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STRENGTH ANALYSIS OF A VENTILATED BRAKE DISC-HUB ASSEMBLY FOR A MULTIUTILITY VEHICLE.

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Abstract - *Disc brakes are exposed to large stresses during routine and hard braking. Disc brakes generate an opposing braking torque there by converting kinetic energy to heat. High-g decelerations typical of passenger vehicles are known to generate stresses of higher magnitude. Thus the strength analysis of disc brakes is more critical for the designing the brake rotor. In this paper the stresses induced in the disc brake due to the braking torque, bolt preload and combined braking torque and bolt preload are simulated using the finite element model of the disc brake together with hub . The disc brake design can be optimized using the results obtained from the simulation. The finite element model comprises of both ventilated disc brake and hub. The brake disc and hub are modeled using 3D tetrahedral elements. The material cast iron is taken for both disc and hub. Rigid body elements and beam elements are used for modeling the bolts. The shell coat is provided over the model for transmitting the brake torque effectively. The bolt preload is applied using thermal effects.*

Key Words: Disc brakes, Ventilated disc rotor with hub, Durability, Bolt Pretension, Braking Torque, Fading

1. INTRODUCTION

Disc brake assembly consist of cast iron disc rotor that rotates with the wheel, caliper assembly attached to the steering knuckle and disc pads made up of friction materials are mounted to the caliper assembly. Warping, scarring, cracking and excessive rusting are the most commonly observed failure in disc brake rotor. This paper presents a failure analysis of disc brake rotors due to the applied braking torque, bolt preload and combined braking torque and bolt preload.

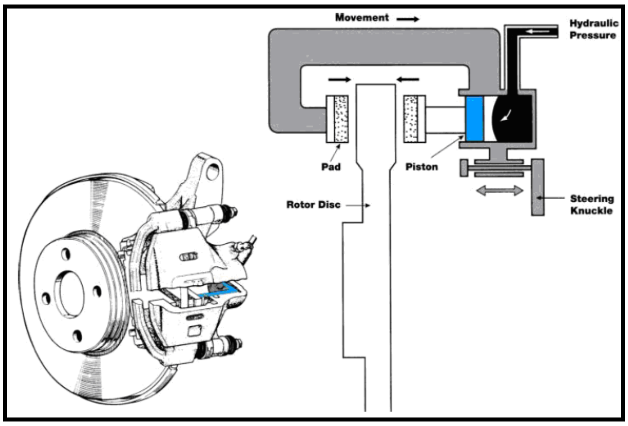


Fig -1: Disc Brake Assembly

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 the braking power as drum brakes, they are less likely to cause a pull.

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.

2. VENTILATED DISC ROTOR

The disc rotor is usually made of grey cast iron and is either solid or ventilated. Grey cast iron is chosen for its relatively high thermal conductivity, high thermal diffusivity and low cost. The brake rotor consists of a hat, or hub which is connected to the wheel and axle and an inboard and outboard braking surface. The outboard braking surface is attached directly to the hat, while the inboard braking surface is attached to the outboard unit by a series of cooling vanes. A small groove is machined around the periphery of the hat-rotor attachment site to relieve the stress concentration associated with the change in section. The inboard disc is not directly attached to the hat; attachment to the hat is through the cooling vanes. The inboard and outboard rotors are squeezed by the brake pads during braking. The subsequent frictional work arrests the rotation of the wheel and generates a substantial amount of heat. The ventilated type disc rotor consists of a wider disc with cooling fins cast through the middle to ensure good cooling. Proper cooling prevents fading and ensures longer pad life. Some ventilated rotors have spiral fins which creates more air flow and better cooling. Spiral finned rotors are directional and are mounted on a specific side of the vehicle. Ventilated rotors are used on the front of all late model Toyotas.

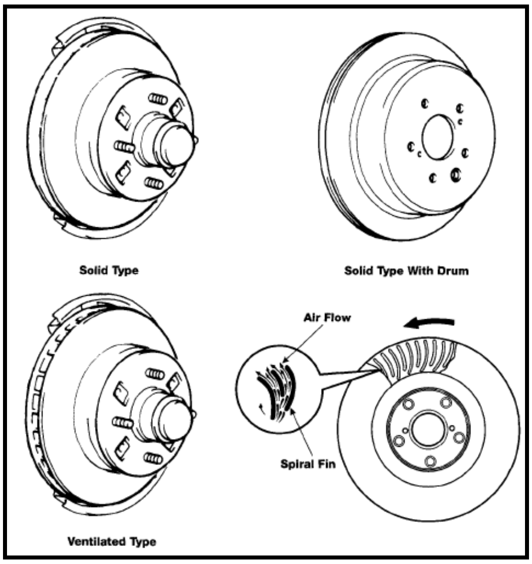


Fig -2: Disc Rotor Types

3. BRAKING TORQUE (M_f or M_b)

During braking, the kinetic energy of the moving vehicle is converted in to heat energy in a timely and repeatable fashion. In order to estimate the stresses induced in the disc brakes that arise during braking, it is necessary to calculate the frictional forces acting on the brake rotors. The kinetic energy of a moving vehicle is expressed as

