NUMERICAL INVESTIGATION OF FLUID FLOW AND HEAT TRANSFER CHARACTERISTICS ON THE AERODYNAMICS OF VENTILATED DISC BRAKE ROTOR USING CFD

by

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Ventilated brake discs are used in high speed vehicles. The brake disc is an important component in the braking system which is expected to withstand and dissipate the heat generated during the braking event. In the present work, an attempt is made to study the effect of vane-shape on the flow-field and heat transfer characteristics for different configurations of vanes and at different speeds numerically. Three types of rotor configurations circular pillared, modified taper radial and diamond pillar vanes were considered for the numerical analysis. A rotor segment of 20° was considered for the numerical analysis due to its rotational symmetry. The pre processing is carried out with the help of ICEM-computational fluid dynamics and analysis is carried out using ANSYS CFX 12.1. The three dimensional flow through the brake rotor vanes has been simulated by solving the appropriate governing equations viz. conservation of mass, momentum and energy using the commercial computational fluid dynamic stool, ANSYS CFX 12. The predicted results have been validated with the results available in the literature. Circular pillar rotor vanes are found to have more uniform pressure and velocity distribution which results in more uniform temperature drop around the vanes. The effect of number and diameter of vanes in the circular pillared rotor is studied and the geometry is optimized for better mass flow and heat dissipation characteristics.

Key words: *computational fluid dynamics, disc brake, rotational symmetry, turbulence, vane shape*

Introduction

Braking system is one of the important safety components of a vehicle. Braking system is mainly used to decelerate vehicles from an initial speed to a given speed. Friction based braking systems are the common device to convert kinetic energy into thermal energy through friction between the brake pads and the rotor faces. The modernization of multi-lane facilities paves the way for high-speed driving of vehicles. The passenger and racing cars require high speed braking system which could not be met with drum braking systems. Excessive thermal loading can result in surface cracking, judder, and high wear of the rubbing surfaces. High temperatures can also lead to overheating of brake fluid, seals and other components. The stopping capability of brake increases by the rate at which heat is dissipated due to forced convection and the thermal capacity of the system.

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Limpert [1] investigated the temperature distribution of solid rotors using Duhamel's theorem. Newcomb and Spurr [2] revealed that the thermal properties vary linearly with temperature. Chi [3] has performed the thermal characteristics of brake discs using analytical approaches on the assumption of laminar flow and simplified geometry. The air flow through the passage of vane rotors is a complex turbulent flow. Thus, how to select better geometrical design variables and improve thermal performance of automotive brake rotors is a task that the vehicle designers are often confronted. This is because it is found that the thermal failure is not only due to high temperatures, but also due to high thermal stresses developed because of non-uniform cooling of rotor vanes. Thus it is difficult to determine the effects of geometries of rotors on thermal performance of disc brakes.

David *et al*. [4] have carried out the analysis for both flow surrounding the brake disc and inside the radial rotor passages using a two-component particle image velocimetry (PIV) system. Most of the previous studies have measured the air flow velocity profile at the exit of the passage of disc brake rotor using pressure probes. In some cases, they have used hot wire anemometry (HWA) to examine the unsteady flow-field at rotor exit [5, 6]. Prediction of average heat transfer coefficient was in the past largely performed by using semi-empirical correlations [7] whereas recent attempts to predict it by computational fluid dynamics (CFD) simulations can be found in the literature [8]. Anders and Bergstrom [9] has performed numerical simulation and compared the results with experimental data. Manohar Reddy [10] has validated the flow field velocities of a simple radial vane with the experimental data measured using PIV and modified the vane shape for maximum heat transfer. Structural analysis of ventilated rotor brake discs using CAE, finite element analysis (FEA) was reported by Amol and Ravi [11]. In their study, they reported the procedure for prediction of thermo-mechanical performance like temperature distribution and heat transfer coefficient estimation. Other studies have shown that inlet and cross-drilled holes can have a beneficial effect on the rate of cooling [12].

