

## Studying the Effect of Different Disc Designs of Floating Caliper Disc Brake on Braking Force

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**ABSTRACT:** Braking systems in general have been considered one of the most important parts in vehicles to improve and maintain safety requirements. Disc brakes are spreading now in most vehicles replacing drum brakes because it has a lot of advantages rather than drum brakes such as easy maintenance, light weight and higher effectiveness. It has also different shape designs which changes the contact area between the disc and the brake pad and that affect the friction coefficient and hence the resulting brake force. In this paper, experimental work was carried out using different designs of brake discs namely normal (ND), grooved (GD), drilled-grooved (DGD) of Floating Caliper Disc Brake (FCDB) and its effect on the braking force. A test rig has been designed and constructed that has a facility of changing various operating conditions such as brake oil pressure and rotational speed, as well as the use of brake discs with different designs. The experiments were conducted on brake oil pressure ranging from 2.5 to 10 bar and rotational speed ranging from 50 to 200 rpm, at 60 seconds of braking. The results showed that the better brake force results from normal disc (ND) but accompanied with higher friction temperature. In order to reduce the friction temperature it will be in the other discs designs. The better final friction temperature was reached in drilled-grooved (DGD) disc. By comparing the drilled-grooved disc relative to the normal disc, it was found that by increasing the brake oil pressure from 2.5 to 10 bar at constant rotational speed 150 rpm, the percentage of reduction in the mean brake force was from 15.8 %, to 9.99 % and the percentage of improvement in the final friction temperature was from 17.14% to 27.78% respectively. Also, by increasing the rotational speed from 50 to 200 rpm at constant brake oil pressure 10 bar, the percentage of reduction in the mean brake force was from 7.75 % to 12.07 % and the percentage of improvement in the final friction temperature was from 19.05% to 30% respectively.

**KEY WORDS:** Disc, Brake, Floating, Caliper, pressure and friction.

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### 1. INTRODUCTION:

A disc brake is known generally as a kind brake that uses calipers to squeeze pairs of pads against a rotor to create a friction that delay the rotation of a rotating part. In addition to that it also reduce the rotational speed or to hold it stationary. The energy of motion is converted into waste heat which must be dispersed. The disk brake is widely used in ground vehicles, especially passenger cars. However, recently the use of this type of brake has been spreading to many other kinds of ground vehicles. It is either hydraulic or pneumatic are the most commonly used and offer better braking performances compared to drum brakes. Currently, disc brakes with ventilated disc rotors are now almost universally used on the front axles of cars and light commercial vehicles because of their ability to dissipate more heat than solid rotors, especially at higher speeds [1-7]. Recent research has shown that ventilated disks have high heat transfer rates due to an increase in turbulence which results in a higher heat transfer coefficient of temperature. It also have a greater resistance to thermal deformation due to the uniform distribution of the material that reduces the thermal stress accumulated inside the rotor, which depends mainly on the geometry of the disc and the optimal configuration of the ventilation channels [8].

Many researchers investigated the effect of operation parameters affecting braking process such as sliding speed, normal force, and contact temperature. In a piece of work, the influence of the disc brake geometry on heat dissipation performance has been studied by Kang and Cho [9]. It was showed that ventilated disc has better braking performance in term of heat dissipation compared to solid disc. Also at smaller speed, ventilated disc can be more controllable. It can provide suitable torque value during braking. However, Gao and Lin [10] showed that the contact temperature is an integral factor reflecting the specific power friction influence of combined effect of load, speed, friction coefficient, and the thermo physical and durability properties of the

materials of a frictional couple. Their experiments showed that the friction coefficient decreased with increasing sliding speed and applied load, but increased with increasing disc temperature up to 300 °C and then decreased above this temperature. The specific wear rate was found to increase with increasing sliding speed and disc temperature [11].

The temperature distribution of rotor disc during braking operation using finite element methods has been investigated and analyzed to predict the temperature distribution on the full and ventilated brake discs and to identify the critical temperature of the rotor [12]. It provided also the heat flux distribution for the two discs. The thermal behaviour of full and ventilated discs in a transient state has been presented. This numerical simulation showed that radial ventilation played a very significant role in cooling of the disc during the braking phase. However, Severin et al. [13] have carried out an experimental study aimed at investigating the effect of the temperature on the coefficient of friction as function of the number of brakes. It was showed that the number of brakes increases the friction temperature as well as the coefficient of friction is not maintained at constant value as a result of the friction temperature increase. In addition to that the coefficient of friction had a minimum value when the temperature at a maximum value. The effect of the sliding speed on the friction coefficient of different materials was investigated experimentally [14]. It was found that the tendency of the friction coefficient to decrease as the sliding speed of the disc during braking increases. A number of different brake pads were evaluated with respect to friction behavior with changing speed the results showed that there is a negative friction –velocity behavior [15]. Most pads showed a slightly higher coefficient of friction at low sliding speeds. In addition to that, it was also found that at braking with constant speed the effect slightly reduced. The effect of sliding speed and normal load on the friction and wear property of an aluminum disc sliding against stainless steel pin has been also experimentally investigated [16]. It was carried out under normal load 10-20N, speed 500-2500r.p.m and relative humidity 70%. It showed that the friction coefficient decreased with increase of sliding speed and normal load, and the wear rates increased with the increase of the sliding speed and normal load. It was proved that the 90-hole cross-drilled pattern improved heat rejection capability of the disc between 8.8% and 20.1% depending on the vehicle speed [17].

A transient thermal, static structural and modal analysis on the disc brake rotor using three different materials and three different 36-hole ventilated profiles like has been carried out in comparison with solid disc [18]. It was noted 9%, 15% and 20% of heat reduction. A new design of 60-holes cross drilled rotor pattern and conducted a thermal analysis on it was adopted as a result of improvement till the 10<sup>th</sup> cycle were at 288.4 °C while the allowable service temperature was 550 °C [19]. So, it will be considered for the new work in this paper to make extra modification to the disc brake rotor and studying the effect of different disc designs of floating caliper disc brake on braking force.

## **II. EXPERIMENTAL WORK AND INSTRUMENTATIONS**

In this study, a laboratory brake test rig was designed and constructed to investigate experimentally the different parameters such as brake force, temperature and friction coefficient during braking process. The test rig has a facility of changing various operating conditions such as rotational speed and brake oil pressure, as well as the use of brake system with different designs. Some modifications were carried out to on the Floating Caliper Disc Brake (FCDB) as well as a replacement of different designs of ventilated discs namely normal, grooved and drilled-grooved disc to investigate the effect of different operating parameters on braking force during braking process.

The brake test rig is constructed from rotational motion generation system, pressurized oil generation system and Floating Caliper Ventilated Disc Brakes system. All of these systems are installed on one chassis as shown in Figure 1. A 3-phase AC electric motor of 10 HP and 1500 rpm was used to rotate the test rig brake at the required rotational speed as clear in figure 2. An inverter of KAS var type model VFC-18 of 50Hz was connected to the AC motor to give extra rotational speed. It controls the rotational speed through changing the inverter frequency. In order to perform the experiments at various speeds, a gear box with differential reduction speed is installed between the electric motor and the brake system to simulate the real vehicle at the different braking conditions.



Figure 1. Test rig components.

The pressurized oil generation system consists mainly of oil reservoir, a master cylinder working through screw with handle, valve and pressure gauge. This system is used to generate a constant oil pressure supplied to piston of floating caliper ventilated disc brake system to apply constant normal force that affect the brake pad as shown in figure 2.

Disc brake system used in this work is a floating caliper ventilated disc brake system of a commercial vehicle of Hyundai Excel as shown in figure 3. The main components of the floating caliper ventilated disc brake system are floating caliper, piston, hub, wheel bearing, rotor disc, and brake pads as shown in figure 4.

