DESIGN AND DEVELOPMENT OF A GRAVITY FEEDS EFFECTIVE BRAKING SYSTEM USING A VERTICALLY MOUNTED MASTER CYLINDER

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Abstract— This work contains the design and analysis of braking system for FSAE/SAE SUPRA. Fundamentals of vehicle statics and dynamics and previous research works are kept in the core of designing and sanalysis of various components of braking system. Along with it, different 'Engineering Software' like SOLIDWORK and ANSYS are utilized for the analysis of brake disc, considering structural and thermal aspects. The guiding factors of the design process were maximum braking power, safe stopping distance, minimal weight and manageable temperatures while maintaining reliability and control. For this first we developed a rationalized method to find out required ratio between front and rear braking forces to achieve maximum braking efficiency and to lock all the four wheels at the same time for the fulfilment of SAE SUPRA rules [4].After that according to data of braking forces, we have analysed the brake disc against the structural and thermal stresses. In this work, every progression of design was assessed and all the results of analysis have been interpreted. The basic aim of this work is to achieve the required standards for brake design in SAE SUPRA and to scale through the dynamic test at the competition. The designing of the whole braking system complied with all of the templates and envelopes required by the SAE SUPRA 2017 rules. The work serves as a guide to developing a desired braking system for high performance race car for the competition.

Key words: Traction Circle, Structural and Thermal Aspects, SAESUPRA, FSAE, Brake Force Distribution, Maximum Braking Efficiency

I. INTRODUCTION

Braking system is an essential part of any automobile. Without the ability to slow and stop our vehicles, we cannot hold control on it and ultimately accidents would occur. So each and every motor vehicle requires a reliable braking system. So for this we have to verify the design of braking system on the basis of calculation. When we go for designing of braking system the first problem comes into mind-"From where the calculation should be started?" In most of the brake design calculations we found that people starts calculation from padel force applied by the driver on the basis of ergonomics. From here calculation progresses in the direction of motion of brake fluid and finally at the tyre-road interface. Braking force generated between tyre and road is checked against the tyre adhesion limit. In some calculations tyre adhesion limit is also ignored. So these calculations are based on hit and trial method and very time consuming. For maximum braking efficiency all the tyres must generate maximum braking force in synchronize manner and it can be achieved by providing proper brake force distribution. It provides the data related to worst condition for analysis of components of braking system such as master cylinder, brake lines and brake disc against structural and thermal stresses.

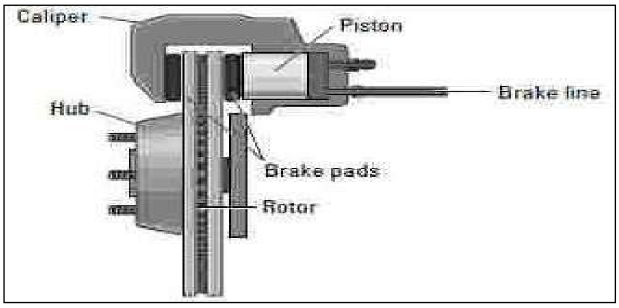

Fig. 1: Disc brake assembly

II. OBJECTIVE

For better controllability of vehicle, it is essential that all the four wheels should be locked-up at the same time. Maximum braking efficiency can be achieved only at particular barking ratio. The second objective of this is to find out the compatibility of the APACHE RTR 180 brake disc (rear) on the basis of structural and thermal analysis.

III. DESIGN METHODOLOGY

The need for the braking system is to increase the safety and manoeuvrability of the vehicle. Design procedure of braking system consists following steps –

- Calculating maximum braking capacity of vehicle means maximum possible deceleration of vehicle based on tyre model.
- Calculate desired braking ratio between front and rear wheels for locking of all the four wheels at the same time.
- Selection of brake disc considering structural and thermal stresses.

IV. DESIGN REQUIREMENTS

The design requirement as per the rules of SAE-SUPRA are as follows-

- The brake pedal must be fabricated from steel or aluminium or machined from steel, aluminium or titanium.
- It must have two independent hydraulic circuits such that in the case of a leak or failure at any point in the system, effective braking power is maintained on at least two wheels.
- Each hydraulic circuit must have its own fluid reserve, either by the use of separate reservoirs or by the use of a dammed, OEM-style reservoir.
- A single brake acting on a limited-slip differential is acceptable.
- The brake system must be capable of locking all fourwheels during the test.

