions are limited by the design constraints set forth by the conventional design. However, there is design latitude in part thicknesses considering dissimilar materials are being used so provided overall width, outside diameter, and rotor mounting bolt pattern limitations are observed. Direct shear stress, $\tau_{\text{avg}}=V/A$, is necessary to determine the diameter and number of fasteners that will attach the friction surfaces to the rotor body. The fastener diameter, number of fasteners, and bolt circle dimensions for mounting the rotor body to the hub is established by the manufacturer of the hub unit. However the width of the body material surrounding the fastener is subject to analysis since

the mechanism is in single shear and will place pressure along a semicircular area of the fastener length in the Y-axis. Finally, normal stress $\sigma=F/A$, is necessary to determine the amount of clamping force is present in the braking moment.

SCOPE OF EFFORT

The scope of the project will involve the mechanical components of a vehicle's hydraulic braking system. The evaluation is only of the mechanical aspect of a hydraulic braking system. The caliper, pads, rotor carrier, and bearings have already been produced by manufacturers and are not subject to evaluation. Due to a multitude of different concepts involving the braking system's friction surface, slight alterations may be made to the rotor body and friction surface dimensions in order to work within the constraints set forth by the initial design criteria.

SUCCESS

The objective of this project is to design a lighter rotor that maintains comparable structural performance to that of commercial rotors currently available. The success criteria are a direct result of the design requirements. Thus, for the brake rotor to be one hundred percent successful, it must meet all of the standards set in the success criteria listed below. Answers requiring numerical test data values will be included along with a pass/fail listing and an explanation supplement.

- Removal of 35% of the total rotating mass.
- Maintain similar, or improved, braking characteristics than that of a conventional design.
- Concentrate on maintaining rigidity around the central pad contact area
- Friction surfaces must be replaceable in order to reduce cost of wear components
- Corrosion resistance of the friction surface will be advantageous to later generation automobiles with regenerative braking as a safety device in emergency braking incidents.
- Subsequent designs may employ variations in friction surface materials to improve frictional and thermal braking characteristics.

However, for the device to be a functional part, it must be able to withstand the forces generated by 4000 revolutions per minute and have the necessary thermodynamic properties to dissipate 125 kilowatts of energy in the form of radiant heat and the integrity to withstand a moment about the hub center axis of which 1300 Newton-Meters are applied.

DESIGN & ANALYSIS

APPROACH

The proof of concept design was developed from observations into current off-the-shelf brake metallurgy; cast iron, carbon composite and derivations and a means to qualify material characteristics.

Cast iron is by far the most common material used for automotive rotors for a number of reasons:

- It has excellent strength at high temperatures.
- Does not warp after severe thermal cycling.
- Inexpensive to manufacture.

However, there are also design limitations to consider:

- During heavy and/or rapid thermal cycling, iron rotors have shown tendencies to fatigue crack in the friction surface areas.
- Use of non-asbestos pad materials aggravates the fatigue tendency.
 - Non-asbestos pad materials have considerably greater coefficients of friction and greater temperature fade points when compared to their asbestos-laden predecessors. [1]

Carbon rotors address many of the shortcomings of traditional grey iron and steel rotors. They are both lightweight, 40-50% lighter is typical, as well as providing superior braking performance at elevated temperatures under track conditions. Typical coefficient of friction is 0.5-0.8, whereas iron is ~0.34-0.42. Carbon composite rotors, though technologically superior to iron, are prohibitively expensive, typically \$1000-3000 per rotor [3], due to the manufacturing process from which they are derived thereby limiting their application to exotic sports cars and less budget minded race teams.

Low-carbon steel rotors are widely used in racing applications. Made primarily from SAE 1080, as well as other proprietary alloys, the cast steel rotor provides excellent strength characteristics and is resistant to cracking. Conversely, under extreme temperature cycling steel rotors will experience some form of warp and/or shrink. Because the steel is the most widely used alternate material to iron, this material shall be the basis for the friction surfaces.

PROPOSED SOLUTION

This project is designed as a system and will address the manufacturing cost of the rotor as a consumable. By substantially reducing the amount of material necessary to replace when maintenance is required the cost of consumables is reduced. Two goals exist for this project; melding two design avenues into a hybrid design that performs satisfactorily in friction and heat dissipation testing requirements set forth by Society of Automotive Engineering tests SAE J2522 (Inertia Dynamometer Test Procedure), ISO 26867 (Friction Behavior for Automotive Brake Systems) and NTSHA FMVSS-135 (Light Vehicle Brake System Standard) yet is cost effective upgrade for the budget minded enthusiast.

DESCRIPTION

Conception of the current design began as an observation of an enthusiast oriented, flywheel/clutch assembly containing a replaceable friction surface on an aluminum flywheel as a means to reduce an inertial moment. By employing a ferrous material as the friction surface to interact with the clutch disc, a suitable lightweight material, aluminum, could be utilized as the primary energy storage facility and therefore a reduction in net parasitic losses from the power plant. In the event that maintenance is necessary, the friction surface is simply unbolted from the flywheel body and a replacement remounted. Further the momentum losses were realized by transferring the same philosophy to other rotating objects within the power train; hence the root endeavor. By reducing the moment of inertia about the brake rotor, thereby the tire/wheel/rotor assembly, the effect translates into an overall reduction in energy required to overcome the change in momentum during acceleration or transient events.

Seen in figures 1 and 2 are subsequent modeled examples of the proposed design consisting of two ferrous material faces mounted to an 6061-T6 Aluminum rotor body and

attached to an off the shelf rotor carrier. From the observed braking event, from 45 m/s to 22 m/s, a theoretical rise in temperature is calculated to be 273°C whereas the vented conventional design would be 212°C from ambient, a ΔT of +53°C (Appendix A6). The resultant change in temperature increase reduces the safety factor when considering future power plant or braking improvements or additional racetracks not yet considered.



Figure 1: Initial Rotor Design with Off-the-Shelf Rotor/Bearing Carrier

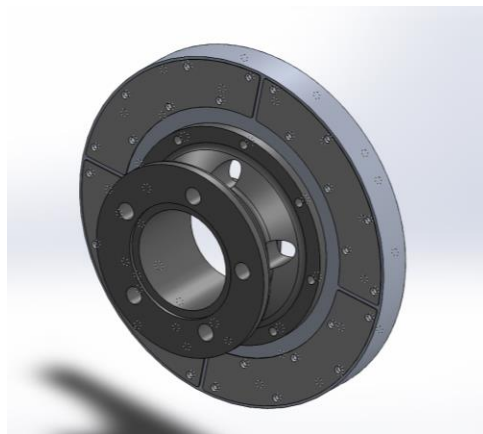


Figure 2: Addition of Segmented Friction Faces

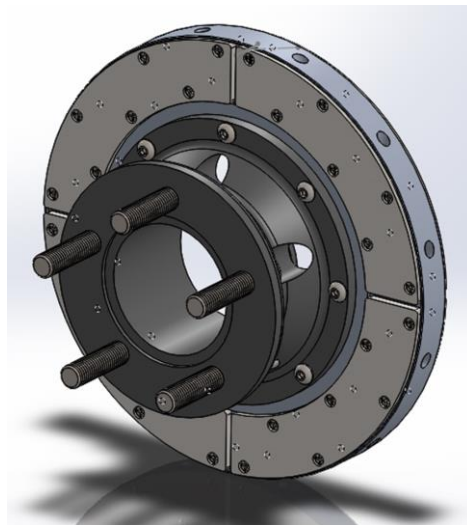


Figure 3: Forced Convection Venting

BENCHMARK

Figures 4 and 5 are examples of an in production aluminum-bodied rotor with a steel clad friction surface. The braking system uses a one-piece design rotor with non-removable friction surfaces cast in place and is meant for a domestic road-going automobile. By observation of the part finishes all non-friction surfaces are coated for corrosion resistance and any pad contact surfaces are machined. The radiating lines on the friction surface are for thermal expansion. The friction surfaces are not intended for replacement thus eliminating one tenant of the requirements set forth. Also note the lack of forced convection venting.

List of benchmark design claims [2]:

- 30% to 50% weight reduction
- Considerably better gas mileage up to 10%
- Faster heat dissipation and lower braking temperatures
- No heat dissipation degradation due to rusting
- Approximately 30% less wear on brake pads
- Faster car acceleration
- More precise steering due to un-sprung weight reduction

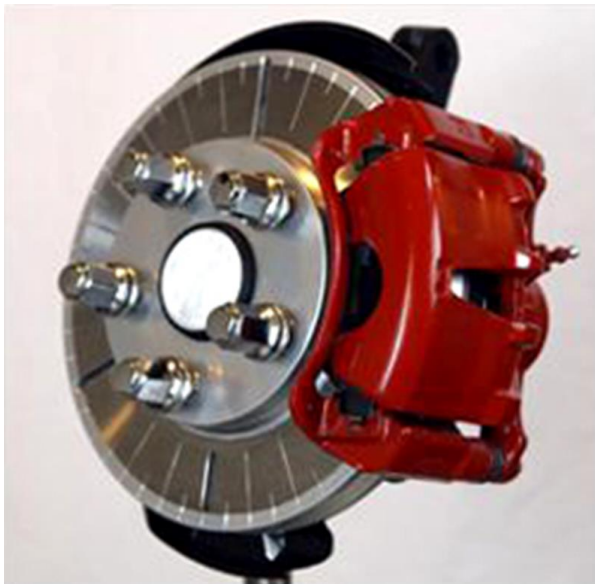


Figure 4: Image Courtesy of LiteBrake Tech, LLC

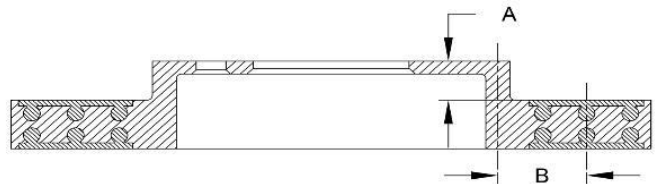


Figure 5: Cutaway Drawing Courtesy of LiteBrake Tech, LLC

PARAMETERS OF INTEREST

The vehicle for which the composite rotors are to be mounted to is a Northwest Series Limited Late Model class race car. There are no on board gauges or meters pertaining to speed or time. Only information the driver has on hand pertains to the engine state of tune and a radio headset to his/her crew relaying total lap times. Therefore any information attained pertaining to the entry/exit and time in the braking zone is simply an assumption and subject to a “fish story” of some sort. Therefore a GPS-based accelerometer will be employed to determine the information necessary to validate the initial assumptions and if the initial thermal energy calculations are correct qualifying the concept as achievable. This in turn will quantify a forced

convection coefficient and determine if indeed for the composite design is equal, or has a competitive advantage, to the conventional iron/steel brake rotor.

Secondary to the assumption questions are the results of the mass reduction and decrease in the moment of inertia. Based on the GPS data, was there a decrease in the on-track braking points? Was the car able to accelerate/decelerate quicker than before?

Finally braking surface temperature peak is of paramount concern. With the composite structure, the point at which solid aluminum becomes a liquid, 675°C, shall be avoided at all costs. For an added margin of safety, if the data indicates a temperature of 600°C, any further testing is aborted until such a temperature can be avoided.

PERFORMANCE PREDICTIONS

These rotors must be a direct replacement for the conventional design and must meet or exceed the parameters set forth. The following predictions compare an internally vented grey iron rotor versus a solid aluminum rotor with steel friction surfaces affixed. Subsequent calculations will draw conclusions between the thermal characteristics of the two materials. It is the prediction that this device will be able to withstand the braking force of 1300 N-m and dissipate 125 kW of energy generated by that force. The device is also predicted to have 34% less rotational inertia, (Appendix A9 and A10), than the conventional design.

Initial calculations indicate a theoretical temperature differences of +53°C is valid, (Appendix A6). Further calculations and later independent testing will confirm whether repeated braking events, two per lap, overcomes the heat dissipation rate of aluminum and reaches the melt temperature, 677°C, of the rotor body. From the predictions, it is determined that this brake rotor concept should be kept to short track racing at light to moderate speeds or high speed circuits with minimal braking zones.

DESCRIPTION OF ANALYSIS

Initial calculations are produced to gain perspective on the parameters regarding the braking event and to gather information for a direct comparison between grey iron and aluminum/steel lumped mass. To simplify, the analysis will be broken down into three categories:

- Mechanical
- Thermodynamic Analysis
- Cost Analysis

SCOPE OF TESTING AND EVALUATION

Scope of testing is limited to the evaluation of un-sprung mass and qualification that the aluminum/hybrid unit preforms as well, or better, than grey iron in temperature rise and thermal dissipation testing through independent test methods. An outside vendor has been selected to perform OEM-level testing once the unit is completed.

