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Finite Element and Analytical Investigation of a Go- Kart Braking System

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Abstract - *Go-karts are open wheeled racing kart, which require efficient, and effective braking systems to guarantee the safety of the driver and the controllability of the vehicle. Due to their light weight, moderate operating speed, and lack of any kind of suspension mechanisms, the rear braking designs are widely embraced to reduce the complexity of a system, low cost and weight. This study is an analytical and finite element study of a rear-mounted hydraulic disc braking system that would be used in a go-kart and on a dry track. Braking system is also analysed with stopping distance, braking time, pedal force, hydraulic pressure, braking force, transfer of weight dynamically, and also with thermal loading. The choice of the material to be used in making the brake disc and pedal is done depending on the mechanical and thermal performance requirements. The Finite Element Analysis (FEA) is used to confirm the analytical predictions by structural and thermal modelling of the brake disc and structural analysis of the brake pedal. The analysis indicates that there is a high level of agreement between the analytical and numerical results and the highest level of deviation is under 5% in the maximum temperature. The braking system shows sufficient factor of safety in structural integrity and thermal overload and is thus suitable in a safe and effective operation of the go-karts. The research gives an approved design methodology, which can be applied as a reference to the development of go-kart braking systems in the future for student events.*

Key Words*: *Go-kart braking system, hydraulic disc brake, finite element analysis, thermal analysis, braking dynamics.

1. INTRODUCTION

Go-karts are low-weight racing cars, but they are mainly used in recreational and competitive motorsport purposes, and simplicity, low price, and performance are the important design factors. In comparison to traditional cars, go-karts usually have a moderate speed, and do not possess complicated systems of suspensions. Therefore, the braking needs of go-karts are varied as compared to full size vehicles. It is common to find most of commercial and competition go-karts use rear-only braking systems since they offer the necessary level of stopping force with the lowest level of system complexity, total weighting, and cost of production [1]. Nevertheless, even being seemingly straightforward, rear-mounted braking systems have to be thoroughly designed and analytically tested in order to achieve sufficient performance, structural integrity, and thermal safety.

Every go-kart braking is determined by the mass, velocity, coefficient of friction of tires and the surface of road, and weight transfer in braking. On dry surfaces, there is sufficient friction to enable braking to be carried out adequately by using a single rear disc without adversely affecting stability [2]. Because go-karts have low center of gravity and short wheelbase, in case of over-braking on the front, the wheels might lock or become unstable. Rear-only braking as such thus offers a fair trade-off between safety and control especially in the entry level and amateur racing use.

The paper is dedicated to the analytical research and finite element confirmation of the rear-mounted hydraulic disc braking system of a go-kart that works in dry conditions on the road. The analysis method commences with the calculation of braking force which is based on the principles of vehicle dynamics. The basic requirements include vehicle mass, initial velocity, desired deceleration, tire, and road coefficient of friction to determine the required rear axle braking torque. These are computed to determine the minimum performance of the braking system [3].

Hydraulic analysis is important in providing assurance that the force exerted on the pedal is transformed to be enough clamping force in the brake calliper. The hydraulic braking system comprises of a master cylinder, brake lines and calliper assembly. The analytical assessment of the relationship between pedal force, master cylinder pressure, and calliper piston force using the Pascal law is done. Adequate sizing of the master cylinder and calliper pistons helps to achieve the right balance of braking power and pedal force without too much effort on the pedal which is essential to the comfort and safety of drivers [4].

The other critical feature of braking system design is thermal analysis because the kinetic energy is transferred to heat during braking. Over a rear mounted disc brake, the heat is dissipated by the brake disc and air around it. Analytical thermal model is used to approximate the increase in temperature during repeated instances of braking. Higher temperatures may result in wear and tear of the brake, material wear or breakages. Thus, the thermal capacity of

the disc material and the ability to evaporate the heat is thoroughly considered so as to maintain a stable performance at the racing conditions [5].

In order to confirm the results of the analytical study, finite element analysis (FEA) is conducted on the most important scientific elements of the braking system, in this case, brake disc and calliper. Structural analysis is done to evaluate stress distribution and deformation at maximum braking loads. The outcomes assist in confirming the fact that the stresses are not beyond permissible material levels, which guarantee the structural integrity and longevity. Thermal FEA is an additional tool of prediction made through analytical analysis by design that simulates the temperature distribution and heat transfer in the brake disc during braking cycles.

Finally, the combination of braking dynamics, hydraulic analysis, thermal modelling and finite element validation would give a complete frame of assessment of rear-mounted hydraulic disc braking systems in go-karts. The results validate that in reducing braking performance to be safe and effective on dry roads; rear-only braking can be either designed or validated to be effective.

2. DESIGN INPUTS AND ASSUMPTIONS

The braking system is developed to suit a go-kart that has a total mass of 170 kg including the driver. The assumed tyre-road friction coefficient is 0.9 whereas the pad-rotor friction coefficient is 0.4. The worst-case scenario is considered to be the greatest vehicle speed of 16.67 m/s and on that basis the braking performance is considered.

The braking system was designed for a lightweight go-kart employing a rear-only braking configuration, intended for operation under dry track conditions. A rear-only braking system was selected to reduce system complexity and overall mass, which is suitable for low-speed go-kart applications where simplicity and steering stability are prioritized.

The coefficient of tyre–road friction was assumed to be 0.9 and was used for braking force calculations. The maximum speed of the go-kart was 16.67 m/s (60 km/h). At this speed, the estimated stopping distance was approximately 15.7 m. At a reduced speed of 8.33 m/s (30 km/h), the stopping distance decreased to about 3.9 m, with a braking time of nearly 1 second.

The braking system was primarily designed considering the worst-case scenario, i.e., braking from the maximum vehicle speed of 16.67 m/s on a dry track surface. The total mass of the system, including the go-kart and the driver, was taken as 170 kg, where the mass of the go-kart was 110 kg and the driver mass was 60 kg.

