Krishnan et al. 2016. *Int. J. Vehicle Structures & Systems*, 8(4), 255-259
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.8.4.13
© 2016. MechAero Foundation for Technical Research & Education Excellence

International Journal of Vehicle Structures & Systems
Available online at www.maftree.org/eja

Numerical Analysis of Novel Design for Ventilated Brake Disc

A. Krishnan[a], S.K. Sampat, H. Vinod, M.S. Kumar and P.K. Aswin Babu

Dept. of Mech. Engg., Amrita School of Engg., Amrita Vishwa Vidyapeetham, Amrita University, India
[a]Corresponding Author, Email: aravindkrishnan@am.amrita.edu

ABSTRACT:

Design of braking system is an important phase in the development of an automobile. During braking, disc brakes accumulate heat energy and high thermal gradients which leads to reduced functionality. This heat has to be dissipated for efficient functioning of brakes. In this paper, a novel design for brake is developed using Solid Works tool. This model aims at maximum dissipation of heat by increasing surface area. A thermal flux analysis of the new model is done to confirm this achievement. A comparative study of other designs of ventilated discs and a plane rotor disc is also done. The calculation and verification of braking force is followed in designing the disc rotor. The main objective is to generate more braking force so as to maximize efficiency. The analyses are done in ANSYS to study and understand the heat flux and temperature that are formed in the disc rotor.

KEYWORDS:

Brake disc design; Numerical simulation; Heat flux analysis; Temperature study; Design optimization

CITATION:

A. Krishnan, S.K. Sampat, H. Vinod, M.S. Kumar and P.K.A. Babu. 2016. Numerical analysis of novel design for ventilated brake disc, *Int. J. Vehicle Structures & Systems*, 8(4), 255-259. doi:10.4273/ijvss.8.4.13.

1. Introduction

Braking system is the most significant safety system of an automobile. Friction brakes are the most commonly used breaking systems. During braking, the kinetic energy of the brakes is converted into heat energy and so, large amount of heat is produced due to friction on the discs and adjacent components. The better the braking more is the heat flux generated and more the thermal stresses. This heat flux can cause the brake fluid to vaporize and can damage other components. This heat flux has to be handled, i.e. carried away quickly from the system. This happens in three ways, conduction, convection and radiation. Convection is the most common method for heat transfer. When heat accumulates on components, it causes overheating and damage to the components. In the past, many studies have been conducted to study the effects of heats and thermal effects. Li, Han et al [1] analysed the mechanisms of thermal fatigue and phase change of steel discs in brake discs.

Belhocime et al [2] conducted a thermal analysis of rotor discs with ventilation. The objective of this paper is to modify the design of the brake disc so as to increase the heat lost by convection. A plain disc is analysed as a brake disc rotor, and a novel design is developed to increase the heat flow through convection by increasing surface. Saiz et al [3] conducted thermal analysis of different designs of brake discs and compared them. Nakatsuji et al [4] describe how cracks, which form around small holes in discs, propagate during overloading. Gao et al [5] studied the thermal fatigue fracture in brake discs. Airflow patterns around ventilated disc patterns were studied by Michael et al [6]. Valvano [7] discusses an analytical method to predict thermal distortion in a disc rotor. Talati et al studied the effect of heat conduction of brake system. In this study, CAD models using Solid Works and ANSYS are used to develop new models and determine the heat flux around each of them. Ventilated disc brakes are used to increase the flow through brakes and carry away heat quickly.

Their effects were studied by Huwng et al [8] and Gallindo et al [9]. In this paper, a novel design based on certain requirements of the vehicle (SAE Baja all-terrain vehicle) is presented. The objective is two-fold, to increase the heat convection and also to reduce the overall weight of the vehicle, keeping the strength of the component in consideration. Reducing weight of the component is achieved by removing material from the component by cutting holes of different shapes. It is also required to increase overall surface area to increase convection effects to help cool the component faster and reduce the thermal effects. This is achieved by providing coolant holes through its thickness. This is an alternative to fins in the past designs. Removing too much material will lead to overall decrease in strength of the component resulting in failure. This design helps reducing thermal flux, without compromising the strength of the component.