ANALYSIS

The sanctioning body rules dictate that the perimeter tube frame chassis car must not weigh more than 2900lbs. Tires are restricted to that of the current NASCAR specification of 28” diameter with a width of 12” and mounted to a 15”x10” steel rim. Rotor dimensions are restricted to 11.75” x 1.25” as noted in the constraints set forth in the proposal introduction.

Entry and Exit speeds as well as time for braking event are all on-track observations. Data logging and telemetry for events are strictly prohibited except for testing events. Instead tachometers with memory features indicate highest RPM achieved. Conversations with drivers as well as collection of RPM, transmission and final drive gear ratios data indicate that the average entry speed of the cars is approximately 100 mph with a fast lap completed in 18-20 seconds. Braking zones are completed in 3-3.5 seconds. It should be noted that current maximum velocities of Pacific Northwest race tracks are approximately 105 mph, 47 m/s. For vehicles to reach 120 mph, 54 m/s, maximum speed, approximately a 37% increase in power is required. Given that future power increases as well as improvements in braking efficacy, it is imperative that the entry speed is moved to 120 mph as an additional safety factor for all thermal and force calculations.

Historically, racing has strived for speed for the sake of reliability with safety factors given the barest minimum. Though the braking system may overheat, it is vitally important that the structural integrity of the braking system must not fail under the given racing conditions. Therefore, the fasteners holding the friction faces to the rotor as well as the rotor to the hub must be able to withstand the radial forces generated at 4000 RPM, 336mph, for a Safety Factor of 2.5.

MECHANICAL

Vehicle Data	Northwest Series Limited Late Model Class
Mass of vehicle, M	1315 kg
Diameter of Wheel/Tire, r_w	0.71 m
Diameter of Brake Disc, r_d	0.30 m
Initial Velocity, V_i	44.7 m/s
Final Velocity, V_f	22.4 m/s
Braking time, T	3 sec
Calculated Deceleration Rate, A_c	-7.4 m/s ²

Basic elements are derived from the given constraints of the project. Deceleration rate, radial and tangential forces about the rotor, angular velocity, shear and normal stresses on fasteners are all calculable from the initial data. For practical purposes, all hydraulic work is performed, no mechanical deflection, and all other friction losses are negated. Deceleration rate of the car is assumed as a linear rate on level ground with no braking embankments. The mass of the vehicle multiplied by the deceleration rate determines the total force required in a single braking event. It is assumed that this force is constant throughout.

$$v_f = v_i + at \rightarrow a = \frac{v_f - v_i}{t} = -7.43 \text{ m/s}^2$$

$$F_{Total} = ma \rightarrow 1315 \text{ kg} \left(-7.43 \text{ m/s}^2 \right) = -9.77 \text{ kN}$$

Determining the actual rotational speed of the rotor assembly was found given the circumference of the tire and converting to a linear rate of travel per revolution before

multiplying by the rate of travel. At 45 m/s, 100 mph, the wheel/tire/rotor assembly is spinning at 1200 RPM (Appendix A1) before entering the braking zone.

The rotor serves as the primary heat sink in the braking system; it is the functional responsibility of the rotor to generate a retarding torque as a function of the brake pad frictional force. Torque is applied to the brake rotor from the force $F = ma$, then that force is considered about the wheel/tire radius arm. $Tq_{rotor} = F_{car}R_{wheel}$. For the sake of simplicity, the rotor is mechanically coupled to the hub and wheel assembly. Because the tire is assumed to be rigidly attached to the wheel, the torque will be constant throughout the entire rotating assembly, $Tq_{tire} = Tq_{wheel} = Tq_{rotor}$.

$$Tq_{rotor} = F_{car}R_{wheel} \rightarrow -9.77kN \left(\frac{0.71m}{2} \right) = 3.47kN * m$$

During the braking event it is assumed that the front brakes will distribute 75% of the total force required to slow the car. Observations indicate a brake bias distribution range between 80/20 to 70/30 brake bias for the front depending on driver “feel.” For the initial calculations, force distribution per rotor is assumed as 75/25 with the resulting torque distribution as such:

$$Tq_{front\ rotor} = \frac{3.47kN*m}{2} (.75) = 1.30kN * m$$

$$Tq_{rear\ rotor} = \frac{3.47kN*m}{2} (.25) = 0.43kN * m$$

As the probability of disintegration is high, it is imperative to determine the radial and tangential forces acting on the rotor and its affixing fasteners for the friction surfaces to the rotor body as well as the rotor body to the hub in order to consider an appropriate safety factor given the maximum speed. All fasteners for attachment are in single shear mechanisms. The fasteners attaching rotor body to the hub serve as the only means of resisting the torque placed about the rotor, therefore the forces generated at the attachment points are assumed as $Tq_{tire} = Tq_{wheel} = Tq_{rotor}$. The diameter and number of fasteners attaching the rotor body to the hat is specified by the manufacturer as an SAE 5/16-inch socket head cap screw. However, the shank length is subject to the material properties of the aluminum rotor. So as not to deform the mating hole in the rotor body, the contact stress must be the same as the cast iron unit.

$$W_{T_{Front}} = Tq / R_{wheel} = \frac{1.30kN*m}{\frac{0.71m}{2}} = 3662N$$

$$W_{T_{Rear}} = Tq / R_{wheel} = \frac{0.43kN*m}{\frac{0.71m}{2}} = 1211N$$

Direct shear for each fastener on the hub is then found and a safety factor is given to determine if the fasteners are within the shear allowable. Determining the stresses found in the rear is inconsequential at this point since all numbers are roughly 1/3 of what the front stresses are. Therefore the front axle placement is the basis for direct comparison to the iron rotor.

$$\tau = \frac{F}{A} = \frac{\frac{3662N}{8}}{3.381e-5 m^2} = 13.5MPa$$

Shear and stress on fasteners attaching friction plates are the only means of attachment of the plates to the rotor body. Given the rotor is assumed to have a peak torque placed about the rotational axis, it is imperative that the fasteners attaching the faces must not shear. Using the same equations as the hub to rotor body, the shear found on each of the 10-24 fasteners was found to be:

$$\tau = \frac{F}{A} = \frac{\frac{3662N}{8}}{1.129e-5 m^2} = 46.3MPa$$

Even with a safety factor of 5, a hardened 82° countersunk screw was found acceptable for the purpose of attaching the plates to the rotor body. Observing the same forces when placed on the eight 5/16” fasteners that attach the rotor to the hub, the safety factor is 12. Therefore the fasteners for mounting are not the limiting factor given the on-track conditions.

THERMODYNAMIC ANALYSIS

Properties of Materials	6061-T6 Aluminum	SAE G3000 Cast Iron
Heat conductivity, λ	166.6 W/m·K	43.4 W/m·K
Density, ρ	2712 kg/m ³	7196 kg/m ³
Specific heat capacity, c_p	897 J/kg·K	449 J/kg·K
Modulus of Elasticity, E	68.9 MPa	96.5 MPa
Poisson's number, ν	0.33	0.294

The braking system exists to convert the car's momentum into thermal energy by pressing a brake pad into the rotating rotor surface creating a moment about the brake rotor. The rotor functions as heat sink by storing heat energy during a relatively low duty cycle braking event and dissipating it to the surrounding air over a given period of time. By using the kinetic energy of the car between the braking zone entry and exit points determines the energy, in Watts, that must be absorbed and dissipated twice per lap.

$$KE_{Total} = \frac{1}{2}mv^2 \rightarrow 984 kJ$$

In the same distribution method as used to determine the torque applied per rotor is used to find the kinetic energy and thus the thermal energy to be dissipated per rotor given the observed braking event. This energy must be dissipated twice per lap for a period of no less than 25 laps.

$$KE_{front} \frac{738 kJ}{2 rotors} = \frac{369 kJ}{3 seconds} = 123kW \text{ per rotor}$$

$$KE_{rear} \frac{246 kJ}{2 rotors} = \frac{123 kJ}{3 seconds} = 41kW \text{ per rotor}$$

Ideally, the kinetic energy produced by the braking event is completely absorbed by the braking system and an increase in temperature is observed. The temperature increase is based on the thermal mass and the specific heat of the material absorbing the energy. In theory, the solid aluminum rotor assembly will store more energy than the conventional grey iron unit that it replaces due to the biased tradeoff between the total volume and the material density; the iron

rotors volume is diminished because of venting channels perpendicular to the axis of rotation but the aluminum rotor is solid yet one-third the density of iron.

The grey iron rotor is assumed as a homogenous casting consisting of a single material with vented inner channels. Therefore the equation for determining the theoretical temperature rise can be applied as follows:

$$T_{Fe} = \frac{(1 - \theta) \left[mg(V_i^2 - V_f^2) \right]}{2 \left[2g\rho_{Fe}c_{Fe}v_{Fe} \right]}$$

$$T_{Fe_{front}} = 180^{\circ}\text{C}$$

$$T_{Fe_{rear}} = 60^{\circ}\text{C}$$

Calculations predict that the aluminum with steel friction surfaces will store more energy with less mass than the grey iron rotor. It must be noted that the difference in theoretical temperature increase lends itself to the vented iron rotor's reduced volume when compared to the "lumped mass" of the solid aluminum with steel segmented plates. It is assumed that the tradeoff for reduced rotating mass will outweigh the penalty that is the increase in temperature.

$$T_{Al+st} = \frac{(1 - \theta) \left[\frac{mg(V_i^2 - V_f^2)}{2g(\rho_{Al}c_{Al}v_{Al} + \rho_{st}c_{st}v_{st})} \right]}$$

$$T_{Fe_{front}} = 233^{\circ}\text{C}$$

$$T_{Fe_{rear}} = 78^{\circ}\text{C}$$

Heat dissipation and convection coefficient calculations determine the performance of a brake system with the prediction of the brake surface temperature. To gain a safe braking system performance, the brake must be sufficiently designed to be able to dissipate the heat generated from the braking process adequately, so that the brake surface temperature is kept within the acceptable operating range for the brake material. Races are usually held during the evening hours. Therefore it is assumed that the ambient temperature will be 20-25°C when these rotors are tested. However it should be noted that daytime racing and testing do occur during daylight hours with ambient temperatures reaching 40°C.

Thermal analysis of cast iron to proposed composites currently uses a two lap event involving two braking zones per lap. Nodes were chosen at the rotor surface as well as the hub mating surface to gauge heat retention within the disc. Solidworks simulation assumes the material is in conduction/convection sequence with still air within proximity of the rotor assembly. Further CFD analysis is necessary to more accurately analyze the heat flow through the vented channels during a 7-second cool down period before being subjected to another 3-second braking event. However the current conclusion is that a late model circle track car is inherently reliant on the brake cooling ducts from the front of the vehicle to aid in cooling.

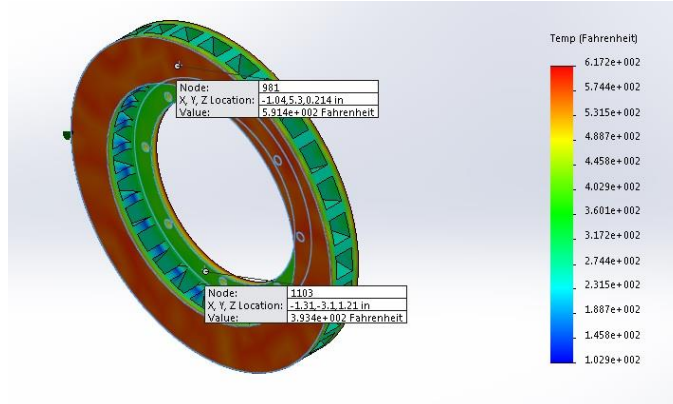


Figure 6: FEA Thermal Analysis indicating a surface temperature of 617°F

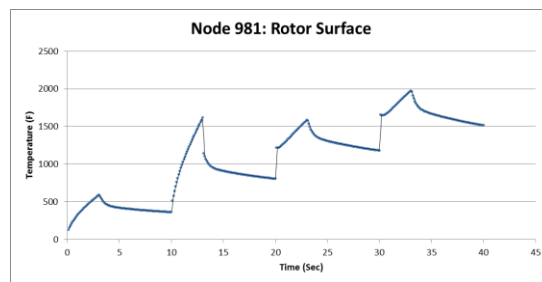


Figure 7: FEA Theoretical Temperature Spikes on Surface

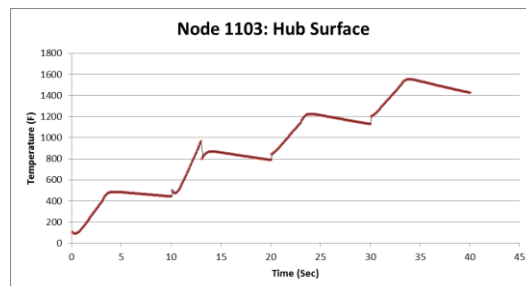


Figure 8: FEA Theoretical Temperature Spikes near Hub Mating Surface

LOAD PATHS

Thermal loads are often much more severe than mechanical loads and also much more difficult to predict accurately in a theoretical sense. Therefore experimental testing is paramount. The assumed load path is from the friction surface to the fasteners as well as the side plating machined into the rotor body. Load is then transferred from the rotor body into the fasteners affixing the rotor to the hub carrier and finally from the hub carrier to the wheel/tire assembly.

