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Thermal Analysis of A Brake Disc Rotor

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Abstract — *The present study examines the phenomenon of heat distribution on the disc while braking. An attempt was made to study numerically, using CFD, the effect of vane-shape on the flow-field and heat transfer characteristics of a disc brake rotor for different configurations and at different speeds. Heat distribution on the brake disc is caused by the change of the kinetic energy into the mechanical energy. The energy change occurs during the process of braking due to the friction between the surfaces of the disc with the caliper pad. Friction results in the increase of temperature. This phenomenon is very important to be highlighted in order to learn the characteristics of heat distribution occurring on the various models of rotor discs. Thermal analysis of the rotor of disc brake is aimed to evaluate the performance of disc brake rotor of a car under different braking conditions. Also, the analysis for the variation in the fin configuration of vented disc brake is intended. Numerical simulations are conducted to determine the desired parameters. The simulation results would be useful for identifying the influence of different models of disc brake with regard to the distribution of heat.*

Keywords*-Rotor Disk, CFD, Brake Rotor*

I. INTRODUCTION

In today's growing automotive market the competition for better performance vehicle is growing enormously. The racing fans involved will surely know the importance of a good brake system not only for safety but also for staying competitive. In motor vehicles, one of the most critical safety systems for human life against accidents is the brake system. For this reason, the design and improvement of brake systems are of great importance. Car braking systems must perform the following fundamental tasks:

- Reduce the speed of the vehicle
- Bring the vehicle to a halt
- Prevent unwanted acceleration during downhill driving
- Keep the vehicle stationary when it is stopped

A brake system must fulfill its function under various working conditions such as iced, wet or dry roads, empty, partly or fully loaded vehicle, straight or curved roads, new or worn friction linings in brakes, experience or inexperience driver, etc. Therefore, a brake system must be designed in such a manner that all its components function efficiently and consistently with each other.[1] Each single system has been studied and developed in order to meet safety requirement. Instead of having air bag, good suspension systems, good handling and safe cornering, there is one most critical system in the vehicle which is brake systems. Without brake system, vehicle will put a passenger in unsafe position.[2]

A brake is a device by means of which artificial resistance is applied on to a moving machine member in order to retard or stop the motion of the member or machine. The main purpose of a brake system is to slow the vehicle or to stop it completely within a reasonable amount of time.[3]

The braking system of a vehicle is in charge of transforming the kinetic energy of movement into heat by means of the friction between the braking elements and dissipating this heat to the atmosphere. The energy which is transformed into heat is characterized by a total heating of the disc and pads during the braking phase. The energy dissipated in the form of heat can generate temperature rise ranging from 300°C to 800°C. The brakes must therefore store and dissipate all this heat into the surroundings before subsequent braking stages to have a good braking efficiency. Also, it has the function of decelerate gradually or quickly the vehicle in order to get the partial or total detention of the car according to the driving needs.[4]

II. AIM AND OBJECTIVE

Study of literature survey covers the various types of disc brake rotors like ventilated, solid, cross drill. Also it covers parameters which affects heat transfer rate of the disc brake rotors like size, pattern, material etc. From the literature review it was found that the frictional heat generated at the interface of the disc and the pads can cause a high temperature during the braking process. The braking performance is significantly affected by the temperature rise in the process of halting the vehicle. This phenomenon attracts the attention to work on this problem.

The aim of this project is to obtain the temperature rise and the temperature behavior of solid and ventilated disc brake rotor of vehicle. Focus has been set in maximizing the flow rate of air by changing the design of rotor to optimize heat transfer.

In achieving this aim, project objectives are set as below:

- To understand fundamental of heat transfer through thermal analysis of disc brake rotor.
- To fabricate an experimental set-up that gives temperature data during braking at different speeds.

- To determine temperature field in a disc brake during the braking phase of a vehicle.
- To study heat dissipation characteristics of different types of brake discs during their operation.
- To improve heat transfer rate of disc brake by changing vane geometry and material.
- To analyze the thermal distribution of vehicle disc brake.
- To validate the temperature results of simulation with experimental results.
- To find out ways that gives better thermal performance by changing the parameters of brake disc rotor.