- The brake pedal shall be designed to withstand a force of 2000 N without any failure of the brake system or pedal box. This may be tested by pressing the pedal with the maximum force that can be exerted by any official when seated normally.
- Calculating maximum braking capacity of vehicle means maximum possible deceleration of vehicle based on tyre model.
- Calculate desired braking ratio between front and rear wheels for locking of all the four wheels at the same time.
- Selection of material and dimensions for brake disc considering structural and thermal stresses.

V. Required Data

Few data required for calculation was taken from the cad model of vehicle and data as obtained by the manufacturer and by various references

- Mass of vehicle M=280kg
- Height of C.G. from ground h(mm) =240
- Distance of C.G. from front axle(mm)=602
- Wheel base WB(mm) = 1600
- Tyre outer diameter(mm)=508
- Diameter of brake disc(mm)=200
- Calliper piston diameter(mm)= 22
- Internal diameter master cylinder (mm) =19.05
- Coefficient of friction between pad and disc =0.4
- Material Properties (Grey cast iron) Thermal conductivity(w/m k) =50
- Density, ρ(kg/m3) =6600
- Specific heat , c (J/Kg)=380
- Thermal expansion ,(10-6 / k)=0.15
- Elastic modulus, E (GPa) =110
- Coefficient of friction,µ =0.4
- Heat transfer coefficient h(w/Km^2)=120
- The cornering data obtained was: (AVON 6.2 R20/13 SLICKS at 0^0 Camber Angle)

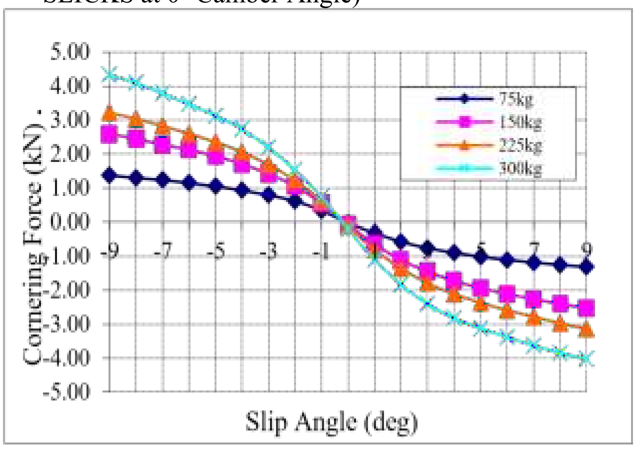

Fig. 2: Relation between cornering force and slip angle[6]

Some abbreviations used further in calculations:

WT= weight transfer D = deceleration h= height of cg from ground

WB= wheel base

R_F= load on front axle in static condition

R_R= load on rear axle axle in static condition

N_F= load on front tyre

N_R= load on rear tyre

R_F'= load on front axle in dynamic condition

R_R'= load on rear axle in dynamic condition

T_F= Maximum possible braking force on front tyre

T_R = Maximum possible braking force on rear tyre

VI. Calculations with Derivation

In static condition normal load on front and rear axle respectively-

R_F& R_R

While in dynamic Condition-

$R_F' = R_F + W.T.$

$R_R' = R_R - W.T.$

Normal/ Vertical load on each tyre -

$N_F = R_F'/2$

$N_R = R_R'/2$

Longitudinal force is of our interest to determine the maximum possible braking force. If only the data related to lateral force is available, we have to use 'Traction Circle' theory.

From that tyre data interpolate the value of maximum possible tractive force at respective normal forces- Let at weight (normal load) W_1& W_2 the value of tractive forces are T_{r1}& T_{r2} respectively. So, for N_R&N_F traction forces will be T_R & T_F.

$W_1 \rightarrow T_{r1}$

$N_F \rightarrow T_F$

$W_2 \rightarrow T_{r2}$

By applying liner interpolation-

$(N_F - W_1)/(W_2 - W_1) = (T_F - T_{r1})/(T_{r2} - T_{r1})$...(1) Similarly,

$(N_R - W_1)/(W_2 - W_1) = (T_F - T_{r1})/(T_{r2} - T_{r1})$….(2)

From here find out value of T_F& T_R.