COST ANALYSIS

The root endeavor of the project was to reduce the cost of the elements requiring replacement during maintenance. By doing so, a root structure could be realized that is lighter and less costly overall to maintain versus a conventional rotor design.

DESIGN SEQUENCE

- Design 1 consists of a solid aluminum rotor with two circular disks, one per side, providing the friction surface. This idea is abandoned due to concerns of warping due to insufficient areas of clearance during the thermal event.
- Design 2 replaces the previous version with quartered segments for the friction surfaces.
- Design 3 reorients the segments into an angular slot for brake pad gas escape.
- Design 4 addresses internal temperature loads with the addition of forced convection holes

TOLERANCES

The tolerances of the device itself when assembled are not as critical as the tolerances of the individual components. The rotor body faces need to be straight and parallel to reduce run out and pedal pulsation the driver will feel. The friction face pockets need to be straight and parallel as well for the same reasoning. Fastener holes between the body and faces will need to be 0.025” oversize to allow for different temperature expansion rates between the dissimilar materials between ambient and a maximum temperature of 670°C.

TECHNICAL RISK ANALYSIS

The braking system is by far the most important safety aspect in the vehicle. With a braking failure the driver cannot maintain control and given the speeds attainable. The vehicle will veer off course, colorful words will be spoken by driver and crew alike and the structural integrity of the car’s framework will be severely compromised in an off track incident. Designing for eventual collision and impact is accomplished with forethought in chassis design; not brake design. However the risk to life and limb is still great for both driver and spectator alike in the event of a system calamity. To prevent loss of life, the safety factor will be increased to allow the vehicle to still be operational, but it will not be able to perform at its peak ability.

OPTIMIZATION

Thermal absorption and dissipative characteristics shall be the mitigating basis for the comparison. The reduction in rotating mass is greatly appealing. However it is the generation of a computer model that best describes the real world thermal event that is being optimized. By utilizing a known rotor material as the friction surface this will greatly aid in depicting a more accurate computer model. Once completed, it is the hope that this project will move further ahead with more radical friction surfaces.

METHODS & CONSTRUCTION

CONSTRUCTION

Design is constrained to the conventional means of mounting and must therefore consist of a rotor “hat” along with a rotor body and two friction surfaces. The surfaces are subject to later development as the project progresses. For prototyping, the friction surfaces will be water

jet cut from low carbon steel and surface ground to the specified thickness. It is the evolution of the motorsport that in the future more suitable materials shall be chosen to supplant the currently chosen friction material. If during the testing phase a more suitable material is found, it is the expectation that such a material will take the place of the low carbon steel. The rotor body is also subject to later development. However for the basis of this proposal, the body shall consist of 6061-T6 aluminum. The hat is a component-off-the-shelf unit and not subject to testing.

The rotor assembly will be built in sections. The rotor body will be CNC milled in house on university equipment. The friction face segments will be water jet cut via an off-site vendor, transported back, and finish ground in house using university equipment. Final assembly is in a nonspecific order. End result will mate eight segments to the rotor body and the rotor body to the rotor hat. Assembly of the finished product will occur at the designated university facility. Two rotor assemblies consist of the completed unit.

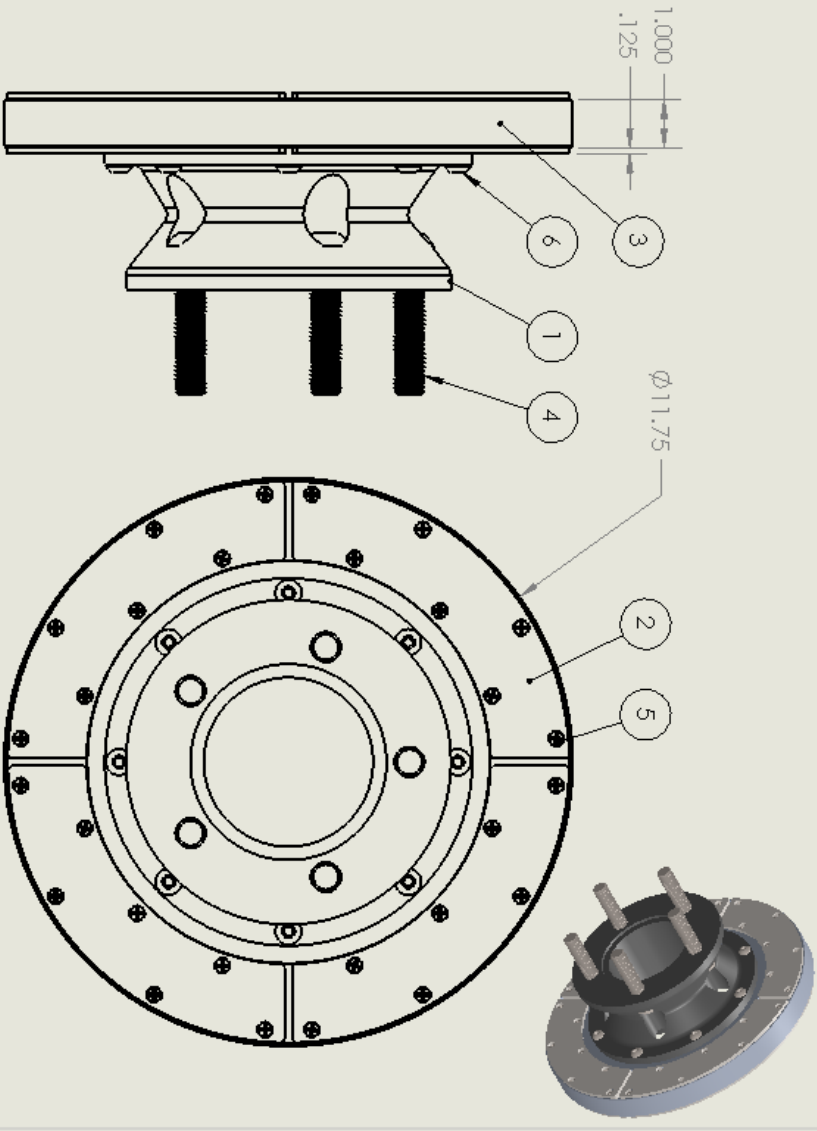
DESCRIPTION

A 24" x 12" x 1.25" block of aluminum will be cut into roughly equal 12" x 12" x 1.25" blocks on the horizontal band saw. The block will then be fastened into a CNC milling station where the first order of operations is to mill out the friction face pockets recesses and drill/tap one sides face mounting holes before pocketing the interior diameter, thereby setting the origin. Once that operation is complete, it will be flipped over, chucked in such a manner so as to relocate the origin and ensuring the material has been seated. Mounting holes for attaching the friction faces are open on the backside. Therefore the drilling and tapping can occur either in one operation or from both sides so as to uniformly locate the friction faces. Facing the material to the specified thickness is the beginning of the opposite face operations. Pocketing for the opposite side friction faces, drilling/tapping the mounting holes, and finally machining the outside diameter.

The friction faces will be array cut on a water jet cutting machine off-site at a local vendor. Once the cutting is complete, the material will be brought back to university facilities where precision grinding will ensure parts flat and parallel. Countersinking the fastener holes is the final operation before assembly to the hub carrier with predrilled fasteners for aircraft safety wire. Balancing of the entire assembly will take place through an offsite source.

PART DRAWINGS

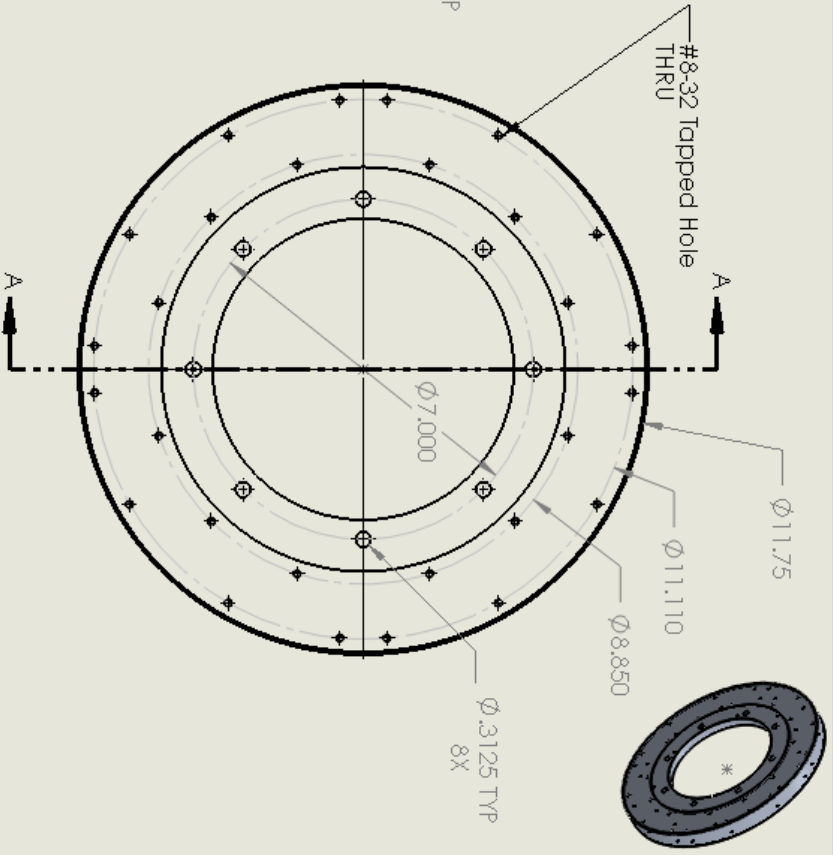
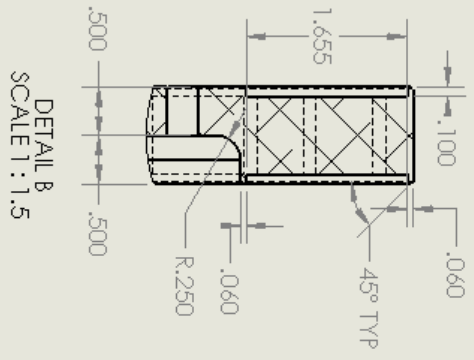
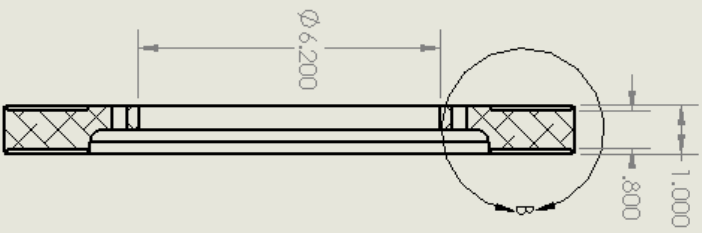
ITEM NO.	PART NUMBER	QTY.
1	Bearing Hub	1
2	11.75 Rotor Friction Face Segment	8
3	11.75 Brake Rotor Prototype Body	1
4	SBHCSCREW 0.625-11X3-HX-S	5
5	CR-FHMS0.164-32x0.5x0.5-S	56
6	SBHCSCREW 0.3125-18x1-HX-S	8
7	Preferred Narrow FW 0.3125	8
8	HHJNUT0.3125-18-D-S	8



UNLESS OTHERWISE SPECIFIED:		DRAWN	NAME	DATE
DIMENSIONS ARE IN INCHES			JWE	1-5-15
TOLERANCES:		CHECKED		
FRACTIONS		ENG APPR.		
ANGLES: MIN CHG. BEH. 4		WRG APPR.		
TWO PLACE DECIMAL 2.0000				
THREE PLACE DECIMAL 2.0000				

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5	4	3	2	1
APPLICATION	USED ON	DO NOT SCALE DRAWING	SHOWN AS ASSEMBLED	SCALE: 1:3
NEXT ASSY	ASSEMBLED	2 ROTORS = 1 SET	SIZE DWG. NO. A	WEIGHT:
				SHEET 1 OF 1



SECTION A-A
SCALE 1 : 3

DETAIL B
SCALE 1 : 1.5

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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ANGULAR DIMENSIONS: BOND 1
TYPE UNLESS OTHERWISE SPECIFIED
THREE PLACE DECIMAL 50.000
MFG APPR.
Q.A.
COMMENTS:
MATERIAL: 6061-T6 ALUMINUM
FINISH: AS MACHINED
DO NOT SCALE DRAWING