III. MATERIAL AND METHODS

The rotating test rig shown in Fig. consists of an ac motor mounted on a base frame, which is connected to a shaft (axle) through belt-pulley mechanism. The shaft rotates on bearings, and is flanged at the opposite end to allow brake discs to be directly attached. A Hydraulic brake caliper applies a braking torque to the disc. While pressing the brake lever, high pressurized oil moves into the caliper from the master cylinder. This high pressurized fluid caused the piston inside the caliper to force the brake pad against the rotor. Speed of the motor can be controlled by the use of speed regulator. The temperature generated during the braking process is measured with the help of laser temperature gun.

Figure 1. Schematic diagram of Rotating test rig

3.1 Design of Motor

$I = K \times m \times r^2$

$= \frac{1}{2} \times m \times r^2$

$= \frac{1}{2} \times 7.36 \times (100.5 \times 10^{-3})^2$

$I =$ **0.03717 kg.m^2**

$\omega = 2\pi N/60$

$= (2\pi \times 1000)/60$

$\omega =$ **104 rad/sec**

$S = 2 \times \pi \times r \times n$

$30 = 2\pi \times 110.5 \times 10\text{-}3 \times n$

$n =$ **15.40 Turns**

$\omega_f^2 = \omega_o^2 + 2 \times \alpha \times \theta$

$0 = (104)^2 + 2 \times \alpha \times 2\,\pi \times 15.40$

$\alpha =$ **-53.79 rad/sec^2**

$\tau = I.\ \alpha$

$= 0.03717 \times 53.79$

$= 1.999$

$\tau =$ **2 N.m**

$P = \dfrac{2\pi \times N \times \tau}{60}$

P = 209.44 watt

P = **0.5 HP**

By considering factor of safety, motor is taken of **1HP**.

3.2 Design of Shaft

$$T = \frac{\pi}{16} \times \tau \times d^3$$

$2000 = \pi/16 \times 66.67 \times d3$

$d = 5.25$ mm

By considering factor of safety, **d = 20 mm**

3.3 Design of Bearing

Length of shaft, L = 700 mm

Diameter of shaft, d = 15 mm

Volume of shaft, $V = \pi \times r2 \times L$

$= \pi \times 7.52 \times 700$

$= 123700$ mm3

$V = 1.24\times10\text{-}4$ m3

Self-weight of shaft, $m = V \times \rho$

$= 1.24 \times 10\text{-}4 \times 7850$

$m = 0.9734$ kg

Weight of shaft, $W = m \times g$

$= 0.9734 \times 9.81$

$= 9.549$

$W = 10$ N

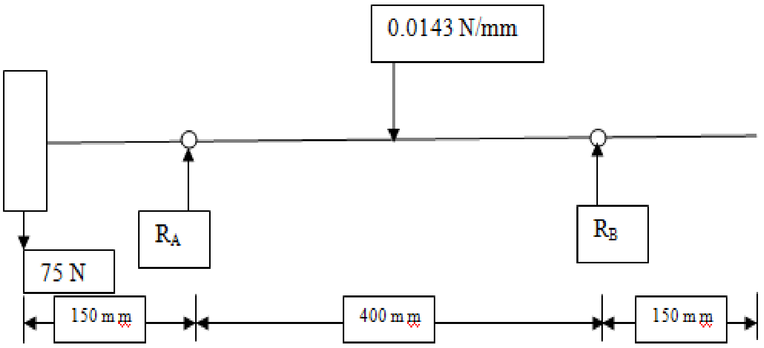

Figure 2. Free body diagram

$R_A + R_B = 85$ N

Moment @ A, $-75 \times 0.150 + 14.3 \times 0.2 \times - R_B \times 0.400 = 0$

$R_B = -21$ N, $R_A = 106$ N

$F_{R1} = R_A = 106$ N, $F_{R2} = R_B = -21$ N

$P_1 = F_{R1}$ & $P_2 = F_{R2}$

For safety purpose, life of bearing can be taken as 100 million revolutions.

