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Comparative Performance Study of Floating Caliber Ventilated Disc Brake and Fixed Caliber Solid Disc Brake

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ABSTRACT : *In this paper, Experimental performance comparison between floating caliper ventilated disc brake unit (FL) and fixed caliper solid disc brake unit (FX) with three different profile designs of brake discs is carried out. The three profiles of discs are normal (ND), grooved (GD) and drilled grooved (DGD) which are used during the tests. In this study, brake test rig was designed to simulate the actual environment of braking process. It was used in experimental comparisons of different brake systems designs performance at different operation parameters during braking process. Brake oil pressure and rotor disc speed are the operation parameters that are varied during the running test. The comparisons are focused on brake force, friction coefficient and final friction temperature. The comparisons are carried out for brake oil pressure values 2.5, 5, 7.5 and 10 [bar] and disc speed 50, 100, 150 and 200 [rpm]. By using the measured parameters from experimental works and mathematical calculations for brake force and friction coefficient, the comparisons were carried out for different brake unit types and rotors designs. The results and conclusion from the comparisons graph and trends show that, at low brake oil pressure 2.5 [bar] and constant disc speed 150 [rpm], the better percentage of improvement for (FL-ND) mean brake force is 30.67% and mean friction coefficient is 27.94% relative to (FX- ND) and the better percentage of reduction of mean final friction temperature for (FL- DGD) relative to (FX- ND) is 31.71%.*

KEYWORDS *Disc, Brake, Caliper, Floating, Fixed, Friction.*

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I. INTRODUCTION

Brakes are one of the most important components of a car as they're the primary source to bring the car to a halt. Failure to brake can result in a disaster and manufacturers are increasingly working to make braking on their vehicles efficient for better passenger safety. Different vehicles use different types of brakes; while some use a basic drum brake, others use a disc brake. Disc brakes are the most widely used as well. These consist of a disc brake rotor which is attached to the wheel, and a caliper which holds the brake pads in place. The master cylinder exerts hydraulic pressure causing the caliper (piston) to clamp the rotor between the disc brake pads causing friction between the pads and the rotor, bringing the car to a halt. The two most popular types of disc brake rotors are solid and ventilated discs. A solid brake disc is the simplest form and consists of a single solid disc. Ventilated discs have cooling vanes between the braking surfaces which allows air to flow through, and has a cooling effect on the disc. There are two types of disc brakes, named after the type of brake caliper used: floating and fixed. A floating caliper (also called sliding) is the most common type. It has one or two pistons. When the brakes are applied, the inner brake pad is forced against the disc while, at the same time, the caliper body moves closer to the rotor. This action forces the outer brake pad against the rotor. The fixed caliper design has one or more pistons mounted on each side of the rotor. The caliper itself doesn't budge: It's rigidly fastened to a brake caliper bracket or the spindle. When the brakes are applied, only the caliper pistons move, pressing the brake pads against the disc. Disc brake pads consist of steel shoes to which the lining is riveted or bonded. They slow the rotor through friction and they wear with normal use.

Studying the relations of different operation parameters of braking process was carried out in many research. E.J. Berger, [1] mention that the brake functional dependence of the friction coefficient upon a large variety of parameters; including sliding speed, acceleration, critical sliding distance, temperature, normal load, humidity, surface preparation and of course material combination. EL-Tayeb and Liew [2] reported that the variations in the friction coefficient are highly dependent on the frictional materials and braking conditions.

A. Yasuhisa [3], S. Guo, et al. [4] and A. Belhocine [5] were considered the normal load, sliding speed, ambient conditions and material to obtain lower friction coefficient measuring the friction and pull-off forces between a metal pin and plate. Ibrahim Ahmed, et al., [6] showed from experimental results that the sliding speed of the rotating disc has negative effect on the performance of the conventional and modified braking systems. M.A. Chowdhary, et al., [7] studied the torque caused by the friction force between the brake pad and the disc rotor on the rotational deceleration of the wheel and rotor. Experiments by Gao and Lin [8] showed that in general the coefficient of friction decreased with increasing sliding speed and applied load, but increased with increasing disc temperature up to 300 °C and then decreased above this temperature. Some experiments by Bogdanovich and Tkachuk [9] presented to thermal phenomena in a sliding contact, revealing that asperities in contact were found between two sliding bodies to undergo the pulse effect of the temperature. H. Jacobsson [10] and B.K. Choi et al., [11] mentioned that the ventilated rotor disc is lighter than the solid rotor disc, and ventilated rotor disc has additional convective heat transfer from its surface. So, the ventilated rotor disc can reduce temperature rise. S.P. Jung, et al. [12] mentioned that the thermal capacity of the ventilated disc is less than that of the solid disc, and during repetitive braking, ventilated disc temperature can rise relatively faster than the solid disc temperature. By using an aluminum disc sliding against stainless steel pin, M. A. Chowdhur, et al., [13] shown the values of friction coefficient decrease with the increase of sliding speed and normal load. On the test stand, W Szczypinski-Sala and J Lubas, [14] compare two cross-drilled brake discs with different geometry, measurement was conducted on the surface of disc and brake pad at various sliding speeds and loads. It was observed that the friction coefficient increased with temperature in the investigated range and the friction coefficient is also related to load and speed. Pandya Nakul Amrish [15] compare structural and thermal analysis results of normal solid disc rotor and a drilled disc rotor. It is observed that for both rotors, the temperature variation is nearly the same, but for the drilled rotor, maximum temperature is slightly lesser as a result of increasing the surface area due to the drilled holes for heat dissipation during braking.