$$K.E = K\ \frac{m\,v^2}{2} \qquad (1)$$

Where K - Inertia Coefficient, m - Mass of Vehicle in Kg.
v - Speed of the Vehicle in m/s
The amount of braking work required to stop the vehicle will be same as that of kinetic energy possessed by the vehicle during braking. The braking work changed in to heat and dissipated in to the surrounding from the disc.

Table -1: Vehicle Data

Vehicle mass (unladen, With Driver, and 2/3 full tank), m	1445 Kg
Vehicle mass (laden)	1980 Kg
Initial Velocity, V_1	160 kmph
Deceleration, a	5.89 m/s2
Time to stop, t	7.55 secs
Effective rolling radius, $r_{rolling}$	300 mm
Load Distribution on Front Axle	70%
Load Distribution on Rear Axle	30%
Inner radii of the friction surface, r_{inner}	122 mm
External radii of the friction surface, r_{outer}	166 mm

3.1 Fading Procedure

The fading procedure for the vehicle is the vehicle stops from 160 kmph to 0 kmph with deceleration of 0.6 g in 32 secs. The stopping distance for the vehicle is given by

$$\Delta x = \frac{V_2^2}{2a} = \frac{(160\,X\,1000/3600)^2}{2X\,5.886} = 167.8\,m \qquad (2)$$

The stopping time for the vehicle is given by

$$\begin{aligned} &V = U + a \times t \\ &44.44 = 0 + 5.886 \times t \\ &t = 7.55 \sec s \end{aligned} \qquad (3)$$

The tangential force acting on one front wheel surface during braking is given by

$$F_{Tangential} = \frac{0.7 \times 1.1 \times mg}{2} = 5457.6\,N \qquad (4)$$

The frictional or braking torque acting on front wheel surface during braking is given by

$$\begin{aligned} M_f &= F_{Tangential} \times r_{rolling} \\ &= 5457.6 \times 300 \\ &= 1.64 \times 10^6\ N - mm. \end{aligned} \qquad (5)$$

The braking torque of is applied at the centre of the disc brake finite element model using the rigid elements.

4. BOLT PRETENSION

The bolts are used for attaching the disc rotor to the hub. The bolts are used for transmitting the braking torque applied to the disc rotor to the wheel hub where the wheel is mounted. During assembling of the disc brake with the wheel hub, the bolts are subjected to pretension and the tensile forces due to the bolt pretension along with the braking torque will induce stresses in the disc rotor. The analysis carried out in this research includes the following load cases (1) Braking Torque (2) Bolt Pretension (3) Combined braking torque and bolt pretension.

Table -2: Disc Hub Assembling

No of bolts used for disc hub assembly	5
Type of bolt used for disc hub assembly	M14
Bolt tightening torque (T_{bolt})	104E+03 N mm
Poisson ratio (γ)	0.28
Thermal expansion coeff of Bolt (α)	6.94E-09 1/ °C
Young's modulus of Bolt Material (E)	120E+06N/mm^2

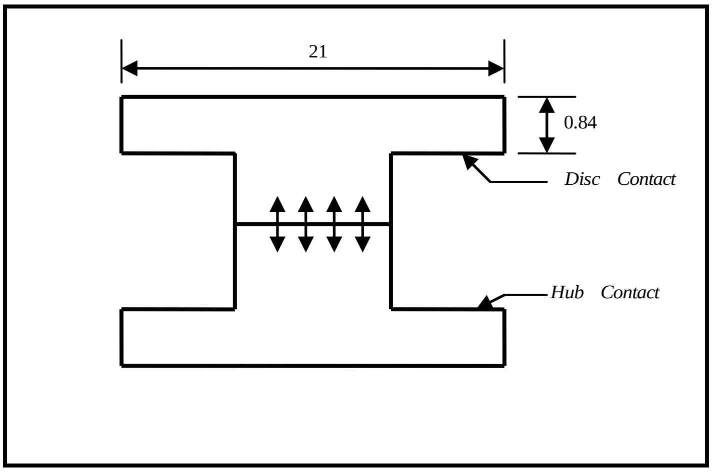


Fig -3: Bolt Pretension

The expression for the normal forces (F_N) and tangential forces (F_T) acting on the bolt during assembling is given as follows.

