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C.A.D. & F.E.M. ANALYSIS OF DISC BRAKE SYSTEM

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ABSTRACT

The aim of my paper is to show how to perform a crashworthiness simulation in the automobile industry using Finite Element Method, Repetitive braking of the vehicle leads to heat generation during each braking event. The resulting rise in temperatures has very significant role in the performance of the braking system. Problems such as premature wear of brake pads and thermal cracking of brake discs are attributed to high temperatures.

Consequently controlling the temperature profiles and thermo-mechanical stresses are critical to proper functioning of the braking system. CAE simulations are often used for evaluating the brake disc design using thermo-mechanical analysis techniques. Conventional approach is to use three dimensional FE models of the brake discs. This approach has major drawbacks of higher pre and post processing as well as solution times. Need is felt to develop a quick and reliable method to evaluate the thermal stresses in brake discs. This paper describes one such approach based on modified FEM axisymmetric analysis.

This paper reviews numerical methods and analysis procedures used in the study of automotive disc brake. It covers Finite element Method approaches in the automotive industry, the complex Contact analysis. The advantages and limitations of each approach will examine. This review can help analysts to choose right methods and make decisions on new areas of method development. It points out some outstanding issues in modelling and analysis of disc brake squeal and proposes new conceptual design of the disk braking system. It is found that the complex Contact analysis is still the approach favoured by the automotive industry.

KEY WORDS:

CAD & FEM Analysis of Disc Brake System; Contact pressure and Thermal Analysis of disc brake; Brake Pad Material Analysis

1. INTRODUCTION

1.1 Disc Brake

A disk brake consists of a cast iron disk bolted to the wheel hub and a stationary housing called caliper. The caliper is connected to some stationary part of the vehicle like the axle casing or the stub axle as is cast in two parts each part containing a piston. In between each piston and the disk there is a friction pad held in position by retaining pins, spring plates etc. passages are drilled in the caliper for the fluid to enter or leave each housing. The passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston. A schematic diagram is shown in the figure 1.1.

A hydraulically activated *disc brake* comprises two opposing pistons each faced with a pad of lining material. When the hydraulic pressure is increased the pads are forced against the rotating metal friction disc, exerting a normal force at each contact. The two normal forces cancel one another axially but cause additive tangential friction forces

Figure 1.Disc Brake components

which oppose the disc's motion and decelerate it.

1.2 Components of Disc Brakes

A disc brake assembly consists of –

- Brake Pedal—Force input to system from driver. Design gives a Mechanical Advantage
- Master Cylinder—Converts force to pressure. Pressure is used to move brake pads into place
- Brake Pads—Provide friction force when in contact with rotor. Works to slow or stop vehicle

- Caliper—Holds pads and squeezes them against rotor
- Rotor—Spins with wheel. When used in conjunction with brake pads, slows vehicle
- Vents—Help provide cooling to brake

1.3 Working principle of disc brake system

When hydraulic pressure is applied to the caliper piston, it forces the inside pad to contact the disc. As pressure increases the caliper moves to the right and causes the outside pad to contact the disc. Braking force is generated by friction between the disc pads as they are squeezed against the disc rotor. Since disc brakes do not use friction between the lining and rotor to increase braking power as drum brakes do, they are less likely to cause a pull.

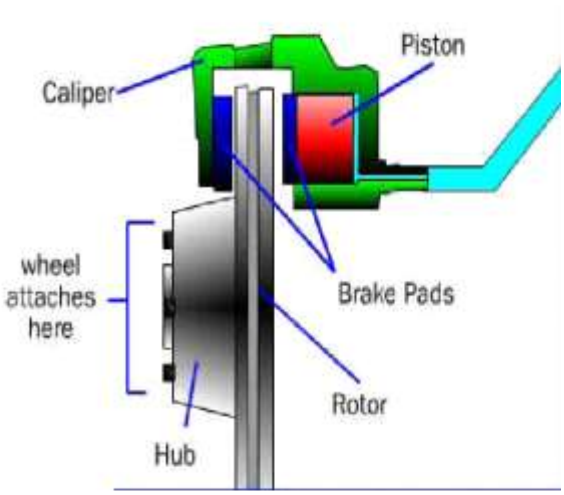


Figure 2. Working principle of disc brake system

The friction surface is constantly exposed to the air, ensuring good heat dissipation, minimizing brake fade. It also allows for self-cleaning as dust and water are thrown off, reducing friction differences.

Unlike drum brakes, disc brakes have limited self-energizing action making it necessary to apply greater hydraulic pressure to obtain sufficient braking force. This is accomplished by increasing the size of the caliper piston. The simple design facilitates easy maintenance and pad replacement.

The disc brake is a lot like the brakes on a bicycle. Bicycle brakes have a caliper, which squeezes the brake pads against the wheel. In a disc brake, the brake pads squeeze the rotor instead of the wheel, and the force is transmitted hydraulically instead of through a cable. Friction between the pads and the disc slows the disc down.

2. WHY CERAMIC BRAKE PADS?

We want our vehicle's brake system to offer smooth, quiet braking capabilities under a wide range of temperature and road conditions. We don't want brake-generated noise and dust annoying us during our daily driving.