II. EXPERIMENTAL TEST RIG

In this study, brake test rig was designed to simulate the actual environment of braking process. It was used in experimental comparisons of different brake systems designs performance at different operation parameters during braking process. The designed brake test rig consists of driving unit, braking units and applying normal force unit. All of these units are arranged and fixed on one rigid steel structure fixed on the ground by anchors to withstand dynamic loads during operation as shown in Fig.1.

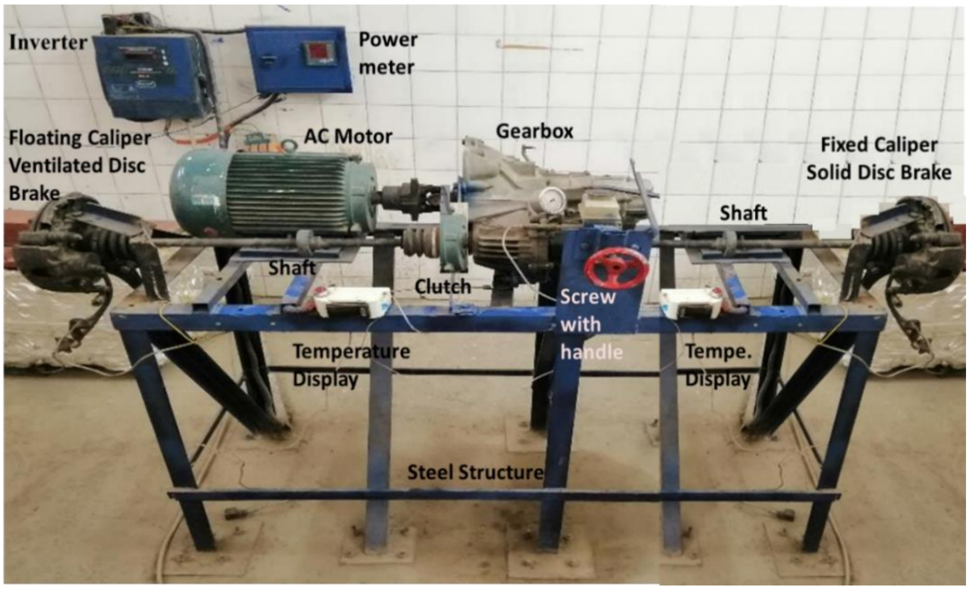

Fig.1. Laboratory brake test rig

The brake test rig driving unit used to rotate rotors of brake systems by using three phase AC electric motor with rated power 10 [HP] and output speed 1500 [rpm]. Rotational speed is transferred to rotors of brake units through a gear box with differential unit of a Hyundai Excel passenger car. This gear box and its differential unit have 5 reduction ratios with dual output shafts to erect two brake rotors units. In order to perform the experiments at various speeds A.C. frequency inverter 50 Hz is connected to the electric motor to control motor speed. The rotational speed of gearbox outlet shafts was measured by using a digital tachometer type (DT6234B) and reflective tape stamped on gearbox outlet shafts. The power consumption by electric motor was measured by using a digital power meter type Schneider PM 1200 as shown in Fig. 2. The applying normal force unit is used as master cylinder of actual brake system operated with screw with handle to adjust the required brake oil pressure and consequently the normal brake force applied to brake pad for braking process. The brake oil pressure values were measured by pressure gauge attached to master cylinder outlet. Directional control valve is used to control the brake oil for working brake unit as shown in Fig. 2.