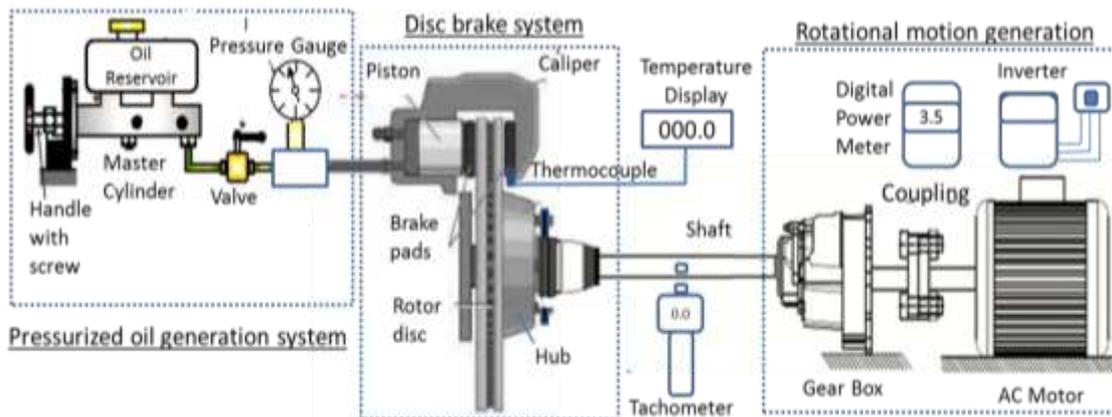


Figure 2. Schematic drawing of the test rig and measuring instrumentations.

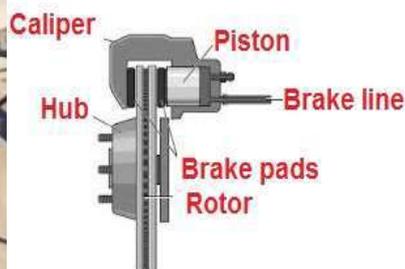


Figure 3. Floating caliper disc brake (FCDB). Figure 4. Main components of FCDB.

However; the specifications of the used disc brake are shown in Table1.

**Table1** specifications of the used disc brake system

Component	Dimension
Disc outer diameter	240 mm
Disc inner diameter	120 mm
Disc thickness	20 mm
Friction material length	110 mm
Friction material width	50 mm
Friction material thickness	10 mm
Pad back plate length	130 mm
Pad back plate thickness	6 mm
Hydraulic piston diameter	50 mm

**2.1 Pressure measurement and normal force calculation**

A pressure gauge was used during the experiments to measure the oil pressure in the range from 2.5 bar to 10 bar. The values of the applied force are determined according to the braking oil pressure through the following equations [2-3];

$$F_n = P \times A \tag{1}$$

Where:

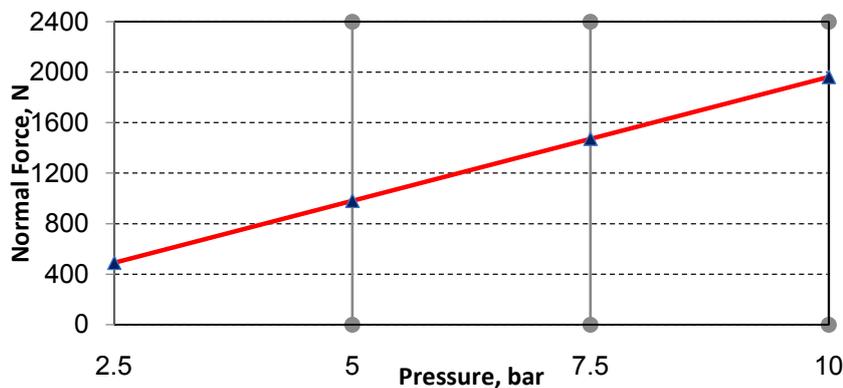
$F_n$  is the normal force, N.

$P$  is the hydraulic oil pressure, bar.

$A$  is the piston area  $= \frac{\pi}{4} D^2$ , mm<sup>2</sup>.

$D$  is the piston diameter, mm.

The applied normal force equivalent to the selected oil pressure is shown in figure 5 which represent the calibration chart of the applied normal force of the used brake system.



**Figure 5.** Brake oil pressure versus the normal force of the used brake system (calibration chart).

**2.2 Speed measurement**

The rotational speed of the rotor disc was measured by using a digital tachometer of (DT6234B) type with arange from 5 to 100000 rpmand accuracy of 0.5%.Four rotational speeds of the rotating discare selected during the test in the range from 50 to 200 rpm.

**2.3 Brake Torque Calculation**

A digital power meter of Schneider PM 1200 type with a range from 20 W to 300 KW and accuracy of 1% of reading for power was used to measure the power consumption by the electric motor during the braking process and no loadcondition as shown in the following equation [2];

$$P_{\text{brake}} = P_{\text{Load}} - P_{\text{no load}} \quad (2)$$

Where:

- $P_{\text{brake}}$  is the brake power, W.
- $P_{\text{Load}}$  is the electric motor power during the braking process, W.
- $P_{\text{no load}}$  is the electric motor power during the operation at no load case, W.

Then, the brake torque was calculated according to the following equation [3];

$$T_{\text{brake}} = P_{\text{brake}} / \omega \quad (3)$$

where :

- $T_{\text{brake}}$  is the braking Torque, Nm.
- $\omega$  is the angular speed of rotating disc  $= \frac{2\pi n}{60}$ , rad/sec.
- $n$  is the rotational speed of the rotating disc, rpm.

#### 2.4 Brake Force and Friction Coefficient Calculations

For a disc brake system there is a pair of brake pads, the brake force of the floating caliper disc brake can be calculated as follow:

$$F_{\text{brake}} = T_{\text{brake}} / 2R_{\text{eff}} \quad (4)$$

Where:

- $F_{\text{brake}}$  is the brake force generated at the contact interface, N.
- $R_{\text{eff}}$  is the effective radius of the brake pad  $= \frac{r_o + r_i}{2}$ , mm.
- $r_o$  is the outer radius of the brake pad, mm.
- $r_i$  is the inner radius of the brake pad, mm.

Then the friction coefficient  $\mu$  can be calculated from the relation between braking force and normal force, which can be given by:

$$\mu = F_{\text{brake}} / F_n \quad (5)$$

#### 2.5 Temperature measurement

A thermocouple of J-Type was used and fixed in the brake pad to measure the friction temperature at the contact area between the brake pad and the brake disc of floating caliper ventilated disc brake system. The thermocouple was connected to temperature display unit as shown clearly in figure 2.

#### 2.6 Experimental works

The experimental works was carried out by using three different designs of brake discs of floating caliper ventilated disc brake on Braking Force. Three kinds of floating caliper disc brake are used as follows;

##### 2.6.1 Normal Disc Brake (ND):

The normal disc brake face of floating caliper ventilated disc brake system is used in the experiment as a standard disc without any modifications and will be referred as ND as shown in the figure (6-A).

##### 2.6.2 Grooved Disc Brake (GD):

Some modification will be made to the disc surfaces of the brake system for both sides by digging 8 grooves (slots) on each side of the disc and will be referred as grooved disc (GD). Each groove has a length of 60 mm, width of 6 mm and a depth of 3 mm as shown in the figure (6-B).

##### 2.6.3 Drilled –Grooved Disc Brake (DGD):

An extra modification to grooved disc (GD) will be made by drilling 48 holes on both side of the disc (24 holes on each side) and will be referred as drilled-grooved disc (DGD). Each hole has a diameter of 6 mm and depth of 3 mm in addition to the 16 grooves on both sides of the disc as shown in the figure (6-C).

The experiments were conducted for each of three different disc brake designs of floating caliper ventilated disc brake system. The experiments were carried out on four brake oil pressure ranging from 2.5 to 10 bar with an increment of 2.5 bar and four rotational speed ranging from 50 to 200 rpm with step of 50 rpm. All experiments were performed at 60 seconds of braking time each, and at initial disc temperature of 24 °C.