To accommodate this, brake friction materials have evolved significantly over the years. They've gone from asbestos to organic to semi-metallic formulations. Each of these materials has proven to have advantages and disadvantages regarding environmental friendliness, wear, noise and stopping capability.

Asbestos pads caused health issues and organic compounds can't always meet a wide range of braking requirements. Unfortunately the steel strands used in semi-metallic pads to provide strength and conduct heat away from rotors also generate noise and are abrasive enough to increase rotor wear.

Since they were first used on a few original equipment applications in 1985, friction materials that contain ceramic formulations have become recognized for their desirable blend of traits. These pads use ceramic compounds and copper fibers in place of the semi-metallic pad's steel fibers. This allows the ceramic pads to handle high brake temperatures with less heat fade, provide faster recovery after the stop, and generate less dust and wear on both the pads and rotors. And from a comfort standpoint, ceramic compounds provide much quieter braking because the ceramic compound helps dampen noise by generating a frequency beyond the human hearing range.

Another characteristic that makes ceramic materials attractive is the absence of noticeable dust. All brake pads produce dust as they wear. The ingredients in ceramic compounds produce a light colored dust that is much less noticeable and less likely to stick to the wheels. Consequently, wheels and tires maintain a cleaner appearance longer.

Ceramic pads meet or exceed all original equipment standards for durability, stopping distance and noise. According to durability tests, ceramic compounds extend brake life compared to most other semi-metallic and organic materials and outlast other premium pad materials by a significant margin - with no sacrifice in noise control, pad life or braking performance.

3. MATHEMATICAL FORMULATION OF THE PROBLEM

CALIPER DISK BRAKES

From the moment of contact until the disk is stopped, the velocity of the disk relative to the brake pads will vary linearly with the disk radius. If the thickness of the lining material removed is denoted by δ and if δ is dependent on the relative velocity and the pressure, as is commonly assumed, then according to the uniform wear assumption,

$$\delta = kpr \tag{1-1}$$

where k is a constant of proportionality. Since the caliper brake pads are usually small enough for their supports to be considered rigid, we shall assume that δ is constant over the brake pad (i.e., the wear is uniform). Whenever these conditions hold, equation (1-1) implies that the pressure increases as the radius decreases, so the maximum pressure is found at the inner radius, r_i. Thus

$$\delta = kp_{\max} r_i \tag{1-2}$$

Elimination of k and δ from equations (1-1) and (1-2) yields

$$p = p_{\max} \frac{r_i}{r} \tag{1-3}$$

With the lining pressure known, we may now calculate the required axial force from

$$F = \int_A p \, da \tag{1-4}$$

and the resulting braking torque from

$$T = \mu \int_A pr \, da \tag{1-5}$$

Evaluation of these integrals is easiest for brake pads with radial and circular boundaries, for which equation (1-4) and (1-5) may be written using a dummy variable ϕ as

$$F = p_{\max} r_i \int_A \frac{1}{r} da = p_{\max} r_i \int_{r_i}^{r_o} \int_0^{\theta} d\phi dr \tag{1-6}$$

$$= p_{\max} r_i \theta (r_o - r_i)$$

and

$$T = \mu p_{\max} r_i \int_A da = \mu p_{\max} r_i \int_{r_i}^{r_o} r \, dr \int_0^{\theta} d\phi \tag{1-7}$$

$$= \mu p_{\max} r_i \frac{\theta}{2} (r_o^2 - r_i^2)$$

From equation (1-7) we find that for the pressure distribution given by relation (1-3) the torque may be easily calculated for any brake pad whose area is known or simply calculated. For a circular pad of diameter d, for example, the torque is given by

$$T = \mu p_{\max} r_i \frac{\pi}{4} d^2 \tag{1-8}$$

Circular pads are often used, nevertheless, in hydraulically activated caliper brakes whenever the hydraulic pressure may be increased relatively cheaply because the pads themselves are supported entirely by the piston face and are therefore cheaper to produce because no additional supporting structure is required. Noncircular pads are used where increasing the pressure may be relatively expensive and where the maximum performance is required for the pressure that is available, as in aircraft brakes.

If we replace $d^2/4$ in equation (1-8) with r_p^2, where r_p is the pad radius ($r_p = d/2$), and also replace r_i in equation (1-8) according to the relation $r_i = r_o - 2r_p$, we have

$$T = \mu \pi p_{\max} (r_o - 2r_p) r_p^2 \tag{1-9}$$

Upon differentiating equation (1-9) with respect to r_p we obtain

$$\frac{dT}{dr_p} = \mu \pi p_{\max} 2r_p (r_o - 3r_p) \tag{1-10}$$

which is equal to zero when $r_p = r_o/3$, indicating an extreme value of T for that pad radius. Since $d T^2/dr_p^2$ is negative at this value of r_p, it follows that T has its maximum value at $r_p = r_o/3$.

