

EXPERIMENTAL ANALYSIS OF BRAKE DISC COOLING CAPACITY

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Abstract: The car brake system performance has to be reliable at wide range of operating conditions. The friction based braking system reliability strongly depends on cooling capability of disc. According to this, different testing is needed to assure the performance of discs during their design and prototyping phase. The contribution presents the study of cooling capacity of different vented brake disc geometries.

Brake disc temperatures during thermal capacity test can exceed 600 C. The temperatures are highly dependent on disc cooling capability. Vented disc have higher cooling capability. There is an open issue which factor has the most significant impact. Is it air flow, air flow distribution, turbulence or ...? Brake discs with the same external geometry but different internal geometry were tested on brake dynamometer and subsequently air flow parameters were analysed in cold condition to attempt to explain different maximum temperatures.

Keywords: brake disc, cooling, flow measurements

1. Introduction

Braking performance is crucial for the safety of the vehicle and its occupants. Braking performance can be influenced by wear, maintenance and braking history. Most vehicles use cast iron discs or drums. Brake drums are used in smaller cars and mostly for rear brakes since the majority of brake force is distributed on front axle (60-70%). The most important task of the brake system is to convert kinetic energy into heat. Braking performance is brake disc temperature dependent. Disc temperature rises with every braking and the braking performance decreases with the temperature. Brake disc is mostly cooled by air, although some heat is transmitted via radiation too. The cooling efficiency was studied by prof. Limpert (Limpert, 1975) and vented disc were proposed and a whole new area of study was opened dedicated to the question which remains actual: Which internal geometry of the disc is the best? Is a straight vanes design, curved vanes design or fins design? Beside the braking efficiency and temperature there are still other requirements to be fulfilled. Minimizing the disc mass, moment of inertia and increasing of resonant frequencies are also important but the production costs have very high priority when design is considered.

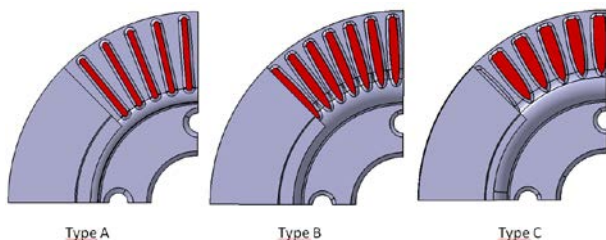


Fig. 1 Different shapes of disc vanes.

2. Problem

Two prototypes with aerofoil shaped vanes (B and C) were produced to find possible replacement for existing disc design with straight vanes (A). The discs were used on a B class car and are presented in Figure 1.

The results on dynamometer test were below expectations and the temperature after thermal capacity test (SAE J2522) was higher than in the case of the original design in case C and the same in case B. A more detailed analysis was performed and some of the results will be presented and discussed.

External disc geometry was the same for all three cases. The distance between the friction plates was 7 mm in type A, 8 mm in type B and 6 mm in type C. The number of vanes was 41 in case of type B and C and 53 in case of type A. The inner diameter of the friction surface was 145 mm for discs A and B and 154 mm for disc C. Vane length in case of type C disc was 4-5 mm shorter.

3. Air flow measurements

The first idea that comes to mind is to measure the air flow through the disc. The disc was placed on a rotating device driven by frequency controlled electro-motor. Most rotating device parts were the same as in appropriate car, although some modifications to the axle were made and the brake pads and the jaws were removed. It is quite complicated to measure the air flow at the disc exit. Air exit velocity in case of radial fan with straight vanes can be approximated by circumferential velocity of the vane tip which was approx. 10 m/s. Low velocities are difficult to measure with traditional methods like Pitot tube due to very low pressure differences. Additionally, exit velocity profile is not constant due to finite number of vanes and their thickness. Most of the studies were performed at 15.7 Hz rotational speed (corresponding to 100 km/h vehicle speed). When multiplying rotational frequency with the number of vanes frequency above 600 Hz can be expected. This frequency makes Pitot based measurements practically impossible. There are some reports of HWA measurements but our first idea was to simplify the mass flow measurements and the disc orientation was changed (friction surface faced outwards instead inwards). This enabled free access to the intake and enabled measurements of the intake velocity profile. Since the disc had no cross drilled holes the conservation of mass was assumed. Anemometer Testo 435-4 was placed on a traverse system and velocity profile in horizontal and vertical plane measured. Program controlling anemometer and traverse system was written in LabVIEW. The measurement results for disc type A are presented in Figure 2.