Figure 6. Three different disc brake designs of floating caliper ventilated disc brake system.

### III. RESULTS AND DISCUSSIONS

#### 3.1 Effect of brake oil pressure on the brake force at rotational speed of 150 rpm.

The effect of brake oil pressure on the brake force of normal (ND), grooved (GD) and drilled-grooved (DGD) discs of floating caliper ventilated disc brake system at rotational speed of 150 rpm is presented in figures 7, 8 and 9. It can be concluded that the brake force is increased with increasing the brake oil pressure of the normal, grooved and drilled-grooved discs at 60 seconds of braking time. The fluctuation of the brake force is due to the variation of the friction coefficient with the braking time.

Figure 8-a shows that the mean brake force of the normal disc brake is higher than the mean brake force of the grooved and drilled-grooved discs brake at the same operation condition and this is because normal disc brake has more surface contact area with the brake pad when the brake is applied.

Figure 8-b shows comparison between the mean brake force for normal disc (ND) brake, grooved disc brake (GD) and drilled grooved disc (DGD) brake. The results in figure 8-b also indicate that the mean brake forces for the normal disc (ND) is reached to 196,403,612 and 821 N at the braking oil pressure of 2.5, 5, 7.5 and 10 bar respectively. However; for the grooved disc (GD) it is reduced to 185,376, 570 and 792 N respectively at the same values of pressures and for the drilled-grooved disc (DGD) the mean brake force is reduced to 165, 341,525 and 739 N at also the same values of oil pressures.

Figure 9 shows the percentage of reduction in the mean brake force of drilled grooved disc (DGD) brake and grooved disc (GD) brake relative to normal disc (ND) brake. The results in figure 9 indicate that by increasing the brake oil pressure from 2.5 to 10 bar and at constant rotational speed 150 [rpm], the percentage of reduction in the mean brake force was from 15.8 %, to 9.99 % for drilled grooved disc (DGD) brake and from 5.61 %, to 3.53 % for grooved disc (GD) brake relative to normal disc (ND) brake.

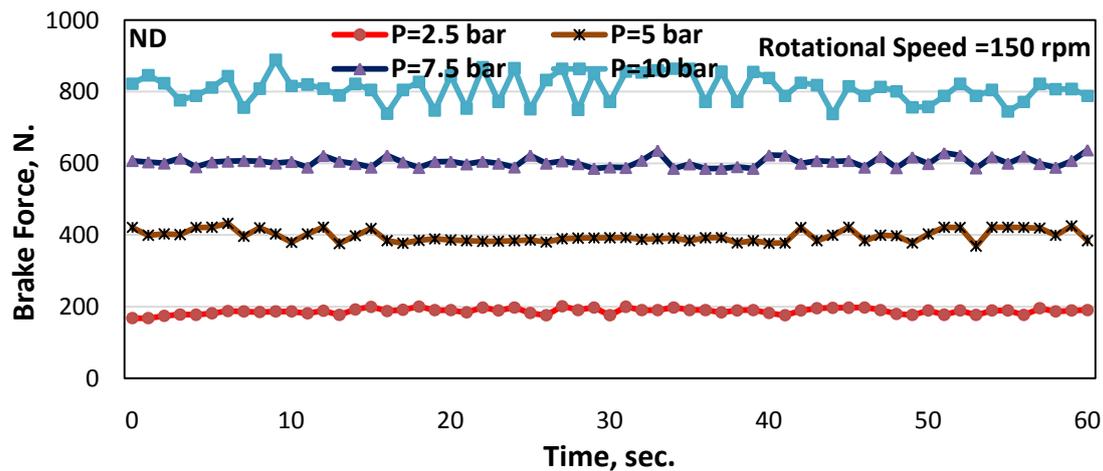


Figure 7-a

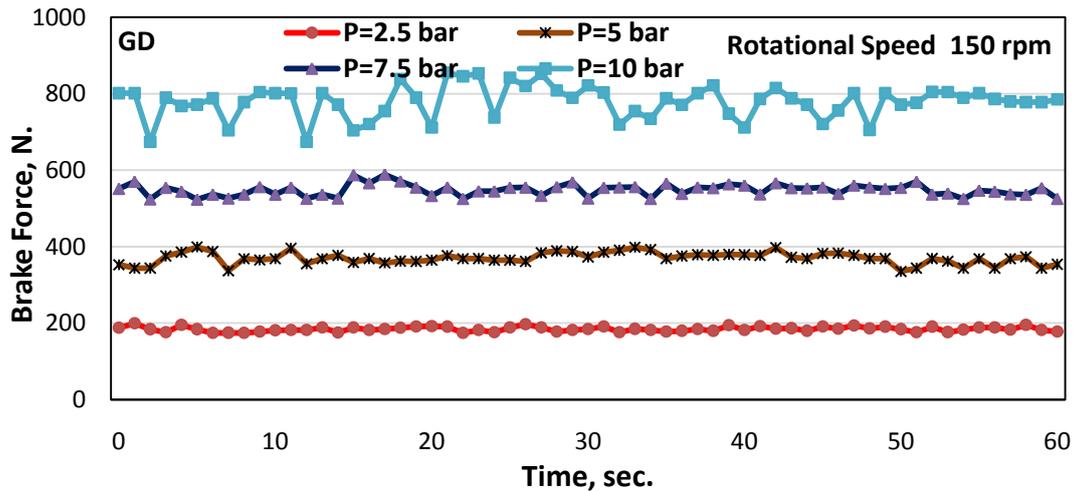


Figure 7-b

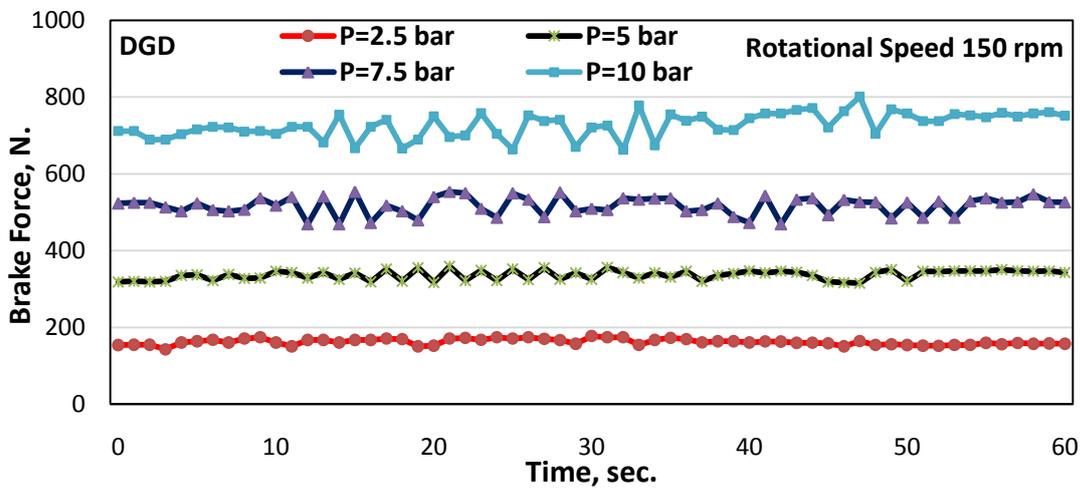


Figure 7-c

Figure 7. Brake force versus time for ND, GD and DGD at different brake pressure and speed of 150rpm.