In the present work, three types of disc brake rotors, circular pillar (CP) modified taper radial vane (MTRV) and diamond pillar (DP) are analyzed for the effect of vane-shape rotor on the flow-field and heat transfer characteristics for different configurations of vanes using commercial CFD tool. The predicted results are validated with the numerical results of circular pillared vane performed by Manohar Reddy [10].Pillared vanes posses the advantage of curved vane rotors and bi-directionality which are mainly developed for high performance applications. It is taken care that the inlet and the exit area of ventilated rotor discs remains almost the same for all the configurations considered. The rotational speed of the rotor is varied from 800 rpm to 1400 rpm which corresponds to approximately 90 to 120 km/h. Two rotor temperatures 700 K and 900 K are considered for analysis. This temperature was calculated based on the absorption of kinetic energy by rotor immediately after applying the brakes [13].

The flow field characteristics like pressure and velocity components are plotted at mid-plane from inlet to exit of the ventilated rotor brake discs. The average mass flow rate and heat dissipation characteristics are also obtained in the present study. The numerical analysis is performed to select better geometric variables viz. the number of vanes and the diameter of vanes in circular pillared configuration for better mass flow and heat dissipation characteristics.

Computational modeling and simulation

Figure 1 shows the overall dimensions of the disc brake rotor which was used for CFD analysis for all the configurations of ventilated rotor brake discs. The dimensions of circular pillar vanes are similar to the model in literature [10] and are shown in fig. 2. The 20 sector model is considered for numerical analysis for all configurations due to the rotational symmetry. The
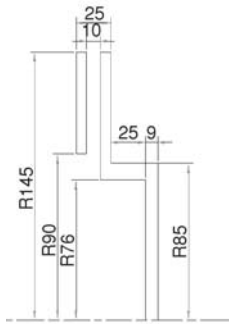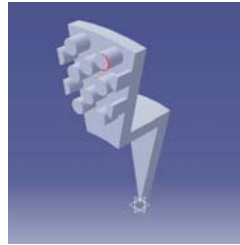
Figure 1. Disc brake rotor

Figure 2. 20 sector of circular pillar vane created in CATIA

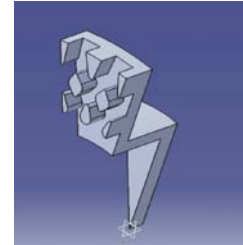**Figure 3. 20 sector of modified taper radial vane created in CATIA**

Figure 4. 20 sector of diamond pillar vane created in CATIA

20° sector of the modified taper radial vane and diamond pillar vane created in CATIA are shown in fig. 3 and fig. 4, respectively.

Grid generation

The 3-D model of the circular pillar vane is imported in parasolid format in ICEM-CFD for mesh generation. In order to capture both the thermal and velocity boundary layers the entire model is discretized using hexahedral mesh elements which are accurate and involve less computation effort for the solver. The generation of hexahedral elements is quite difficult due to the complexities of rotor vane shapes. Fine control on the hexahedral mesh near the wall surface allows us to capture the boundary layer gradient accurately. The entire geometry is divided into three fluid domains FLUID_STATOR, FLUID_ROTOR_OUTER, and FLUID_ROTOR_INNER. The discretised model is checked to have a minimum angle of 27° with minimum determinant of 0.65. The meshes are checked for free of errors like volume and surface orientation and minimum required quality in order to achieve better convergence and are exported to ANSYS CFX pre-processor. The fluid mesh around circular pillar, modified tapered radial vane and diamond pillar vanes are shown in figs. 5, 6, and 7, respectively. The study was initially carried out with 1, 67,000 nodes and 4, 95,000 nodes. It is found that the variation in the numerical results between 1, 67,000 nodes and 3, 84,000 nodes is quite significant whereas the variation is negligible between 3, 84,000 and 4, 95,000 nodes. Thus the numerical results are independent of grid after 3, 84,000 nodes for circular pillared vanes. Similar grid independence studies are carried out for all the configurations considered for the analysis.