$C_1 = P_1 \times (L_{10})^{1/3} \times$ Load factor

$= 106 \times (100)^{1/3} \times 2$

$C_1 = 969$ N

Therefore we can select bearing No. 6002 for shaft diameter of **20 mm.**

3.4 Numerical Simulation

In the present work, nine types of bi-directional configurations of the disc brake rotors i.e. which can be used for both sides (left and right) of a vehicle are considered for investigation. They are analyzed for the passage heat transfer coefficient and rate of heat transfer at different speeds using FLUENT, a CFD code. The conventional type of the ventilated disc brake configuration is the straight radial vane (SRV) rotor is shown in figure. Tapered radial vane (TRV) rotor and modified tapered radial vane (MTRV) rotor are shown in figure. Also, pillared rotors are shown in figure. Circular pillared (CP), and diamond pillared (DP) and modified diamond pillared (MDP) rotor configurations. Tapered vane (TP), curved vane (CV) and cross elliptical (CE) rotors are the proposed designs of this work and shown in figure. Show the overall dimensions of the rotor maintained for each configuration considered in the numerical study is of a typical passenger car.

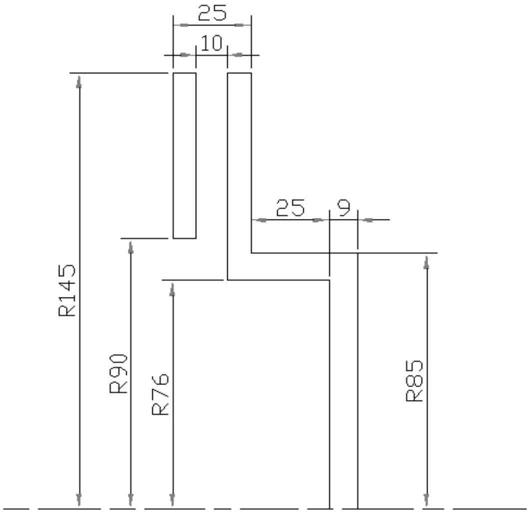

Figure 3. Geometry of the disc brake rotor [7]