5

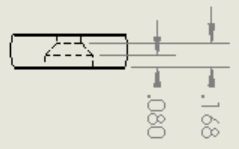
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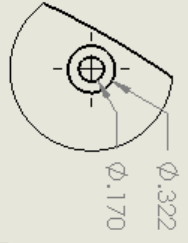
2

1

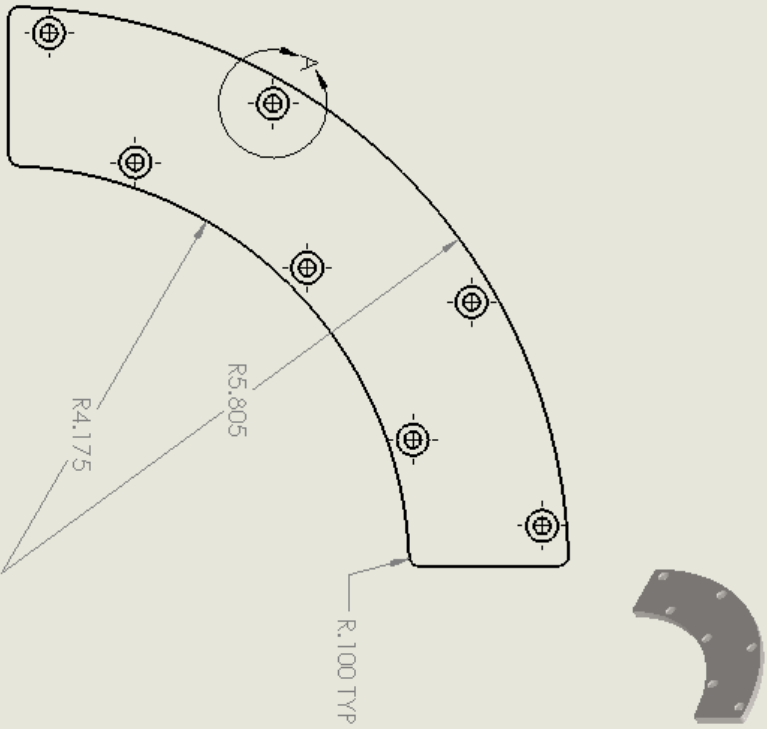
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ANGULAR DIMENSIONS: BOND 1 TYPE UNLESS OTHERWISE SPECIFIED THREE PLACE DECIMAL 50.000 MFG APPR. Q.A. COMMENTS:	NAME	DATE	TITLE:
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UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ANGULAR DIMENSIONS: BOND 1 TYPE UNLESS OTHERWISE SPECIFIED THREE PLACE DECIMAL 50.000 MFG APPR. Q.A. COMMENTS:			SIZE DWG. NO. A
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DETAIL B
SCALE 1:1



DETAIL A
SCALE 1:1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FOR DIMENSIONS
ANGULAR DIMS: BND 1
AND DECIMAL 50/100
THREE PLACE DECIMAL 50/100

INTERFERENCE FIT
IDENTIFICATION TRANSISTIVE

MATERIAL:
SAE 1080 HR

FINISH:
GROUND

DO NOT SCALE DRAWING

UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DRAWN	JWE	10-2-15
CHECKED		
ENG APPR.		
MFG APPR.		

WATERJET CUT
AND GROUND
TO SPEC WIDTH

TITLE:
11.75 ROTOR FRICTION
FACE SEGMENT

SIZE DWG. NO. REV
SCALE: 2:3 WEIGHT: SHEET 1 OF 1

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PROHIBITED.

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4

3

2

1

TESTING METHOD

INTRODUCTION

Braking components are vital pieces to making the race car achieve better efficiency during the races. If the rotor cannot absorb and dissipate the heat generated from friction before the next braking event, the driver will lose faith in the components and not push the vehicle to its ultimate performance point. If the rotors are the source of a frictional loss before, or after the braking event, the car will slow down. Sources from which losses could emanate are from induced pad drag due to radial run out. The run out will then push the caliper pistons further back in their bores thereby giving more pedal travel. If there is axial run out, this could be mistaken for a flat tire, wheel bearing spall, or other imbalance will in turn will cause the driver to slow down or be conscious that the car is not performing as well as it should. Thus, it is of the utmost importance that the brake rotors be evaluated for balance and symmetry to achieve maximum efficiency.

If the disc brake design fails the thermal analysis process, then a new design will be selected as the replacement. The performance analysis is repeated until the disc brake has met all the design requirements and the disc will be installed on the racecar later on for substantive testing. If the disc brake fails in testing then the data will be analyzed for the methods of the failure. If a redesign is necessary from the testing, a new disc brake shall be produced.

METHODS

Testing procedures involve four phases of criteria; confirming the dimensions so that the unit is a direct replacement and will mount to a components-off-the-shelf hub assembly, quantifying an overall reduction of static mass as well as moment of inertia, and finally on-vehicle testing for thermal tests and confirmation of forced convection coefficients, and finally independent testing if time and budget permit. If any unforeseen issues are found during testing, reassess material selection and design before moving forward with the next phase of testing.

DIMENSIONALITY

Structure is constrained by the dimensions, 11.75" diameter and 1.25" width and serves as a direct replacement rotor for a circle track racecar. The disc brake must mount to a purchased hub assembly with eight 5/16" fasteners on a 7" bolt circle. Confirmation of such dimensionality is the use of two hubs from different manufacturers to ensure the industry standard for fitment is maintained.

OFF VEHICLE TESTING

Record material mass, volume and calculate density. Perform recalculations, if necessary, for dependent energy and thermal transfer equations if mass, volume or density is significantly skewed. Once done, a comparison between the iron and the composite rotor to gauge a static mass difference noting any difference between the theoretical and actual.

To quantify a theoretical reduction in inertia, a machine base is necessary to mount rotor. Once done, a piece of TIG welding rod is inserted into the apparatus vertically though the origin so as to suspend the mount and rotor. This will serve as a torsion bar. The rotor/mount will then be rotated 90° in order to place a torque on the welding rod/torsion bar. Once released, the user will log the time necessary to reach 10 cycles. An evaluation on the percentage reduction per given time difference will note any change in the moment of inertia.

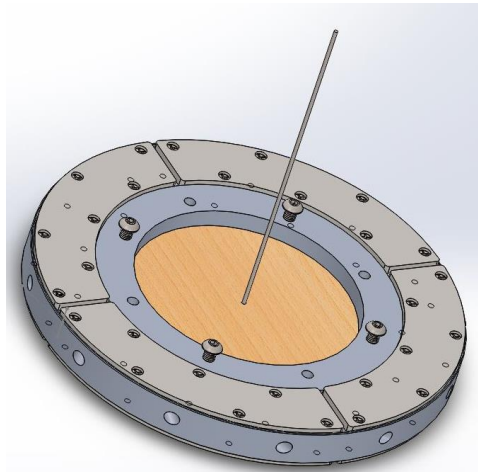


Figure 9: Illustration of Inertia Test Fixture

ON VEHICLE TESTING

The vehicle will be equipped with a GPS-based accelerometer and infrared non-contact pyrometers to verify assumptions of initial/final velocity, deceleration rate, time between braking events, temperature rise versus lap. Throughout the repeated braking condition, the disc brake rotor is subjected to continuous heating and cooling process. During braking, frictional heat load is subjected to the rotor surface through conduction. After the brake is released, the rotor is then allowed to cool through convection process. The heat transfer process repeats until the end of a 20 lap period and data is collected. Because of the nature of competition, the chosen test driver and team are subject to non-disclosure agreements as well as liability waivers in the event that a catastrophic event occurs stemming from the construction of the brake rotors.



Figure 10: GTechPro RR Fanatic



Figure 11: Amprobe IR-750

THIRD PARTY TESTING

If the on-car testing is to satisfaction, the rotors will then move onto third party testing. A suitable testing facility has already been chosen and is awaiting the deliverables.

DELIVERABLES

Two rotor bodies along with 16 friction faces shall be assembled along with corresponding hubs mounted and ready for off-vehicle testing by no later than March 16th, 2015. On-vehicle testing shall be completed no later than May 10th, 2015 so that raw data may be considered before SOURCE presentation and in-class presentations soon there to follow.

PROJECT MANAGEMENT

COST AND BUDGET

The list of raw materials and fasteners required to produce two brake rotors is broken down in list form in the following section. Estimated cost for materials alone is \$283.20. Labor hours for CNC machining as well as water jet cutting are as of yet not calculated. But the predicted number of hours necessary to produce a working prototype is approximately 51 hours.

Third party dynamometer rates quoted from a telephone conversation with a representative of Link Engineering, in Detroit Michigan, including shipping charges, was between \$750-1200. Testing equipment includes GPS-based accelerometer and non-contact infrared thermometer. Total estimate, before dynamometer rates is approximately \$600.

On track testing and materials will rely on a GPS-based accelerometer and two non-contact infrared thermometers mounted on the car's frame rails to monitor temperature rise. All testing equipment has data logging capability and the use of this data will culminate in the final report. Gross approximate cost to produce and test a working prototype is approximately \$2000.

ESTIMATED PARTS LIST AND BUDGET

Item #	Description	Source	Model	Price/Cost	Misc. Info	Quantity	Subtotal \$	Actual \$ w/ tax
1	Aluminum Plate	Haskins	13" x 13" x 1.5"	81.21/ea.	6061-T6	2	\$162.42	\$175.74
2	Water jet cut	Haskins	Cut Charge	\$30	Add'l cost for Al plate	1	\$30	\$32.46
3	Steel Plate	Yakima Steel Fab	18" x 24" x .25"	\$31.15	1018 plate	1	\$31.15	\$33.70
4	Fastener	McMaster-Carr	Countersunk	\$7.42/100	91771A196	56	\$7.42	\$8.03
5	Fastener	McMaster-Carr	Shouldered	\$1.54/ea.	91264A242	16	\$24.64	\$26.66
6	Fastener	McMaster-Carr	Locking nut	\$6.31/100	91837A014	16	\$6.31	\$6.83

Estimated Total: \$283.42

SCHEDULE

Parts are scheduled for a delivery date no later than March 16th, 2015. This will insure that on car testing will proceed in accordance to opening testing sessions on racetrack grounds. Once testing is completed, rotors will then be shipped to a third part testing facility to qualify thermodynamic and heat transfer calculations.

Event #	Description	Duration (days)	Start	Finish	Predecessors
1	Analysis	73	9/24/2014	1/2/2015	
2	Materials Quotes	15	11/20/2014	12/1/2014	
3	Materials Ordering	10	12/17/2014	12/31/2014	2
4	CNC Programming	5	1/10/2015	1/15/2015	1
5	Waterjet Cutting	4	1/15/2015	1/20/2015	3
6	CNC Setup	4	1/19/2015	1/23/2015	4
7	Machining	5	1/26/2015	2/2/2015	6
8	Quality Control	1	2/3/2015	2/4/2015	
9	Assembly	2	2/5/2015	2/7/2015	5,7
10	Off-Site Testing	20	2/9/2015	3/9/2015	9
11	On-Car Testing	7	3/13/2015	3/23/2015	10
12	Presentation Prep	74	2/9/2015	5/22/2015	
13	Delivery	1	3/24/2015	3/24/2015	11
14	Presentation	1	5/25/2015	6/4/2015	12

Figure 12 Gantt chart of Estimated Start/Finish and Completion Dates

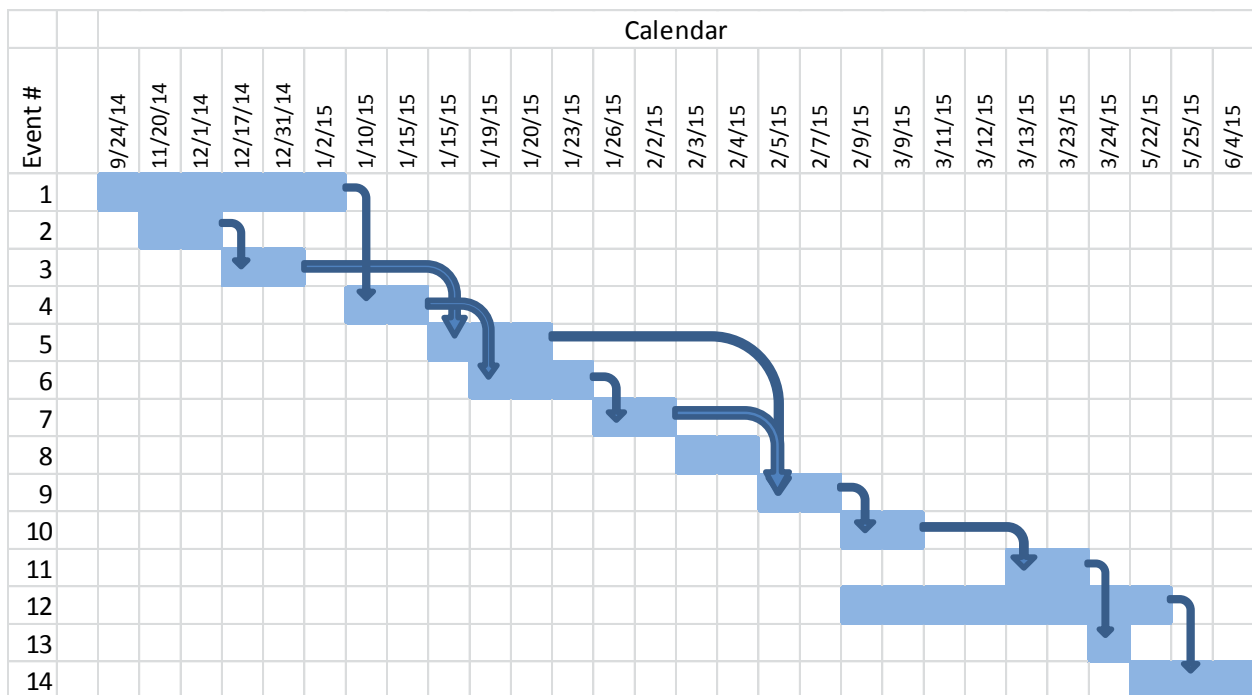


Figure 13 Gantt chart Timeline

DISCUSSION

The disc brake rotor is made from gray cast iron material which provides good wear resistance with high thermal conductivity and the production cost is low compared to other high performance disc brake rotor materials such as cast steel, Metal Matrix Composite (MMC), carbon composites and ceramic based composites. Although advanced brake materials such as aluminum metal matrix composite offer significant weight advantages compared with the traditional cast iron rotor, the aluminum metal matrix composite material has a much lower maximum operating temperature which limits its application.