Governing equations

The 3-D flow through rotor vanes was simulated by solving the appropriate governing equations viz. conservation of mass, momentum, and energy using ANSYS CFX 12.1 code. Turbulence is taken care by Shear Stress Transport (SST) *k-ω* model of
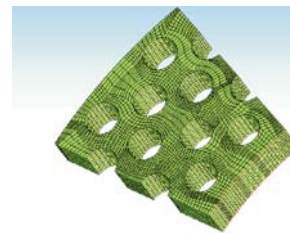
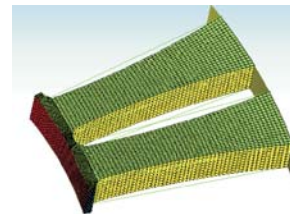

Figure 5. Hexahedral fluid mesh of circular pillar vane

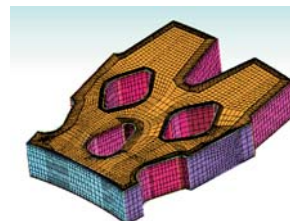**Figure 6. Hexahedral fluid mesh of modified taper radial vane**
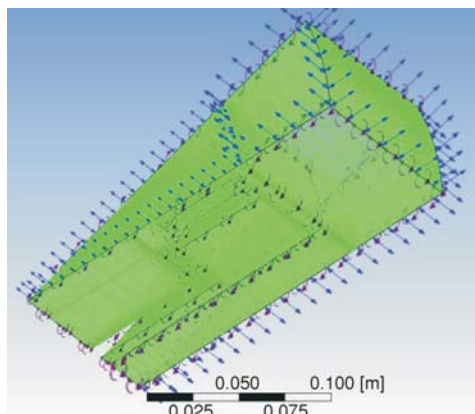

Figure 7. Hexahedral fluid mesh of diamond pillar vane

Figure 8. Fluid domain of CP vanes created in ANSYS CFX 12.1 pre-processor

closure which has a blending function that supports Standard k-ω near the wall and Standard k-ω elsewhere.

Boundary conditions

The fluid region of the ventilated brake disc is divided into three regions namely FLUID_STATOR, FLUID_ ROTOR_OUTER, and FLUID_STATOR. The fluid domain of CP ventilated rotor disc with boundaries specified in ANSYS CFX 12.1 pre-processor are shown in fig. 8. The connectivity between these domains is established by creating necessary fluid interfaces in the solver. The advantage of creating these domains separately helps in grid refinement in the FLUID_ROTOR_INNER compared to FLUID_ROTOR_OUTER and to have better control on the number of mesh nodes generated. The flow physics through the ventilated brake disc is quite complex as it contains both rotating and stationary domains. These domains are properly interfaced by FROZEN_ROTOR interface available in the ANSYS CFX solver, for which hexahedral mesh is relatively better compared to tetrahedral elements.

The solid domain of rotor vanes is not generated as the rotors are assumed to have isothermal temperatures in this study. 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, large enough to avoid the effect of boundary layer. 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. The stator domain was considered three times the rotor diameter. The flow is assumed to be steady and incompressible ideal gas. Ambient temperature and pressure are assumed as 298 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 rpm with smooth surface. For the analysis, moving frame of reference is considered, and buoyancy and radiation effects are neglected.

Validation

The predicted mass flow rate results of circular pillar vanes are compared with the results of Manohar Reddy *et al*. [10]. The mass flow rate predicted for various speeds from 800 rpm to 1400 rpm in steps of 200 rpm are given in tab. 1. The experimentally validated numerical data obtained from literature of Manohar Reddy [10] are tabulated in tab. 1 along with numerical readings predicted with ANSYS CFX 12.1 by this study. A very good agreement between the numerical and ex-

Table 1. Mass flow rate of circular pillared vanes

Speed [rpm]	Numerical mass flow rate [gs^{-1}]	Literature mass flow rate [5], [gs^{-1}]	Error [%]
800	0.7	0.72	–2.8
1000	0.98	0.96	2.1
1200	1.18	1.23	–4.1
1400	1.37	1.32	3.8

perimentally validated numerical data is obtained. The maximum deviation in mass flow rate is less than 5% and the same is plotted as shown in fig. 9.

Results and discussion

Mass flow rate and heat dissipation of isothermal rotors

Mass flow rate and heat dissipation are often the common factors while selecting the vane configuration. Table 2 shows the predicted mass flow rate and heat dissipation of CP, MTRV and DP at 1000 rpm.