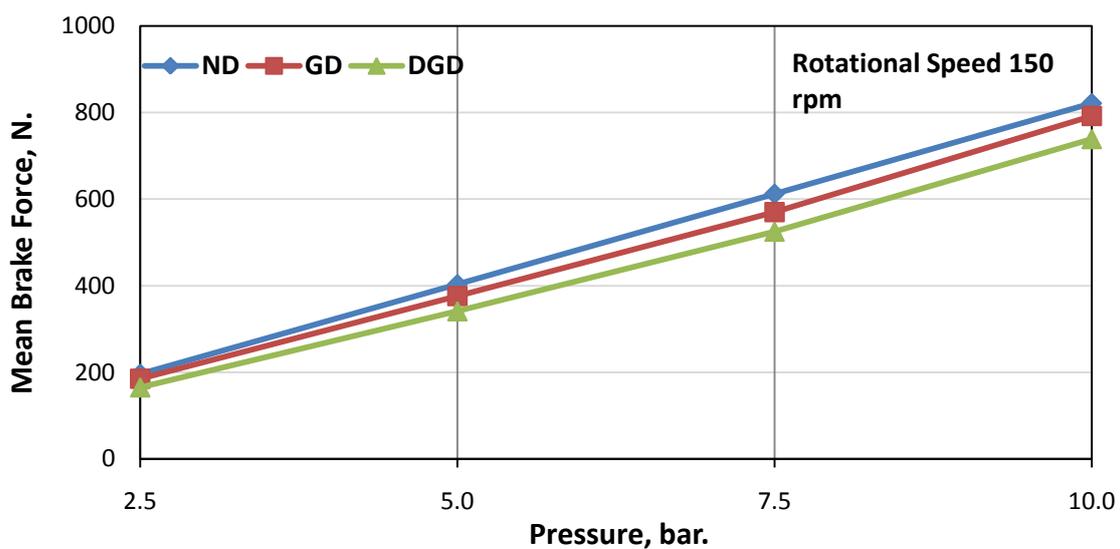


Figure 8-a. Mean brake force versus brake oil pressure for ND, GD and DGD at rotational speed of 150 rpm.

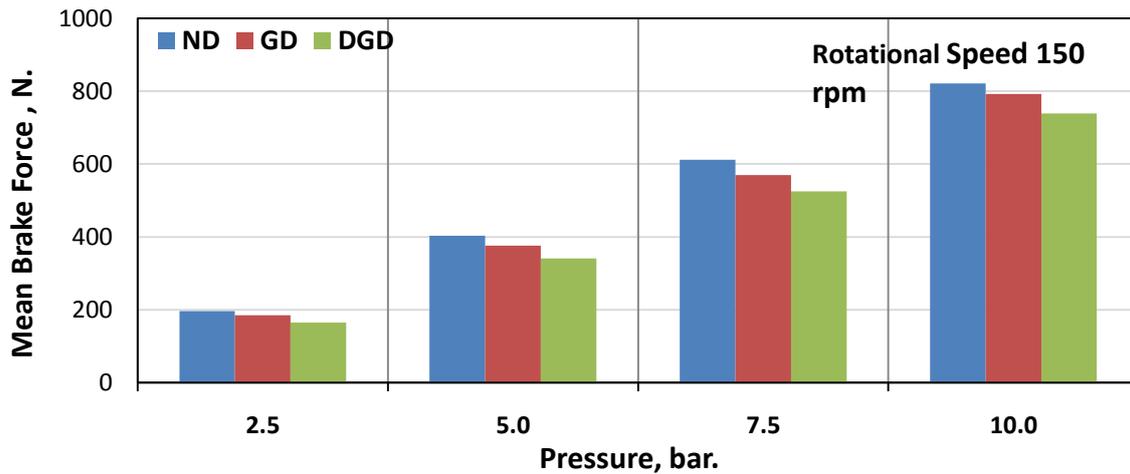


Figure 8-b Mean brake force comparison at different brake oil pressure for ND, GD and DGD at rotational speed of 150 rpm.

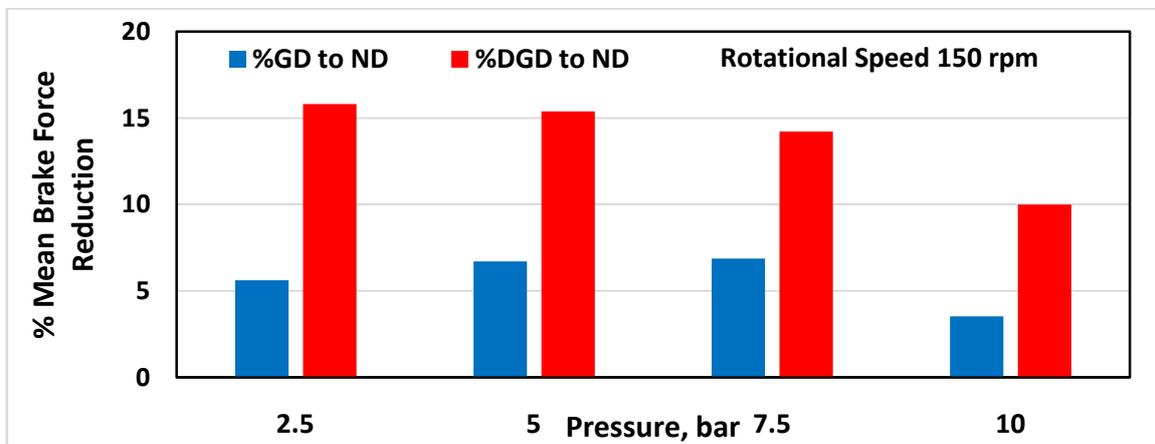


Figure 9. Mean brake force reduction percentage of GD and DGD compared to ND for different brake oil pressure at rotational speed of 150 rpm.

### 3.2 Effect of brake oil pressure on the mean friction coefficient at rotational speed of 150 rpm.

Figures 10 and 11 show the mean friction coefficient versus the brake oil pressure at rotational speed of 150 rpm for the three kinds of used discs mentioned previously. The results indicated that the increase in the brake oil pressure causes an increase of the mean friction coefficient for the normal, grooved and drilled-grooved discs. The increase of the brake oil pressure from 2.5 to 10 bar causes an increase of the mean friction coefficient from 0.40 to 0.42 for the normal disc and from 0.38 to 0.40 for the grooved disc and from 0.34 to 0.38 for the drilled-grooved disc. The results in figures 10 and 11 also indicated that, the mean friction coefficient of the normal disc is higher than the mean friction coefficient of the grooved and drilled-grooved discs at the selected brake oil pressure values 2.5, 5, 7.5 and 10 bar respectively.

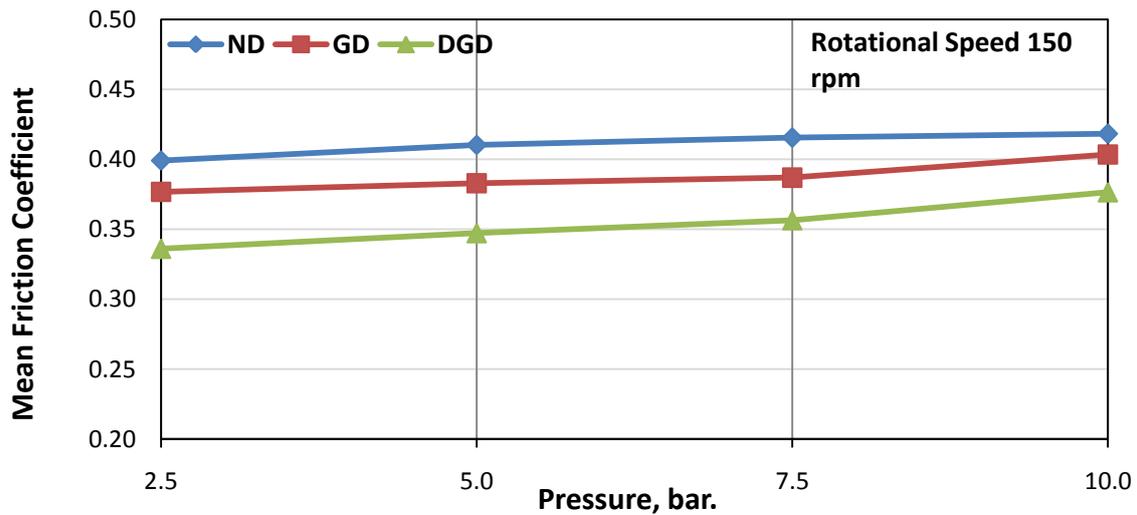


Figure 10. Mean friction coefficient versus brake pressure for N, G and DG discs brake at rotational speed of 150 rpm.