Several issues limit the application for an aluminum bodied rotor. For one, much like the MMC rotor the low temperature threshold is of paramount concern and must be addressed for higher duty environments such as motorsport.

CONCLUSION

Brakes are an essential part of a racecar, but in general large performance gains are not made here. It is important to design a system that is well balanced, offers tuning potential during testing, and is reliable. Reducing the amount of mass on one aspect of the vehicle is important, but a reduction of the moment of inertia about the wheel is significant from an acceleration or deceleration standpoint. It would be exciting to see further research look into the heat capacity of rotors more and balance analytical work with real world testing.

ACKNOWLEDGEMENTS

I would like to thank Central Washington University for providing me the means to see my dream of a motorsport developed part through. I would like to especially thank three professors, Charles Pringle, Roger Beardsley and above all else Dr. Craig Johnson for all the times of that I wanted to quit and transfer into the art department.

There are no shortage of support staff and friends that have helped along the way. Matt Burvee was instrumental in acquiring materials and expertise when it came time to produce the initial set of rotors and talk me down off the ledge when I broke all those fragile 10-24 spiral taps. Friends such as Eric Johnson for helping with the thermal calculations, Nate Wilhelm for proofreading and small parts allocation, Trevor Bergstrom for Gantt chart development, Jeremy Dickson for this help with the plasma torch cutting table, and Tom Lee for his professional guidance in the realm of performance braking.

I'd like to thank some of the professional support I've received along the way; Lenard Bartheld for his consummate professional support and the use of his machine shop. His relentless snickering at my over-engineered approach was reminder to keep everything simple and cheap. For on-vehicle testing, Rick and Eric Matthews for the use of their award winning race car when others flaked out or were leery of their performance.

My final acknowledgement goes to John Ellsworth for saving my entire project when he located my lost thumb drive in a flower pot in front of the Ace Hardware in Ellensburg, WA.

Without everyone's persistent pushing to strive for more there is no me.

APPENDIX A – ANALYSES

A1

John Ewert MET 495 Oct 2nd 2014 3

Nascar tire spec
 28" x 12" x 15"
 200 mph

$C = \pi D$
 $\pi(28 \text{ in}) = 87.96 \text{ in} \rightarrow .0014 \text{ mi/rev}$

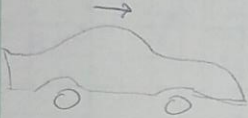
Conversion
 $200 \text{ mi/hr} \left[\frac{1 \text{ hr}}{60 \text{ min}} \right] = 3.3 \text{ mi/min} \left[\frac{1 \text{ rev}}{.0014 \text{ mi}} \right] = 2400.97 \text{ rev/min}$

$250 \text{ mi/hr} = 4.1\bar{6} \text{ mi/min} \left[\frac{1 \text{ rev}}{.0014 \text{ mi}} \right] = 3001.21 \text{ rev/min} \left(\frac{\pi}{1 \text{ rev}} \right) \left(\frac{1 \text{ min}}{60 \text{ sec}} \right)$
 315 rad/sec

Observed Track Speed at Yak Speedway = 150 mph
 $150 \text{ mi/hr} = 2.5 \text{ mi/min} \left[\frac{1 \text{ rev}}{.0014 \text{ mi}} \right] = 1800 \text{ rev/min}$

S.F. = $\frac{3000}{1800} = 1.6$

increased mpg
 lighter weight
 better off/on response
 brake/cool

John Eward	MET 485	
<p>Senior Project: Aluminum brake rotor with replaceable friction surfaces</p>		
<ul style="list-style-type: none"> - Less static & rotating mass - Cheaper to maintain - Lower melting point = reduced operating range - reduction of moment of inertia 		
<p>Proof of concept: match friction material to that of conventional design</p>		
<p>Evolution: Carbon Ceramic replaceable pads at friction of full C/C rotors.</p>		
	<p>$m = 2900 \text{ lbs} \rightarrow 1315 \text{ Kg}$</p> <p>Brake zone 3-3.5 sec</p> <p>Apex to next Brake zone $\rightarrow 17 \text{ sec}$</p> <p>Total lap time = 18-20 sec.</p>	
<p>$v_i = 100 \text{ mph} = 44.7 \text{ m/s}$</p> <p>$v_f = 50 \text{ mph} = 22.4 \text{ m/s}$</p>		
<p>Assumed brake bias 75% front, 25% rear</p>		
<p>Kinetic Energy, $KE = \frac{1}{2} m v^2 \rightarrow \frac{1}{2} m (v_i^2 - v_f^2)$</p> $= \frac{1}{2} (1315 \text{ Kg}) \left[(44.7 \text{ m/s})^2 - (22.4 \text{ m/s})^2 \right]$		
<p>$KE_{\text{Total}} = 983837 \text{ Joules} \rightarrow 984 \text{ KJ}$</p>		
<p>Split between front & rear individual rotors</p>		
<p>$KE_{\text{front}} = \frac{984 \text{ KJ} (0.75)}{2} = 368939 \text{ Joules} \rightarrow 369 \text{ KJ}$</p>		
<p>$KE_{\text{rear}} = \frac{984 \text{ KJ} (0.25)}{2} = 122980 \text{ Joules} \rightarrow 123 \text{ KJ}$</p>		
<p>Assume 3.5 sec Braking zone</p>		
<p>Front = $\frac{369 \text{ KJ}}{3.5 \text{ sec}} = 105.5 \text{ KW}$</p>		
<p>Rear = $\frac{123 \text{ KJ}}{3.5 \text{ sec}} = 35.2 \text{ KW}$</p>		

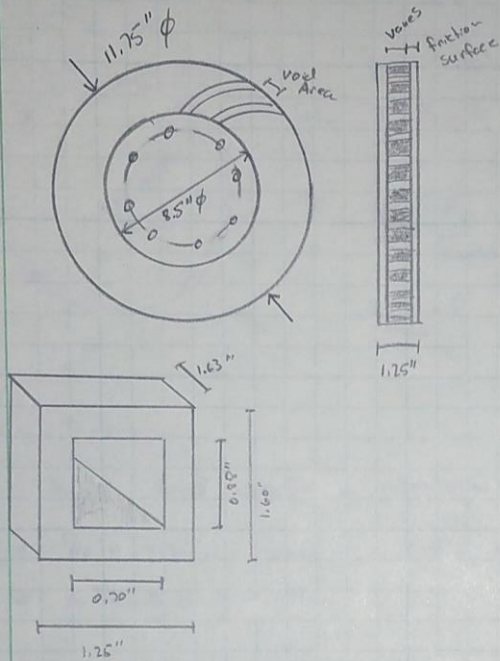
John Ewert

MET 495

Composite Brake Rotor Design

- replace cast iron with Al (body) & Steel (friction surfaces)

Picture



Basis: Late Model Circle Track Race car

C.O.T.S: bearing carrier is mounting surface for rotor via 8 x 7" bolt pattern on internal diameter
utilizes 4-piston caliper with 2 pads total

Constraints: must be direct replacement for conventional rotor

Assumptions: Voids/Void area are straight rather than curved

Typical rotor contains 29 voids

$$\text{Vol}_{\text{rotor}} = \left[\pi \left[\frac{\text{Dia}_{\text{out}}}{2} \right]^2 - \pi \left[\frac{\text{Dia}_{\text{inner}}}{2} \right]^2 \right] \cdot \text{width} = \text{Void Volume}$$

$$= \left[\pi \left[\frac{11.75 \text{ in}}{2} \right]^2 - \pi \left[\frac{8.5 \text{ in}}{2} \right]^2 \right] \cdot 1.25 \text{ in} = \text{Void Volume}$$

$$64.61 \text{ in}^3 - 29.12 \text{ in}^3 = 35.49 \text{ in}^3 \rightarrow \underline{\underline{581 \times 10^{-6} \text{ m}^3}}$$

$$1 \text{ m}^3 = 61024 \text{ in}^3$$

Void Volume

$$V_{\text{outer}} = 1.25 \text{ in} \cdot 1.60 \text{ in} \cdot 1.63 \text{ in} = 3.26 \text{ in}^3$$

$$V_{\text{inner}} = 0.70 \text{ in} \cdot 0.88 \text{ in} \cdot 1.63 \text{ in} = 1.00 \text{ in}^3 = 29 \text{ voids} = 29.12 \text{ in}^3$$

$$\text{Rotor Mass: } m_{\text{Iron}} = \rho \text{ Vol} = 7820 \text{ kg/m}^3 \cdot 581 \times 10^{-6} \text{ m}^3 = 4.57 \text{ Kg} \rightarrow 10.1 \text{ lbs}$$

$$m_{\text{Al+st}} = \rho \text{ Vol} = 2712 \text{ kg/m}^3 \cdot 405 \times 10^{-6} \text{ m}^3 + 7860 \text{ kg/m}^3 \cdot 176 \times 10^{-6} \text{ m}^3 = 2.48 \text{ Kg} \rightarrow 5.5 \text{ lbs}$$

JohnEvent	MET 455	Oct 19 th 2014
Forces about the wheel		
Given:	Car mass = 3400 lbs \rightarrow 1542 Kg car velocity $v_i = 150$ mph \rightarrow 67.056 m/s $v_f = 50$ mph \rightarrow 22.352 m/s Time = 4sec Tire Diameter = 28" \rightarrow 0.7112m Nascar spec Assume 75% front brake bias 25% rear	Find: Moment placed about each wheel during a braking event
Soln:	Decel Rate $v_f = v_o + a_T T \rightarrow a_T = \frac{v_f - v_o}{T}$ $\frac{(22.352 - 67.056) \text{ m/s}}{4 \text{ sec}} = -11.18 \text{ m/s}^2$ Force from Decel $F_T = m a_T$ $(1542 \text{ Kg})(-11.18 \text{ m/s}^2) = -17.2 \text{ KN}$	
	Force Distribution $F_{\text{front}} = \frac{(-17233 \text{ N}) \cdot 0.75}{2} = -6462 \text{ N}$ per wheel on front axle $F_{\text{rear}} = \frac{(-17233 \text{ N}) \cdot 0.25}{2} = -2154 \text{ N}$ per wheel on rear axle	
	Moment about each wheel $m = F r$ $m_{\text{front}} = (-6462 \text{ N}) \left(\frac{0.7112 \text{ m}}{2} \right) = -2298 \text{ N}\cdot\text{m}$ $m_{\text{rear}} = (-2154 \text{ N}) \left(\frac{0.7112 \text{ m}}{2} \right) = -766 \text{ N}\cdot\text{m}$	
	Assumes $\mu = 1.0$ between track & tire on level surface with no banking	
	Conversion: 1 ft-lb = 1.35 N·m 2298 N·m = 1695 ft-lbs 766 N·m = 565 ft-lbs	