Bolt tightening torque (T_{bolt})
=Tangential force (F_T) X Bolt Radius (6)

104E+03 = Tangential force (F_T) X (14/2)

Tangential force (F_T) = 15E+03 N.

Tangential force (F_T)
= Poisson ratio (γ) X Normal force (F_N) (7)

15E+03 = 0.28 X Normal force (FN)

Normal force (F_N) = 53.57 E+03 N.

The normal force of 53.57 E+03 N is applied as a bolt preload in the beam elements of disc brake finite element model.

5. FEM MODEL -DISC ROTOR AND HUB

As the braking torque is shared by both the disc rotor and hub, the FEM model includes both the rotor and hub assembly. The FEM model of the disc rotor and hub comprises of second order tetrahedral elements. The solver that uses second order elements consume more time but ensures high accuracy. In this research, the disc rotor and hub are modelled by using second order tetrahedral elements for the accurate representation of given geometrical surfaces.

The bolts used for assembling the disc rotor with hub are modeled by beam and RBE2 elements. The RBE2 defines a rigid body with one independent node and an arbitrary number of dependent nodes. RBE2 elements are normally used to model areas that are very stiff compared to the adjoining structure in order to prevent numerical ill-conditioning and to simplify the model.

The braking torque of is applied at the centre of the disc brake finite element model using the RBE3 elements. RBE3 is an interpolation element used to model the motion of one node (the dependent node) as the least square weighted average motion of an arbitrary number of independent nodes. The intended use of the RBE3 is to transmit forces and moments from a reference node to several non-collinear nodes without adding any stiffness to the structure. The solid elements have only 3 degree of freedom, so transmitting the rotation of braking torque will need shell elements. A two dimensional shell coat is provided over the FEM model of the disc and hub surface to transmit the braking torque from the centre of RBE3 element to the disc and hub surface.

6. LOADS AND BOUNDARY CONDITION

The tetrahedral elements of the disc surface in the contact pad region are constrained in the translational and vertical degree of freedom. The rear side of hub with axle shaft is constrained in lateral direction. The braking torque is applied at centre of the disc-hub finite element model using the rigid elements RBE 3.The bolt preload for disc hub is applied by thermal effects using Nastran.

Thermal loads may be used to represent preloading of elements in MSC/Nastran because a preload entry does not exist. The temperature ΔT required to apply to the model to yield bolt normal force of 53.57 E+03 N is given by

$$\frac{F_N}{A*E*\alpha} = \Delta T \qquad (8)$$

Where A – bolt area (π * 14^2/4 = 153 mm^2)

$$\frac{53.57E3}{153*120E6*6.94E-9} = 418^o\ C$$

The temperature ΔT = 418 °C required for bolt pretension is applied using TEMPRB card in MSC/Nastran. The TEMPRB assumes that the bolt is modeled by a Rod, Bar, Beam, Conrod or Tube element.

The required bulk data entries in MSC/Nastran are given by

Table -3: Temperature input using TEMPRB card

1	2	3	4	5	6	7
TEMPRB	SID	EID1	TA	TB		
TEMPRB	12367	4566	- 418	- 418		

The material data entries are given by

Table -4: Material Properties for Grey Cast Iron.

1	2	3	4	5	6	7	8
MAT1	MID	E	G	NU	RHO	ALPHA	TREF
MAT1	1000	120 E+06		0.3	3714	6.94 E-09	0

The analysis is carried out initially by running a single load case applying only the thermal load for bolt pretension. The element forces are checked to verify the bolt normal force of 53.57 E+03 N. Similarly the analysis is carried out by running a load case of braking torque alone. After verifying the results in previous cases, the analysis for load case of combined braking torque and bolt pretension is done to determine the stresses induced in disc-hub assembly.

7. RESULTS AND DISCUSSION

The analysis is done initially for the bolt pretension alone with disc hub model. The bolt normal force is applied as a temperature difference using TEMPRB card. The stress produced is of lower magnitude which is well below the yield limit of grey cast iron. The contour stress plot for the bolt preload is shown in the fig 4.