DESIGN FORMULATION

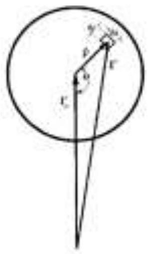


Figure 3. Geometric relation between ρ θ r and r_c Arc length at radius r over angular increment dθ is rdθ

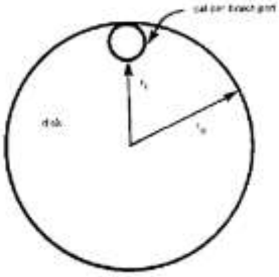


Figure 4. Circular lining pad of caliper disk brake

4. METHODOLOGY

A Structural analysis calculates deformations, stresses, and strains on model in response to specified constraints. A static analysis gives certain information about model. For example, a static analysis tells us if the material in our model will stand stress and if the part will break (stress analysis), where the part will break (strain analysis), and how much the shape of the model changes (deformation analysis).

In this project work two different pad material – Ceramic & composite Fiber are analyzed using FEM Software ANSYS. The present study can provide a useful design tool and improve the brake performance of disk brake system based on the strength and rigidity criteria.

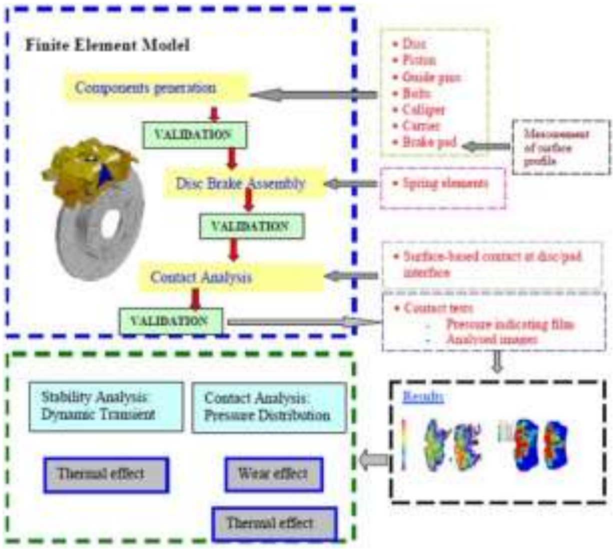


Figure 5. Methodology

5. SOLID MODELING

The essential difference between Pro/ENGINEER and traditional CAD systems is that models created in Pro/ENGINEER exist as three-dimensional solids. Other 3-D modelers represent only the surface boundaries of the model. These models are the complete solid. This not only facilitates the creation of realistic geometry, but also allows for accurate model calculations, such as those for mass properties.

Design Consideration

- Brake Power
- Larger diameter rotors more will be brake power with the same amount of clamp force than a smaller diameter rotor.
- The higher the coefficient of friction for the pad, the more brake power will be generated
- Dynamic Coefficient Of Friction
- type of material used for the brake rotor.
- Speed Sensitive – Coefficient of friction typically drops as the speed of the vehicle increases
- Pressure Sensitive - Coefficient of friction typically drops as more clamp force is generated.
- Temperature Sensitive - Coefficient of friction typically drops as the temperature of the brake system increases.
- Surface Area – The more surface area available on a brake system, the better heat dissipation will be via convection.
- Material Selection– Material selection is important in trying to control where the heat dissipates once generated
- Wear –wear is proportional to pressure intensity(p) and relative velocity (v) which is proportional to radius. Thus

  W=k p r
- Thermal Mass –Must have enough material mass to properly handle the temperatures during braking applications. This is limited by size and weight.
- The brake systems on vehicles must be capable of absorbing a lot more horsepower than the engine typically produces because the heat (power) that is generated when braking occurs over a short period of time