John Event

MET 495

Shear on Fasteners

Given: T_b on Rotor/wheel/Tire = 1300 N·m

Find: Shear on fasteners

Diameter of Tire = 0.71m

Using 5/16" screw for hub

Using 10-24 screw on face

Forces on faces

$$W_T = \frac{T}{D/2} = \frac{1300 \text{ N}\cdot\text{m}}{\frac{0.71\text{m}}{2}} = \frac{3662 \text{ N}}{7 \text{ fasteners per plate}} = 523 \text{ N per fastener}$$

Forces on hub

$$W_T = \frac{T}{D/2} = \frac{1300 \text{ N}\cdot\text{m}}{\frac{0.71\text{m}}{2}} = \frac{3662 \text{ N}}{8 \text{ fasteners on hub}} = 458 \text{ N per fastener}$$

Shear of fasteners

5/16 tensile stress Area, $A_T = 0.0524 \text{ in}^2 \rightarrow 3.381 \times 10^{-5} \text{ m}^2$
 matt pg. 633

10-24 tensile stress Area, $A_T = 0.0175 \text{ in}^2 \rightarrow 1.129 \times 10^{-5} \text{ m}^2$

$$\tau = \frac{F}{A_T} \quad 5/16 \rightarrow \frac{458 \text{ N}}{3.381 \times 10^{-5} \text{ m}^2} = 13.5 \text{ MPa}$$

$$10-24 \rightarrow \frac{523 \text{ N}}{1.129 \times 10^{-5} \text{ m}^2} = 46.3 \text{ MPa}$$

Grade 5 yield 92 Ksi \rightarrow 558 MPaGrade 2 yield 57 Ksi \rightarrow 393 MPa

Safety Factor: if Grade 5 if Grade 2

5/16 41.3 29.1

10-24 12.1 8.5

John Ewert

MET 495

Theoretical Temp Rise for Aluminum with Steel Plates

6061-T6 Aluminum

$$\rho_{Al} = 2712 \text{ kg/m}^3$$

$$c_{Al} = 897 \text{ J/kg}\cdot\text{C}$$

$$v_i = 44.7 \text{ m/s}$$

$$v_f = 22.4 \text{ m/s}$$

SAE 1018 Steel

$$\rho_{st} = 7860 \text{ kg/m}^3$$

$$c_{st} = 434 \text{ J/kg}\cdot\text{C}$$

$$m = 1315 \text{ kg}$$

75% front } Brake bias
25% rear }

$$Vol_{st} = \pi \left[\frac{11.63 \text{ in}}{2} \right]^2 - \pi \left[\frac{8.62 \text{ in}}{2} \right]^2 = 0.225 \text{ in} = 10.8 \text{ in}^2 \rightarrow 176 \times 10^{-6}$$

$$Vol_{Al} = 581 \times 10^{-6} \text{ m}^3 - 176 \times 10^{-6} \text{ m}^3 = 405 \times 10^{-6} \text{ m}^3$$

$$T_{Al+St} = \frac{1-\epsilon}{2} \left[\frac{mg(v_i^2 - v_f^2)}{2g(\rho_{Al} c_{Al} Vol_{Al} + \rho_{st} c_{st} Vol_{st})} \right]$$

$$= \frac{1-0.25}{2} \cdot \frac{(1315 \text{ kg})(9.81 \text{ m/s}^2) \left[(44.7 \text{ m/s})^2 - (22.4 \text{ m/s})^2 \right]}{2(9.81 \text{ m/s}^2) \left[(2712 \text{ kg/m}^3)(897 \text{ J/kg}\cdot\text{C})(405 \times 10^{-6} \text{ m}^3) + (7860 \text{ kg/m}^3)(434 \text{ J/kg}\cdot\text{C})(176 \times 10^{-6} \text{ m}^3) \right]}$$

620.47 total

$$T_{Al+St} = 233^\circ\text{C} \rightarrow 451^\circ\text{F}$$

$$= 78^\circ\text{C} \rightarrow 172^\circ\text{F}$$

$$\Delta T = T_{Al+St} - T_{Iron}$$

$$\Delta T_{Front} = +52^\circ\text{C}$$

$$\Delta T_{Rear} = +18^\circ\text{C}$$

If Subjected to 150 MPH (67.1 m/s)

$$T_{Al+St} = 622^\circ\text{C} \rightarrow 1152^\circ\text{F}$$

$$= 207^\circ\text{C} \rightarrow 405^\circ\text{F}$$

$$\Delta T_{Front} = +138^\circ\text{C}$$

$$\Delta T_{Rear} = +46^\circ\text{C}$$

165.9 Total

Melting point of Aluminum: $677^\circ\text{C} \rightarrow 1250^\circ\text{F}$

John Evers

MET 495

Theoretical Temp Rise for Iron Rotor

$$T_{\text{Iron}} = \frac{1-\phi}{2} \left[\frac{w(V_i^2 - V_f^2)}{2g \rho r c_p V_r} \right]$$

Corner Entry Speed, $V_i = 100 \text{ mph} \rightarrow 44.7 \text{ m/s}$ Exit Speed, $V_f = 50 \text{ mph} \rightarrow 22.4 \text{ m/s}$ Specific Gravity, $\rho_{\text{Iron}} = 7820 \text{ kg/m}^3$ Coeff of part, $C_{\text{Iron}} = 449 \text{ J/kg}\cdot\text{C}$ Volume, $V_{\text{rotor}} = 581 \times 10^{-6} \text{ m}^3$ Mass, $m = 1542 \text{ kg}$ Brake bias, $\phi = \begin{matrix} 0.25 \\ 0.75 \end{matrix}$

$$T_{\text{Iron}} = \frac{1-(0.25)}{2} \left[\frac{(1542 \text{ kg})(9.81 \text{ m/s}^2) \left[(44.7 \text{ m/s})^2 - (22.4 \text{ m/s})^2 \right]}{2(9.81 \text{ m/s}^2)(7820 \text{ kg/m}^3)(449 \text{ J/kg}\cdot\text{C})(581 \times 10^{-6} \text{ m}^3)} \right]$$

$$T_{\text{Iron}} = 181 \text{ }^\circ\text{C} \rightarrow 358 \text{ }^\circ\text{F} \text{ if used on front}$$

$$60 \text{ }^\circ\text{C} \rightarrow 141 \text{ }^\circ\text{F} \text{ if used on rear}$$

if subjected to 150 MPH (67.1 m/s)

$$T_{\text{iron}} = 484 \text{ }^\circ\text{C} \rightarrow 903 \text{ }^\circ\text{F} \text{ front}$$

$$= 161 \text{ }^\circ\text{C} \rightarrow 322 \text{ }^\circ\text{F} \text{ rear}$$

482 Total

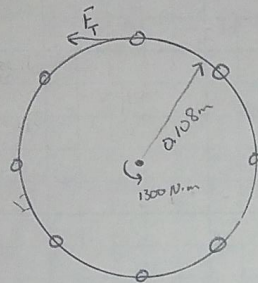
1289 Total

Given: Rotor bolts (8x) in single shear

$$d = 5/16" \rightarrow 0.0794 \text{ m}$$

$$T_b = 1300 \text{ N}\cdot\text{m}$$

Find: Specify width of flange on rotor body



$$\text{Soln: } T_b = F_b \cdot r \rightarrow \frac{F_b}{\text{Total}} = \frac{T_b}{r} = \frac{1300 \text{ N}\cdot\text{m}}{0.108 \text{ m}} = 12042 \text{ N}$$

$$F_{\text{Total}} = \frac{F_{\text{bolt}}}{8 \text{ bolts}} = 1506 \text{ N per bolt}$$

$$\text{S.F.} = 2.5 \rightarrow F_{\text{max}} = F_{\text{bolt}} \cdot \text{S.F.} \rightarrow 1506 \cdot 2.5$$

$$F_{\text{max}} = 3764 \text{ N}$$

Flange Dimensions

- Assume RC8 class fit on rotor hat

$$d = 5/16" \rightarrow .3125 \text{ in} + 0.0022 \text{ in} = .3147 \text{ in} \rightarrow 0.00799 \text{ m}$$

$$w = 3/4" \rightarrow .75 \text{ in} \rightarrow 0.0191 \text{ m}$$

Mott
Fig 3-29

$$\frac{d}{w} = 0.417 \rightarrow K_T = 2.9$$

$$\sigma_{\text{max}} = K_T \frac{F}{A} = 2.9 \frac{3764 \text{ N}}{\pi \left[\frac{(0.0191 \text{ m})^2}{4} \right]} = 218 \text{ MPa}$$

$$\sigma_{\text{Allowable}} \text{ for Al 6061-T6} = 276 \text{ MPa}$$

$\sigma_{\text{max}} < \sigma_{\text{Allow}}$ \therefore A 5/16" hole in the Aluminum is acceptable for the load.

$$\sigma_{\text{max}} = \frac{F}{A} \rightarrow \frac{F}{(w-d)T} \rightarrow T = \frac{F}{(w-d)\sigma_{\text{max}}} = \frac{3764 \text{ N}}{(0.0191 - 0.00799) 276 \times 10^6} = 0.0012 \text{ m}$$

$$= 0.048 \text{ in}$$

Any hole longer than 0.048 in will support the load for this application

John Ewert MET 495 Oct 2nd 2014

Hybrid brake rotor design

Notes:

- Basis: Local Sportsman class constraints
- 15" steel wheel
- 4-piston caliper
- ∅ 12.12" x 1.25"
- C.O.T.S. Rotor/wheel bearing mounting hat
- original cast from grey iron
- $\rho = 7.86 \text{ g/cm}^3 @ 20^\circ\text{C}$ sae.org

Moment of Inertia

* assume for now rotor is solid, neglect void area from vanes

Disk $m_d = \rho_d V_d = 7.86 \text{ g/cm}^3 [\pi (12.19 \text{ in})^2 (1.25 \text{ in})]$
 convert $7860 \text{ kg/m}^3 [\pi (0.3096 \text{ m})^2 (0.03175 \text{ m})]$
 $7860 \text{ kg/m}^3 (.00956 \text{ m}^3)$
 $m_d = 75.16 \text{ Kg}$

Steels rotor
#11810
p.541

$(I_o)_d = \frac{1}{2} m_d r_d^2 + m_d d^2$
 $= \frac{1}{2} (75.16 \text{ Kg}) (0.3096 \text{ m})^2 + (75.16 \text{ Kg}) (0.3096 \text{ m})^2 = 10.81 \text{ Kg} \cdot \text{m}^2$

hat $m_h = \rho_h V_h = 7860 \text{ kg/m}^3 [\pi (1.5 \text{ in})^2 (1.25 \text{ in})]$
 $7860 \text{ kg/m}^3 [\pi (0.2159 \text{ m})^2 (0.03175 \text{ m})]$
 $m_h = 36.54 \text{ Kg}$

$(I_o)_h = \frac{1}{2} m_h r_h^2 + m_h d^2$
 $\frac{1}{2} (36.54 \text{ Kg}) (0.2159 \text{ m})^2 + (36.54 \text{ Kg}) (0.3096 \text{ m})^2 = 4.35 \text{ Kg} \cdot \text{m}^2$

$I_o = (I_o)_d + (I_o)_h$
 $10.81 + 4.35 = 6.46 \text{ Kg} \cdot \text{m}^2 \text{ for grey iron}$