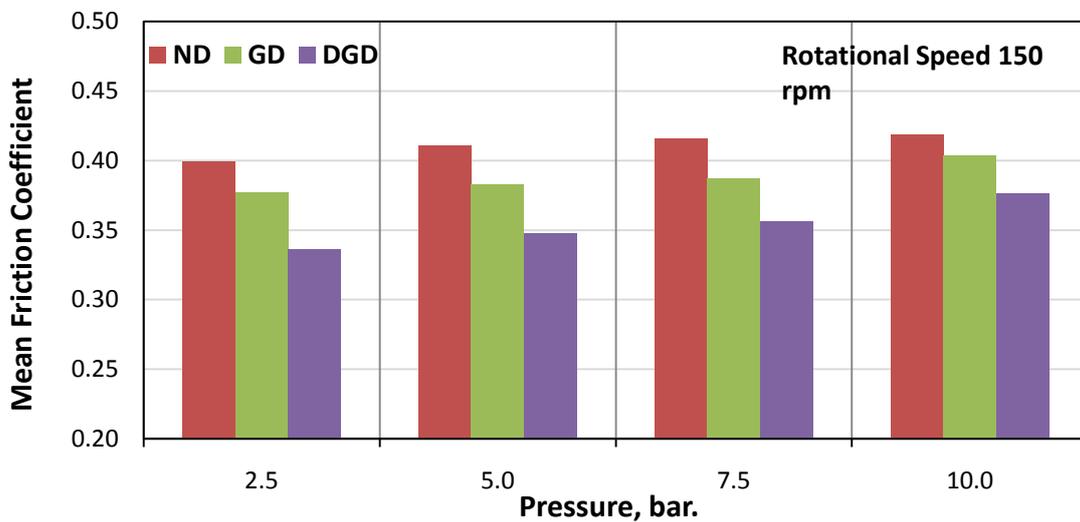


Figure 11. Mean friction coefficient comparison at different brake oil pressure for N, G and DG discs at rotational speed of 150 rpm.

### 3.3 Effect of rotational speed on the brake force at brake oil pressure of 10 bar.

The effect of rotational speed of the disc on the brake force of normal (ND), grooved (GD) and drilled-grooved (DGD) discs brake at brake oil pressure of 10 bar is presented in figures 12 and 13. It can be seen that the increase of the rotational speed of the rotating disc causes a decrease in the mean brake force of the normal, grooved and drilled-grooved discs. Figure 12-a shows that the mean brake force of the normal disc brake is higher than the mean brake force of the grooved and drilled-grooved discs brake at the same operation condition. Figure 12-b shows comparison between the mean brake force for normal disc (ND) brake, grooved disc brake (GD) and drilled grooved disc (DGD) brake. The results in figure 12-b also indicate that the increase of the rotational speed from 50 rpm to 200 rpm causes a decrease in the mean brake force of the normal disc by 8.3% and for the grooved disc by 10.3% and for the drilled-grooved disc by 12.6%. Figure 13 shows the percentage of reduction in the mean brake force of drilled grooved disc (DGD) brake and grooved disc (GD) brake relative to normal disc (ND) brake. The results in figure 13 indicate that by increasing the rotational speed from 50 to 200 rpm and at constant brake oil pressure of 10 bar, the percentage of reduction in the mean brake force was from 7.75% to 12.07% for drilled grooved disc (DGD) brake and from 3.65% to 5.72% for grooved disc (GD) brake relative to normal disc (ND) brake.

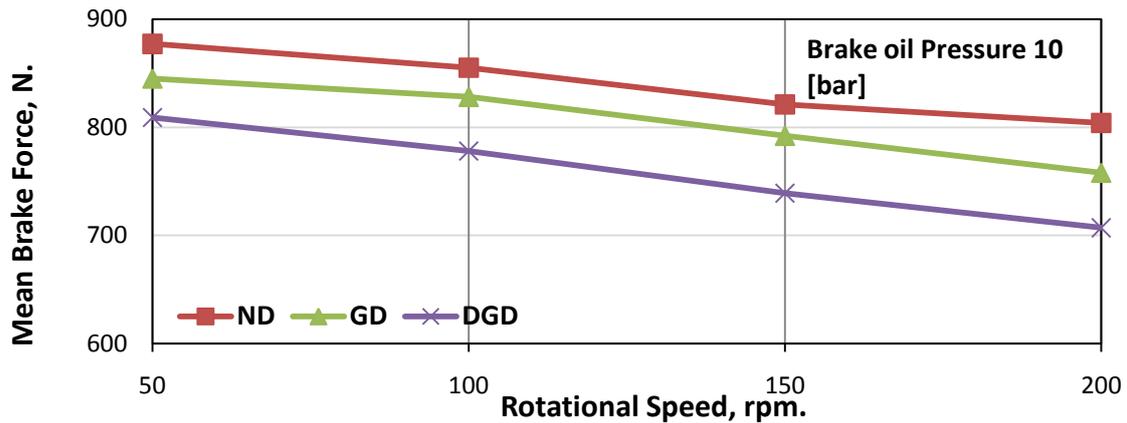


Figure 12. Mean brake force versus rotational speed for N, G and DG discs brake at brake oil pressure of 10 bar.

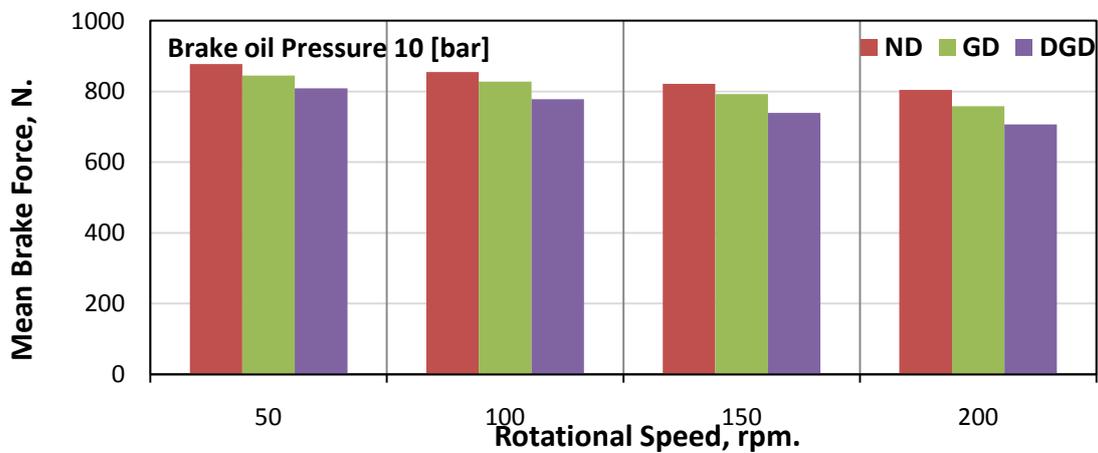


Figure 13. Mean brake force comparison at different rotational speed for N, G and DG discs at brake oil pressure of 10 bar.