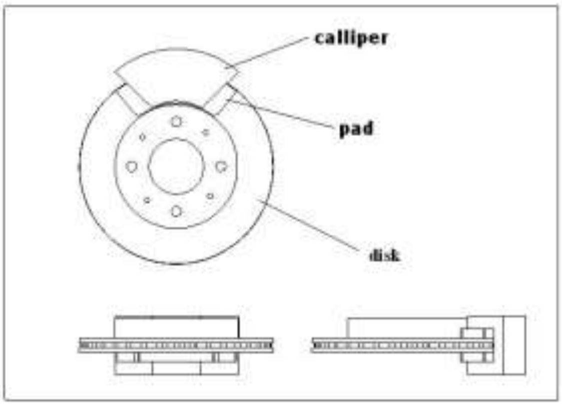


Figure 6 .CAD Prototype_2D of DISC BRAKE

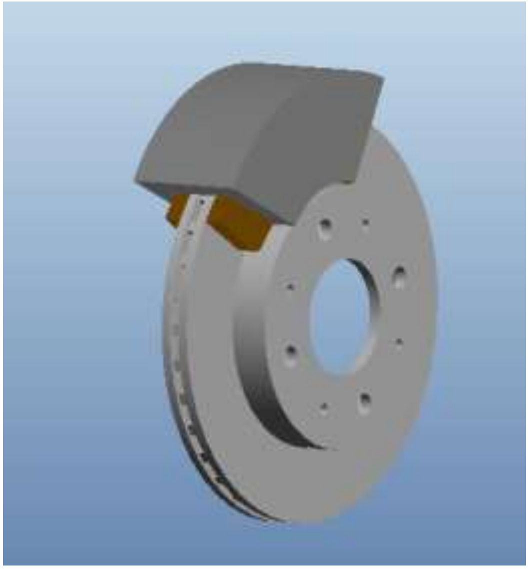
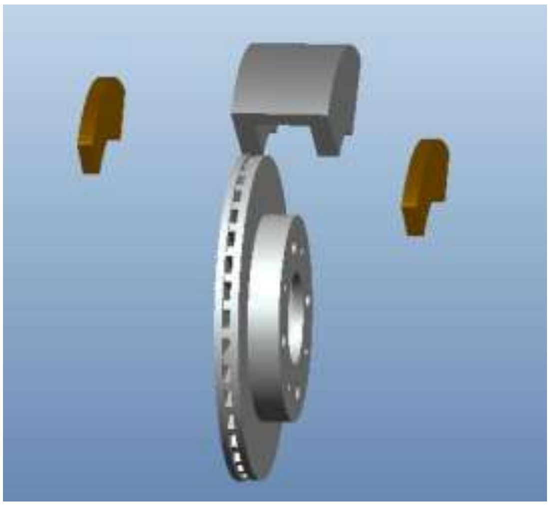


Figure 7 .CAD Prototype of DISC BRAKE

Figure 8. Exploded- CAD Prototype of Disc Brake



6. INTERPRETATION OF THE FINITE ELEMENT METHOD

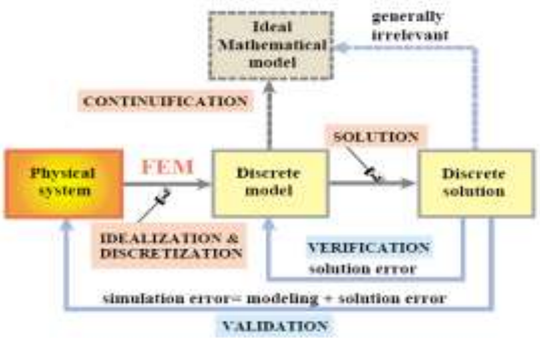


Figure9 . Physical Interpretation of FEM

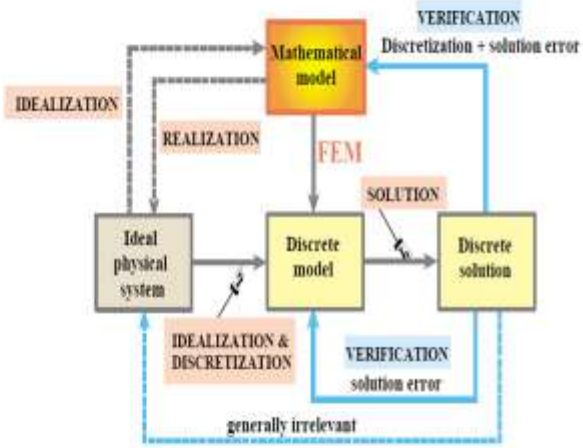


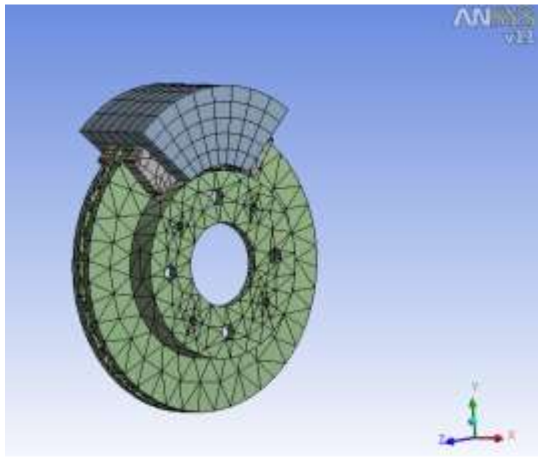
Figure 10 Mathematical Interpretation of FEM

7. FE MESH GENERATION OF DISC BRAKE

To create a subassembly or an assembly, you must place a base component or feature, then attach additional components to the base and to each other. We cannot attach components to an exploded assembly. We must unexplode it first.

We can add components to an assembly in the following ways:

- Attach a component parametrically by specifying its position relative to the base component or other components in the assembly.
- Attach a component nonparametrically using the **Package** command in the COMPONENT menu. Use packaging as a temporary means to include the component in the assembly; then finalize its location with assembly instructions.



Nodes	11926
Elements	4865

Figure 11 Meshing

8. STRUCTURAL ANALYSIS

Structural analysis is probably the most common application of the finite element method. The term *structural* (or *structure*) implies not only civil engineering structures such as bridges and buildings, but also naval, aeronautical, and mechanical structures such as ship hulls, aircraft bodies, and machine housings, as well as mechanical components such as pistons, machine parts, and tools.

Thermal analysis to calculate the heat flux, temperature gradient & temperature variation.

For finite element Analysis of Engine Head, SOLID92-Tetrahedral element has been used. Total number of elements generated-4066.

8.1 Types of Solution Methods

Two solution methods are available for solving structural problems.

- The h-method
- the p-method.

The h-method can be used for any type of analysis, but the p-method can be used only for linear structural static analyses. Depending on the problem to be solved, the h-method usually requires a finer mesh than the p-method. The p-method provides an excellent way to solve a problem to a desired level of accuracy while using a coarse mesh.