Maximum possible braking force = $2(T_F + T_R)$

Braking force required = $M*D_{MAX}$

For limiting condition/ verge of skid

F_{req} = Maximum Possible Braking Force

$M.D_{max} = 2(T_F + T_R)$

$= 2[T_{r1} + (N_F - W_1)(T_{r2} - T_{r1})/(W_2 - W_1) + T_{r1} + (N_R - W_1)(T_{r2} - T_{r1})/(W_2 - W_1)]$

$= 2[2T_{r1} + (N_F + N_R - 2W_1)(T_{r2} - T_{r1})/(W_2 - W_1)]$

$= 4T_{R1} + \{(R'_F + R'_R - 4W_1)(T_{r2} - T_{r1})/(W_2 - W_1)\}$

$= 4T_{R1} + \{(R_F + R_R - 4W_1)(T_{r2} - T_{r1})/(W_2 - W_1)\}$

$$\boxed{D_{MAX} = \frac{1}{M}\{4T_R + \frac{R_F + R_R - 4W_1}{W_2 - W_1}(T_{r2} - T_{r1})\}}$$

This is the general expression for maximum deceleration.

For given data,

$D_{MAX} = 16.847 m/s^2 = 1.717g$

Required braking force ratio –

Putting the vehicle date in equation 1 & 2 we will get the following expressions – $T_F = 0.9285 + 0.302\,D_{MAX}$

$T_R = 1.427 - 0.302\,D_{MAX}$

On substituting--- D= 16.847m/s^2

T_F=2895.67NandT$_R$=1.8155N

So, percentage of braking force required of front axle % F_{BF} = $T_F/(T_F + T_R)$ *100... (3)

Percentage of braking force required of front axle

% F_{BR}= $T_R/(T_F + T_R)$ *100...(4)

By putting values of T_F and T_R we will get –

% F_{BF} : % F_{BR} = 61.43 : 38.53

It is not possible to achieve perfect lockup of all the four wheels so in this case it is desirable to lock-up front wheels slightly before the rear wheels to avoid oversteering condition otherwise in case of rear lockup first the vehicle start to spin on and only a skilled driver can control the vehicle by counter steering until front wheels are not locked. So acceptable braking ratio taking all the above factors in consideration is 62:38.

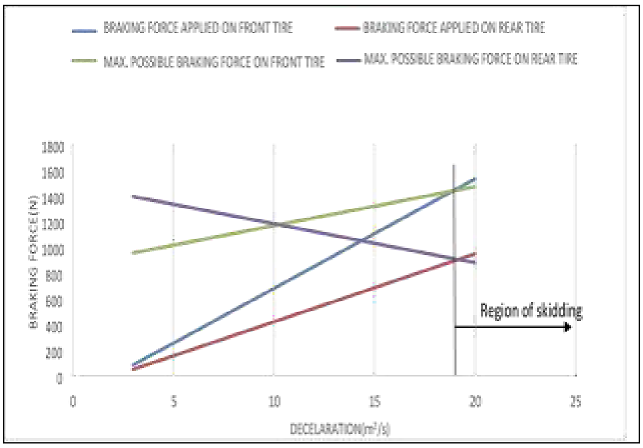

Fig. 3: Graph showing variation of braking force with deceleration and safe region of braking

VII. Analysis

The kinetic energy of vehicle get dissipated into atmosphere through the brake disc and it is achieved by intense rubbing action between brake disc and pads. So brake disc should be capable enough to withstand under such extreme conditions. For this it becomes an essential part of braking system designing to analyze the strength of disc against structural and thermal stresses and on the basis of analysis optimum dimensions of brake disc can be obtained.

Here we are checking the compatibility of 'APACHE RTR 180'rear brake disc with our system because it is easily available, easy to mount and proper outer diameter to assemble with the rim.

First step for any analysis is to make a 3D model of the component in the respective software.