The mass flow rate in MTRV is around 200% more than CP. The mass flow rate in DP is around 60% more than CP. The improved mass flow rate in MTRV is attributed to less vane hindrances for the fluid flow compared to CP and DP. Though CP has lesser mass flow rate compared to MTRV, the heat dissipation in CP and MTRV are almost the same. The above is because that CP has larger surface area compared to MTRV. Heat dissipation in DP is around 25% more than CP and MTRV but the mass flow rate lies in between CP and MTRV.
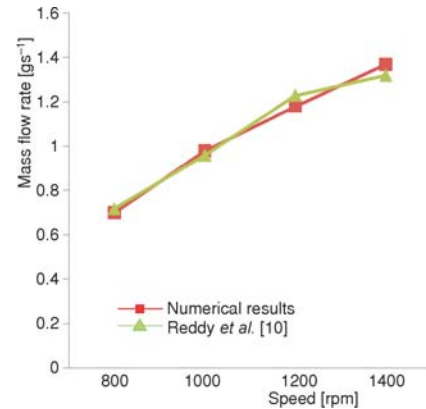

Figure 9. Mass flow rate of centre pillared vanes

Table 2. Mass flow rate of vane configurations at 1000 rpm

Vane configuration	Mass flow rate at [gs^{-1}]	Heat dissipation [W]
CP	0.98	238.5
MTRV	3.19	245.0
DP	1.59	301.7

Velocity comparison of isothermal rotors

The uniform distribution of air mass flow around the rotor vanes is essential for uniform temperature drop around the vanes. The velocity and pressure contours are plotted in the mid-plane which comprises of rotating and stationary fluid domains. Figure 10 shows the velocity distribution on mid plane for CP, MTRV, and DP at a speed of 1000 rpm. In MTRV, the average velocity in the mid plane is higher but the distribution is non uniform.

It was found that the velocity magnitude increases from zero near the hub to the maximum near the tip of the vanes for all the three types of ventilated brake discs.
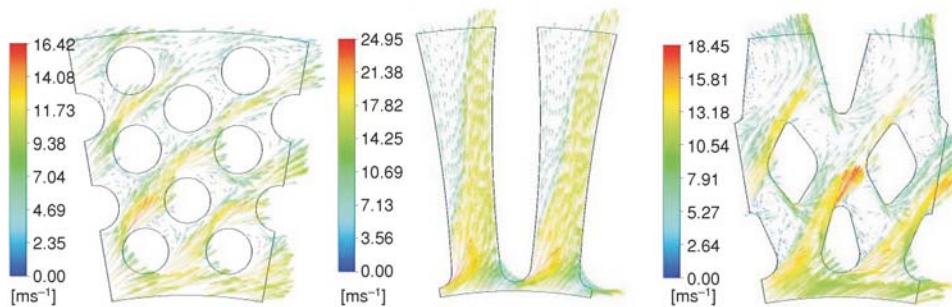

Figure 10. Velocity distribution on mid plane for CP, MTRV, and DP at 1000 rpm
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Rajagopal, T. K., *et al.*: Numerical Investigation of Fluid Flow and Heat Transfer ...

672   THERMAL SCIENCE: Year 2014, Vol. 18, No. 2, pp. 667-675

In MTRV, the velocity at the right side of the vane is relatively less compared to the velocity at the right hand side. This will lead to thermal failure not because of high temperature but due to high temperature gradients exists between the sides of the rotor vanes. The velocity distribution in CP is more uniform around the vanes. The velocity distribution in DP is uniform than MTRV. The velocity distribution of CP is more uniform than DP and MTRV. Even though the average velocity is higher in MTRV and DP, CP is more preferred since the velocity distribution is uniform. This will result in even cooling of vanes. As stated earlier in this work, it is well established that not only the high temperature that causes the failure but also the temperature gradient which would result in failure of the rotor vanes.