ANSYS automatically calculates all measures valid for a static analysis. Following points are important when specifying loads and constraint sets for static analyses:

- If you delete a constraint or load set that you included in an analysis, you also delete that set from the analysis.

  Even if you create a new set with the same name as the set you deleted, you must edit the analysis and reselect the set. Otherwise, you may invalidate the analysis and any design studies in which you included the analysis.

ANSYS calculates results separately for each load set you include in the analysis

8.2 Contact pressure and Thermal Analysis

Table 1 Static Structural Results

Object Name	*Total Deformation*	*Equivalent Stress*
State	Solved	
Scope		
Geometry	All Bodies	
Definition		
Type	Total Deformation	Equivalent (von-Mises) Stress
Display Time	End Time	
Results		
Minimum	0. mm	0. MPa
Maximum	2.049e-002 mm	6.2171 MPa
Minimum Occurs On	Solid	
Maximum Occurs On	Solid	
Information		
Time	1. s	
Load Step	1	
Substep	1	
Iteration Number	1	

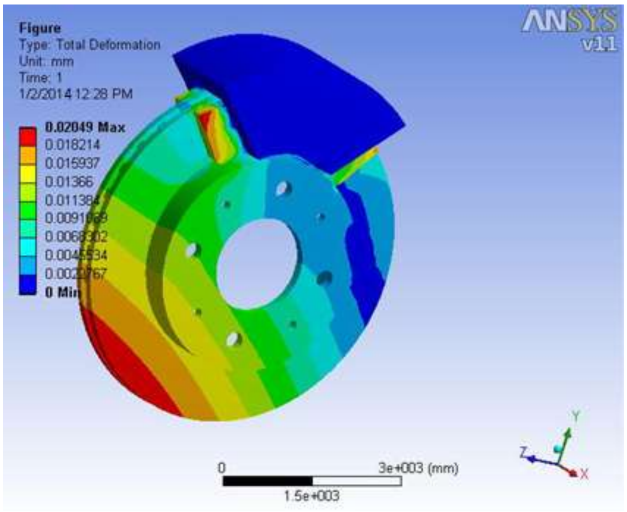


Figure12. Static Structural Total Deformation

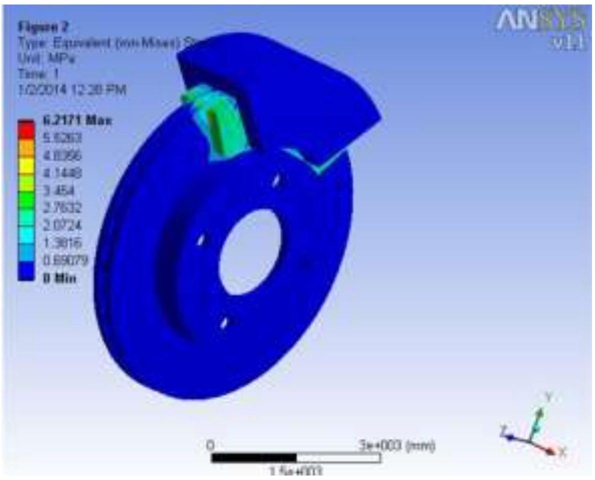


Figure 13. Static Structural Equivalent Stress

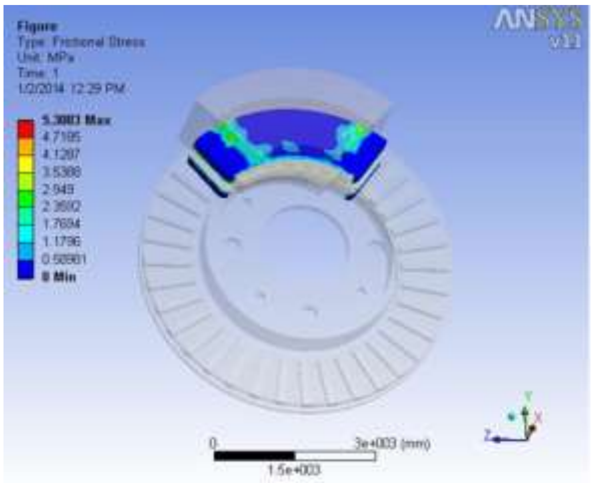


Figure 14. Static Structural Frictional Stress

8.3 Steady State Thermal Analysis

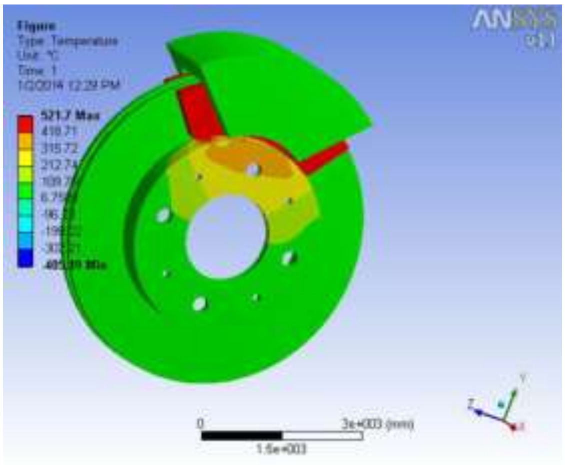


Figure15. Steady state thermal_Temp

Table2 Steady State Thermal Results

Object Name	*Temperature*	*Total Heat Flux*
State	Solved	
Scope		
Geometry	All Bodies	
Definition		
Type	Temperature	Total Heat Flux
Display Time	End Time	
Results		
Minimum	-405.19 °C	1.282e-009 W/mm²
Maximum	521.7 °C	0.81433 W/mm²
Minimum Occurs On	Solid	
Maximum Occurs On	Solid	
Information		
Time	1. s	
Load Step	1	
Substep	1	
Iteration Number	1	