2. Theory

The analytical study and theory behind the calculation of heat flux and breaking forces are discussed in this section. The expression for heat flux in a brake disc is obtained using two approaches. A general analytical energy based solution and a specific equation obtained by Reimpel [10].

2.1. Braking force

All modern disc brakes systems rely on brake pads pressing on both sides of a brake rotor to increase the rolling resistance and slow the car down. The amount of frictional force is found by multiply the force pushing the pad into the rotor by the coefficient of friction of the pad. The force slowing the brake disc is given by,

$$F_{rotor} = 2C_{f\,pad}F_{pad} \tag{1}$$

The braking calculations are based on a set of conditions set from an experimental track to test the automobile. The details of the track and set are shown below. The stopping distance is measured for the same acceleration for six different speeds. The coefficient of friction on the road is determined to be 0.8. The deceleration is obtained to be 7.845 m/s^2. Braking force is 2354N. The other brake calculations were done to obtain the dynamic weight transfer, brake torque and the clamping force are calculated based on the type of braking, pads used and locking type. The characteristics of braking are summarized in Table 2.

Table 1: Brake test details

Velocity (m/s)	Acceleration (m/s^2)	Stopping distance (m)
30	7.84532	4.4254
40	7.84532	7.8687
45	7.84532	9.9581
50	7.84532	12.2924
55	7.84532	14.8742
60	7.84532	17.7019

Table 2: Braking characteristics

Braking circuit	Calculations
Diagonal split (X split)	Dynamic Weight transfer Front: 204.4kg, Rear: 95.59kg
1” master cylinder, Stroke = 16.5mm	Tyre rolling radius = 11.5”
Ventilated and petal rotor, diameter = 200mm	Brake torque, Front: 234.29Nm, Rear: 109.57Nm
DOT4 Brake fluid	Caliper clamping force
Brake caliper, Bore 21, 2 piston	Simultaneous locking
Brake caliper pad, Mean braking radius: 99.5 mm, Area: 692.35 mm^2, Friction coefficient: 0.4	Pedal Force: 356, Pedal ratio: 6:1, Pedal Travel: 99 mm

Total braking force is given by,

$$F = \mu mg \tag{2}$$

Based on $F = ma$, the deceleration is given by,

$$a = \mu g \tag{3}$$

The weight transferred during braking is given by,

$$\Delta W = \frac{h}{l}\frac{w}{g}d_x \tag{4}$$

Where w is weight, h is the height of center of gravity (cg), l is the wheelbase, C is the distance of cg from rear axle and b is the distance of cg from front axle. The static front and rear loads are given by,

$$W_{fs} = wc/l \tag{5}$$

$$W_{rs} = wb/l \tag{6}$$

Dynamic front and rear axle loads are given by,

$$W_{fs} + \Delta w \tag{7}$$

$$W_{rs} + \Delta w \tag{8}$$

The braking torque is given by,

$$T = FR_t \tag{9}$$

Where R_t is the tyre rolling radius. The pressure inside master cylinder is given by,

$$P = F_p b / A_p \tag{10}$$

Where F_p is the pedal force and A_p is the area of piston. The force on the caliper is given by,

$$F_c = PA_c \tag{11}$$

Where A_c is the area of piston of brake caliper. The braking force by the caliper is given by,

$$N = \mu F_c \tag{12}$$

The brake effective radius is given by,

$$R_{eff} = R - R_c \tag{13}$$

Where R is the outer radius of disc rotor and R_c is the radius of piston of caliper. Equating braking torques,

$$FR_t = NR_{eff} \tag{14}$$

From Eqn. (14), we get the minimum required size of the brake disc.