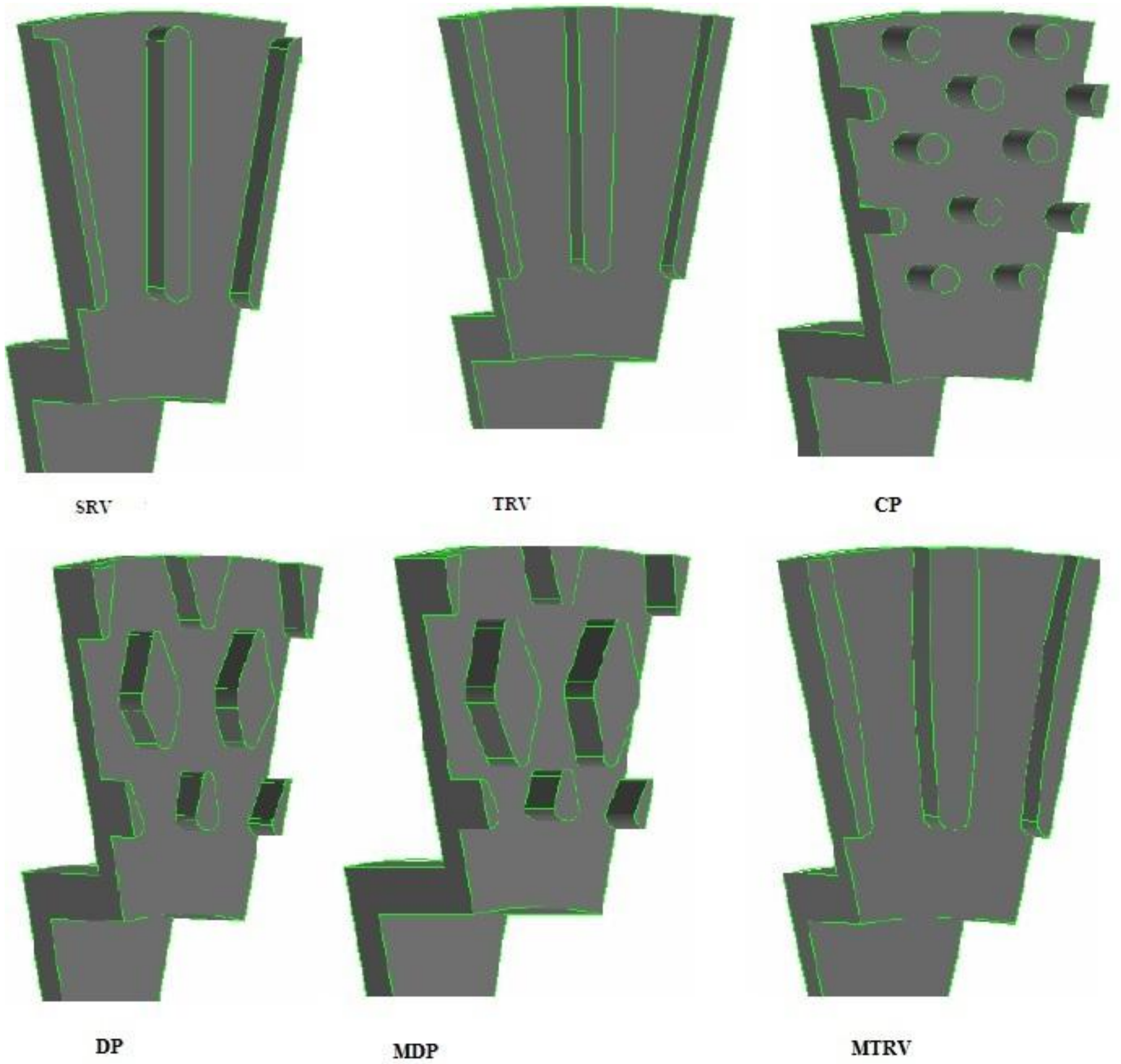

Figure 4. Existing design of rotor disk

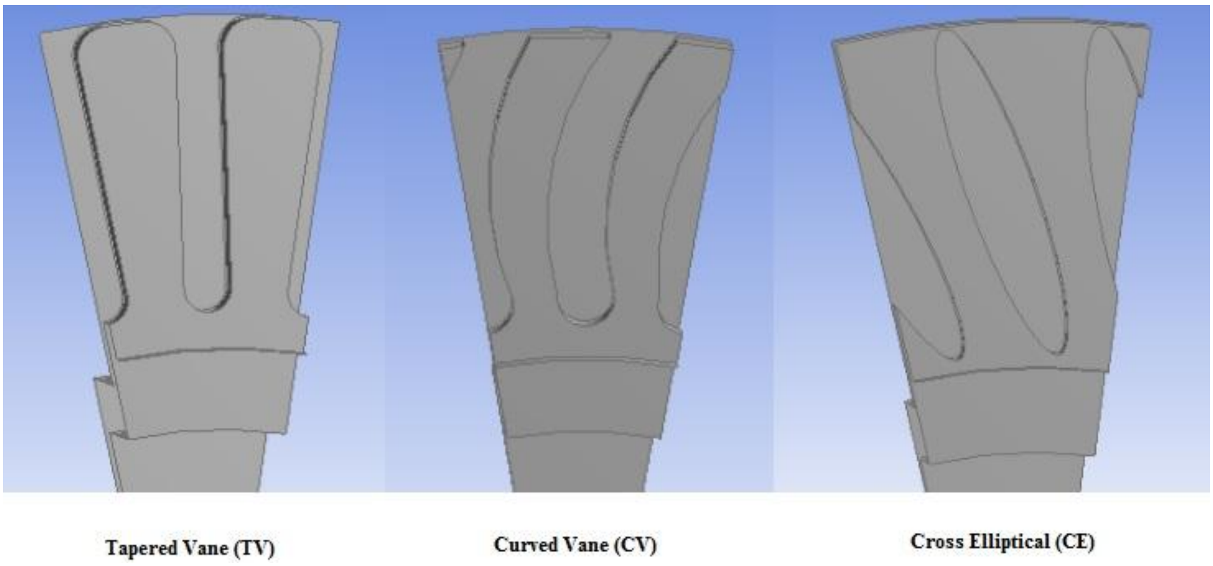

Figure 5. Proposed Designs of Rotor Disk

Rotor geometries are chosen in such a way that the ventilated passage at inlet and outlet are almost the same for all the configurations. For the present study, rotational speed of the rotor is varied from 800 to 1600 rpm, which is approximately 90 to 160 km/h for a typical four-wheeled vehicle.

The rotational periodic nature of the disc brake rotor has enabled the consideration of only a segment of it rather than complete rotor for the analysis. As each of the rotors investigated have 36 passages, a 20° segment of the rotor is modeled. Periodic boundaries are applied to either side of the segment to represent the entire rotor. The rotors are treated as spinning in an infinite environment by a rotating frame of reference and the application of an open boundary condition to the extent of the domain. For the open boundary condition, pressure of one atmosphere is specified at the periphery of the domain and the direction of the flow crossing the boundary is implicitly determined. The computational domain is a 20° segment of a cylinder with radius 290 mm and length 120 mm, which is large enough to eliminate any effect that the boundary may have on the flow through the rotor passages.

