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Topological Optimization of Automobile Rotor Disk Brake

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Abstract
An automobile disk brake is an essential mechanical system used to slow down and stop the vehicle. During the operation of the brake, a high amount of kinetic energy will generate a high amount of heat energy and forces. The generated heat increases the rotor temperature, which leads to disk brake rotor deformation due to the combined effect of mechanical forces and temperature. During the study, a disk brake rotor will be investigated for frictional forces acting on it and heat generated due to friction between calliper pads and rotor surface. Brake rotor dimensions can be optimized for strength with a reduction in weight using advance topology optimization method. Topology optimization is very useful engineering technique, especially at the concept design stage. Topology optimization is able to produce reliable and satisfactory results with the verified structural model. Topological optimization will be performed on ANSYS 18.1 software. New disk brake rotor shape will be produced with the Creo 3.0 based on the topology optimization result. The new disc brake rotor from a topology optimization result will compare with the traditional concept model and topology optimization base model. It will analyze that a new rotor will not fail during an experiment test, and these results will be verified with a fabricated real sample under the durability condition.

Keywords- *Disk brake, Ceramic materials, ANSYS, Topological optimization.*

I. INTRODUCTION

A vehicle requires a brake system to stop or adjust its speed with changing road and traffic conditions. The basic principle used in braking systems is to convert the kinetic energy of a vehicle into some other form of energy. As the brake linings contact the drums/rotors, they create friction which produces the heat energy. The intensity of the heat is proportional to the vehicle speed, the weight of the vehicle, and the quickness of the stop. Faster speeds, heavier vehicles, and quicker stops equal more heat. Disc-style brakes development and use began in England in the 1890s. The first calliper-type automobile disc brake was patented by Frederick William Lanchester in his Birmingham, UK factory in 1902 and used successfully on Lanchester cars. The disc brake is a wheel brake which slows the rotation of the wheel by the friction caused by pushing the brake. Pads against a brake disc with a set of callipers. The brake disc or rotor is usually made of cast iron, but may in some cases be made of composites such as reinforced carbon-carbon or ceramic matrix composites. Figure 1 shows the typical constriction of a floating calliper disk brake. The disk brake containsbrake pads, brake disk, housing, yoke and piston.

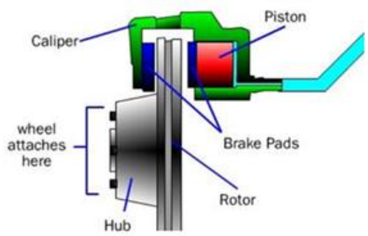

Figure 1: A typical constriction of disk brake

II. DISK BRAKE ROTOR

Brake disc, also called brake rotor, is fixed to the axle, so it rotates with the same speed as the wheel. The braking power of a disc brake is determined by the rate at which kinetic energy is converted into heat due to frictional forces between the pad and the disc. For an efficient brake design, it is also important that heat is dissipated as quickly as possible otherwise the temperature of a disc might rise and affect the performance of a disc brake. So to get optimum performance in demanding applications, ventilation is introduced in the brake disc, which increases the cooling rate.

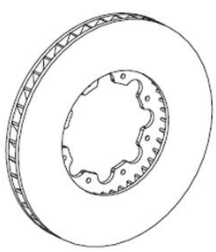

Figure 2 Schematic representation of different brake discs

In the present study, An investigation of disc brake squeal is performed by using the new complex eigenvalue capability of the finite element (F.E.) software ABAQUS version 6.4 [13]. This F.E. method uses nonlinear static analysis to calculate the friction coupling before the complicated eigenvalue extraction, as opposed to the direct matrix input approach that requires the user to tailor the friction coupling to the stiffness matrix. Thus, the effect of non-uniform contact and other nonlinear effects are incorporated. A systematic analysis is done to investigate the effects of system parameters, such as the hydraulic pressure, the rotational velocity of the disc, the friction coefficient of the contact interactions between the pads and the disc, the stiffness of the disc, and the stiffness of the backplates of the pads, on the disc squeal. The simulations performed in this work present a guideline to reduce the squealing noise of the disc brake system.

III. LITERATURE REVIEW

Friction-induced vibrations in automotive disc brakes are of substantial interest for academic research as wells for the industry. The numerous customer complaints due to brake noise cause high warranty costs in the automotive industry. To enable silent brakes to be developed, noise, vibration and harshness (NVH) engineers analyze these phenomena using computational and experimental simulations as well as vehicle tests. In the automotive industry, computational simulations have become increasingly important because of shorter product development processes as well as cost reduction necessities.