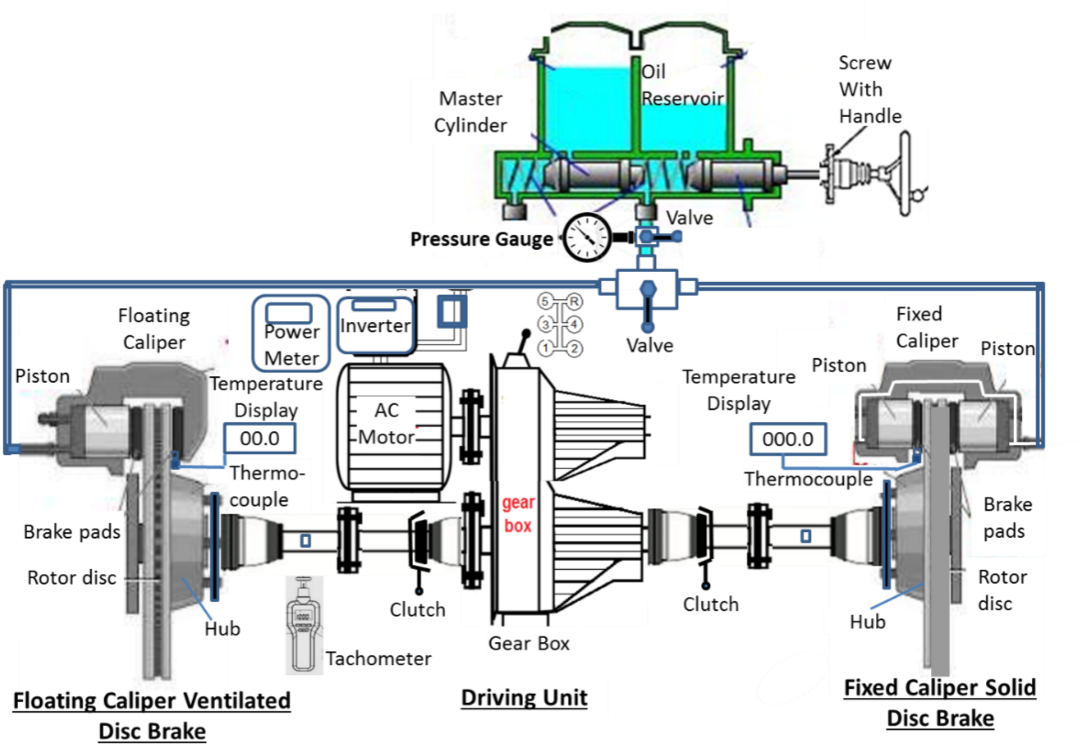

Fig. 2. Test rig components with measuring instrumentation

In this paper, two braking units of a commercial vehicle were coupled to the dual output shafts of gearbox. A floating caliper ventilated disc brake unit in left side and a fixed caliper solid disc brake unit in right side. Each brake unit is attached with a thermocouple of J-Type fixed with the brake pad and connected to temperature display to show the measured friction temperature. A mechanical clutches were used to engage and disengage motion transmission from gearbox output shaft to driven shafts connected to brake rotors units in left and/or right side.

III. BRAKE FORCE AND FRICTION COEFFICIENT CALCULATIONS

In disc brake system, brake force (F_{brake}) acting on pair of brake pads for fixed and floating calipers as shown in Fig. 3 was calculated as follow:

$$F_{brake} = T_{brake}/ 2R_m \quad [N] \qquad (1)$$

Where: R_m: Mean radius of the brake pad, $R_m = \frac{Ro+Ri}{2}$, mm.
R_o: Outside radius of the brake pad, mm.
R_i: Inside radius of the brake pad, mm.

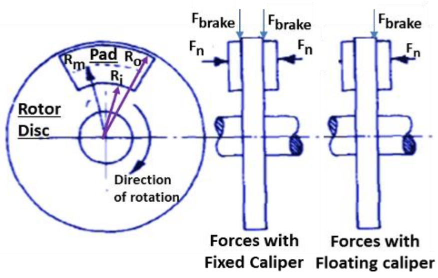

Fig. 3. Forces acting on rotor/pad

The brake torque (T_{brake}) was calculated as follow:

$$T_{brake} = P_{brake}/ \omega \quad [Nm] \qquad (2)$$

Where: ω: Disc angular velocity, $\omega = \frac{2\pi n}{60}$, rad/sec.
n: Disc rotational speed, rpm.

The consumed power of the electric motor was measured by power meter during operation at no load ($P_{no\ load}$) and at braking action (P_{Load}) to calculate the actual brake power (P_{brake}) as follow:

$$P_{brake} = P_{Load} - P_{no\ load} \quad [W] \qquad (3)$$

Where: P_{Load} : Consumed power at braking action, W.
$P_{no\ load}$: Consumed power at no load, W.

Then the friction coefficient (μ) between brake pads and disc was calculated based on the relation between braking force (F_{brake}) and normal force (F_n), as follow:

$$\mu = F_{brake} / F_n \qquad (4)$$

The value of the normal force (F_n) on brake pad is calculated from the value of applied brake oil pressure to the piston from the following equations:

$$F_n = P \times A \quad [N] \qquad (5)$$

Where: P : Hydraulic oil pressure, bar.

A : Piston area, $A = \frac{\pi}{4} D^2$, mm². (The area is duplicated for fixed caliper with two pistons)

D : Piston diameter, mm.

IV. DISC BRAKE TRIBOLOGICAL PROPERTIES

The tribological properties of the disc brake friction couple materials have a major influence on the brake system operation. Friction surface usually contains a series of micro peaks or valleys as shown Fig. 4. The micro peaks are named asperities. Friction force between the pad and disc rotor surfaces results from the meshing between micro peaks and valleys and this lead to deformation, and shearing the asperities [16]. The friction force depends on the size of actual contact area [17]. The number and size of the contact points will increase with increasing braking pressure. Under a low braking pressure, the friction films are hard to be formed on the interface surfaces. With increasing braking pressure, the asperities on the interface contact surfaces are deformed and broken forming some debris which staved easily forming some loose granular films increases the actual contact area. Thus, the friction resistance is increased, which results in an increment of friction coefficient [18]. When disc velocity increases, the asperities contact time is reduced and consequently the deformation time of asperities is reduced which results in reduction of real area of contact and this leads to decreases of friction coefficient [19]. Since the kinetic energy is transferred into heat energy during the braking process, the friction heat causes a temperature rise which will affect the friction. So, with increasing braking pressure and initial braking speed, the friction temperature will increase.