Accordingly, the total weight acting on the vehicle during braking was calculated as:

$$\text{Weight} = m \times g$$
$$= 170 \times 9.81 = 1667.70\ \text{N}$$

This total weight was used for evaluating braking forces, stopping distance, and stability during deceleration. Vehicle design parameters such as wheelbase and center of gravity height were considered and validated through analytical evaluation.

Nomenclature:

- g – Acceleration due to gravity (m/s^2)
- μ – Friction coefficient between road and tyre
- u – Initial speed (m/s)
- v – Final speed (m/s)
- h – Height of centre of gravity (m)
- m – Total mass (vehicle & driver) (kg)
- L – Wheel base (m)
- t – Braking time (s)
- a – Maximum achievable deceleration (m/s^2)
- Q – Heat generation rate (W)
- Cp – Specific heat capacity (J/kg-K)
- ΔW – Dynamic weight transfer (N)

Table -1: Design Inputs and Parameters

Parameter	Value
Total mass (vehicle + driver)	170kg
Weight Distribution (Front: Rear)	40:60
Wheelbase	1.3716m
Height of CG	0.298m
Tyre road friction coefficient	0.9
Pad disc friction coefficient	0.4
Pedal effort	450N
Pedal Ratio	4:1

2.1 The analysis of the braking performance is conducted.

Analysis of braking performance is a necessary element of determining the ability to stop and safety of a go-kart when

under working conditions. In the current research, it is assumed that the vehicle attains a speed of 16.67 m/s on a dry road. The maximum possible deceleration is determined by taking the tire road friction and vehicle mass as 8.83 m/s 2. This is in a reasonable braking range of dry asphalt and it means that the tires are able to efficiently transmit braking force without early wheel lock-off [6]. The minimum theoretical stopping distance has been calculated using classical kinematic relations and it is 15.74 m, with a braking time of 1.88 s. These findings indicate that the braking system can be used to offer fast deceleration to be applied in competitive go-kart uses.

The fact that the stopping distance was calculated is especially meaningful because there were no suspension systems and electronic braking systems in go-karts. Mechanical design and adequate distribution of forces are important in the realization of efficient braking performance [7]. Moreover, it is important to keep the level of braking within the scope of friction in order to avoid frictional skidding and the loss of directional stability. The analytical findings, in general, validate the effectiveness of the chosen braking parameters that can be used to offer effective and controllable deceleration of a lightweight rear-braking dominated car [8].

2.2 Hydraulic and Braking Torque Study.

The hydraulic analysis of the braking system is done to make sure that the effort exerted by the driver on the pedal is efficiently turned into the area of an adequate amount of braking force at the rear disc. This experiment produces a force of 450 N at the pedal and a 4:1 pedal ratio which produces an amplified force that is passed onto the master cylinder.

1. Pedal force

   = Pedal effort x Pedal ratio

   = 450 x 4

   = 1800 N

2. Brake line pressure

   = Pedal force/Area of master cylinder

   = 1800/ 0.000506707

   =3552348.79 Pa

3. Clamping force (Two piston cylinder)
   = Brake line pressure x Area of calliper piston x 2

   =3552348.79 x 0.000706 x 2

   = 5015.91 N

4. **Brake torque developed**

   = Clamping force x Effective rotor radius x Coefficient of friction between pad and rotor

   = 5015.91 x 0.0675 x 0.4

   =135.43 N-m

The pressure of the brake line is determined by application of the hydraulic principles that are based on the Pascal law to be 3.55 MPa. This pressure applies a total clamping force on the calliper pistons to create a total of 5015.91 N on the brake disc [9]. When the resulting clamping force is multiplied by the effective disc radius and coefficient of friction, the resultant braking torque is obtained at the rear axle, 135.43 N m.

5. **Brake torque required**

   = Dynamic weight on rear axle x rolling radius of tyre x μ

   = 674.48 x 0.1397 x 0.9

   = 84.80 N-m

6. **Factor of safety (disc)**
   =Brake torque developed / Brake torque required
   = 135.43 / 84.80
   = 1.60

Rolling radius was taken from standard go-kart tyre specification. The braking dynamics as well as the effective rear axle load give the number 84.80 N-m, which is the required braking torque. Division of the available torque with the required torque gives a factor of safety of 1.60, which means that the braking system has plenty other capacity in the event of dynamic braking [10]. Such safety margin is critical to take into consideration the differences in the friction, thermal and wear of the parts. Hydraulic and torque analysis will assure the fact that the chosen brake parts are properly sized and can provide the adequate performance of the braking system [11].

2.3 Weight Transfer Analysis

Distribution of weight during braking has a major effect on the efficiency of braking especially in a car like go-kart that is not equipped with a suspension system. During the deceleration, the inertial force causes a part of weight of the vehicle resting on the rear axle to shift to the front axle. In the analysis, the weight transfer during braking is dynamic and is determined to be 326.14 N in the present analysis.

Consequently, the rear axle load decreases to 674.48 N as compared to its static value. This decrease in rear axle load has a direct impact on maximum braking force that can be successfully used without locking wheels [12].

7. Static weight on front axle

=0.4 x m x g= 0.4 x 170 x 9.81

= 667.08 N

8. Static weight on rear axle=0.6 x m x g

= 0.6 x 170 x 9.81

= 1000.62 N

9. Dynamic weight transfer
= (m x a x h)/ L

= (170 x 8.83 x 0.298) / 1.3716

= 326.14 N

10. Dynamic weight on front axle

= Static weight on front axle + ΔW

= 667.08 + 326.14

= 993.22 N

11. Dynamic weight on rear axle

= Static weight on rear axle - ΔW

= 1000.62 – 326.14

= 674.48 N

Go-kart has a rear-only brake system which means that the dynamic rear axle load accuracy is extremely important to predict the necessary braking torque. We use the lower rear load as a basis on which braking force and torque requirements are calculated. Under consideration of weight transfer may lead to enhanced braking capacity and undermined stability [13]. As shown by the analysis, the braking system can provide sufficient torque with a satisfactory factor of safety even when the rear axle load is less than the nominal. The inclusion of the weight transfer effects therefore makes the design of the braking system a realistic and conservative one that would be applicable in high-deceleration conditions.