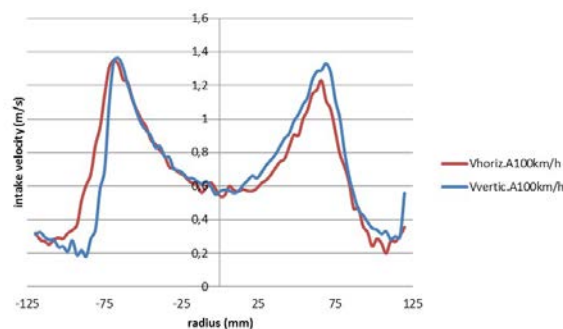


Fig. 2 Intake velocity profile type A disc at 100 km/h.

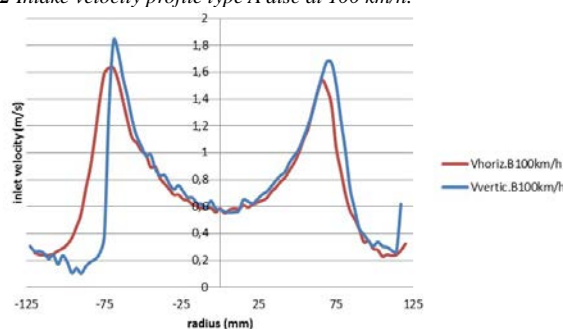


Fig. 3 : Intake velocity profile type B disc at 100 km/h.

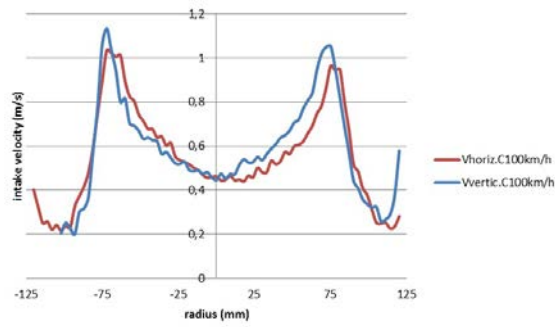


Fig. 4 Intake velocity profile type C disc at 100 km/h.

The procedure was repeated for several rotational speeds and the results for original disc (type A) are presented in Figure 5.

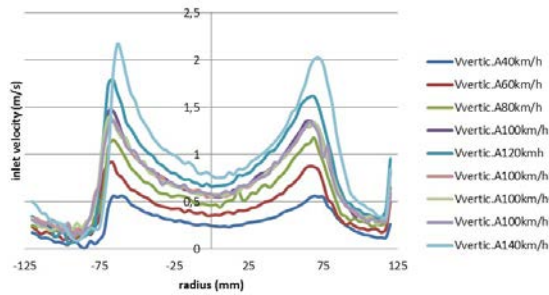


Fig. 5 Comparison of velocity profiles at different rotational speeds.

Velocity profile was used to calculate volumetric flow. Results for volumetric flow are presented and compared in Figure 6.

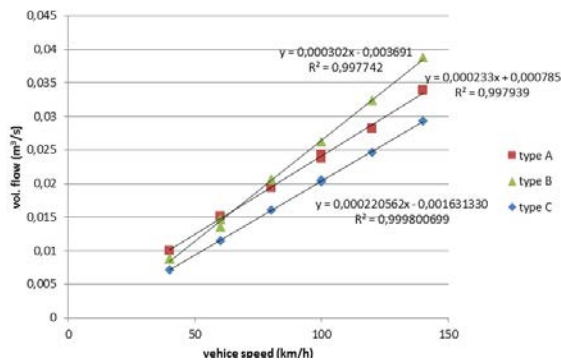


Fig. 6 Comparison of volumetric flow at different vehicle speeds.

Results presented in Figure 6 can be used to explain poor performance of type C disc since this type has the lowest volumetric flow and consequently lower heat dissipation. During testing according to SAE J2522 higher velocities prevail so Figure 6 cannot give a clue why there was practically identical maximum temperature in cases A and B.

Velocity distribution

Brake discs were tested again. This time they were oriented the same as in a car. Exit velocity was measured by Particle Image Velocimetry – PIV. Due to the time and other limitations only approx. 25 degrees of the disc were analyzed as shown in Figure 6. Velocity field derived from the PIV images (Figure 7) is presented in Cartesian coordinate as shown in Figure 8.

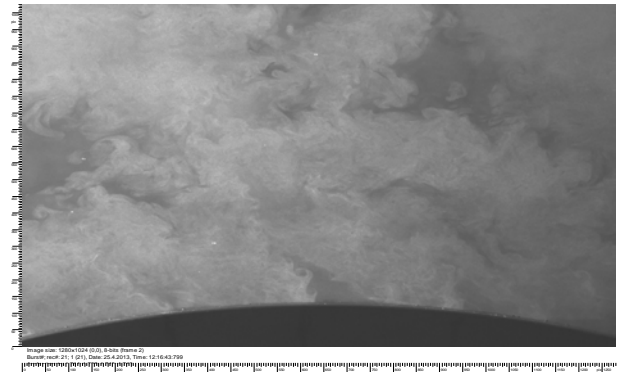


Fig. 7 PIV image of the disc-flow visualization.