Composite materials replace conventional materials in so many fields due to their lightweight and secure processing. Nowadays, hybrid composite driveshafts are also used in replacement of the steel and aluminium for the preparation of these composites automotive parts. Synthetic fibres mainly carbon, glass, Kevlar has satisfactory strength properties coupled with relatively low cost, recyclability and biodegradability and are being used in automotive industries, construction as well as in packaging industries with few drawbacks. The low density of fibres allows fabrication of composites that gives excellent mechanical properties with a low specific mass. The increased interest in the use of fibre among researchers and technologist's has been well known. In the automotive industry, brake squeal has become a significant cost factor because of customer dissatisfaction. In North America, up to one billion dollars p.a. were spent on noise, vibration and harness (NVH) issues. From the literature, it is observed that many researcher and automobile industries are working on the reduction of noise and vibration.

S. Oberst and J. C.S. Lai studied the influence of geometrical parameters (namely, the number and location of slots) of brake pads on brake squeal noise. Four different brakes lining geometry were prepared (i) basic configuration without any slot (ii) basic configuration modified with a vertical slot in the mid-surfaces (iii) basic configuration with two slots and (iv) basic configuration with diagonals slot. This study reveals for the first time that severe nonlinearity is directly correlated with brake squeal and could be the reason for lousy noise performance.

T. Jearsiripongkul and D. Hochlenertstudied the mathematical-mechanical models for studying the dynamics of the brake of modern passenger's cars. A simplified model for the dynamics of a floating calliper disk brake is presented. The model includes the brake disk, modelled as a flexible rotating plate, calliper and brake pads. In the model, all the prominent features of squeal are reproduced, such as e.g. independence of the frequency on the speed, etc. For a moderately wide frequency range (1-5 kHz), the transverse vibration of the disk plays a significant role in a squeal. The pad stiffness and damping coefficient are modelled by distributed nonlinear springs and linear dampers, respectively. The development and laboratory implementation of the active squeal control goes along with a more profound understanding of brake squeal and better modelling of the phenomena, ultimately leading improvements in the design of disk brakes.

M. Nouby and K. Srinivasaninvestigated the influence of brake design parameters on brakes squeal. They studied by modifying the various structure of brake pad to reduce the squeal. The finite element method (F.E.M.) is used to simulate and predict the disc brake squeal using a complex eigenvalue analysis. An approach to examining the disc brake squeal based on the complex eigenvalue analysis is proposed in which a positive real part indicates that the corresponding Eigenmode is unstable and in turn, squeal may occur. From the several simulations done by complex eigenvalues analysis, it is observed that a higher coefficient of friction increases the likelihood of squeal. The squeal can be reduced by decreasing the stiffness of the backplates of the pads. The chamfer provided significant squeal reduction. To explain the effect of slot configurations on squeal, the understanding of the pressure contact distribution between the pad and rotor is required.

L. Rudolfexamined, the study of fade in conventional disc brakes, results from two basic causes. (1) The brake pads overheat, reducing their coefficient of friction which decreases braking ability, and (2) Excessive heat in the brake pads is transferred via the hydraulic pistons to the brake fluid, which boils and produces bubbles in the brake lines. The full circle disc Brake resists these fade inducing caused by (1) Distributing in-pad heat over a greater area and conducting heat both away from and through the brake pads into the brake body structure to enable more efficient heat dissipation, and (2) isolating the hydraulic cylinder from the brake pads so that direct heat is not transferred to the brake fluid.

IV. DESIGN AND ANALYSIS OF DISK BRAKE ROTOR

Grey cast iron is used for Maruti Suzuki Ecco passenger vehicle in disk brake rotor applications. The supplier gives the material properties of the grey cast iron.

The disk brake rotor is developed using Creo parametric 4.0 software using exiting dimension of Ecco disk brake rotor as given in table 4.4. All the dimension presented in table 4.4 is measured using a vernier calliper. Figure 4.1 and 4.2 show the 3D model of the disk brake rotor and 2D drawing of the rotor, respectively.