Pressure comparison of isothermal rotors

Figure 11 shows the pressure distribution on mid plane for CP, MTRV, and DP at speeds of 1000 rpm, respectively. The average pressure drop in CP is less compared to DP and MTRV. The pressure variation is almost uniform around the vanes for CP. The pressure variation is non uniform around MTRV and DP rotor vanes. This uniform pressure distribution of CP helps in uniform thermal cooling of rotor vanes and more uniform distribution of mass flow around the vanes of CP.
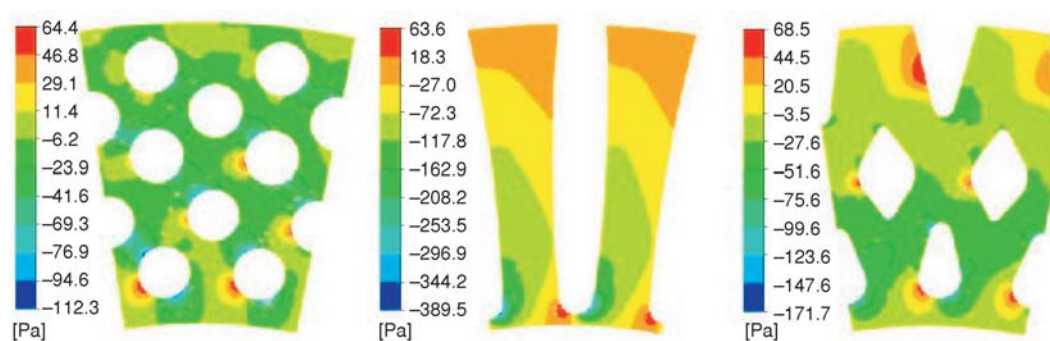

Figure 11. Pressure distribution on mid plane for CP, MTRV, and DP at 1000 rpm
(for color image see journal web site)

Effect of number of pillars

The uniform velocity and pressure distribution around the vanes makes CP the preferred configuration disc brake compared to MTRV and DP. Since the air mass flow rate is less around the pillars compared to MTRV and DP an extensive numerical study is carried out to enhance the mass flow rate by analyzing the effect of number of pillars on fluid flow characteristics.

The basic circular pillared brake disc had a diameter of 8 mm and the number of pillars is 12. Two case studies were carried out by varying the diameter of circular pillar vanes from 8 mm to 6 mm and 5 mm, respectively. The number of pillars was increased to 17 and 22 pillars for therby keeping the overall rotor mass constant. The analysis were carried out for different speeds of 800 rpm, 1200 rpm, and 1400 rpm for all the configurations as mentioned below in tab. 3. The mass flow rate and heat dissipation for different pillar diameters at various speeds are tabulated in tab. 3. It is to be noted that the isothermal surface temperature of rotor vanes were specified as 700 K for 800 rpm and 1200 rpm where as the surface temperature of rotor vanes are specified as 900 K for 1400 rpm.

Table 3. Mass flow rate and heat dissipation comparison for different pillar diameters

Pillar diameter [mm]/ number of pillars	Speed [rpm]	Mass flow rate [gs^{-1}]	Heat dissipation [W]
8/12	800	0.72	203.7
	1200	1.23	277.0
	1400	1.32	469.0
6/17	800	1.01	210.6
	1200	1.53	286.5
	1400	1.78	484.2
5/22	800	0.97	224.4
	1200	1.46	288.2
	1400	1.71	488.6

The mass flow rate increases with increase in rotational speed of disc brake for all the configurations. It could be observed that the mass flow rate for 6 mm diameter pillar is higher than that of 8 mm and 5 mm diameter pillar configurations. The heat dissipation increases with decrease in pillar diameter. This is because the number of pillars increases as the diameter decreases and hence more surface area to volume exists for better heat dissipation.   The heat dissipation for 5 mm diameter pillar is better than 8 mm diameter pillar configuration. The heat dissipation for 5 mm diameter pillar is better than 6 mm diameter pillar at lower speeds but tends to become negligible at higher speeds.