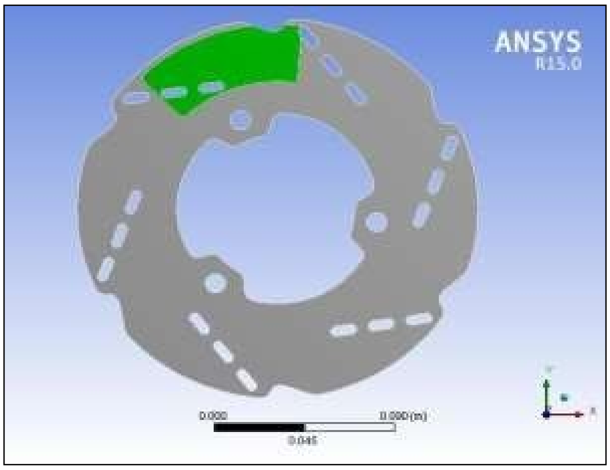

Fig. 4: Solid model of brake disc with pad impression

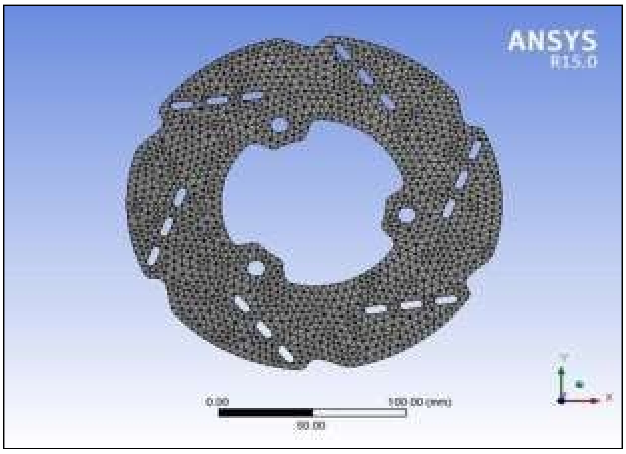

Fig. 5: Meshed model of the disc

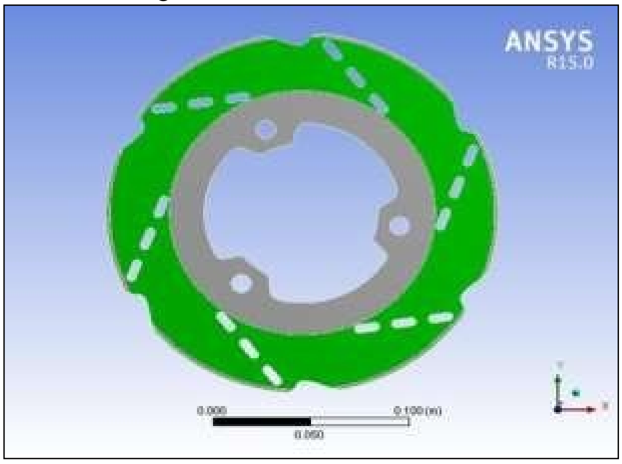

Fig. 6: Model showing the rubbing ring

A. *Structural Analysis*

The analysis showing validation of the structure to be used in our vehicle that is to check its durability, factor of safety and stiffness. Greater the stiffness greater is the durability of the component.

Maximum braking force is acting on front wheels so if all the four discs are identical, it is enough to analyze only a front disc. First we have to calculate all the forces acting on the disc.

Braking force on front axle T_F = 2895.67N

Braking force on one front tyre=2895.67/2 = 1447.83N

Braking torque on tyre = Braking force * radius of tyre

= 1447.83 * 0.254

= 367.75N-m Braking torque on tyre = Braking torque on disc for new disc 'Uniform Pressure Theory' will be used.

Effective radius or mean friction radius $r_B = \frac{2}{3}(r_o^3 - r_i^3)/(r_o^2 - r_i^2)$

as r_o = 98mm and r_i = 68mm therefore we have,

r_B=83.90mm

Braking torque on disc = $F_{clamp} * r_B$

Where, F_{clamp} = Clamping force

$367.75 = F_{clamp} * (83.90*10^{-3})$

F_{clamp}= 4383.005N

Normal force exerted by calliper on one side of disc ,

Fcal = Fclamp / 2

F_{cal} = 2191.502N

No. of piston per calliper = 2

$F_{Piston} = F_{cal}/2$ = 1095.751N

Pressure inside piston of calliper, $P_{Piston} = F_{Piston}$ / Area of piston

$=1095.75/(\frac{11}{4}*.01905^2)$

$=3844427.17 N/m^2$

=3.844MPa

Pressure generated inside piston of calliper which will be equal to pressure inside master cylinder in ideal case means neglecting pressure losses throughout the brake line.

If you have internal diameter of calliper cylinder and master cylinder you can easily find out optimum the pedal ratio on the basis on ergonomic pedal force.