2.2. Heat flux

It can be assumed that all the kinetic energy lost is converted into heat energy during braking. Hence,

$$\Delta KE = \frac{1}{2}m\left(v_f^2 - v_i^2\right) \tag{15}$$

Also,

$$\left(v_f^2 - v_i^2\right) = 2ad \tag{16}$$

Where a is the deceleration and d is the linear distance travelled.

$$\Delta KE = mad \tag{17}$$

From Jazar [11], total thermal energy is a function of change in KE,

$$Q_{wheel} = \frac{1}{2}f\Delta KE \tag{18}$$

where f is the dynamic load distribution factor. Hence heat flux $q = Q_{wheel}/\Delta t$ and Δt is the wheel running time. The initial heat flux q_0 on the rotor face can also be obtained by the expression given by Reimpel,

$$Q_0 = \frac{(1-\phi)}{2}\frac{mgv_0 z}{2A_d\varepsilon_p} \tag{19}$$

Where $z = a/g$ is the braking effectiveness, ϕ is the rate distribution of the braking forces between the front and rear axles, A_d is the disc surface area swept by a break pad, v_0 is the initial speed of the vehicle, ε_p is factor load distributed on the disc surface, m is the mass of the vehicle and g = 9.81 m/s^2 is the acceleration of gravity.

3. Design

In this paper two designs are considered, chosen with the intention of reduction of temperature, stresses and weight. A plain outer edge design with lower number of coolant holes and a petal outer edge design with more number of coolant holes are developed. These designs are shown in Figs. 1 and 2. The other major difference is the thickness. The thickness of the baseline design was 4 mm which was reduced to 3.4 mm to reduce weight. The outer edge is modified to petal-shape and the center cavity is also modified to reduce weight. Various designs proposed in the past use methods to provide better convection by increasing surface area or providing holes to let airflow. Saiz et al [3] shows four different designs of ventilated braked discs. Fins are used to increase surface area, straight fins, curved fins and pillar shaped fins are studied and their effects on heat flux are described. In this paper we provide holes on the surface which let air flow and act as coolant for the disc. Air can flow through these coolant holes and heat is carried away. The holes are aligned in a curved shape from the center of the disc to the edge. The effect of holes as coolant is studied by Feng et al [13].

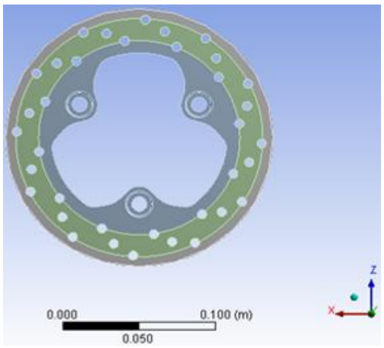

Fig. 1: Proposed baseline design

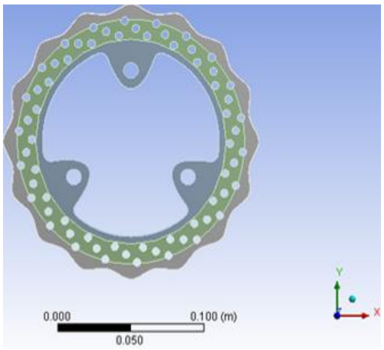

Fig. 2: Proposed improved design

When designing the brake disc rotor, the weight of the component is also to be considered. Weight is reduced by taking out material from the center. This is done after a feasibility study on how much material should be removed. The disc is clamped using a 3 hole clamp, hence the shape contains three clamp holes and cut outs in between. This shape increases the surface area compared to a plain hole. Further, material is removed from the outer edge keeping the total diameter constant using the outer petal shape. The shape is arbitrarily chosen to achieve the objective of reduced weight. Both these models are being studied in this paper. The next factor considered is strength of the disc. This has to be considered when the disc involves thin sections, which is a result of cutting out large sections of materials from a disc. Most modern brake disc designs have better convection achieved by removing material from the center. However, this creates very thin sections on the disc, which reduces the strength of the component, making it susceptible to high stresses or bending. Hence it is decided not to go for thin sections. The design consists of a center cavity with three clamp holes. The shape between clamp holes ensures maximum surface area. Keeping the total diameter constant, material is removed from the outer edge making the edge in the shape of flower petals.