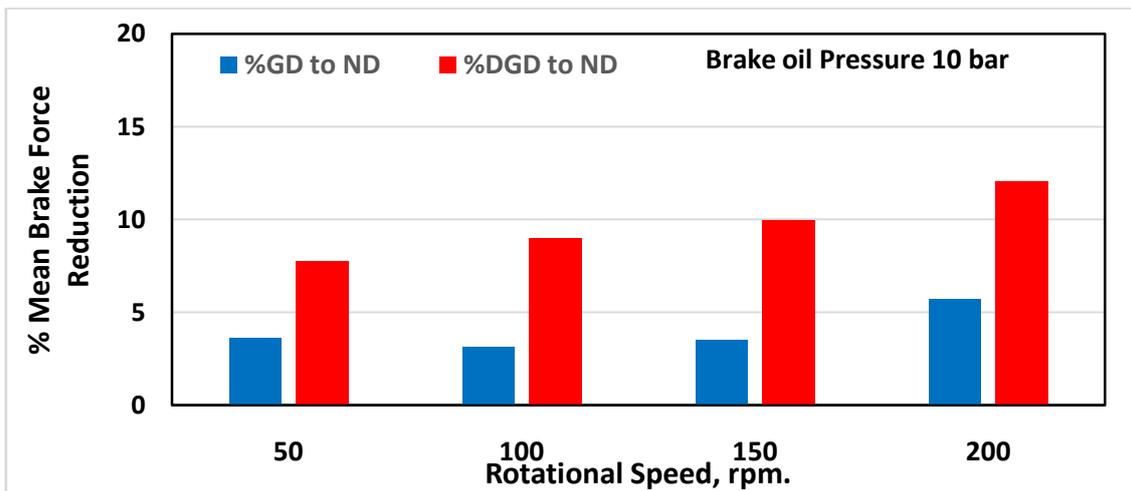


Figure 13. Mean brake force reduction percentage of GD and DGD compared to ND for different rotational speed at brake oil pressure 10 bar.

### 3.4 Effect of rotational speed on the mean friction coefficient at brake oil pressure of 10 bar.

Figures 14 and 15 show the mean friction coefficient of the normal (ND), grooved (GD) and drilled-grooved (DGD) discs versus the rotational speed at brake oil pressure of 10 bar. The results indicated that the increase of the rotational speed of the rotating disc brake causes a decrease in the mean friction coefficient for the three types of used discs. The increase of rotational speed from 50 to 200 rpm causes a decrease in the mean friction coefficient from 0.45 to 0.41 for the normal disc and from 0.43 to 0.39 for the grooved disc and from

0.41 to 0.36 for the drilled-grooved disc. It also indicated that, the mean friction coefficient of the normal disc is higher than the mean friction coefficient of the grooved and drilled-grooved discs at each constant speed because normal disc brake has more surface contact area with the brake pad during applying the brake.

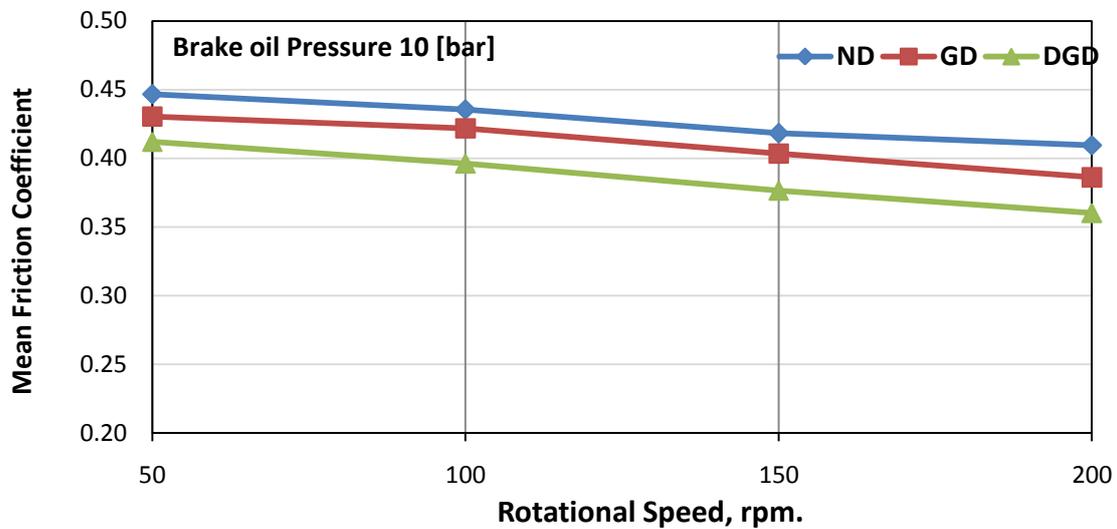


Figure 14. Mean friction coefficient versus rotational speed for N, G and DG discs brake at brake oil pressure of 10 bar.

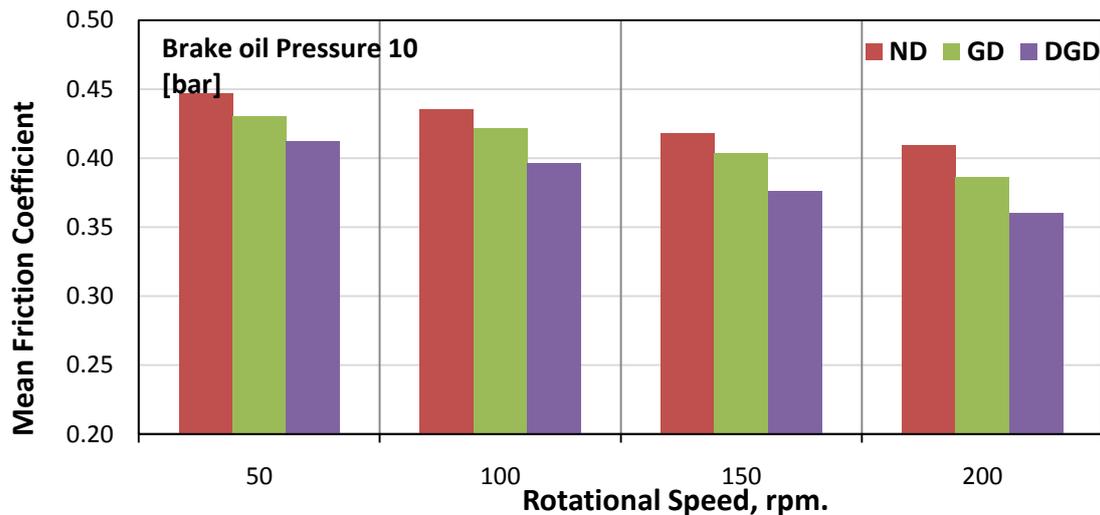


Figure 15. Mean friction coefficient comparison at different rotational speed for N, G and DG discs brake at brake oil pressure of 10 bar.

### 3.5 Effect of disc design types on the final friction temperature:

The effect of the disc design types of normal (ND), grooved (GD) and drilled-grooved (DGD) discs on the final friction temperature for rotational speed of the rotating disc and brake oil pressure are presented in figures 16, 17 and 18. The experiments were carried out on four brake oil pressure ranging from 2.5 to 10 bar with increment of 2.5 bar and four rotational speed ranging from 50 to 200 rpm with step of 50 rpm. The results indicated that, the final friction temperature increases with increasing of the rotational speed of the rotating disc and also by increasing brake oil pressure as clear obviously in figures 16, 17 and 18.

The results presented in Figure 19 show the variation of temperature due to friction for the normal, grooved and drilled-grooved discs versus the brake oil pressure at rotational speed of 150 rpm. It shows that increasing the brake oil pressure from 2.5 bar to 10 bar causes an increase in the mean value of final friction temperature from 35 °C to 54 °C for normal disc, from 32 °C to 46 °C for grooved disc and from 29 °C to 39 °C for the drilled-grooved disc. Figure 19 also indicates that, the final friction temperature of the normal disc is higher than the final friction temperature of the grooved and drilled-grooved discs at the selected brake oil pressure values.

Figures 20 and 21 show the percentage of enhancement in final friction temperature during braking process of drilled grooved disc (DGD) brake and grooved disc (GD) brake relative to normal disc (ND) brake. The results in figure 20 indicate that by increasing the brake oil pressure from 2.5 to 10 bar at constant rotational speed 150 rpm, the percentage of improvement in the final friction temperature was from 17.14% to 27.78% for drilled grooved disc (DGD) brake and from 8.57% to 14.8% for grooved disc (GD) brake relative to normal disc (ND) brake. The results in figure 21 indicate that by increasing the rotational disc speed from 50 to 200 rpm at constant brake oil pressure 10 bar, the percentage of improvement in the final friction temperature from 19.05% to 30% for drilled grooved disc (DGD) brake and from 9.52% to 11.67% for grooved disc (GD) brake relative to normal disc (ND) brake. From figures 20 and 21 it was concluded that there was good enhancement in final friction temperature during braking process in the drilled-grooved disc because it has a good heat dissipation and removing ability of brake wear particles.