The flow is assumed to be steady and incompressible ideal gas. Ambient temperature and pressure are assumed as 300 K and 101325 Pa respectively. Rotor walls are assumed at constant temperature of 700 K for 800, 1000 and 1200 rpm and 900 K for 1400 and 1600 rpm with smooth surface. For the analysis, moving frame of reference is considered, and buoyancy and radiation effects are neglected.

While braking the kinetic energy of rotor will gets converted into the heat energy.

$v = \frac{\pi DN}{60}$

$v = \frac{\pi \times 0.23 \times 500}{60}$

$v =$ **6.02 m/s**

Kinetic energy of rotor:

$K.E = \frac{1}{2} \times m \times \frac{(v_2^2 - v_1^2)}{t_2 - t_1}$

$= \frac{1}{2} \times 7.36 \times \frac{6.02^2 - 0}{5 - 0}$

K.E. = **26.5 watt**

Heat energy:

$Q = h \times A \times (T_s - T_a)$

Rotor area:

$A = \frac{\pi}{4} (D^2 - d^2)$

$A = \frac{\pi}{4} (0.23 - 0.15)$

$A = 0.0232 \text{ m}^2$

Here,

$T_s = 42°$

$T_a = 25°$

$26.5 = h \times 0.023 \times 17$

h = 67.78 w/m²k

IV. RESULT AND DISCUSION

Simulation results for the rotor are shown in below figures. The figure shows the graphical representation of Passage heat transfer coefficient Vs rotor rpm. From the graph it can be seen that the passage heat transfer coefficient increases as the rotor RPM increases for all the design patterns but it has higher value for the new proposed designs.