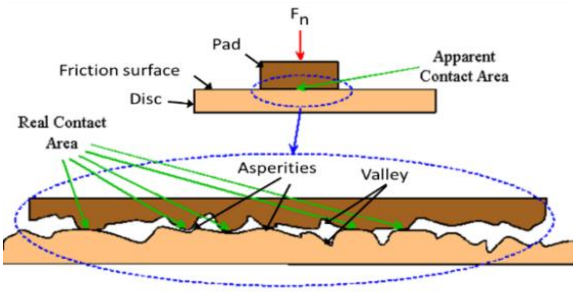

Fig. 4. Disc/Pad Friction Surface Tribology

V. EXPERIMENTAL WORKS

In this paper, experimental comparison works were done between a floating caliper ventilated disc brake unit (FL) of a commercial vehicle (Hyundai Excel) and a fixed caliper solid disc brake unit (FX) of a commercial vehicle (Mercedes Benz W124). The specifications of the main components of the floating caliper ventilated disc brake (FL) and fixed caliper solid disc brake (FX) units used in this paper is shown in Table (1)

Table (1) Specifications of the main components of the floating caliper ventilated disc brake and fixed caliper solid disc brake units

Component	Floating caliper ventilated disc brake unit (FL	Fixed caliper solid disc brake unit (FX)
Disc outer diameter[mm]	240	278
Disc inner diameter[mm]	120	168
Disc thickness[mm]	20	9.1
Friction material length[mm]	110	60
Friction material width[mm]	50	40
Friction material thickness[mm]	10	9
Pad back plate length[mm]	130	61.6
Pad back plate thickness[mm]	6	4
Hydraulic piston diameter[mm]	50	35

Experimental comparisons of three different profile designs of brake discs were carried out for both floating caliper ventilated disc brake (FL) and fixed caliper solid disc brake (FX) on same operating parameters. The three different profile designs of brake discs were as follow:

Normal disc brake profile (ND):

The normal disk brake profile (ND) of both floating caliper ventilated disc brake (FL-ND) and fixed caliper solid disc brake (FX-ND) units is a flat surface without any modifications (does not have any drills, notches or grooves at the disc) as shown in Fig.5, so, this design shape has more contact surface area touching the brake pads during braking and thus, have a better braking power.

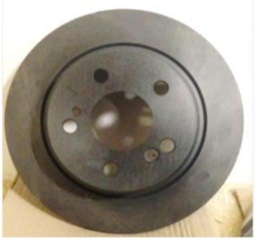

Fig. 5 Normal disc brake profile (ND)

Grooved disc brake profile (GD):

The grooved disc brake profile (GD) of both floating caliper ventilated disc brake (FL-GD) and fixed caliper solid disc brake (FX-GD) units had been modified from both sides to have 3 inclined curved grooves on disc surface. Each groove has a length of 45 mm, width of 3 mm and a depth of 3 mm as shown in Fig. 6.

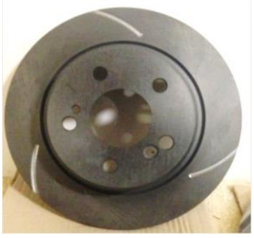

Fig. 6 Grooved disc brake profile (GD)

Drilled- grooved disc brake profile (DGD):

The drilled grooved disc brake profile (DGD) of both floating caliper ventilated disc brake (FL-DGD)and fixed caliper solid disc brake (FX-DGD) units had been modified from both sides to have 36 circular drilled holes with diameter of 6 mm and 3 inclined grooves on disc surface. Each groove has a length of 45 mm, width of 3 mm and a depth of 3 mm as shown in Fig. 7. Although circular drilled and inclined grooves reduces the surface contact area between the pads and rotors, they help the heat to dissipate as the overall surface of the rotor increases keeping the brake performance.

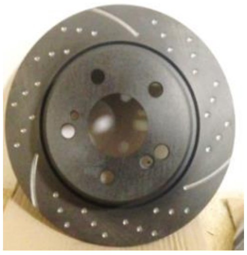

Fig. 7 Drilled grooved disc brake profile (DGD)

In this paper, experimental comparison was conducted by erecting a floating caliper ventilated disc brake unit (FL) of a commercial vehicle (Hyundai Excel) with selected disc profile and a fixed caliper solid disc brake unit (FX) of a commercial vehicle (Mercedes Benz W124) with selected disc profile as shown in Fig. 1. By using the clutch, the motion was transferred to the selected brake unit. The experiments were performed at brake oil pressure 2.5, 5, 7.5 and 10 [bar] and at four rotational disc speed 50, 100, 150 and 200 [rpm]. Before starting the tests, bedding in procedure was applied by initial running at moderate braking with approximated time of two hours with 10 start/stop to approximate the real contact area from the nominal contact area and to have stable coefficient of friction [20]. To keep same operating conditions, starting the run with initial brake pad temperature 24 C° and fixing braking duration 60 [sec]. By using the measured parameters from experimental works and mathematical calculations for brake force and coefficient of friction, comparisons of brake oil pressure and rotational speed on brake force, coefficient of friction and final friction temperature for both (FL) and (FX) units with different discs profiles were presented.

VI. EXPERIMENTAL RESULTS AND DISCUSSION

1. Comparison of (FL) and (FX) units based on varying brake oil pressure at constant disc speed

The comparison between (FL) and (FX) units with three different profile designs of brake discs was carried out by varying brake oil pressure at constant disc speed.

A. Influence of varying brake oil pressure on brake force at constant disc speed

The Brake force of both (FL) and (FX) units in time domain for brake oil pressure 2.5, 5, 7.5 and 10 [bar] at constant disc speed 150 [rpm] is presented in Fig. 8 for (FL-ND) and (FX-ND), in Fig. 9 for (FL-GD) and (FX-GD) and in Fig.10 for (FL-DGD) and (FX-DGD). It can be seen that the brake force is increased with increasing the brake oil pressure of (FL) and (FX) units for all profiles of (ND, GD and DGD) discs. Also as the brake oil pressure increased, a fluctuation in brake force trend is observed due to the variation of the friction coefficient with the braking time.