Area of brake pad of APACHE RTR 180 = $0.002327 mm^2$

Uniform pressure exerted on each side of disc

$= F_{cal}$ / Area of brake pad

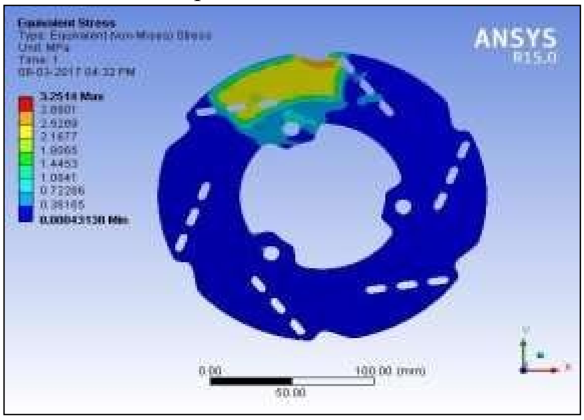

Fig. 3: Stress variation over disc

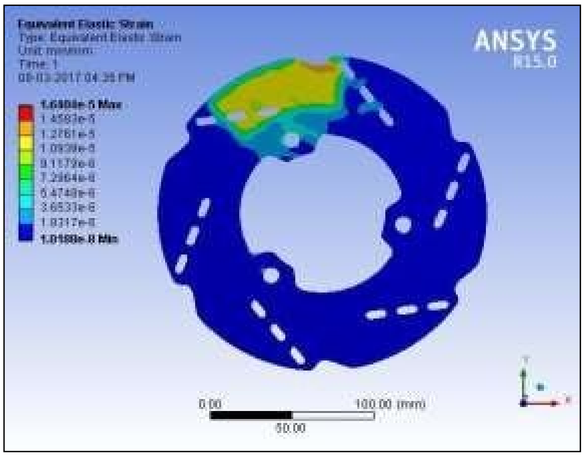

Fig. 4: Strain variation on disc

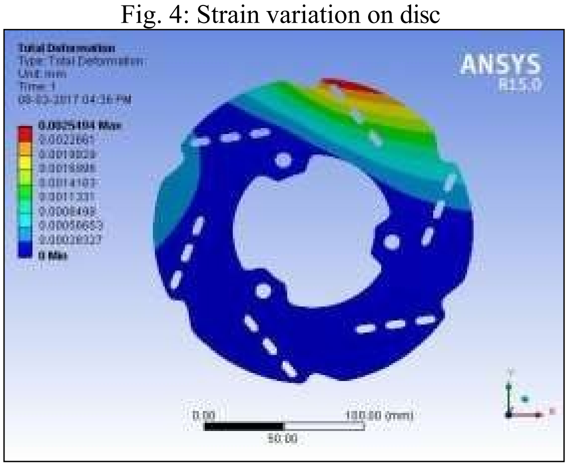

Fig. 5: Deformation of the disc

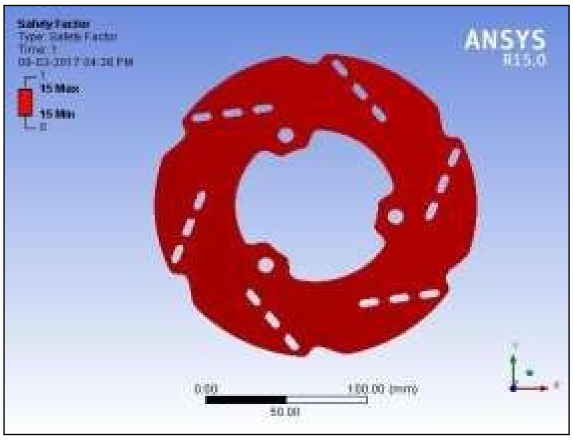

Fig. 6: Diagram showing factor of safety

B. Thermal Analysis

During braking process the disc has to save about 80-90% of kinetic energy of vehicle as heat. Only 10- 20 % of thermal energy can be delivered as free & forced convection and radiation into the environment. Therefore the thermal analysis of brake disc is essential for the optimum dimensioning of brake disc.