3. ANALYTICAL BRAKING CALCULATIONS

Analytical calculations are carried out to estimate the performance parameters of braking performance such as deceleration, stopping distance, braking time, hydraulic pressure, braking torque and dynamic weight transfer. These computations are the basis of the design of the braking system and make sure that the chosen elements work within the safe operating mechanical and thermal ranges under the worst-case operating conditions.

The highest possible deceleration obtained is 8.83 m/s^2 and the minimum stopping distance is 15.74 m with a braking time of 1.88 s. These values are controlled mainly by the tyre to road friction coefficient and are the theoretical maximum performance of the vehicle in dry track conditions. The stopping distance both proves the fact that the braking system is capable of slowing the vehicle down safely under the normal conditions of the go-kart track, and the braking time guarantees that the driver will have enough time to respond and control the vehicle in the event of emergency braking.

Hydraulic analysis means that a pedal force of 1800 N produces a brake line pressure of 3.55 MPa, and this produces a clamping force of 5015.91 N at the calliper. This hydraulic amplification of the pedal ratio and master cylinder sizing gives assured development of sufficient braking force without the driver putting too much effort on the brake pedal. This driver ergonomic and braking balance is also crucial in the case of racing, where too much driver fatigue might arise with repeated cycles of braking when the forces on the pedals are too strong.

The braking force generated in the rear disc is 135.43 Nm, which is more than the necessary braking force of 84.80 Nm and this gives a factor of safety of 1.60. This margin helps to provide consistent braking behaviour in the transient condition like small surface irregularities, friction change with temperature, and part wear. It is necessary to ensure that a factor of safety exceeding unity is maintained so that brake fade or torque deficiency does not occur even when it is required to sustain more than the normal braking under extended or strenuous braking conditions.

Transfer of weight during braking is dynamic and results in a reduced rear axle load of 674.48 N that is put into the design to provide a conservative design. Dynamic load redistribution, especially in the rear-only braking, should also be taken into consideration because the unloading of the rear axle may exceed the maximum traction. The braking system is developed to accommodate dynamic effects in the analytical model and hence to drive safely in real-life conditions as opposed to an idealized model based on a minimal set of assumptions.

3.1 Braking Performance Analysis

Go-kart braking is one of the most important factors in the overall safety, control and adherence to the requirements of competitions. In lightweight cars like the go-kart, the tyre-road friction is the main factor that controls the performance

of the brakes, mass of the car, weight distribution, and the efficiency of the braking system. Analytical assessment offers a credible way of forecasting braking performance in an idealized situation and as a reference point in the further numerical verification [14].

In the current analysis, the performance of braking is measured by measuring the most critical parameters that are maximum deceleration that could be achieved, stopping distance, braking time and braking torque. The greatest deceleration is constrained by the tyre-road friction and it has a direct proportional effect on the shortest stopping distance. In dry track conditions, the deceleration calculated shows that the braking system is going to operate near the frictional limit ensuring that the available traction is fully utilized without causing a wheel lock-up situation. The associated stopping distance and braking time are indicative of the fact that the system is capable of decelerating the vehicle in a safe manner with regards to racing and recreational usage [15].

The hydraulic braking performance is studied through the analysis of the pedal effort, hydraulic pressure generation and the calliper clamping force. Hydraulic amplification is adequate, making sure that there is enough braking torque produced with easy driver input, this is required in order to be consistent in repeated braking cycles. Moreover, the braking torque is compared to the rear axle load which is dynamically reduced to consider the effects of weight transfer during deceleration. This has the benefit of conserving design and avoiding excessively claiming the capability of braking in rear only designs.

The transfer of weight through dynamic would have major impact on braking performance as it decreases normal load on the rear axle hence limiting traction available. This effect is important to add to the analytical model to increase the braking torque estimation and increase the reliability of the system. On the whole, the analytical assessment of braking performance demonstrates that the developed braking system addresses the requirement on safety, efficiency, and performance, which is the strong basis of the finite element validation and experimental implementation [16].

The maximum achievable deceleration is calculated using the tire–road friction coefficient:

$$a = \mu g = 0.9 \times 9.81 = 8.83 m/s2$$

The minimum theoretical stopping distance is:

$$s = \frac{v^2}{2a} = \frac{16.67^2}{2 \times 8.83} = 15.74m$$

The corresponding braking time is:

$$t = \frac{v}{a} = \frac{16.67}{8.83} = 1.88s$$

These values confirm effective braking performance within frictional limits.

3.2 Hydraulic and Braking Torque Analysis

The hydraulic braking system is critical in transforming the activity of drivers into effective force of braking at the wheel. When used in go-karts, the system should have enough braking power as well as easy, reliable, and lightweight. Hydraulic analysis is based on the analysis of pedal force transmission, formation of brake line pressure, calliper clamping force, and the braking torque obtained to guarantee safe and efficient working conditions [17].

The driver increases the force applied by the pedals through the pedal ratio that is applied to the master cylinder to create hydraulic pressure in the brake lines. To obtain the best relationship between the pedal motion and the braking, it is important to have a proper size of the master cylinder and the calliper piston regions. Sufficient hydraulic pressure also gives uniform calliper clamping force, and frictional contact between the brake pads and disc. Overpressure can cause wear and less modulation of the components and under pressure can affect the effectiveness of the braking.