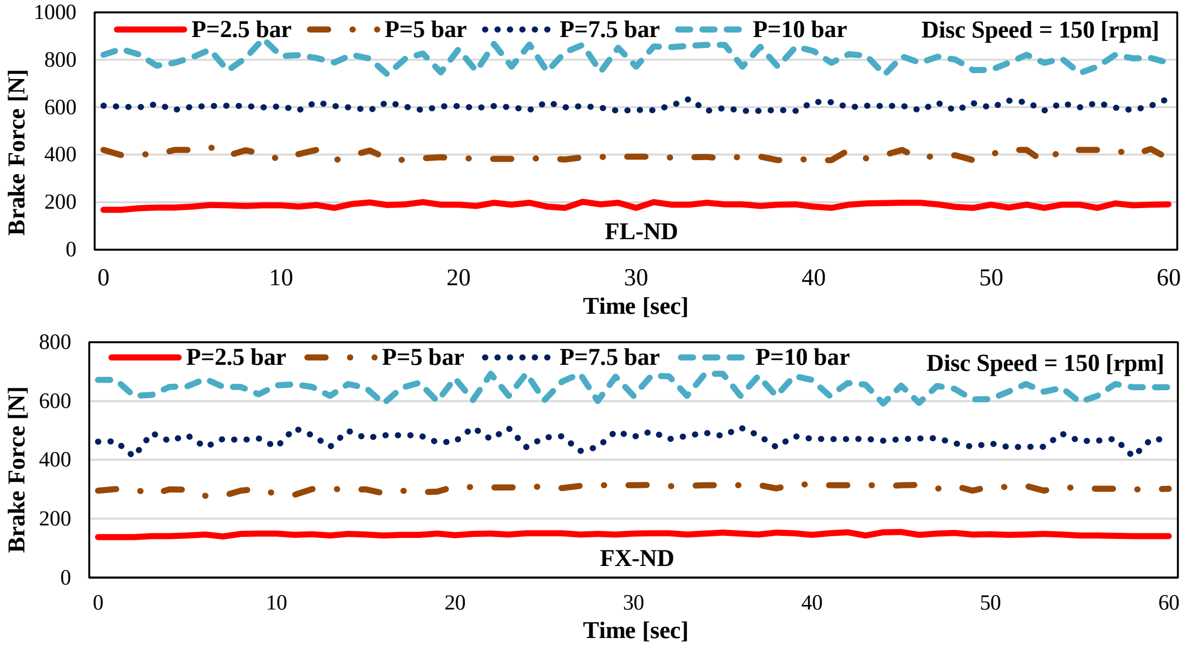

Fig. 8 (FL-ND) and (FX-ND) brake force vs. time for different brake oil pressure at disc speed 150 [rpm]

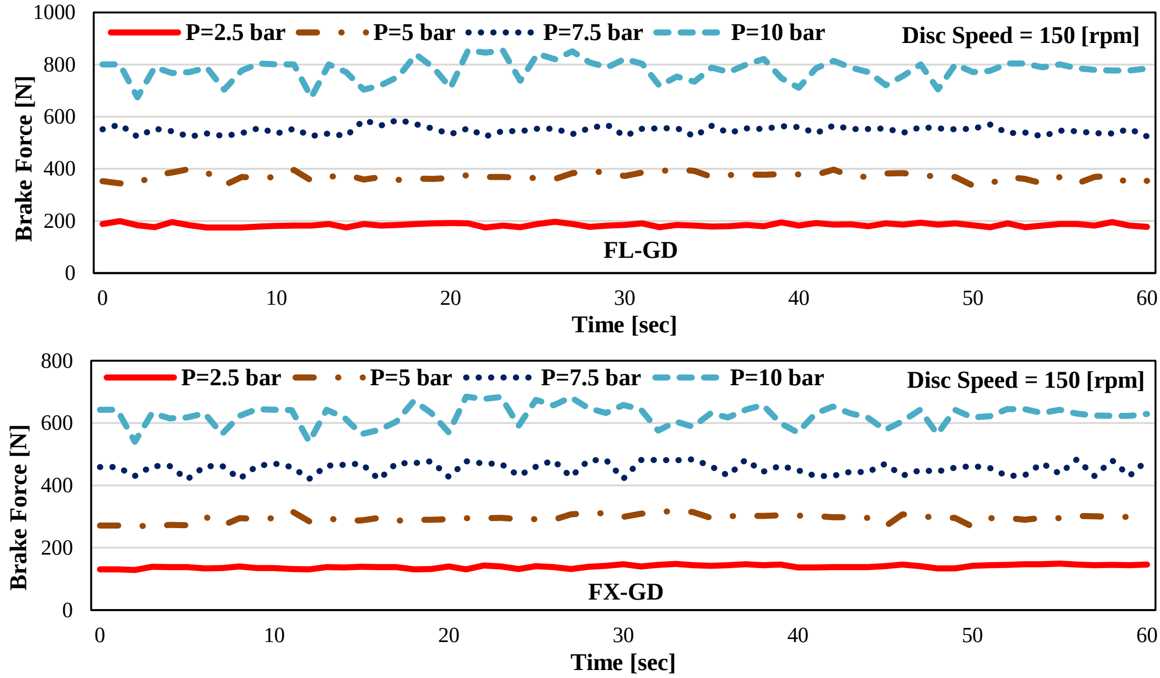

Fig. 9 (FL-GD) and (FX-GD) brake force vs. time for different brake oil pressure at disc speed 150 [rpm]