Sol. Exam	MET 495	Oct 2nd 2014
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Aluminum
 - assume 6061 density $\rightarrow \rho = 2.698 \text{ g/cm}^3$

$$\underline{\underline{Disk}} = 2.698 \text{ kg/m}^3 \left[\pi (0.3096 \text{ m})^2 (0.03175 \text{ m}) \right]$$

$$= 25.80 \text{ kg}$$

$$(I_o)_d = \frac{1}{2} m_d r_d^2 + m_d d^2$$

$$= \frac{1}{2} (25.80 \text{ kg}) (0.3096 \text{ m})^2 + (25.80 \text{ kg}) (0.3096 \text{ m})^2$$

$$= 3.71 \text{ kg} \cdot \text{m}^2$$

Hole = $2.698 \text{ kg/m}^3 \left[\pi (0.2159 \text{ m})^2 (0.03175 \text{ m}) \right]$

$$= 12.54 \text{ kg}$$

$$(I_o)_h = \frac{1}{2} m_h r_h^2 + m_h d^2$$

$$= \frac{1}{2} (12.54 \text{ kg}) (0.2159 \text{ m})^2 + (12.54 \text{ kg}) (0.3096 \text{ m})^2$$

$$= 1.49 \text{ kg} \cdot \text{m}^2$$

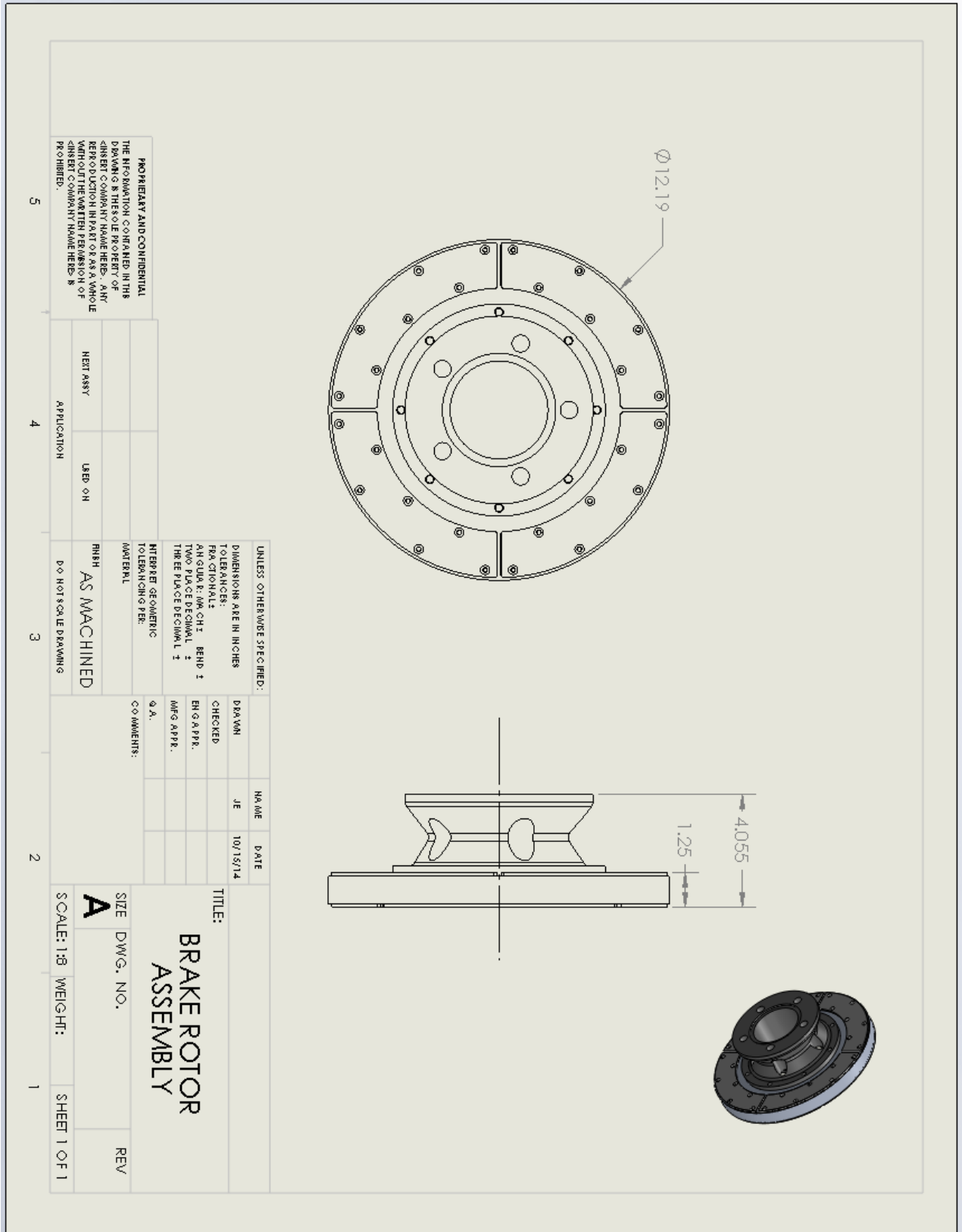
$$I_o = (I_o)_d - (I_o)_h$$

$$3.71 - 1.49 = \boxed{2.22 \text{ kg} \cdot \text{m}^2 \text{ for Aluminum}}$$

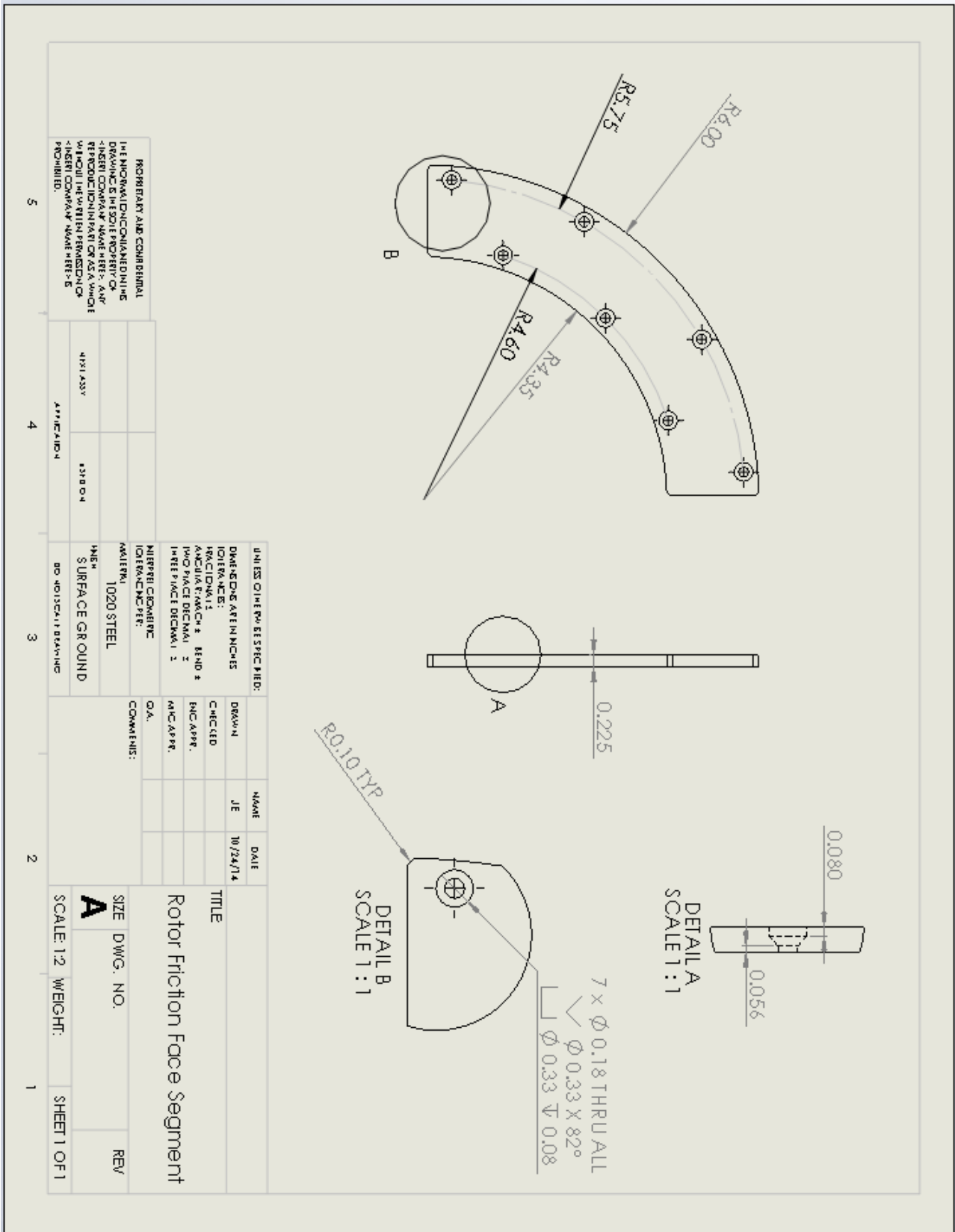
$$\% \text{ change} = \left[\frac{2.22 - 6.46}{6.46} \right] \cdot 100 = \boxed{65.4\%}$$

APPENDIX B – ASSEMBLY DRAWINGS

Design 2 Assembly Drawing



Segmented Friction Face Drawing



APPENDIX C – PARTS LIST

Item #	Description	Source	Model	Price/Cost	Misc. Info	Quantity	Subtotal \$	Actual \$ w/ tax
1	Aluminum Plate	Haskins	24" x 12" x 1.25"	\$151.83	6061-T6	1	\$151.83	\$175.74
2	Aluminum Plate	Haskins	12" x 12" x 1.0"	\$71.23	6061-T6	1	\$71.23	\$77.07
3	Steel Plate	Yakima Steel Fab	18" x 24" x .25"	\$31.15	SAE 1018 CR plate	1	\$31.15	\$33.70
4	Water jet Cutting	Perrault Fabrication	Cut Charge	\$125/hr.		0.5	\$62.50	\$67.63
5	Steel Plate	Western Metal	16" x 20" x .25"	\$31.15	SAE 1018 HR plate	1	\$16.35	\$17.64
6	Fastener	Tacoma Screw	10-24 #2 Drive Undercut Hardened	\$5.49/100	91099A265	112	\$10.98	\$11.88
7	Fastener	Tacoma Screw	Cap Screw 5/16 - 18 1-1/4" Thread	\$7.16/25	90462A991	16	\$7.16	\$7.75
8	Fastener	Tacoma Screw	USS Flat Washer 5/16"	\$2.28/100	90108A412	16	\$2.30	\$2.49
9	Fastener	Tacoma Screw	Distorted-Thread Centerlock Nut 5/16-18	\$11.16/50	91837A016	16	\$11.32	\$12.25

Total: \$406.15

PURCHASED TESTING EQUIPMENT

Item #	Description	Source	Model	Price/Cost	Misc. Info	Quantity	Subtotal \$	Actual \$ w/ tax
1	Accelerometer	EBay	G-Tech	150.00	502-RR	1	150.00	162.50
2	Infrared Thermometer	Amazon	Amprobe	226.94	IR-750	2	453.88	491.10
3	Laptop	Bestway Pawn	Acer	\$75	Aspire D255	1	\$62	\$73
4	Track Rental	Yakima Speedway	Asphalt	\$50/hr. + \$10/person	½ mile oval	4 hours 5 people	\$250.00	\$250.00

Total: \$803.60

Grand Total: \$1209.75

APPENDIX D – SCHEDULE

Event #	Description	Start	Finish	Trailers	Duration (days)	Est. Hours	Act. Hours
1	Analysis	9/24/2014	1/2/2015		73	4	22
2	Materials Quotes	11/20/2014	12/1/2014		15	2	7
3	Materials Ordering	12/17/2014	12/31/2014	2	10	2	3
4	CNC Programming	1/10/2015	1/15/2015	1	5	5	15
5	Waterjet Cutting	1/15/2015	1/20/2015	3	4	2	7
6	CNC Setup	1/19/2015	1/23/2015	4	4	3	2
7	On-Site Machining	1/26/2015	2/2/2015	6	5	8	24
8	Outsourced Machining				5	2	4
9	Quality Control	2/3/2015	2/4/2015		1	1	1
10	Assembly	2/5/2015	2/7/2015	5,7	2	2	5
11	Finished Prototype		3/15/2015				
12	Off-Site Testing	2/9/2015	3/9/2015	9	20	6	8
13	On-Car Testing	3/15/2015	4/23/2015	10	7	2	14
14	Presentation Prep	2/9/2015	5/22/2015		74	10	32
15	Delivery	3/24/2015	3/24/2015	11	1	1	1
16	Presentation	5/25/2015	6/4/2015	12	1	1	1
					Total	51	146

APPENDIX E – EVALUATION SHEET

Iron Rotor

Lap #	Entry Speed (Vi)	Exit Speed (Vf)	Time in Brake Zone	Accel	Left Side Rotor Temp	Right Side Rotor Temp	Cool down time	Left Side Rotor Temp	Right Side Rotor Temp
1A									
1B									
2A									
2B									

Composite Rotor

Lap #	Entry Speed (Vi)	Exit Speed (Vf)	Time in Brake Zone	Accel	Left Side Rotor Temp	Right Side Rotor Temp	Cool down time	Left Side Rotor Temp	Right Side Rotor Temp
1A									
1B									
2A									
2B									

APPENDIX F – TESTING REPORT

Initial testing of static mass as well as moment of inertia produced a 19.6% decrease in the moment of inertia and 1.3lbs mass reduction. Once a final contour was placed along the radial edge, further decreases in static mass and inertia were realized, 23.8% and 1.55lbs respectively.

Since several driver interviews were necessary to choose the velocities it was imperative to validate such assumptions. The result is the GPS telemetry data from a GTechPro RR Fanatic performance meter, manufactured by Tesla Electronics, LLC. This, along with an Amprobe IR-750 data logging infrared gun were used extensively to gather the required information that is the results seen.