The braking torque is produced as a result of the force of clamping the calliper, the radius of the disc, and also the coefficient of friction between the rotor and the brake pad. The calculated braking torque should be greater than the amount needed to decelerate the vehicle under the dynamic loading condition and in the rear braking system only the weight transfer towards the rear decreases the rear axle traction. Measuring braking torque at dynamically reduced normal load provides conservative design, and eliminates the possibility of wheel lock-up or brake fade when using aggressive braking manoeuvres.

In general, the analysis of hydraulic and braking torque proves that the designed system is capable of ensuring adequate braking force with a sufficient safety margin, which will allow it to address the issue of vehicle deceleration and driver control when operating in adverse conditions [18].

3.3 Weight Transfer Analysis

Weight shift during braking is a vital dynamic event which plays a very important role in vehicle stability and braking performance particularly in low-weight cars like the go-karts. During the deceleration, the inertia causes a shift in the forward load which results in the increased normal force on the front axle and the decrease of the load on the rear axle. This repositioning of the vertical load has a direct impact on the traction available at the wheels and would have to be thought out in the design of braking systems [19].

With rear-only systems the weight transfer is particularly critical to evaluate, since an excessive amount of unloading of the back axle can restrict the amount of braking force available and substantially raise the chances of wheel lock-up. The analysis of weight transfer is based on the analytical way and taken into account the vehicle mass, deceleration, centre of gravity height, and wheelbase. A combination of these parameters will define the extent of load transfer during braking which is dynamic. Dynamic effects will be considered to make sure that the braking torque calculations are not made by assuming a static load.

The decrease in axial load of the rear when braking happens affects directly the maximum possible braking force since the friction between tyres and roads increases with the normal load. Through consideration of this decrease, the braking system can be designed in a conservative manner to work within the traction limits during extreme deceleration. This helps increase the controllability of the vehicle, as well as reducing instability that occurs during panic braking or high-speed deceleration situations.

Altogether, weight transfer analysis is a crucial source of information on the dynamics of braking and is a crucial factor in achieving safe and predictable braking performance. Weight transfer considerations in the design of the analysis enhance the trustworthiness of braking systems mounted on the rear and aid in verifying the braking torque and safety margins by the means of the numerical analysis and experiments [20].

Dynamic weight transfer during braking is given by:

$$\Delta W = \frac{mah}{L} = 326.14N$$

The reduced rear axle load is used to compute the braking torque requirement, ensuring realistic and conservative design conditions.

4. THERMAL ANALYSIS

The thermal analysis of the braking system is necessary to test the capability of the brake disc to safely release the heat produced during braking without undergoing thermal failure and excessive brake life. When the vehicle is decelerating, a large percentage of kinetic energy is converted to thermal energy through the friction of the brake pads on the disc. When the brake disc is used in lightweight vehicles (e.g. go-karts) high transient heat fluxes are experienced due to the low mass and area of the surface.

The thermal analysis is conducted in the current study with the assumption that the significant proportion of the kinetic energy of the vehicle is converted into heat at the brake disc during a single braking event. The heat rate is calculated depending on the mass of vehicle, the initial velocity and the braking time. The heat flux on the disc pad contacting surface is then determined and applied in estimating temperature increase of the disc material. These are the calculations that give a first estimate of the maximum operating temperature when the braking is severe.

The expected temperature increment is then cumulated to the ambient temperature in order to estimate the final disc temperature. This value is a critical input in the assessment of the suitability of materials, as well as in defining the thermal boundary conditions to a finite element calculation. It is necessary to ensure that the highest disc temperature is kept at a reasonable level to ensure the same friction properties, to eliminate the possibility of material degradation, and to avoid thermal cracking.

All in all, the analytical thermal analysis provides information that is useful in the understanding the thermal behaviour of the braking system and forms a foundation of the numerical validation of the thermal behaviour through a finite element thermal analysis, therefore, being able to ensure safe and reliable braking behaviour.

The kinetic energy of the go-kart is:

1. **Kinetic energy**

   $=1/2 \times m \times (\text{initial speed})^2$

   $= 0.5 \times 170 \times 16.67 \times 16.67$

   $= 23620.56$ J

2. **Heat generation rate, Q**

   90% Kinetic energy is converted into heat,

   = (0.9 x Kinetic energy) / t

   = 0.9 x 23620.56/1.88

   =11307.71 W

3. **Heat flux**

   =Q/Surface area of disc in contact with pad

   $=Q / [2 \times pi \times (0.0825^2 - 0.0525^2)]$

   $=11307.71 / [2 \times pi \times (0.0825^2 - 0.0525^2)]$

   $= 444364.93$ W/m^2

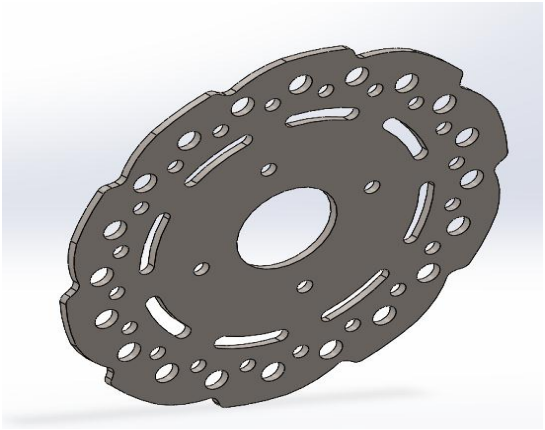

Fig.1: Brake disc geometry

5. MATERIAL SELECTION

The choice of material used in the brake disc is an important consideration when designing a braking system since the disc is prone to repeated mechanical loads, high contact pressure and harsh thermal treatments during braking. The material of brake disc should have high hardness, wear resistance, high thermal conductivity, and enough tensile strength in high temperature to guarantee the reliability of braking performance and service life [21]. This is what causes them to be less so that they can be used in braking processes in which consistency of friction and resistance to surface degradation are essential. Moreover, austenitic grades are also more prone to loss of friction stability at higher temperatures and this may have an undesirable impact on constant braking [22].