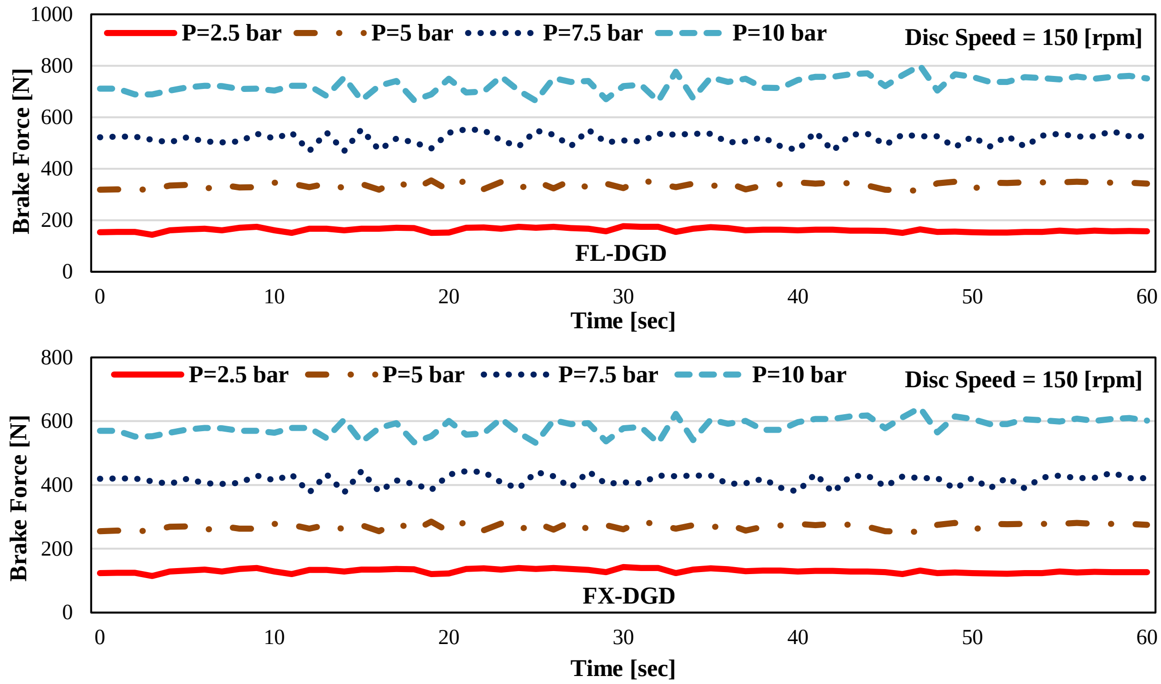

Fig.10 (FL-DGD) and (FX-DGD) brake force vs. time for different brake oil pressure at disc speed 150 [rpm]

The comparison of (FL) and (FX) mean brake force of all selected different designs of brake discs rotors is shown in Fig. 11. It is found that the mean brake force of all selected different designs of brake discs rotors are increased with increasing the brake oil pressure. The mean brake force of (FL) type is higher than (FX) type at the same operation condition.

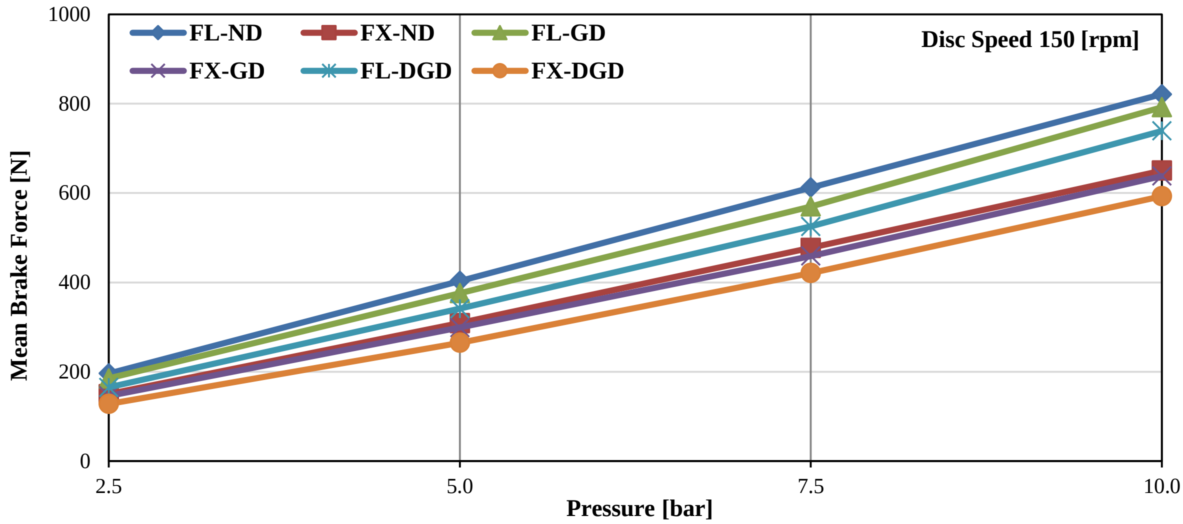

Fig. 11 (FL) and (FX) mean brake force vs. brake oil pressure for different discs profiles at disc speed 150 [rpm]