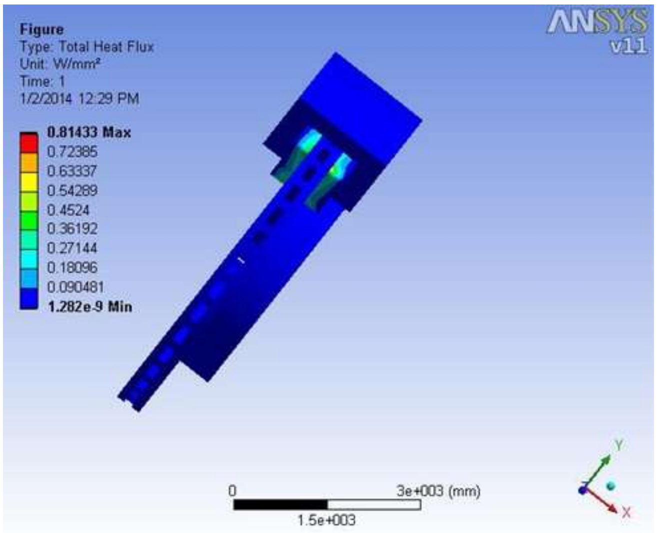


Figure16. Steady state thermal Total Heat Flux

9. CERAMIC CALIBRE ANALYSIS

9.1 Steady State Thermal Analysis

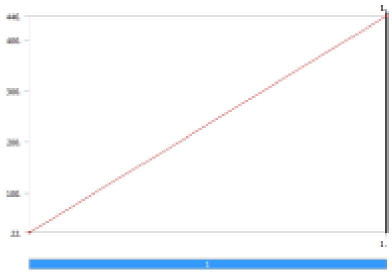


Figure17. Steady state thermal _temperature

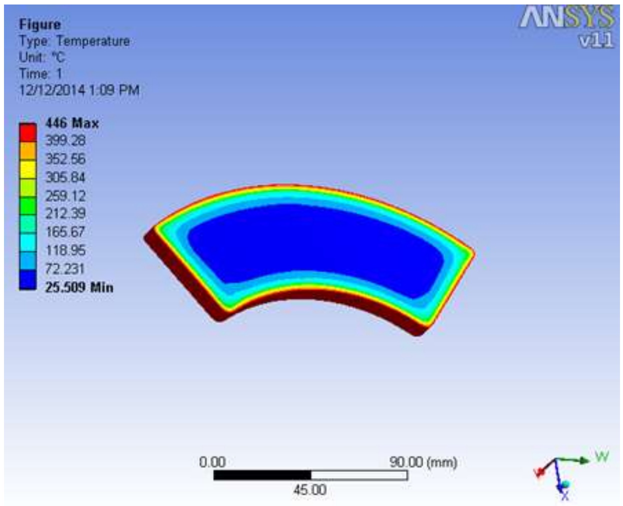


Figure 18. Steady state thermal_Temp

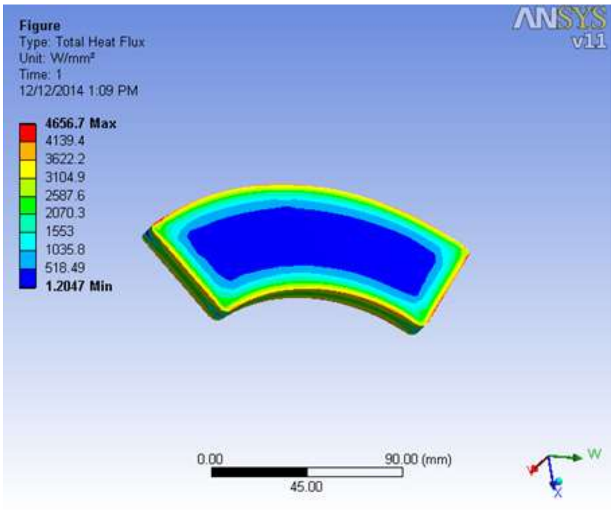


Figure 19. Steady state thermal Total Heat Flux

9.2 **Transient Thermal Analysis**

Table3 Transient Thermal Results

Object Name	*Temperature*	*Total Heat Flux*
State	Solved	
Scope		
Geometry	All Bodies	
Definition		
Type	Temperature	Total Heat Flux
Display Time	End Time	
Results		
Minimum	145.49 °C	8.0467 W/mm²
Maximum	446. °C	4369.4 W/mm²
Information		
Time	1. s	
Load Step	1	
Substep	17	
Iteration Number	17	