5.1 Brake Disc Material

SS 410 was also chosen as the brake disc material compared to SS 303, SS 304 and SS 321 because it has high hardness, wear resistance and the strength at high temperature. Although austenitic grades (SS 303, SS 304, and SS 321) are good in corrosion resistance, they cannot be used in braking due to low friction stability and wear properties. Martensitic structure of SS 410 also makes it more appropriate in high-friction and thermally loaded brake discs [23].

5.1.1 Frictional Heat Generation Calculation

Frictional heat generation

= μ × sliding velocity of disc surface × t × clamping force

The disc is exposed to excessive contact stresses and quick changes in temperatures many times in braking systems due to the frictional heat generation. In these circumstances, low hardness and low wear resistance materials are prone to wear degradation on the surface being used and this causes braking performance that is not consistent and shortened service life. Austenitic stainless steels, this is a ductile and corrosion resistant material that typically has a lower hardness, and is susceptible to adhesive wear and surface glazing with repeated braking. These impacts have the ability of decreasing the coefficient of friction and undermining the braking performance [24].

5.1.2 Temperature Rise in Disc

$$\Delta T = \frac{Q \cdot t}{m \cdot c_p}$$

Where:

- m= mass of brake disc (kg)
- c_p= specific heat capacity of SS 410

4. **Rise in temperature**

   = (Q * t) / (Mass of rotor * C_p)

   = (11307.71 * 1.88) / (0.41319*500)]

   = 102.90 K

5. **Final disc temperature**

   = Rise in temperature + ambient temperature

   = 102.90 + 303.15

   =406.05 K

Thermal analysis is conducted assuming that 90% of the vehicle's kinetic energy is converted into heat at the brake disc. The total kinetic energy at maximum speed is calculated as 23.62 kJ. The resulting heat generation rate is 11.31 kW, producing a heat flux of 444,364.93 W/m^2.

The analytically predicted temperature rise of the disc is 102.9 K, resulting in a final disc temperature of approximately 406 K. These values are used as boundary conditions for finite element thermal analysis.

Assuming 90% conversion to heat, the heat generation rate is 11.31 kW. The analytical temperature rise of the brake disc is calculated as 102.9 K, resulting in a final disc temperature of approximately 406.05 K.

Purpose:

- Predict how much the disc heats up during braking.
- Compare SS 410 vs austenitic steels: higher thermal conductivity and mechanical stability of SS 410 → lower ΔT → better braking performance.

SS 410 has a higher level of mechanical strength and hardness that is provided by martensitic microstructure that increases resistance to wear and surface deformation. Also, SS 410 has high thermal conductivity compared to austenitic grades, and thus it dissipates more heat produced in braking. This feature aids in curbing overheating and minimises

chances of overheating and cracking. Moreover, SS 410 is mechanically stable in high temperature and thus it does not fail to hold its structure during aggressive or repeated braking. These properties combined together make SS 410 an efficient and economical material of choice in the brake discs of such lightweight and performance-based applications as go-karts [25].

Table 2: Properties of Brake Disc Materials

Material	Density (kg/m^3)	Tensile strength (ultimate)(MPa)	Thermal conductivity (W/m-c)	Specific heat capacity (J/kg-K)
SS303	7830	620	16.3	500
SS304	8000	515	16.2	530
SS321	8027	586	16.1	530
SS410	7750	840	24.9	500

Interpretation:

The relative material information shows that there are apparent variations in mechanical and thermal appropriateness in use as brakes discs. SS303, SS304 and SS321 have close tensile strength and low thermal conductivity and thus, poor capability to cool down heat when braking repeatedly. The austenitic stainless steels are not very resistant to wear and surface deformation at high contact stresses because they have moderate yield and ultimate strengths, although they have high corrosion resistance. Instead, SS410 has the highest yield and final tensile strength (840 MPa), which means that it has better resistance to mechanical loading and wear. Moreover, SS410 has much greater thermal conductivity (24.9 W/m-K), and thus, the disc can dissipate heat much more efficiently and experience fewer thermal gradients. Its thermal performance is also stable due to its similar specific heat capacity. In general, the information proves that SS410 is the best combination of strength, wear resistance, and thermal efficiency, so it is the most appropriate among the applicants to a thermally loaded and mechanically loaded brake disc.

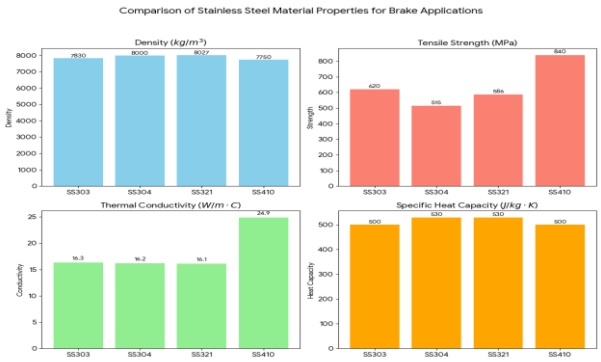

Fig.2: Comparison of disc material properties

5.2 Brake Pedal Material

The brake pedal is a structurally important part of the braking system, as it directly converts the driver input into the hydraulic circuit and is exposed to the repeated bending and compressive loads in the process of operating. The choice of materials used on the brake pedal should then be based on the need to have high yield strength, stiffness, fatigue, and durability that will provide safe and reliable service during the life period of the vehicle [26].

AISI 4130 alloy steel was chosen to be the brake pedal as an alloy in the current research when compared to 6065-T6/T8 alloy aluminium and AISI 1018 steel. Despite the low density of aluminium alloys, relative low density has a disadvantage in the form of low yield and ultimate tensile strengths, which restrict their application to high loads braking. Aluminium pedals can be over-deformed in extreme or panic brake situations and this reduces pedal feel and driver control. Correspondingly, although AISI 1018 steel has sufficient manufacturability and relatively good strength, its lower yield strength value gives out reduced safety margin to repetitive high-stress loading [27].