The percentage of improvement of mean brake force of (FL) with respect to (FX) for different disc brake rotors profiles is shown in Fig. 12. It is found that at constant disc speed 150 [rpm], increasing the brake oil pressure from 2.5 [bar] to 10 [bar], results in mean brake force improvement percentage of (FL) from 30.67% to 26.31% for (ND), from 26.71% to 24.14% for (GD) and from 28.91% to 24.62% for (DGD) respectively relative to corresponding profile in (FX). Also, the percentage of improvement at low brake oil pressure 2.5 [bar] is more than high brake oil pressure 10 [bar] and the percentage of improvement in (ND) is more than (GD) and (DGD) of (FL) with respect to (FX) and the percentage of improvement in (DGD) is more than (GD) of (FL) with respect to (FX).

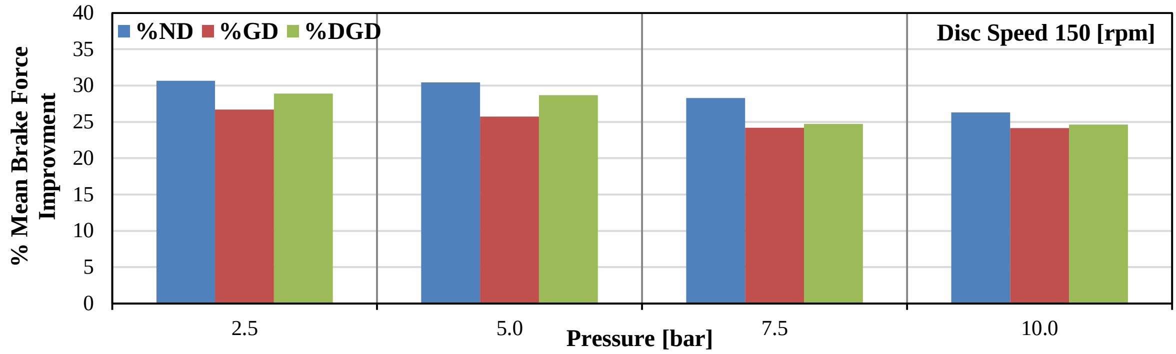

Fig. 12 Mean brake force improvement percentage of (FL)rotors profiles w.r.t. corresponding (FX) rotors profiles for different brake oil pressure at disc speed 150 [rpm]

B. Influence of brake oil pressure on friction coefficient at constant disc speed

The comparison of mean friction coefficients of (FL) and (FX) units for brake oil pressure 2.5, 5, 7.5 and 10 [bar] at constant disc speed 150 [rpm] is presented in Fig. 13-14. From Fig. 13 it is found that the mean friction coefficient of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are increased with increasing the brake oil pressure. The increase of the brake oil pressure from 2.5 to 10 [bar] causes an increase the mean friction coefficient from 0.399 to 0.418 for (FL-ND), from 0.377 to 0.403 for (FL-GD), from 0.336 to 0.376 for (FL-DGD), from 0.312 to 0.338 for (FX-ND), from 0.304 to 0.332 for (FX-GD) and from 0.266 to 0.308 for (FX-DGD). The results also indicated that the mean friction coefficient of (FL) types is higher than the (FX) types at the same operation condition

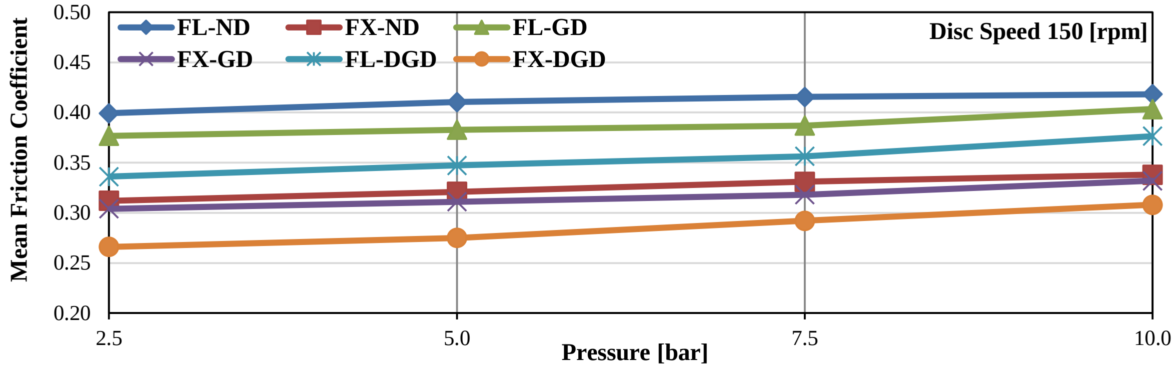

Fig. 13 (FL) and (FX) mean friction coefficients vs. brake oil pressures for different discs profiles at disc speed 150 [rpm]

Fig. 14 shows the percentage of improvement of mean friction coefficient of (FL) with respect to (FX) for different disc brake rotors profiles. It is found that at constant disc speed 150 [rpm], increasing the brake oil pressure from 2.5 [bar] to 10 [bar],results in mean friction coefficient improvement percentage of (FL) from 27.94% to 23.74% for (ND), from 23.94% to 21.53% for (GD) and from 26.33%, to 22.23% for (DGD) relative to corresponding profile in (FX). It is noticed that the percentage of improvement in low brake oil pressure 2.5 [bar] is more than high brake oil pressure 10 [bar]. Also, the percentage of improvement in (ND) is more than (GD) and (DGD) of (FL) with respect to (FX) and the percentage of improvement in (DGD) is more than (GD) of (FL) with respect to (FX).