4. Material

The widely used brake rotor material is cast iron which consumes much fuel due to its high specific gravity. The braking system is a vital safety component of ground-based transportation systems; hence the structural materials used in brakes should have some combination of properties such as good compressive strength, higher friction coefficient, wear resistant, light weight, good thermal capacity and economically viable [14]. The materials used for brake discs are cast iron, titanium alloys and Aluminum-Metal matrix composites (AMC). Grey cast iron is usually used with dissolved carbon within its matrix between 2% and 4.5%. It is economically viable, thermally stable and easy to manufacture. Titanium alloys have a potential of reducing total weight of brake disc significantly. It has high temperature strength and better resistance to corrosion. AMC materials having a lower density and higher thermal conductivity as compared to the conventionally used gray cast irons are expected to result in weight reduction of up to 50-60% in brake systems.

The repeated braking of the AMC brake rotor lowered the friction coefficient μ and caused significant wear of the brake pad. The friction properties of the AMC brake disc are thus remarkable poorer than those of conventional brake disc. Hence the material chosen for brake disc is AISI 321 annealed stainless steel, which rectifies the short comings of both cast iron and Ti alloy. The properties are shown in Table 3.

Table 3: Material properties

Material	AISI 321 stainless steel
Density	0.01g/mm^3
Poisson's ratio	0.27
Young's Modulus	200GPa

5. CAD model and analysis

The CAD model was created using Solid Works. The total diameter of the disc is 180mm and the thickness is 3.40mm. The hole diameters are 5mm. The clamp diameters and the center cavity dimensions are 16.80mm. A triangular mesh was applied and the element size was fixed after a convergence test. The meshed models for the baseline and improved designs are shown in Fig. 3 and Fig. 4. The simulation of the braking action was setup in ANSYS in a static thermal module on the models described above. The analysis was done by fixing the center node and providing an angular velocity of around the component. Convection was applied as the boundary condition with the atmospheric air set to a temperature of 20°C. The analyses have been done, the one on the model with a plain outer disc and the other on the model with flower petal design on the outer edge. The simulation is done to obtain the temperature patterns and heat flux patterns on the brake discs. The next simulation was done to obtain the stresses by considering both thermal stresses and the braking force on the discs. This gives the stress contours on the disc.