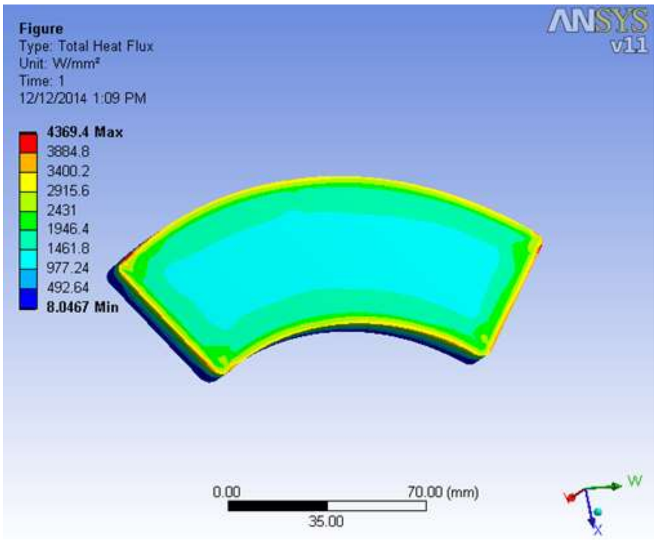


Figure 20. Transient thermal Total Heat Flux

9.3 RESULTS

In this paper two different pad material – Ceramic & composite Fiber are analyzed using FEM Software ANSYS following are details of structural Analysis:

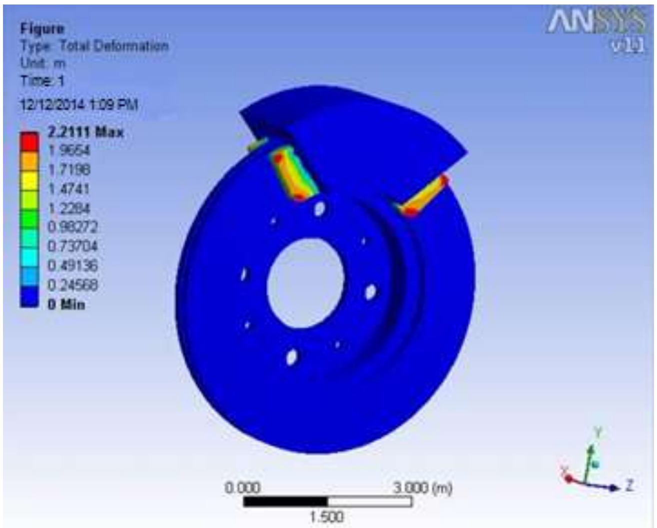


Figure 21
ceramic deformation results

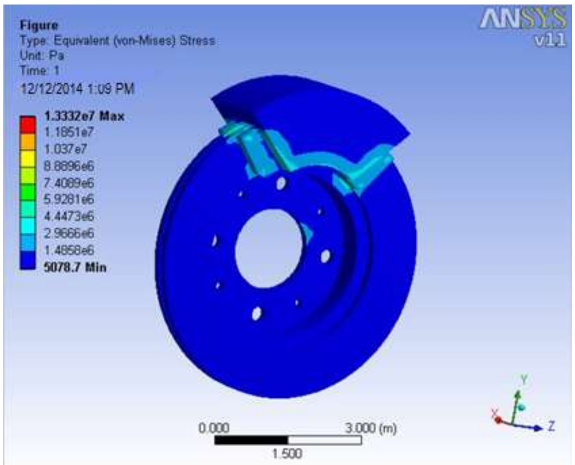


Figure 22
Ceramic stress results

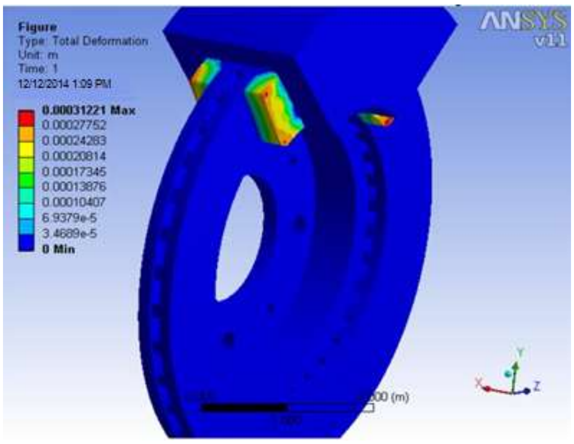


Figure 23
Composite fiber material deformation results

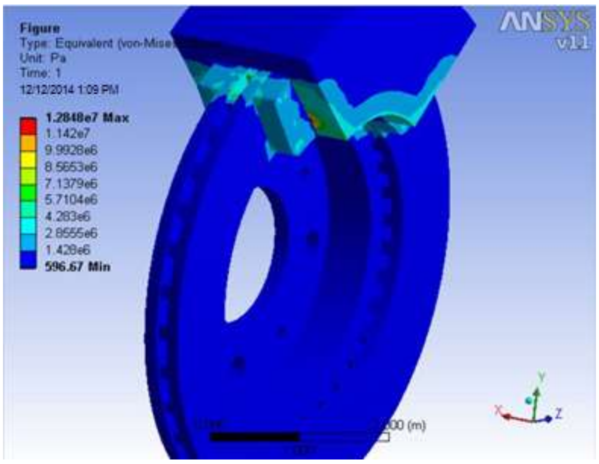


Figure 24
Composite fiber material stress results

these materials has proven to have advantages and disadvantages regarding environmental friendliness, wear, noise and stopping capability.