AISI 4130 alloy steel has very larger yield and ultimate tensile strengths with a high modulus of elasticity leading to better stiffness and least elastic deformation of the material under the influence. It has a better fatigue resistance especially its usefulness in braking components which are subjected to a cyclic loading. Also, AISI 4130 is also mechanically stable in order to be used in different loading conditions with the same pedal response and greater driver confidence. All these characteristics enable AISI 4130 as a strong and dependable material option to use in a structurally sensitive brake pedal exercise in high performance cars like go-karts [28].

Table 3: Mechanical properties of brake pedal materials

Material	Density (kg/m^3)	Young's modulus (GPa)	Ultimate Tensile Strength (MPa).
6065-T6	2800	68	310
6065-T8	2800	68	390
AISI1018	7870	205	440
AISI4130	7850	210	832.62

The comparison of the materials shows that there are serious differences in mechanical performance which are applicable in the use of brake pedals. Aluminium alloys 6065-T6 and 6065-T8 are low density and moderate strength materials that can be used in lightweight design; but with a relatively low Young's modulus (68 GPa), they have poor stiffness properties and therefore high levels of elastic

deformation when under braking forces. Despite the fact that 6065-T8 is stronger than 6065-T6 with respect to strength, its yield and ultimate strengths are considerably lower when compared to that of steel alloys. AISI 1018 steel is much more rigid and has greater strength although its yield strength is relatively low so that there is not so much safety margin related to the high load cyclic braking. On the contrary, AISI 4130 exhibits the greatest yield and ultimate tensile strength and still has high modulus of elasticity, which guarantees low levels of deformation, high load-carrying ability as well as increased fatigue resistance. All these properties allow AISI 4130 to be the most appropriate material to a structurally important brake pedal [28].

6. FINITE ELEMENT ANALYSIS

The analytical design is checked with finite element simulations that are performed with calculated braking loads and thermal input to examine the structural and thermal integrity of the important parts of braking. Finite element method allows the stress distribution, deformation and temperature gradients to be evaluated in a detailed manner which would otherwise be impossible to evaluate within the simplified cases of analytical models.

In this paper three dimensional models of both brake disc and brake pedal are constructed and their discretization done with suitable mesh densities to guarantee accuracy of the solution as well as maintain efficiency in computation. The properties of the material to be used in the experiment are allocated to the chosen disc and pedal materials in accordance with experimentally reported values. Boundary conditions are used to make replications of real operating conditions such as fixed mounting areas, applied braking forces, and thermal heat flux in consequence of analytical minimization. The interaction between the disc and the brake pad in contact is idealized to give the idea of frictional heat generation in the process of braking.

Thermal analysis is conducted to measure peak temperature conditions throughout the disc surface and structural assessment is carried out to measure induced stresses and deformations under braking loads. Finally, the finite element outcomes are then compared with analytical predictions to determine correlation and confirm the assumption of the analytical modelling. Such analytical-numerical methodology is a sure way to provide good performance evaluation and verify the severity and strength of the braking system in extreme braking conditions.

6.1 Brake Disc Analysis

Thermal analysis shows a maximum disc temperature of 387.38 K which is close to the analytical values. The maximum von Mises stress obtained by structural analysis is 340.14 MPa and the factor of safety is 2.24 and the deformation obtained is very low.

The result of a thermal analysis by using the finite element method is the temperature distribution which shows the peak values to be concentrated in the pad-disc contact area where the heat generation due to friction is greatest. The smoothness of temperature distribution in the disc thickness indicates that the material has a high ability to conduct heat, and thus it is unlikely to have hot spots on one hand. It is found that the maximum temperature that is predicted is far below the critical temperature limits of SS410, so no friction behaviour is unstable and a risk to thermal distortion or brake fade during extreme braking events is limited.

Structural analysis shows that the stress concentrations are mainly found in and around the mounting holes and hub interface which are of importance because of the limitations of loads transmission. Although these concentrations of stress are localized, they will only reach a very low value of the Von Mises stress which is well below the yield strength of SS410. The low deformation suggests that the disc is dimensionally stable under peak loading braking forces, retains effective pad contact and has uniform braking performance. In general, the findings of the finite element analysis indicate that the brake disc design is structurally and thermally sound and supports the assumptions of the analytical design and illustrates the sufficient safety margins to operate go-karts in reality.

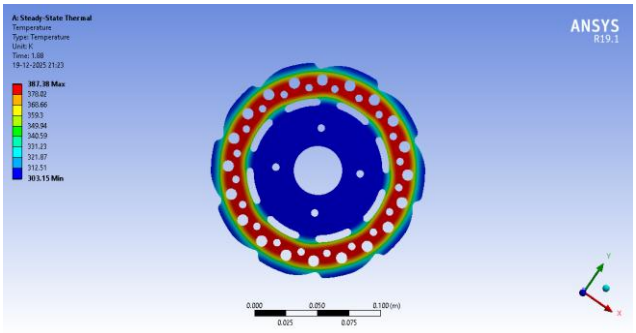

Fig. 3: Brake disc temperature distribution

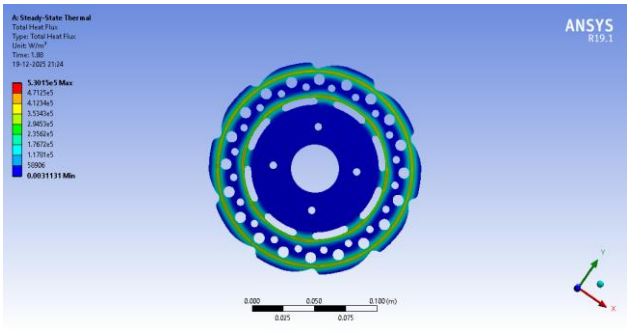

Fig.4: Brake disc total heat flux

Specifically, it visualizes the Total Heat Flux across the rotor's geometry. Based on the simulation data shown, here is a suggested title and a breakdown of what the results indicate.