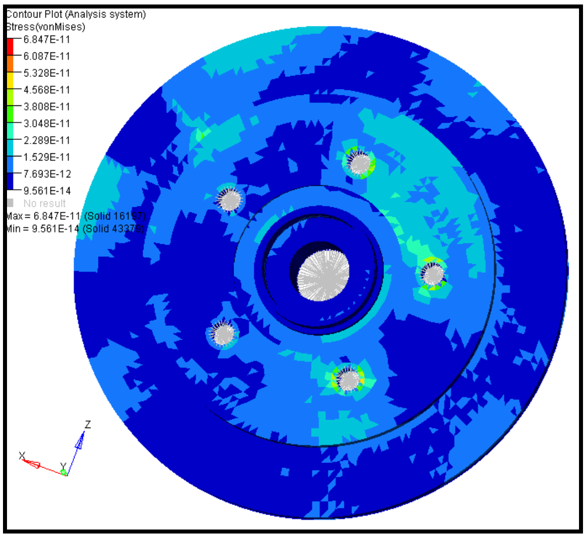
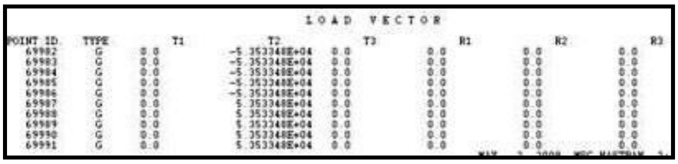


Fig -4: Vonmises Stress Plot for Bolt Preload

The bolt normal force applied using thermal effects can be verified in printed result file fo6 in Nastran. The load vector in fig 5 shows the bolt normal force 53.57E+03 N which is applied initially using TEMPRB card as the temperature difference of 418 ° C.



LOAD VECTOR

POINT ID.	TYPE	T1	T2	T3	R1	R2	R3
69982	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69983	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69984	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69985	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69986	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69987	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69988	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69989	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69990	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69991	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0

Fig -5: Load Vector Showing Normal Force of 53.57 E+03 N for Bolt Preload

Finally the analysis is done for the combined load case of braking torque and bolt preload. The braking torque is applied at centre of brake disc and hub. The max stress of 73 N/mm² occurs in contact pad region which is below the yield limit of grey cast iron. The induced stress ensures that disc hub design is safe for the applied load of combined braking torque and bolt preload. The contour stress plot for the combined load case of braking torque and bolt preload is shown in the fig 5.

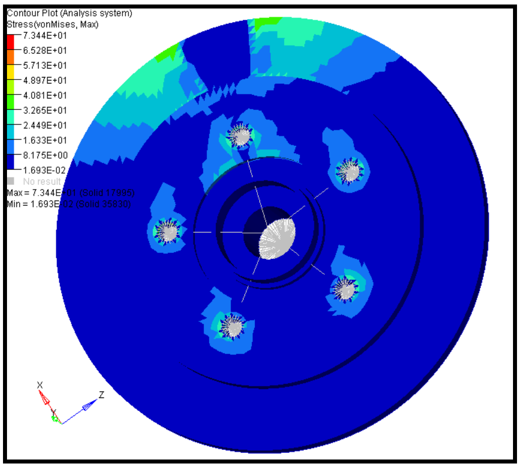
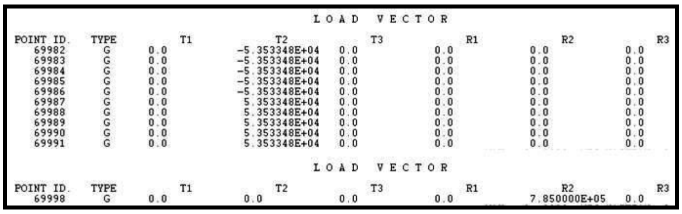


Fig -5: Vonmises Stress Plot for Combined Braking Torque and Bolt Preload

The bolt normal force and braking torque can be verified in printed result file fo6 as shown in fig 6. The load vector in fig 6 shows the bolt normal force 53.57E+03 N and braking torque 0.785E+06 N mm for rear wheels.



LOAD VECTOR

POINT ID.	TYPE	T1	T2	T3	R1	R2	R3
69982	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69983	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69984	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69985	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69986	G	0.0	-5.353348E+04	0.0	0.0	0.0	0.0
69987	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69988	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69989	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69990	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0
69991	G	0.0	5.353348E+04	0.0	0.0	0.0	0.0

LOAD VECTOR

POINT ID.	TYPE	T1	T2	T3	R1	R2	R3
69998	G	0.0	0.0	0.0	0.0	7.850000E+05	0.0

Fig -5: Load Vector Showing both Bolt Preload and Braking Torque

8. CONCLUSIONS

This research paper depicts the strength analysis of ventilated brake disc hub assembly subjected to braking torque and bolt pretension. The induced stresses for bolt pretension is negligible compared to the braking torque. The stress induced for the combined bolt pretension and braking torque for rear is 73 N/mm ^{2} which is satisfying the acceptance criteria. The Load vector showing the bolt pretension and braking torque verifies the applied load. The analysis done here can be used for the virtual design validation of disc brake – hub assembly for the load cases described. The disc brake design can also be optimized based on the results of the structural analysis.

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