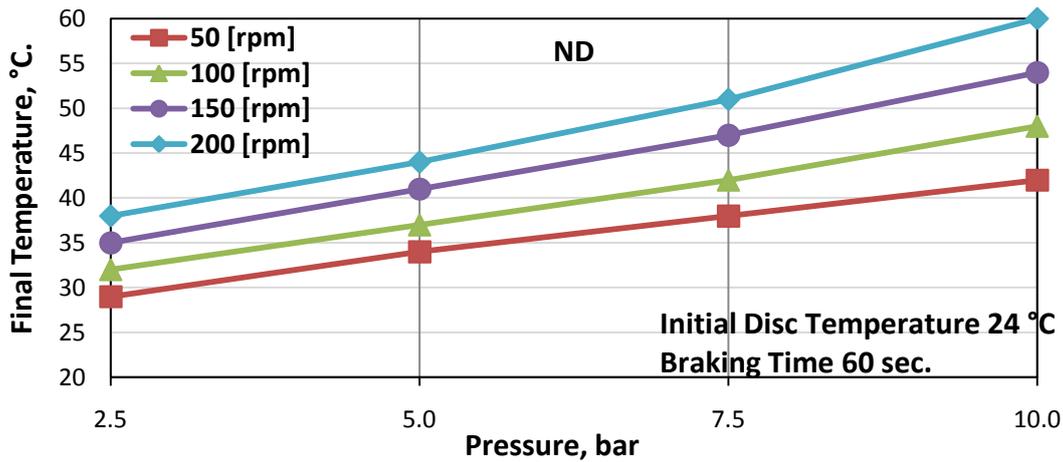


Figure 16. Final friction temperature versus brake oil pressure for (ND) brake with various rotational speeds.

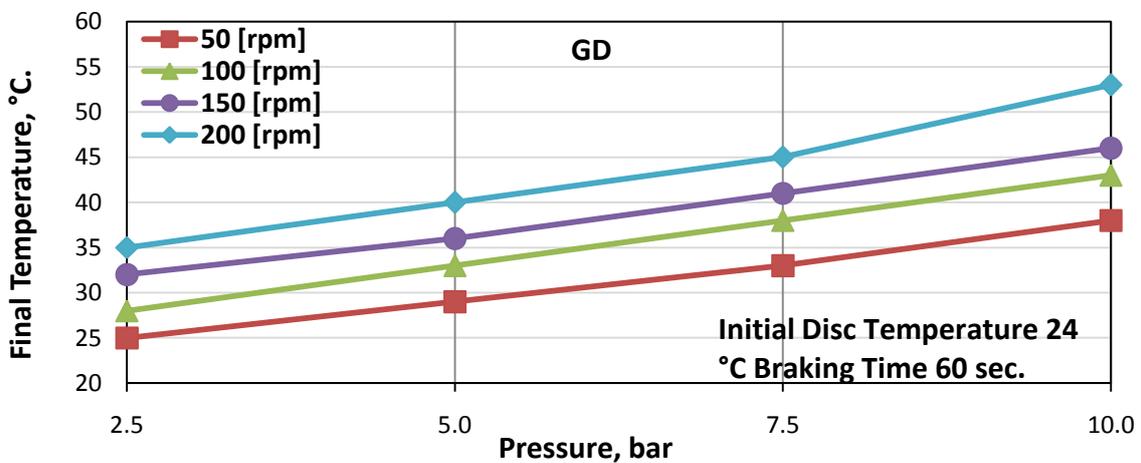


Figure 17. Final friction temperature versus brake oil pressure for (GD) brake with various rotational speeds.

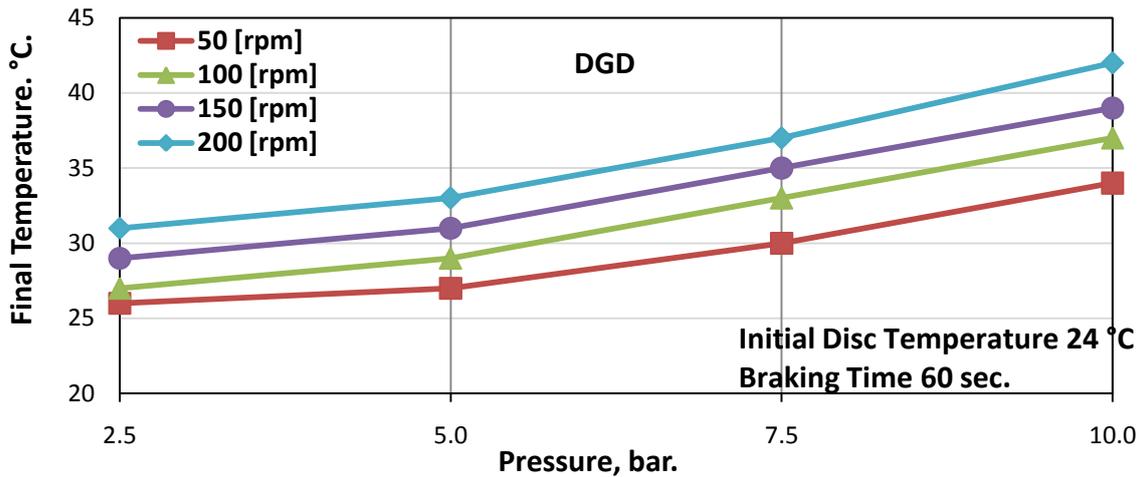


Figure 18. Final temperature versus brake oil pressure for (DGD) brake with various rotational speeds.

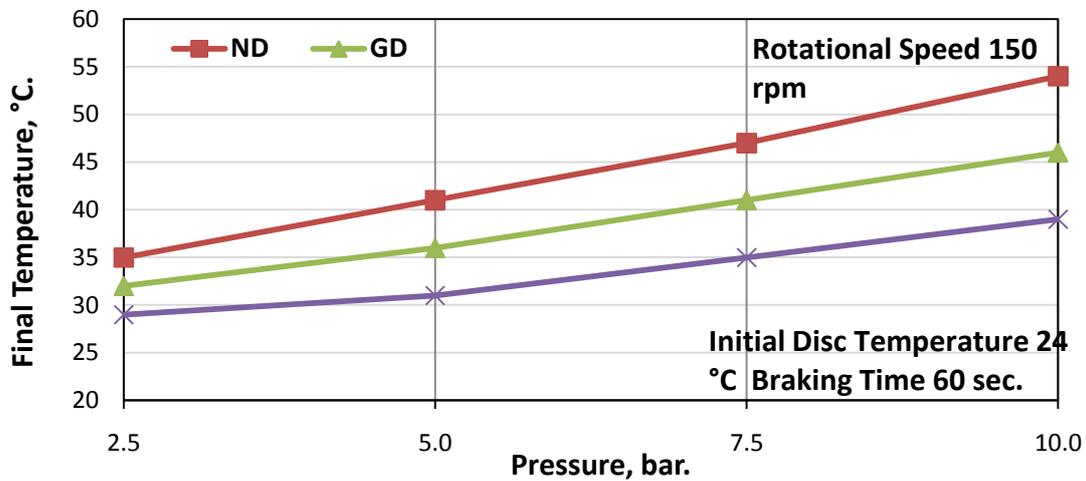


Figure 19-a. Final friction temperature versus brake oil pressure for N, G and DG discs brake at rotational speed of 150 rpm.