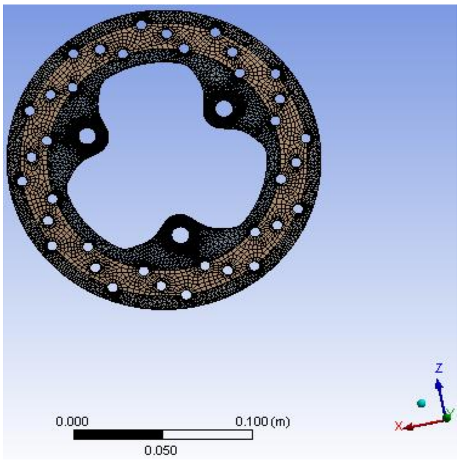

Fig. 3: Mesh of baseline design

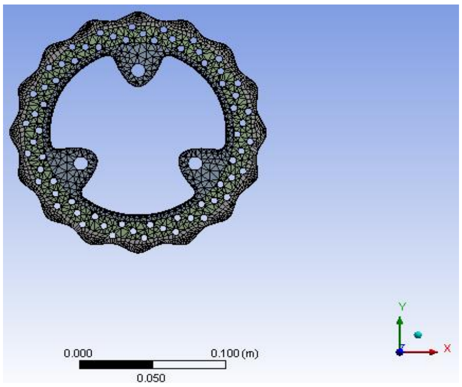

Fig. 4: Mesh of improved design

6. Results and discussions

The plots for maximum temperatures are shown in Figs. 5 and 6. Both the designs are analyzed and we see that the maximum temperatures are much more for the old design. The new design has been successful in reducing both the maximum temperature and the heat flux in the component. The maximum temperatures are found on the outer edge where the brake pads are in contact with the disc. The brake pads are placed slightly away from the outer edge in the improved design. This is done to avoid slippage due to the petal shape outer edge. The maximum temperatures are 62.325°C and 36.898°C for the baseline design and new design respectively. This shows a 40% decrease in temperature for the new design. Fig. 7 and Fig. 8 show the heat flux on the brake disc. Predictably, the heat flux is lower for the improved design. The coolant holes in the improved design are more in number. So the lower value of heat flux can be accounted to more number of holes and a higher surface area on the outer edge. The maximum values for heat flux are 1.0296×10^5 W/m^2 and 53895 W/m^2 for the two designs respectively.