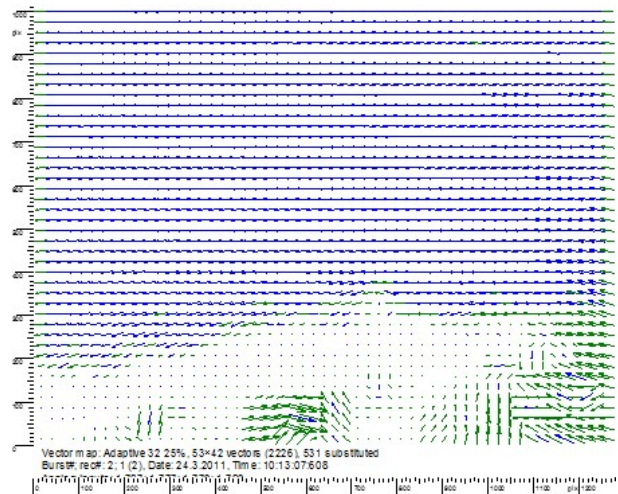


Fig. 8 PIV velocity field-cartesian coordinate system.

At least 240 images were averaged and radial velocity component at desired radius extracted. The procedure was repeated in several layers beginning at outward friction surface and ending at inward friction surface. The layers between friction plates were placed in 1mm interval and two layers were located in the middle of the friction plate. Radial velocity profiles are presented in Figure 9.

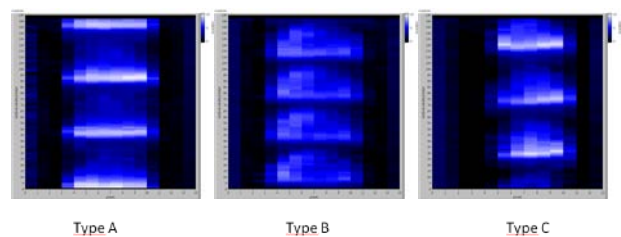


Fig. 9 Radial velocity profile.

By comparing results presented in Figure 9 it possible to observe different radial velocity distribution. The distribution in type A disc is along the vane, while in case of type B and type C it is concentrated in the middle of the channel. This fact can offer possible explanation for lower cooling capacity of type B and C discs. It is desirable to have cooling flow near hot surfaces. The hottest surfaces are friction planes and vanes.

4. Validation and comparison

Radial velocity field was used to calculate volumetric flow. It was assumed that there is no significant difference between the vanes and that 3 or 4 vanes covered by PIV analysis are representative. All internal surfaces are un-machined and sand-cast.

The results for volumetric flow at 100 km/h are presented in Table 1.

Table 1: Comparison of the results.

	vol.flow A (m ³ /s)	vol.flow B (m ³ /s)	vol.flow C (m ³ /s)
PIV	0.025268	0.026803	0.019168
anemometer	0.024063	0.026267	0.020346

As presented in Table 2 quite good agreement between the measurement methods was achieved for internal air flow which contributes the major part in convective cooling of the brake disc.

5. Influence of external disc diameter

We were fortunate to have at our disposal an enlarged version of the original disc A. The number and shape of the vanes is the same. The main difference is increase of external diameter of the disc by 22 mm. The comparison of mass flow is presented in Figure 10.

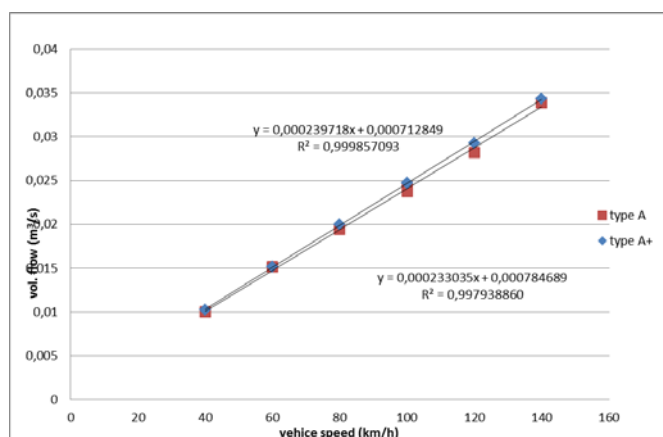


Fig. 10 Comparison of volumetric flow type A and A+

As presented in Figure 10 there is a less than 3% increase in air flow although the diameter increase is more than 8%. The vane length was increased by approx. 11 mm. The distance between the friction plates was 7 mm (the same as in case A) although casting tolerances must be considered. Simple expansion of the disc does not increase its cooling capacity much.