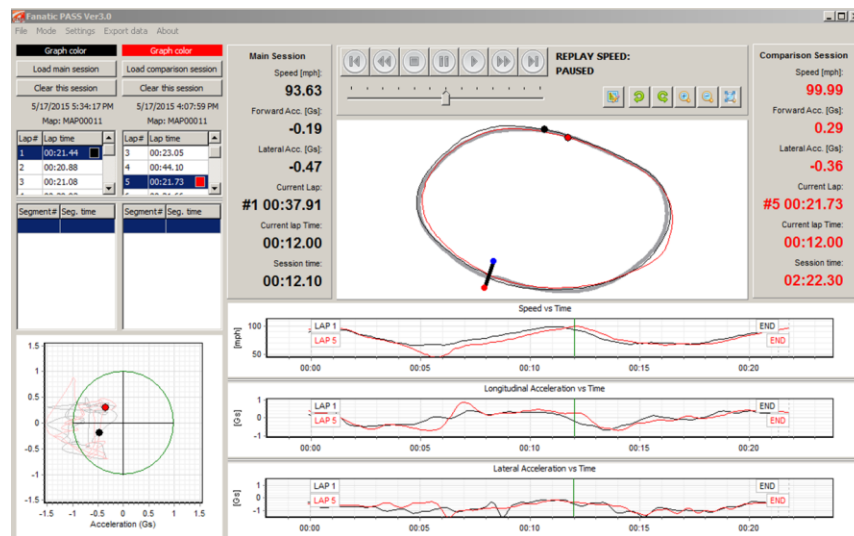


Figure 14: GPS Data logging Software of Yakima Speedway 1/2mi Asphalt Oval

The sample data below is part of the 57 laps acquired from the iron rotor test periods and 74 laps from the composite rotor test periods respectively. Two drivers were used in the testing. Given the varied styles and brake/throttle inputs between the two, it was imperative to judge the composite rotor performance on one driver's style alone. Actual entry and exit velocities were very close to those chosen for the assumptions, 100.51 actual vs 100 estimated and 49.60 actual vs 50 estimated respectively. The assumed time in the braking zones, estimated 3 seconds, and the cool down period, estimated 7 seconds, are reversed when observing the telemetry data skewing the assumed amount of thermal input thus a recalculation of the initial analyses is necessary.

Location on vehicle was of paramount concern. Assuming a 75%/25% front/rear brake bias, noted that both drivers experiment with ranges between 80/20 and 65/35 respectively, the on-car testing placed the rotors on the front axle for the greatest amount of thermal loading. The front was calculated to be subjected to 123kW of kinetic energy whereas if the rotors and been placed on the rear axle, a calculated 41kW of kinetic energy was the result.



Figure 15: Fabricated Rotor Mounted to Car

Predictions for temperature increase from ambient were close to actual for the given track conditions and ambient temperature, 23°C, and driver input. Comparison between the steel rotor versus the fabricated unit indicate a peak temperature of ~315°C and 370°C respectively, $\Delta T = 55^\circ\text{C}$. Final temperature entering the braking zone was 163°C and 99°C respectively, $\Delta T = 64^\circ\text{C}$. The calculated energy dissipation yielded a theoretical temperature rise of the composite rotor, when mounted to the front axle will be +53°C hotter than cast iron unit.

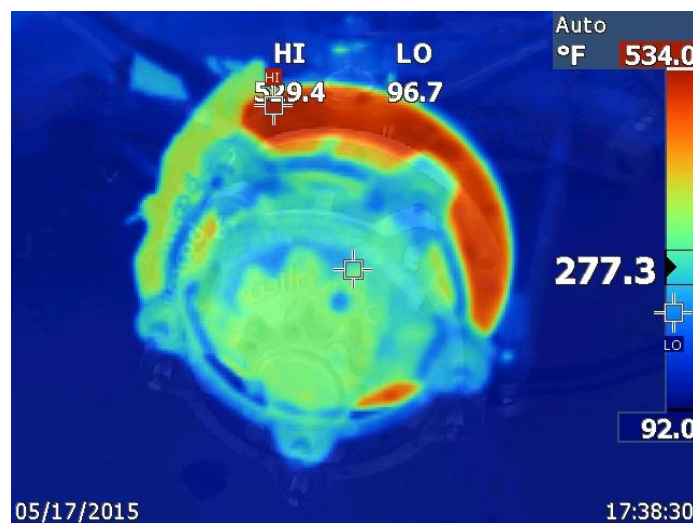


Figure 16: Fluke infrared camera captured image after session 1

Testing was necessary because a Nusslet convection coefficient number is not easily calculable for a partially shrouded, rotating disc. Therefore direct testing of the unit was in order to calculate a convection coefficient for the two dissimilar materials and establish a new ΔT for the remaining thermal energy before the addition at the next braking event.

Failure of fabricated rotors was due to warping on Lap 74 on session 3. On lap 63 of session 3, the driver was instructed to drive more aggressively and brake deeper into deceleration zones. The result was approximately 2.5m further into the braking zone and a +105°C spike in peak temperature, 495°C, was noted in the logged data. 11 laps later, the driver radioed that there

was a pulsation in the pedal and testing was ceased. Post analysis of the offending rotor, both are warped but to varying degrees, indicate that the hub fastener tolerances were too tight given the elevated temperatures experienced. What was given a max clearance of 0.015", standard on the iron rotor, should have been given 0.060" clearance to account for the difference in the coefficients of thermal expansion.



Figure 17: Rotor Condition after 1st Lapping Session

Concerns arose in the selection of the friction face material. Specified was SAE 1080 HR to closely match the metallurgy of a donated rotor in hardness, spark test and grain structure. After producing a complete set of friction faces, it was later revealed that the supplier had not delivered what was specified. Instead they had delivered SAE 1018 CR since it was the spec material was unavailable. Having no choice, the irregularities in mass and density over assumed metallurgy was reevaluated to the delivered material. Nevertheless, the friction faces performed admirably. Both drivers felt confident in the rotor package and were able to drive the car as if a mass-produced brake unit was installed. Final thickness on fabricated rotor was -0.003" from the original dimensional width of 1.253".

APPENDIX G – TESTING DATA

Below are tables of selected lap time throughout the four-hour testing session at Yakima Speedway.

Iron Rotor

Lap #	Entry Speed V_i (mph)	Exit Speed V_f (mph)	Time in Braking Zone (sec)	Accel, a	Cool down time (sec)	Peak Temp (°F)	Low Temp (°F)
1A	99.53	68.11	6.4	-4.91	3.7	583.8	304.5
1B	99.96	65.80	6.4	-5.34	3.5	588.7	312.7
2A	99.28	67.24	6.8	-4.71	3.3	591.6	308.6
2B	98.91	62.45	5.9	-6.18	5.4	601.3	313.3
3A	98.27	62.66	7.0	-5.09	4.3	605.8	315.3
3B	100.08	65.61	6.0	-5.75	4.9	607.8	315.2
4A	100.51	66.15	6.7	-5.13	4.6	608.4	327.7
4B	98.94	66.68	6.1	-5.29	4.5	609.7	313.2
5A	100.36	67.16	5.7	-5.82	5.0	610.0	322.1
5B	99.05	63.90	6.6	-5.33	4.0	615.3	323.3

Average	99.489	65.576	6.36	-5.35	4.32	595.04	315.59
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Composite Rotor

Lap #	Entry Speed V_i (mph)	Exit Speed V_f (mph)	Time in Braking Zone (sec)	Accel, a	Cool down time (sec)	Rotor Peak Temp (°F)	Low Temp (°F)
1A	99.67	57.62	5.7	-7.38	4.9	682.7	228.6
1B	100.36	67.83	5.3	-6.14	5.4	686.4	232.0
2A	99.52	64.31	5.7	-6.18	4.6	693.1	226.1
2B	99.31	49.60	4.8	-10.36	4.7	703.5	229.4
3A	99.06	62.22	5.6	-6.58	5.0	710.7	232.1
3B	99.16	67.73	5.8	-5.42	4.3	716.3	233.6
4A	100.35	65.72	5.8	-5.97	4.2	725.9	235.9
4B	99.08	51.98	5.2	-9.06	5.6	728.4	232.9
5A	100.28	67.65	5.7	-5.72	4.6	740.6	246.5
5B	100.11	64.83	5.4	-6.53	4.7	764.7	257.7

Average	99.69	61.949	5.5	-6.93	4.8	715.23	237.48
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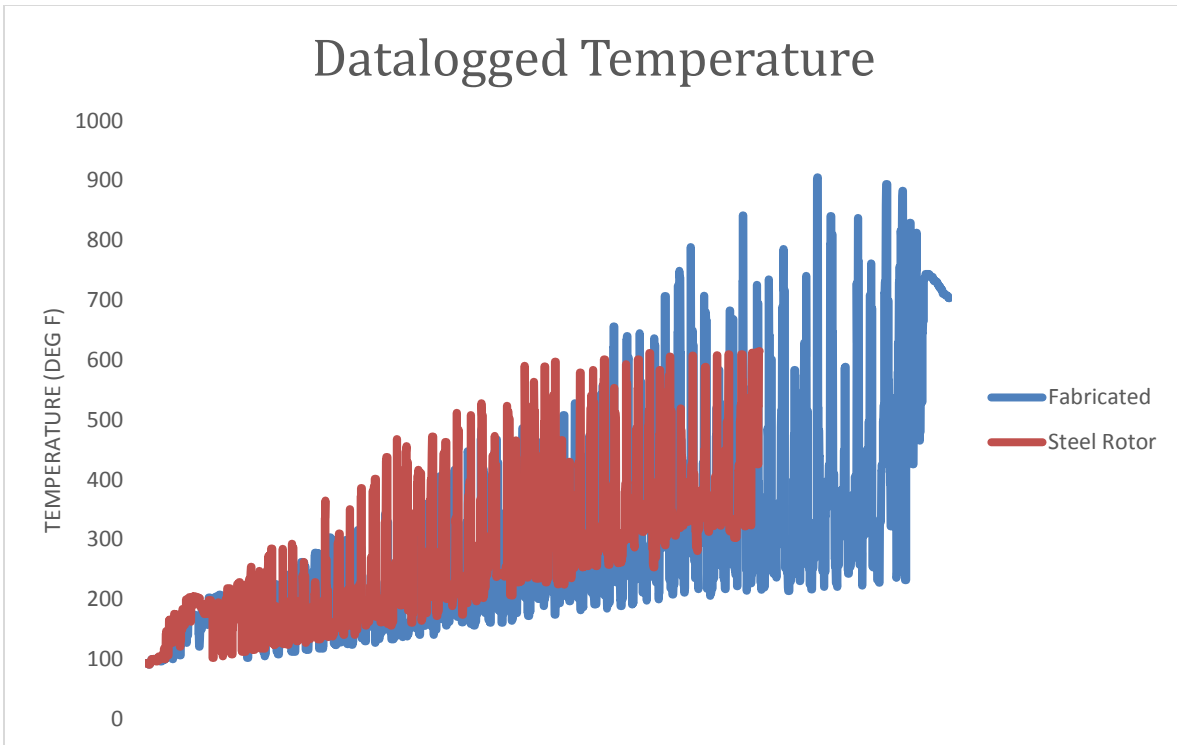


Figure 18: Raw Data of 2nd Iron rotor session vs. 3rd Composite rotor session

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RESUME

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SUMMARY

Skills in fabrication, welding, machining, simulation and analysis.
Experience in structural fabrication and mechanical design.
Diagnosis and repair of mechanical, electrical and hydraulic systems
Highly motivated, disciplined, and resourceful.
Productive interaction with people of varied experience levels

EDUCATION

2010-Present

Central Washington University

Mechanical Engineering Current GPA: 3.49

Manufacturing Specialization

Class Level: Post-Baccalaureate Senior

Mechanical Design

Project Cost Analysis

Finite Element Analysis

Fluid Dynamics

Strength of Materials

Hydraulics and Pneumatics

Manufacturing Processes

CNC Programming

Lean Manufacturing

Tool

Design

2004

Central Washington University

Bachelor of Arts Biology

GPA: 3.06

Chemistry Minor

1996

West Valley High School

GPA: 3.24

EXPERIENCE

- Composite lightweight brake rotor design for Senior Project
- Failure analysis of forklift mast assembly to determine manufacturer quality control errors
- Performed modifications to the department's portable casting unit to increase heat absorption capacity.
- Designed, prototyped and implemented modifications to increase fatigue strength of temperature probes.
- Sheetmetal design and fabrication for a local motorsports dealership.
- Machine design and tool fabrication for a local automotive repair company.

TECHNICAL SKILLS

CATIA V5, SolidWorks (CWSA 2011), AutoCAD, BobCAD, MDesign, Mastercam X8, Microsoft Office, JAVA, Machining (Mill, Lathe, CNC operations), Composites layup, Sheet metal fabrication, Welding (MIG, TIG, Oxy-Acetylene, spot, etc.), Basic electronic design/fabrication,

AFFILIATIONS

(2010-Present) ASME CWU Chapter (2011-Present) Yakima SolidWorks Users Group

(2011-Present) SME CWU Chapter (2012-Present) Yakima Tool Share