Figure 12 shows the velocity on the mid plane at 1200 rpm. The velocity magnitude at the mid-plane has increased with decrease in pillar diameter. The velocity magnitude of 6 mm diameter pillar is higher than that of 5 mm diameter pillar as shown in fig. 12. Moreover the velocity distribution of 6 mm diameter pillar is more uniform compared to 5 mm diameter pillar and this accounts for more mass flow rate of the former compared to the later. Similar velocity contours were obtained for other speeds of 800 rpm and 1400 rpm also. Figure 13 shows the pressure distribution on the mid plane at 1200 rpm.   The pressure distribution of 6 mm diameter pillar is more uniform that 5 mm diameter pillar as shown in fig. 13 for 1200 rpm speed of circular pillared vanes. It is found that the pressure drop between the rotor vanes is decreased   as the number of circular pillar increases from 8 to 22. This is because of the less hindrances offered to
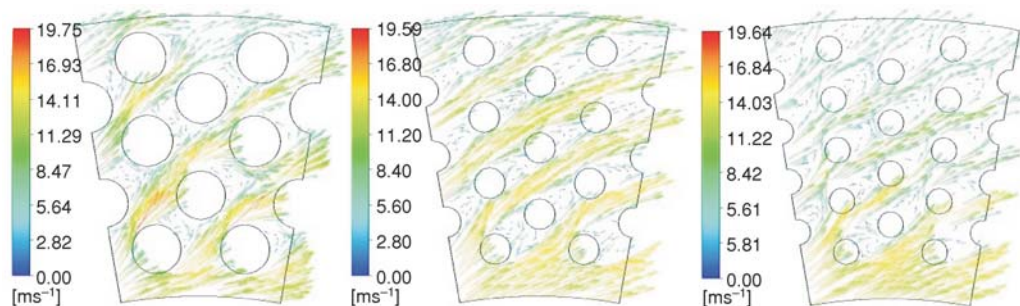

Figure 12. Velocity distribution on mid plane for pillar 8, 6, and 5 mm at 1200 rpm
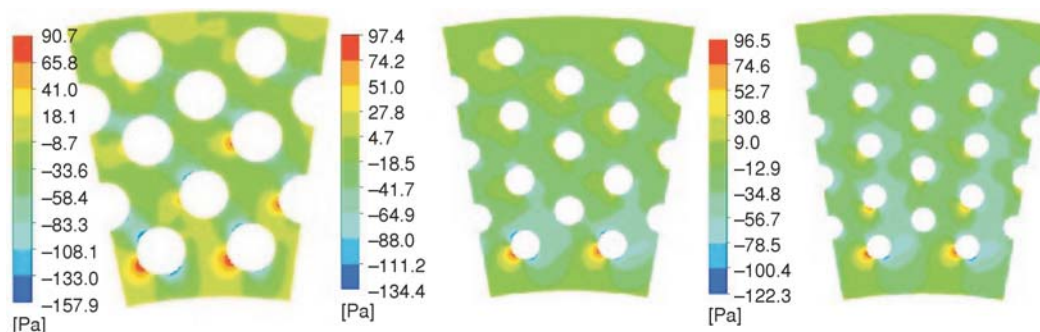(for color image see journal web site)

Figure 13. Pressure distribution on mid plane for pillar 8, 6, and 5 mm at 1200 rpm
(for color image see journal web site)

the flow due to decrease in the diameter of the circular pillared vanes. Hence from the study, it could be concluded that the 6 mm diameter circular pillared vanes offers better heat dissipation characteristics with more uniform pressure and velocity distribution.

Conclusions

In this present work, heat transfer characteristics of circular pillar, modified taper radial vane and diamond pillar vanes are analyzed. The following conclusions are drawn.
- Mass flow rate is considerably higher in modified taper radial vane compared to circular pillar rotor vanes.
- Heat dissipation in diamond pillared vane is around 25% higher as that of circular pillar and modified taper radial vanes.
- Circular pillar rotor vanes have more uniform pressure and velocity distribution which results in more uniform temperature drop around the vanes. This will ensure uniform and even cooling of rotor vanes. It would avoid the thermal failure of rotor vanes due to large thermal gradients between the rotor vanes as expected for MTRV and DP vanes. Due to these advantages, circular pillared vanes are preferred than modified taper radial vanes and diamond pillar vanes.
- The diameter and number of pillars in circular pillar vanes are modified for better mass flow rate and heat dissipation characteristics. It is found that the velocity and pressure distribution is uniform even when the diameter is reduced to 6 mm from 8 mm and this trend becomes less uniform when it is decreased further to 5 mm diameter.
- The 6 mm diameter with 17 circular pillared vanes has around 20% better mass flow rate and around 4% better heat dissipation characteristics compared to 8 mm diameter and 12 circular vane rotor brake discs.

Limitations

The results presented in this paper are on the assumptions of uniform temperature on the rotor surface. A varying heat flux model if incorporated for the rotor surface would closely approximate to actual conditions. It is currently under study.

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THERMAL SCIENCE: Year 2014, Vol. 18, No. 2, pp. 667-675

675

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