6. Influence of the internal vane diameter

A very similar disc (according to A+) was studied too. This disc had the same external diameter but only 40 vanes and the internal diameter of the friction plate 154 mm (like disc C). The air flow measurements are presented in Figure 11.

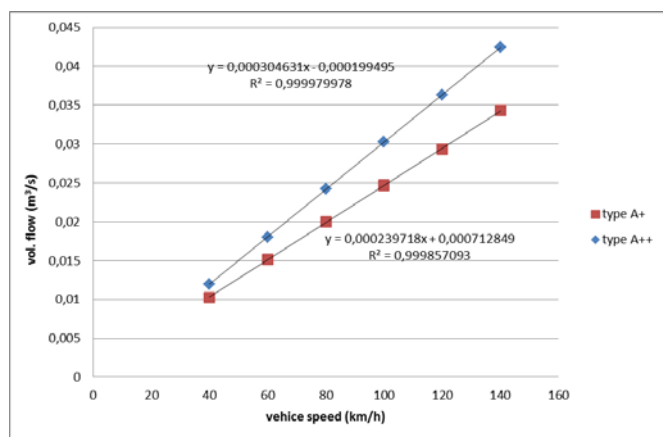


Fig. 11 Comparison of volumetric flow type A+ and A++

As presented in Figure 11 internal vane diameter has significant influence on air flow. The difference between the numbers of vanes

(41 and 40) was neglected. Vane length was reduced by approx. 5 mm (according to A+). External vane diameter was the same. The distance between the friction surfaces was again 7 mm.

7. Discussion

Air volumetric flow and radial exit velocity distribution were acquired during an attempt to explain unexpected results during thermal capacity test. Volumetric air flow is an important factor, but velocity distribution must not be neglected. It is convenient to have high velocity near hot surfaces even if it may cause lower cumulative flow. The radius between the vanes and the friction plates should be minimal since larger radius causes air flow concentration in the middle of the channel which is not desirable from the cooling point of view.

External vane diameter has lower influence on mass flow than internal diameter. The cooling air channel is defined by friction plates and two consecutive vanes. Since all vanes are oriented to the axis of rotation the channel has the bottleneck at vane internal diameter. Air flow is limited by the smallest distance between the vanes. Longer vanes (at limited external diameter) may increase pressure difference but they reduce intake area which at least in our case showed to be more influential.

Linear dependence between air flow and vehicle speed was observed in all presented cases. This fact is a bit surprising since quadratic dependence is expected in closed channels. Possible explanation could be existence of the recirculation area on the suction side of the vane. PIV visualization of recirculation area is presented in Figure 12

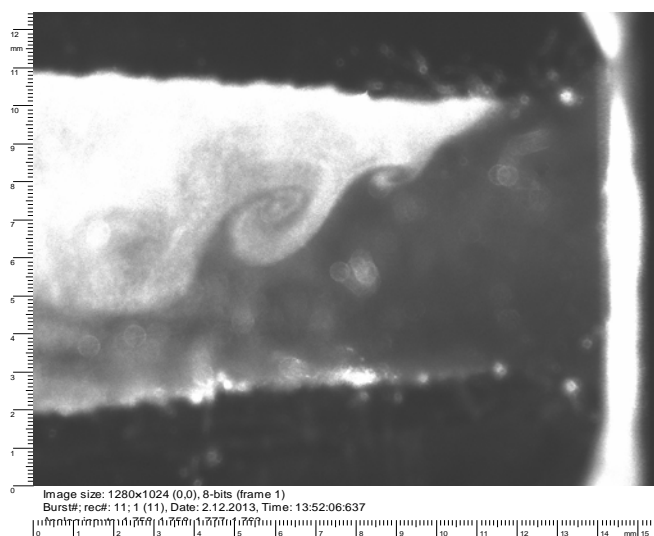


Fig. 12 PIV image of recirculation area in the inter vane channel

White (fog) in figure 12 represents recirculation area. Disc was rotated clockwise. Recirculation area reduces the mass flow and heat transfer on the suction side of the vanes due to the lower air velocity and temperature difference.

It might be beneficial to modify the disc design to reduce recirculation area and maybe achieve better cooling capacity with lower air flow and consequently reduce acoustic emissions and deposition of moisture (corrosion) in recirculation areas.

5. References

Limpert, Rudolf. (1975). Cooling Analysis of Disc Brake Rotors. <http://dx.doi.org/10.4271/751014>