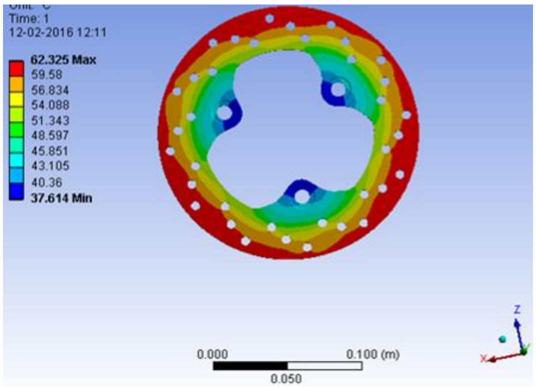

Fig. 5: Maximum temperature in baseline design

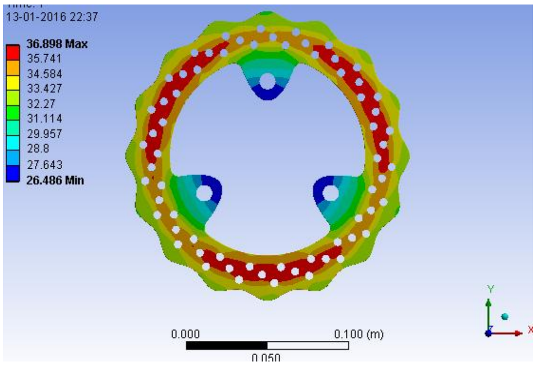

Fig. 6: Maximum temperature in improved design

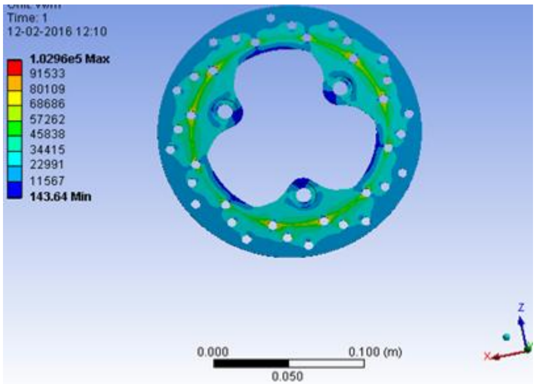

Fig. 7: Heat flux patterns in baseline design

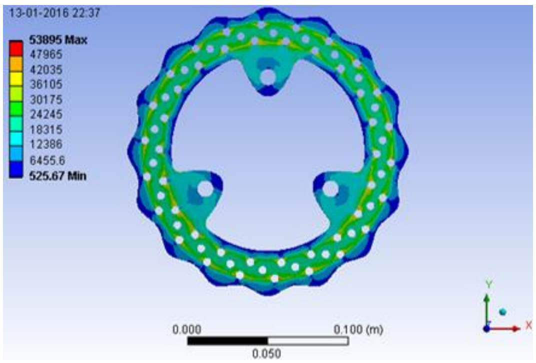

Fig. 8: Heat flux patterns in improved design

7. Conclusions

From the above results, we can conclude that the newly developed design has reduced the maximum temperature and heat flux of the disc during braking. The various factors that contribute to this are number of holes, surface area and thickness. The maximum temperature and heat flux attained on a disc can be reduced by increasing the total surface area. This is done by changing the shape of the center cavity and introducing the petal shape outer ring. The heat flux on the ring can be reduced by designing more coolant holes, allowing more air to pass through the components, hence carrying away more heat. The weight of the component was reduced significantly by removing material from both the center cavity and the petal shaped outer edge. This helps overall weight reduction and decreases fuel consumption.

REFERENCES:

[1] Z. Li, J. Han, Z. Yang and W. Li. 2015. Analyzing the mechanisms of thermal fatigue and phase change of steel used in brake discs, *Engg. Failure Analysis*, 57, 202-218. https://doi.org/10.1016/j.engfailanal.2015.07.002.

[2] A. Belhocine and B. Mostefa. 2012. Simulation of fully coupled thermomechanical analysis of automotive brake discs, *Simulation*, 88(8), 921-935. https://doi.org/10.1177/0037549711429975.

[3] C.B. Saiz, T. Ingrassia, V. Nigrella and V. Ricotta. 2015. Thermal stress analysis of different full and ventilated disc brakes, *Frattura ed Integrità Strutturale*, 34, 608-621.

[4] T. Nakatsuji, K. Okubo, T. Fujii, M. Sasada and Y. Noguchi. 2002. Study on crack initiation at small holes of one-piece brake discs, *SAE Technical Paper 2002-01-0926*.

[5] C.H. Gao and X.Z. Lin. 2002. Transient temperature field analysis of a brake in a non-axisymmetric three-dimensional model, *J. Materials Processing Tech.*, 129(1), 513-517. https://doi.org/10.1016/S0924-0136(02)00622-2.

[6] M. Hudson and R. Ruhl. 1997. Ventilated brake rotor air flow investigation, *SAE Technical Paper 971033*.

[7] T. Valvano and K. Lee. 2000. An analytical method to predict thermal distortion of a brake rotor, *SAE Techinical Paper 2000-01-0445*.

[8] P. Hwang and X. Wu. 2010. Investigation of temperature and thermal stress in ventilated disc brake based on 3D thermomechanical coupling model, *J. Mech. Science and Tech.*, 24(1), 81-84. https://doi.org/10.1007/s12206-009-1116-7.

[9] C.H. Galindo-Lopez and M. Tirovic. 2008. Understanding and improving the convective cooling of brake discs with radial vanes, *J. Automobile Engg.*, 222(7), 1211-1229. https://doi.org/10.1243/09544070JAUTO594.

[10] J. Reimpel. 1998. *Technologie De Freinage*, Vogel Verlag, Wurzburg.

[11] R.N. Jazar. 2013. *Vehicle Dynamics: Theory and Application*, Springer Science & Business Media.

[12] H.B. Yan, S.S. Feng, X.H. Yang and T.J. Lu. 2015. Roll of cross-drilled holes in enhanced cooling of ventilated brake discs, *Applied Thermal Engg.*, 91(5), 318-333. https://doi.org/10.1016/j.applthermaleng.2015.08.042.

[13] M.A. Malaque, S. Dyuti and M.M. Rahman. 2010. Material selection method in design of automotive brake disc, *Proc. World Congress on Engg.*, London, U.K.

[14] P.J. Blau, B.C. Jolly, J. Qu, W.H. Peter and C.A. Blue. 2007. Tribological investigation of titanium-based materials for brakes, *Wear*, 263(7-12), 1202-1211. https://doi.org/10.1016/j.wear.2006.12.015.