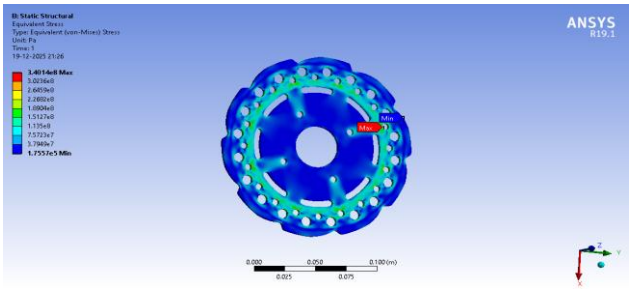

Fig.5: Brake disc von mises stress distribution

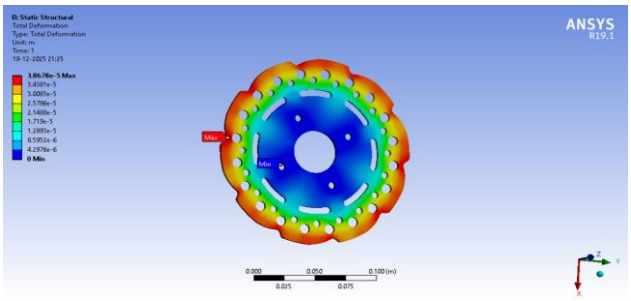

Fig.6: Brake disc total deformation

The image shows a Static Structural Analysis of the same perforated petal-type brake disc using ANSYS R19.1. While the first image focused on heat flow, this one visualizes the physical response of the material to mechanical and thermal loads.

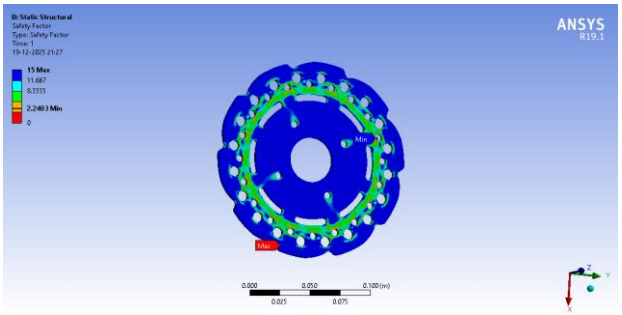

Fig.7: Brake disc factor of safety

6.2 Brake Pedal Analysis

Structural analysis of the brake pedal indicates a maximum Von Mises stress of 240.82 MPa with a factor of safety of 1.91.

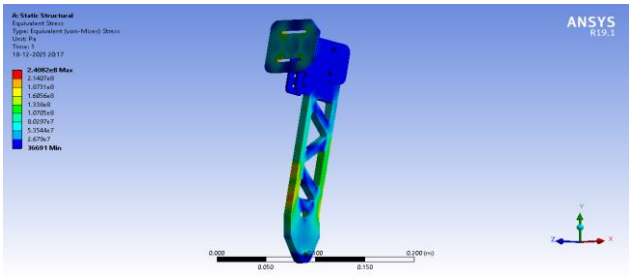

Fig.8: Brake pedal stress distribution

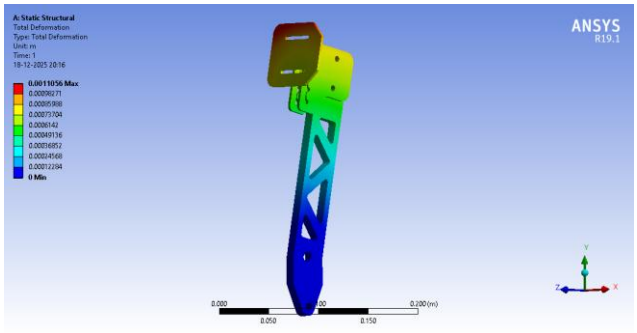

Fig.9: Brake pedal total deformation

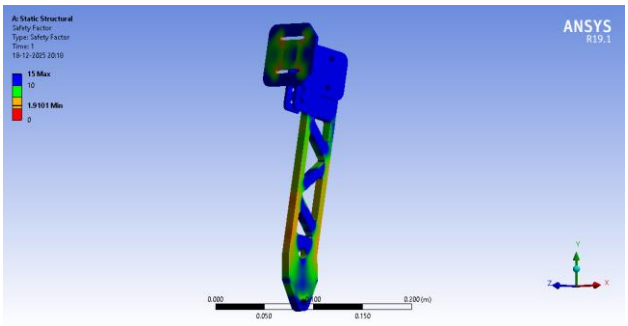

Fig.10: Brake pedal factor of safety

6.2.1 Analysis Results

Finite element and thermal analysis of the brake disc under the same braking conditions showed that SS 410 experiences a maximum surface temperature of around 387.38 K, with contact stresses well below the material's yield strength. In contrast, austenitic grades (SS 303, SS 304, and SS 321) under identical conditions showed slightly higher surface temperatures due to lower thermal conductivity, and the maximum von-Mises stress approached their lower yield limits. The martensitic structure of SS 410 provided higher hardness and wear resistance, reducing surface deformation and ensuring consistent friction performance. Thermal simulations also confirmed better heat dissipation in SS 410, minimizing the risk of thermal cracking during repeated braking cycles.

6.2.2 Comparison and Conclusion

The analytical calculation and simulation results align closely, confirming that SS 410 is superior for go-kart brake discs. While both methods predict moderate thermal loads, the combination of higher mechanical strength, hardness, and thermal conductivity makes SS 410 more capable of withstanding repeated braking stresses compared to austenitic stainless steels. This comparison validates the choice of SS 410 as an efficient, reliable, and cost-effective material for lightweight, performance-oriented braking applications.