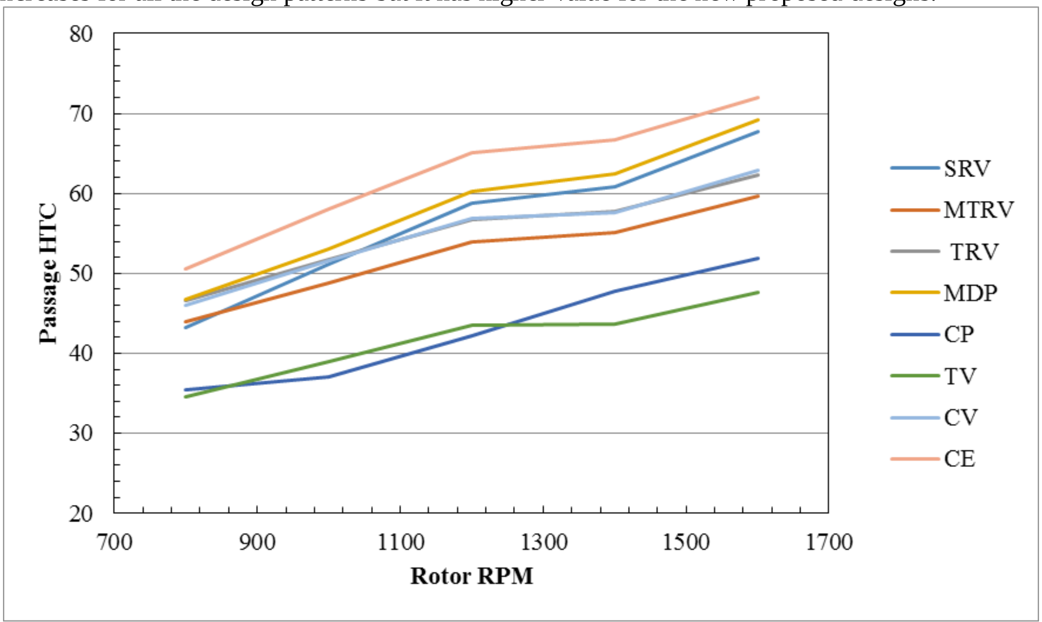

Figure 6. Passage HTC v/s Rotor RPM

Below figure shows the graphical representation of Overall heat transfer coefficient Vs rotor rpm. It can be clearly seen that for the designs curved vane and cross elliptical, the overall heat transfer coefficient increases more rapidly than the other rotor designs. The newly proposed design curved vanes shows similar result like other five rotor designs.

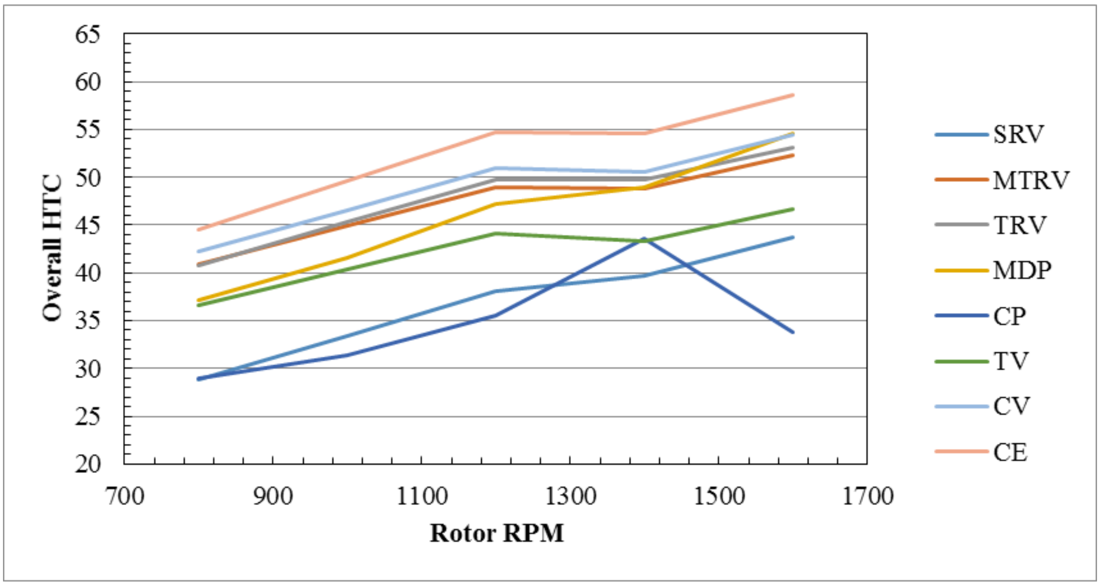

Figure 7. Overall HTC v/s Rotor RPM

Variation of heat dissipation with different rotor speed is highlighted in following figure. It can be easily viewed from the graph that the new proposed design i.e. curved vane and cross elliptical are having better heat dissipation characteristics then the other rotors.