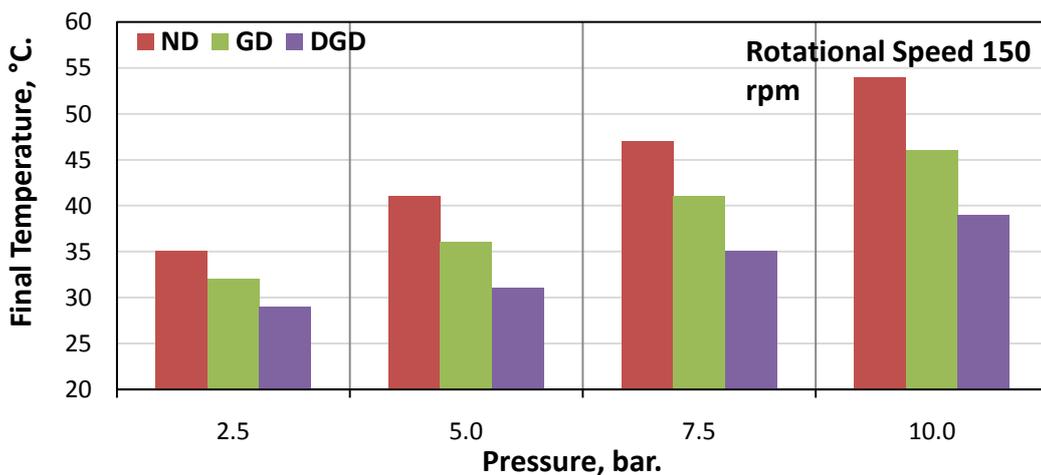


Figure 19-b. Final friction temperature comparison at different brake oil pressure for N, G and DG discs at rotational speed of 150 rpm.

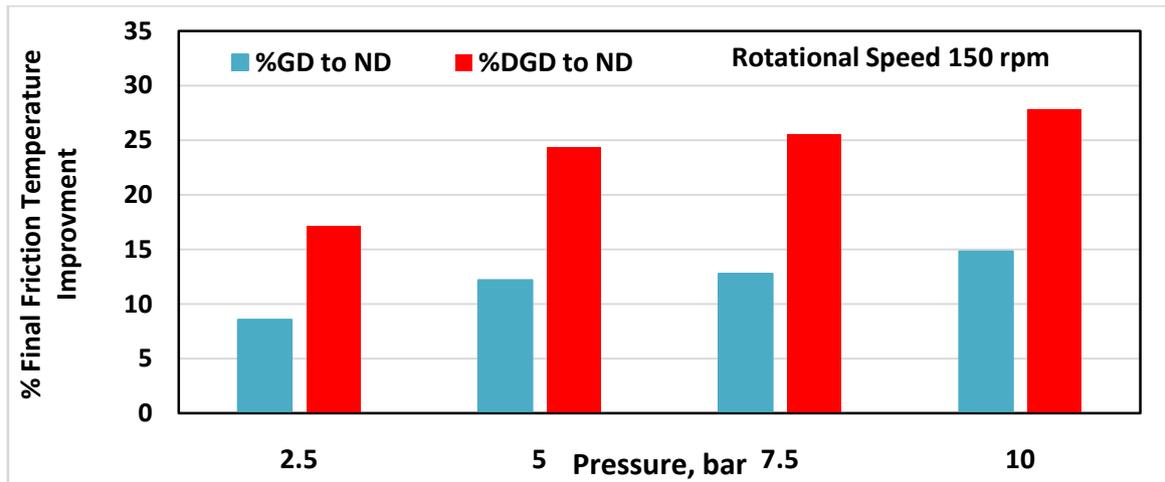


Figure 20. Final friction temperature improvement percentage of GD and DGD compared to ND for different brake oil pressure at rotational speed 150 rpm.

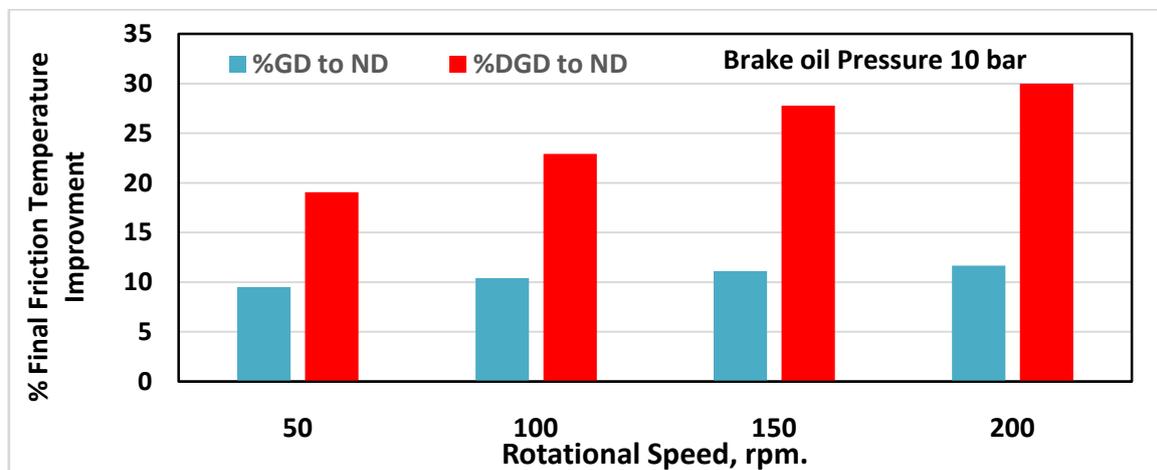


Figure 21. Final friction temperature improvement percentage of GD and DGD compared to ND for different rotational speed at brake oil pressure 10 bar.

#### IV. CONCLUSION:

From the results of the present study, the important conclusions are as follows:

1. The mean friction coefficient and the braking force increase with increasing brake oil pressure for normal, grooved and drilled-grooved discs of floating caliber ventilated disc brake. Also the mean friction coefficient of the normal disc is higher than the mean friction coefficient of the grooved and drilled-grooved discs and consequently the braking force of normal disc is more than grooved and drilled-grooved disc.
2. Increasing brake oil pressure from 2.5 to 10 bar at constant rotational speed 150 rpm causes a percentage reduction in the mean brake force from 15.8 %, to 9.99 % for drilled grooved disc (DGD) brake and from 5.61 %, to 3.53 % for grooved disc (GD) brake relative to normal disc (ND) brake.
3. The increase of brake rotational disc speed at constant pressure causes a decrease of the mean friction coefficient and consequently a decrease of the mean brake force of the normal, grooved and drilled-grooved discs.
4. Increasing the rotational disc speed from 50 to 200 rpm at constant brake oil pressure 10 bar causes a percentage reduction in the mean brake force from 7.75 %, to 12.07 % for drilled grooved disc (DGD) brake and from 3.65 %, to 5.72 % for grooved disc (GD) brake relative to normal disc (ND) brake.
5. The friction temperature increases with increasing brake oil pressure and disc rotational speed. The final friction temperature of the normal disc (ND) is higher than the final friction temperature of the grooved (GD) and drilled-grooved (DGD) discs at the selected brake oil pressure values.
6. Increasing brake oil pressure from 2.5 to 10 bar at constant rotational speed 150 rpm causes a percentage improvement in the final friction temperature from 17.14% to 27.78% for drilled grooved disc (DGD) brake and from 8.57% to 14.8% for grooved disc (GD) brake relative to normal disc (ND) brake.

7. Increasing the rotational disc speed from 50 to 200 rpm at constant brake oil pressure 10 bar causes a percentage improvement in the final friction temperature from 19.05% to 30% for drilled grooved disc (DGD) brake and from 9.52% to 11.67% for grooved disc (GD) brake relative to normal disc (ND) brake.

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Ibrahim Ahmed "Studying the Effect of Different Disc Designs of Floating Caliper Disc Brake on Braking Force" International Journal of Engineering Inventions, Vol. 08, No. 4, 2019, pp. 18-32