Table 4.4 Dimension of the grey cast iron disk brake rotor

Parameter name	Dimension
The outer diameter of the rotor disc	232 mm
The inner diameter of rotor discs	125 mm
Hole diameter	60 mm
The thickness of the rotor disc	17 mm
Calliper piston diameter	44 mm
Mass of disc	4.42 kg

Figure 3:3D model of a disk brake rotor in Creo 4.0

The structural analysis has been done on ANSYS software by the static structural module, as depicted in figure 4.3. Materials properties of the grey cast iron have entered manually from table 4.3, as shown in figure 4.4. By using the above design calculations, the modelling of the disc brake rotor is done as below, and its simple geometry is shown below. The model was imported in ANSYS, as shown in figure 4.5. In software, after entering the material properties, the geometry option was selected. For the analysis, the surface was suppressed, and only solid geometry is selected for further analysis.

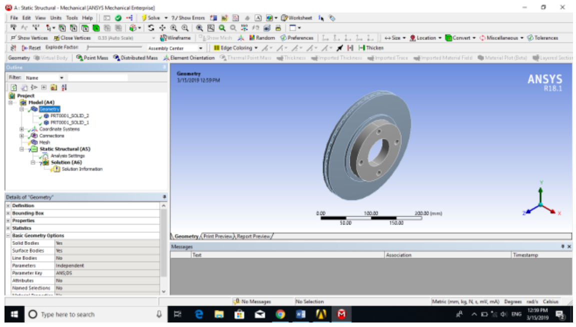

FIGURE 43D DRAWING IN ANSYS 18.1 SOFTWARE

Meshing is the process in which geometry is spatially discredited into elements and nodes. Results of the analysis also depend upon the numbers of nodes and element selected in the analysis. Mesh was refined to get good convergence of the load and displacement results. In the present study following mesh type and size has been selected, as demonstrated in figure 4.6. Meshing size is refined at the hole where the disc brake rotor is fixed with the wheel. The smooth mesh type is selected to get good to converge in the result, but the larger mesh size also takes more time to get a solution. Figure 4.7 shows the refined mesh model at holes and face in ANSYS 18.1 software.

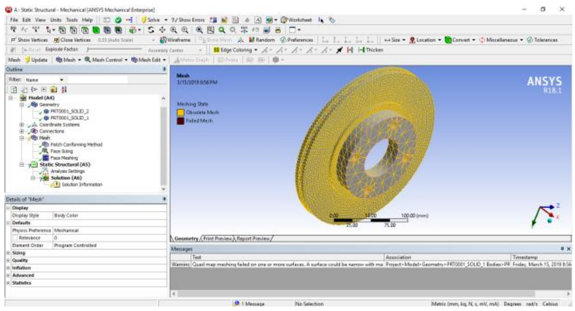

FIGURE 4.9 meshed disc rotor in ansys 18.1 software

Maximum load condition for disc brake rotor occurs while applying the brake to deacceleration the moving vehicle. The disc brake rotor is connected with the wheel by bolts behaves as a fixed body offering zero displacements and withstand during a braking operation. Hence significant boundary conditions that may apply for analysis are (i) gravity/weight acting downward (ii) rotation velocity/moment and (iii) fixed support. As in case of a brake is applied by the driver which transfer to brake rotor by piston arrangement, brake rotor is fixed at the wheel by the bolt is considered as fixed, which is having zero displacements in all the direction and braking torque is applied at both the side of the disc as shown in figure 4.8.

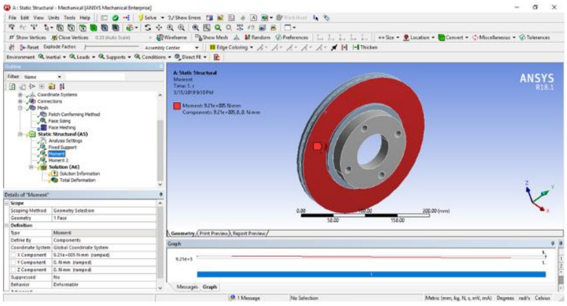

Figure 4.8 Boundary and loading condition in ANSYS 18.1

IV. RESULTS AND DISCUSSION

In the present F.E.A. study total deformation, equivalent stress, an equivalent strain is considered for evaluating the results. The total deformation of the grey cast iron rotor is calculated and the values obtained are the maximum deformation is 2.009 e-5 m, and the minimum deformation is 0, as shown in figure 4.9.