Initial speed of vehicle u = 100 km/hr

Final speed of vehicle v = 0

$D_{MAX} = 16.847 m/s^2$

Using the equation, $v = u - Dt$

From here, t = 1.648 s

Weight on front axle $R_F' = R_F + W.T.$

$R_F = 105.35$ kg

Where,

W.T. =(D/g)(h/WB)*W

So. $R_F' = 105.35 + 72.114 = 177.464$ kg

Assuming 90% of K.E. of vehicle is stored by disc , k= 0.9

Energy generated during braking on a front disc

K.E. = $k*(R_F'/2)*(v^2 - u^2)/2$

K.E. = 0.9*(177.464/2)*(100*100/2)

K.E. = 30809.72 J

Conversion of K.E. into heat per second,P= K.E. / t

P= 18695.21 W

Heat flux,h = 18695.21 / Total ssurface area of disc

$h = 597481.9431 W/m^2$

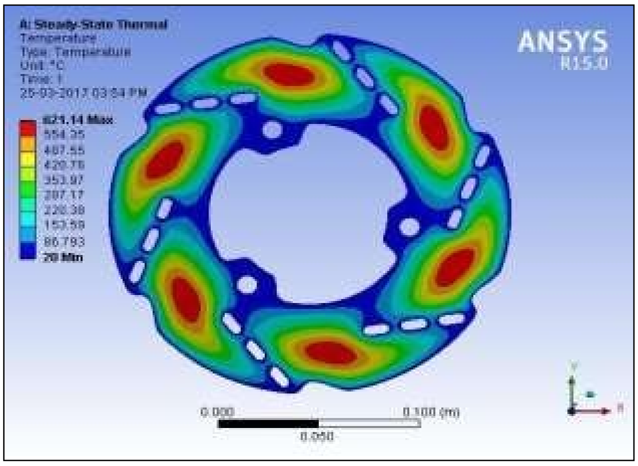

Fig. 7: Temperature Distribution

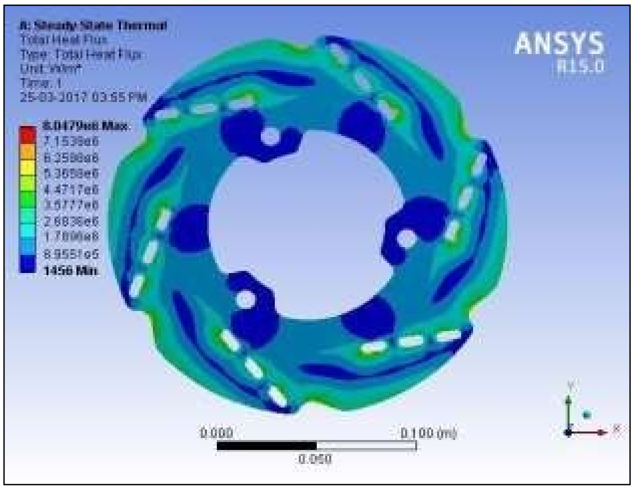

Fig. 8: Total heat flux

VIII. Interpretation of ANSYS Results

A. Structural analysis

Fig 7 & Fig 8 show distribution of compressive stresses and strain induced by the contact of brake pads. And can observe that the maximum I ntensity of stress and strain is located at the under-cut area of brake disc. And due to its maximum deformation is also taking place at the same location as depicted from Fig 9. Fig 10 shows factor of safety, which is coming out to be highly reliable according to the design.

	Equivalent Stress	Equivalent Elastic Strain	Total Deformation
MIN.	4.3138e-004 MPa	1.0188e-008 mm/mm	0. mm
MAX.	3.2514 MPa	1.6404e-005 mm/mm	2.5494e-003 mm

Table 1: Extreme stress - strain data obtained by structural analysis

B. Thermal analysis

Fig 11 shows the temperature distribution across the brake disc and the maximum temperature that can be developed is 621.14^{0}C, which is much lower than the melting point temperature of the material. Fig 12 indicates that the maximum intensity of heat flux is located at the edges and the drilled portion of the disc due to more surface area.

	Temperature	Total Heat Flux
Minimun	20. °C	1456. W/m²
Maximum	621.14 °C	8.0479e+006 W/m²

Table 2: Extreme temperature and heat flux data obtained by thermal analysis

IX. Conclusion

Under harsh conditions, every part of the braking system, including the master cylinder, brake disc, brake lines, etc., is tested. We can see that the chosen brake disc is suitable with our system's requirements from the structural and thermal analyses. Along with it comes a streamlined approach to choosing or designing any part, particularly the brake disc's proportions.

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