7. RESULTS AND DISCUSSION

Analytical and numerical results show strong agreement, validating the braking system design. The percentage error between analytical and FEA temperature results is approximately 4.6%, confirming the accuracy of simplifying assumptions. Both disc and pedal exhibit acceptable stress levels and deformation under severe braking conditions.

The analytical evaluation of the braking system yielded a maximum achievable deceleration of 8.83 m/s^2, resulting in a minimum theoretical stopping distance of 15.74 m from an initial speed of 16.67 m/s. The corresponding braking time was 1.88 s, confirming effective deceleration within tyre–road friction limits.

Hydraulic analysis showed that a pedal effort of 450 N, amplified through a 4:1 pedal ratio, generated a brake line pressure of 3.55 MPa and a calliper clamping force of 5015.91 N. This produced a braking torque of 135.43 N-m at the rear disc. The required braking torque based on dynamic rear axle load was 84.80 N-m, resulting in a factor of safety for disc of 1.60.

Dynamic weight transfer during braking was calculated as 326.14 N, reducing the rear axle load to 674.48 N.

Analytical thermal calculations indicated a heat generation rate of 11.31 kW, producing a heat flux of 444364.93 W/m^2, and a disc temperature rise of 102.9 K, corresponding to an analytically predicted final disc temperature of approximately 406.05 K.

Finite Element thermal analysis predicted a maximum disc temperature of 387.38 K. The percentage error between analytical and numerical temperature predictions was calculated as 4.59%, indicating good agreement and validating the simplifying assumptions adopted in the analytical thermal model.

Finite Element structural analysis of the brake disc showed a peak Von Mises stress of 340.14 MPa, resulting in a factor of safety of 2.24, with a maximum deformation of 0.0386 mm.

Finite Element structural analysis of brake pedal yielded a maximum von Mises stress of 240.82 MPa and a factor of safety of 1.91, confirming adequate structural strength and stiffness.

8. SUGGESTIONS AND RECOMMENDATIONS

8.1 SUGGESTIONS

Test Tracking and Experimental Validation.

Though the safety and effectiveness of the braking system are verified by the analytical and the finite element results, there is a strong recommendation of conducting experimental validation of the braking system by controlled track testing. The real time data that can be given under repeated braking cycles can be measured with the instrumented testing through temperature sensors and strain gauges on disc temperature increase, pedal deformation, and braking efficiency [29]. This kind of testing would be useful in detecting short-term effects like brake fade, vibration, and noise, which cannot be measured with simulations alone. Numerical models can also be further calibrated by experimental data in order to achieve better prediction.

Alternative Brake Disc Designs Investigation.

Future research can discuss other brake disc designs like a ventilated, drilled, or slotted disc to improve heat dissipation and diminish thermal gradient. One method that can be used in such a design is optimization to minimize the mass of disc and retain structural integrity. Comparison of various disc profiles would have an insight into how to enhance thermal performance and increase the life of the brake parts without adding complications to the system [30].

State of the Art Material and Coating Investigations.

The application of surface coatings group or the use of advanced composite materials to brake discs and pads can be researched in order to bring about improved wear resistance and stability of friction. Anti-wear or thermal barrier coating may decrease the degradation of surfaces and increase the performance in long-term braking. Also looking at other pad materials could enhance consistency in friction and also limit thermal stress on the disc [31].

8.2 Recommendations

Implementation of Combined Analytical-FEA Design Approach.

Future designs of the go-kart braking system are suggested to follow a combination of analytical and finite element approach as illustrated in the present case [32]. This method can be sure of proper estimation of braking performance and prescient detection of structural or thermal problems thus saving development time and cost.

Dynamic Effects in Design.

Designers are advised to always use dynamic weight transfer and thermal loading effects when designing braking systems. Failure to pay attention to them may result in excessive braking performance estimates and low safety factors, especially when braking is applied at the rear [33].

Student and Racing Application Standardization.

The design methodology and material choice used in this paper are approved as a standard framework of competitions and entry-level racing among students. This can be enhanced by standardization of such design practices to enhance safety,

reliability and consistency of performance across go-kart platforms [34].

9. CONCLUSIONS

The paper provided a detailed analytical and finite element research of a rear mounted hydraulic braking system to be used in a go-kart in a dry track environment. The main aim was to test the braking performance, structure, and thermal behaviour to make sure that the operation is safe and reliable under extreme conditions, that is, when braking is required. The initial analytical calculations were used to estimate important parameters of braking such as maximum deceleration, stopping distance, braking time, hydraulic pressure, braking torque and dynamic weight transfer. Such calculations determined that the braking system can allow providing effective deceleration on the basis of tyre-road-friction limits with sufficient safety margins.

Hydraulic and braking torque measurements established the ratio of the pedals used, master cylinder, and calliper design achieves the required braking torque and can be operated by a driver with reasonable effort. The importance to dynamic weight transfer effects was to guarantee the conservative yet realistic approximation of braking capability of rear axle in especially the rear only braking geometry. Thermal analysis indicated that the brake disc was safe to absorb the amount of heat produced on braking without overheating the material thus reducing weak brake fade and thermal failure.

The choice of the material was an important factor in the reliability of systems. SS410 stainless steel was also found as a perfect material of the brake disc because of its high strength, wear resistance and thermal conductivity and AISI 4130 alloy steel was used in the brake pedal to give a high stiffness and fatigue resistance as well as structural safety. The prediction made through analytical means was confirmed through Finite element simulation which indicated that the temperature predictions were close to the actual values and that all stresses and deformations occurring in the brake disc and pedal were within acceptable limits. The factors of safety obtained, show strong structural behaviour with extreme braking.

In general, the approach to analysis and numerical technique used in this paper give a credible framework of the design and validation of go-kart braking systems. The results of the present research may be utilized in the future as a beneficial source of referencing regarding future student and professional level go-karts and the methodology can be applied to other braking systems on light vehicles.

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