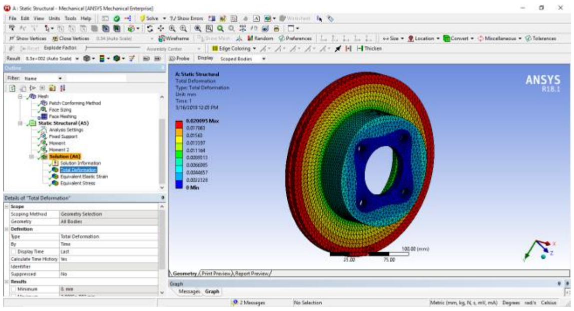

Figure 4.9 Total deformation of grey cast iron rotor disc

The equivalent elastic strain of the grey cast iron rotor is calculated and the values obtained are the maximum strain is 0.0005185, and the minimum deformation is 6.8733e-8 as depicted in figure 4.10.

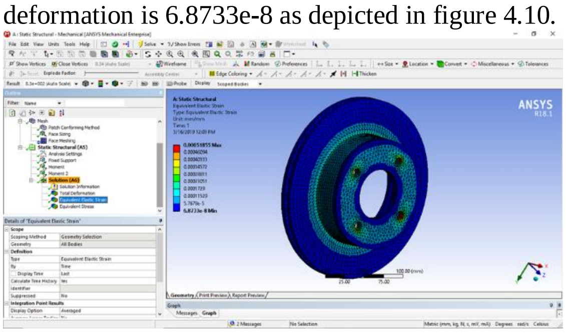

Figure 4.10 Equivalent elastic strain of grey cast iron

The equivalent elastic stress of the grey cast iron rotor is calculated and the values obtained are the maximum stress is 64.686 MPa, and the minimum stress is 0.0040816 Pa, as shown in figure 4.11.

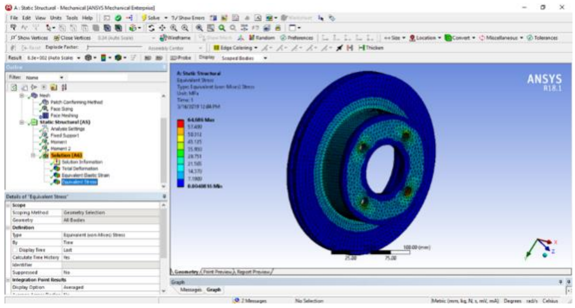

Figure 4.11 Equivalent elastic stress of grey cast iron rotor disc

A. *Weight Reduction of the rotor disc*

The design pattern chosen for the brake disc has plenty of room for alterations, to make it even more lightweight. For example, more number of holes could be added, or the diameter of existing holes could be enlarged, with that the thickness of the ventilation slots could also be increased. Other than the pattern, the thickness of the brake disc itself could be reduced. In other methods is to replace conventional materials by composite materials to reduce the weight. The brake disc made of Cast iron is having weight about 4.42 kg, but the material chosen for our design is ceramic-carbon composite materials has weight about 65% lighter than cast iron. Therefore the weight of the brake disc is e to be about 1.51 kg, as shown in table 4.6 and figure 4.16.

Table 4.6 Comparison between grey cast iron and ceramic-carbon composite rotor

	Grey cast iron	Ceramic-Carbon Composite
Weight	4.42 Kg	1.51 Kg
Maximum Deflection	2.009 e-5 m	2.367 e-5 m
Equivalent elastic strain	0.0005185 mm/mm	0.00068605 mm/mm
Equivalent elastic stress	64.686 MPa	65.067 MPa

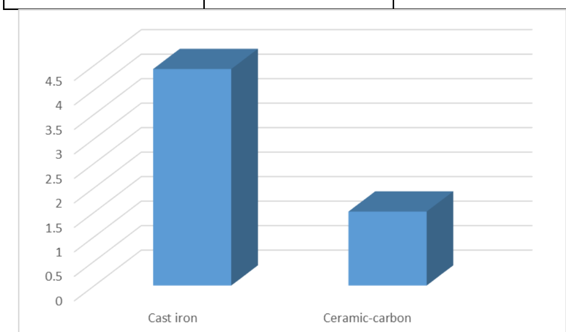

Figure 4.16 Weight comparison of cast iron and ceramic-carbon rotor

V. CONCLUSION

Topological optimization was done on ANSYS 18.1 for disk brake rotor having materials of grey cast iron. Grey cast iron has a weight of 4.42 kg which may be replaced by a ceramic composite material for disk brake having a weight of 1.51 kg materials.

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