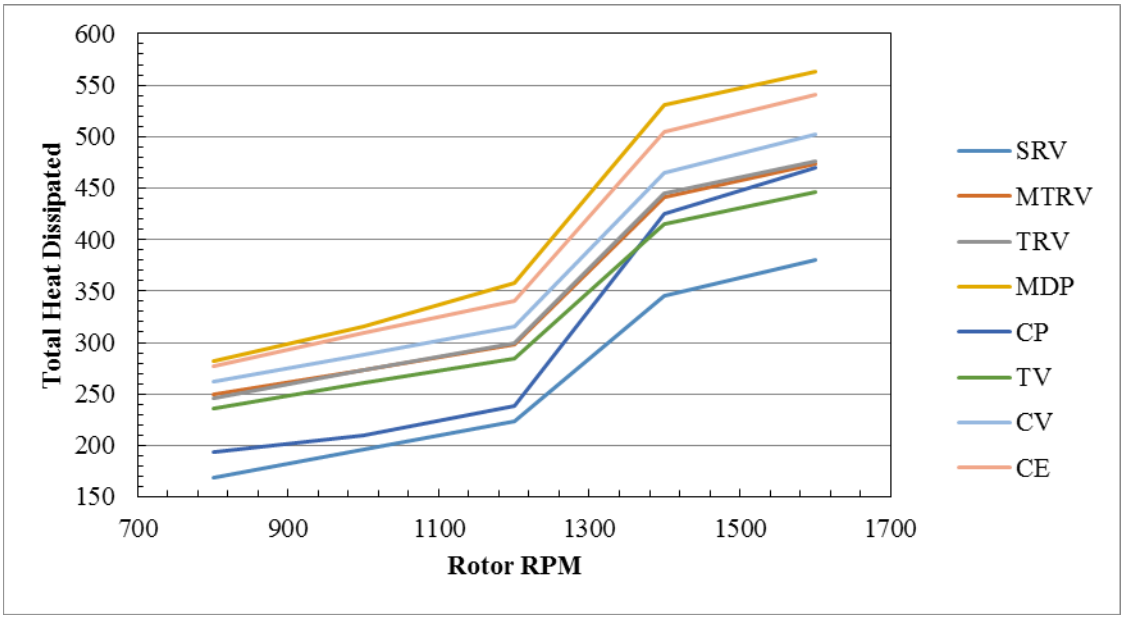

Figure 8. Total Heat Dissipated v/s Rotor RPM

During braking the heat is generated on the rotor which is measured with the help of temperature detector. The following results were taken on the test rig shown in figure 1. Digital tachometer was used to measure the rpm of rotor. Measured temperatures are shown in below table.

Table 1 Experimental data

Sr. No	Speed(RPM)	Temperature
1	500	40
2	600	41.5
3	700	42.9
4	800	43.6

V. CONCLUSION

Disc brake design plays as an important role in heat transfer as other variable like plate & vane thickness, fin material and flow pattern. In the present work, the thermal characteristics of different disc brake rotors are studied extensively by the successful application of both experimental and numerical techniques.

Conclusion from the present study can be stated as follows.

- Study of literature review gives immense ideas about the different rotor geometries.
- Contact time between air flow and vanes (time between air inlet and outlet flow through vanes) is also important factor in heat transfer from Disc rotor.
- With the test rig been fabricated temperature of different rotors can easily be detected.
- Newly proposed design cross elliptical has the highest overall heat transfer coefficient as well as passage heat transfer coefficient characteristics.
- Another newly proposed design curved vane has higher overall heat transfer coefficient compare to five designs of paper.
- There is a scope of improvement in heat transfer in ventilated disc brake if vane is angled and of alternate length other than straight radial vane.

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