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Another characteristic that makes ceramic materials attractive is the absence of noticeable dust. All brake pads produce dust as they wear. The ingredients in ceramic compounds produce a light colored dust that is much less noticeable and less likely to stick to the wheels. Consequently, wheels and tires maintain a cleaner appearance longer.

Ceramic pads meet or exceed all original equipment standards for durability, stopping distance and noise. According to durability tests, ceramic compounds extend brake life compared to most other semi-metallic and organic materials and outlast other premium pad materials by a significant margin - with no sacrifice in noise control, pad life or braking performance.

10. CONCLUSION

1 .Investigated the effects of the rotating speed of the disk and the material properties on wear behaviours.

2. To reduce the solution time for the problem in FEA by utilizing ANSYS's contemporary advantages in contact and thermo-mechanical problems.

Why Ceramic Brake Pads are recommended?

We want our vehicle's brake system to offer smooth, quiet braking capabilities under a wide range of temperature and road conditions. We don't want brake-generated noise and dust annoying us during our daily driving.

To accommodate this, brake friction materials have evolved significantly over the years. They've gone from asbestos to organic to semi-metallic formulations. Each of

11. REFERENCES

1. ***Piotr GRZEŚ***, *Finite Element Analysis of Disc Temperature During Braking Process, Faculty of Mechanical Engineering, Białystok Technical University, ul. Wiejska 45 C, 15-351 Białystok*
2. ***Abd Rahim Abu-Bakar, Huajiang Ouyang,*** *Prediction of Disc Brake Contact Pressure Distributions by Finite Element Analysis, Jurnal Teknologi, 43(A) Dis. 2005: 21–36 © Universiti Teknologi Malaysia*
3. ***M. Nouby, D. Mathivanan, K. Srinivasan***, *A combined approach of complex eigenvalue analysis and design of experiments (DOE) to study disc brake squeal, International Journal of Engineering, Science and Technology Vol. 1, No. 1, 2009, pp. 254-271*
4. ***P. Liu a, H. Zheng a, C. Cai a, Y.Y. Wang a, C. Lu a,K.H. Ang b, G.R. Liu,*** *Analysis of disc brake squeal using the complex eigenvalue method, ScienceDirect, Applied Acoustics 68 (2007) 603–615*

5. **Rajendra Pohane, R. G. Choudhari**, *Design and Finite Element Analysis of Disc Brake, International J. of Engg. Research & Indu. Appls. (IJERIA). ISSN 0974-1518, Vol.4, No. I (February 2011), pp 147-158*
6. **H Mazidi, S.Jalalifar, J. Chakhoo**, *Mathematical Model of heat conduction in a disc brake system during braking, Asian journal of Applied Science 4(2): 119-136,2011, ISSN 1996-3343 / DOI: 10,3923/ajaps,2011,119,136*
7. **V.M.M.Thilak, R.Krishnaraj, Dr.M.Sakthivel,** *Transient Thermal and Structural Analysis of the Rotor Disc of Disc Brake , International Journal of Scientific & Engineering Research Volume 2, Issue 8, August-2011 ISSN 2229-5518*
8. **Prashant Chavan, Amol Apte** , *Axisymmetric analysis of bolted disc brake assembly to evaluate thermal stresses TATA motors ltd. Pimpri, Pune-411018. India 91-20-5613 3159*
9. **Q Cao1, M I Friswell, H Ouyang, J E Mottershead1 and S James**, *Car Disc Brake Squeal:Theoretical and Experimental Study Materials Science Forum Vols. 440-441 (2003) pp. 269-276 © (2003) Trans Tech Publications, Switzerland,*
10. **S. P. Jung, T. W. Park, J. H. Lee, W. H. Kim, and W. S Chung,** *Finite Element Analysis of Themalelastic Instability of Disc Brakes, Proceedings of the World Congress on Engineering 2010 Vol II WCE 2010, June 30 - July 2, 2010, London, U.K.*
11. **Huajiang Ouyang, Wayne Nack, Yongbin Yuan, Frank Chen,** *Numerical analysis of automotive disc brake squeal, Int. J. Vehicle Noise and Vibration, Vol. 1, Nos. 3/4, 2005*
12. **Hao Xing,** *Squeal Analysis of Disc Brake System, Beijing FEAonline Engineering Co.,Ltd. Beijing, China, 4th ANSA & μETA International Conference*
13. **A Söderberg, U Sellgren, S Andersson**, *Using finite element analysis to predict the brake pressure needed for effective rotor cleaning in disc brakes, 2008-01-2565*
14. **Sivarao, M. Amarnath, M.S.Rizal, A.Kamely,** *An Investigation Toward Development Of Economical Brake Lining Wear Alert System, International Journal of Engineering & Technology IJET Vol: 9 No: 9*