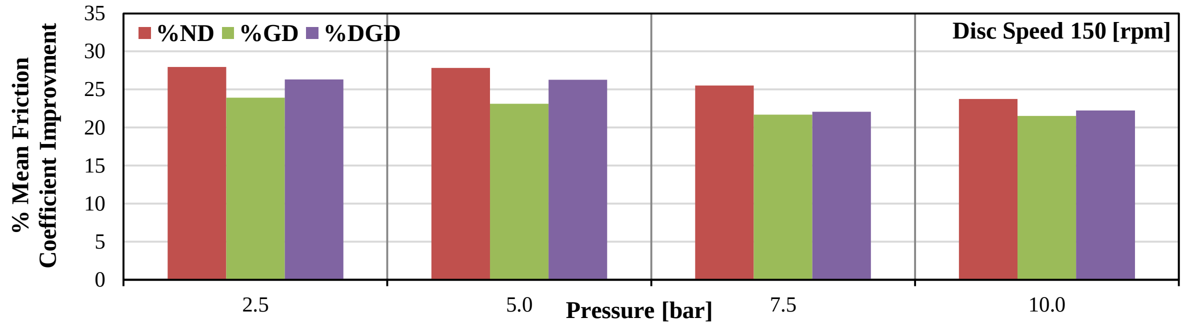

Fig. 14 Mean friction coefficients improvement percentage of (FL) rotors profiles w.r.t. corresponding (FX) rotors profiles for different brake oil pressure at disc speed 150 [rpm]

C. Influence of brake oil pressure on final friction temperature at constant disc speed

The comparison of mean final friction temperature of (FL) and (FX) units for brake oil pressure 2.5, 5, 7.5 and 10 [bar] at constant disc speed 150 [rpm] is presented in Fig. 15-16. From Fig. 15, it is found that the mean final friction temperature of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are increased with increasing the brake oil pressure. The mean final friction temperature of the fixed caliper type is higher than the floating caliper type at the same operation condition.

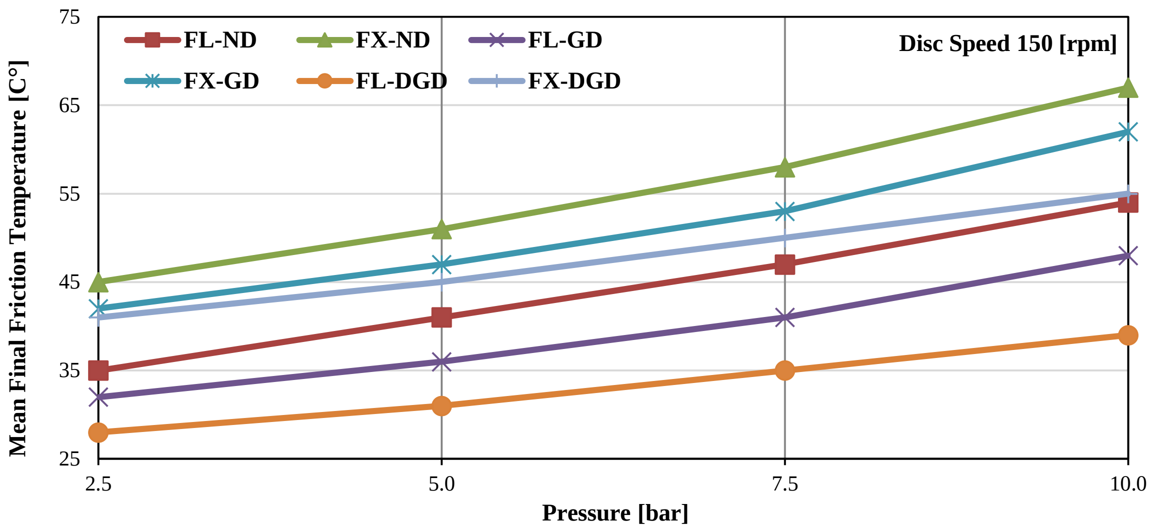

Fig. 15 (FL) and (FX) mean final friction temperature vs. brake oil pressure for different discs profiles at disc speed 150 [rpm]

Fig. 16 shows the percentage of reduction of mean final friction temperature of (FL) with respect to (FX) for different disc brake rotors profiles. The percentage of reduction of (FL) mean final friction temperature is reduced from 22.22% to 18.18% for (ND), from 23.81% to 20.00% for (GD) and from 31.71% to 29.09% for (DGD) relative to corresponding profile in (FX) for brake oil pressure from 2.5 [bar] to 10 [bar] at constant disc speed 150 [rpm]. Also, It is noticed from Fig. 16 that the percentage of reduction at low brake oil pressure 2.5 [bar] is slightly more than at high brake oil pressure 10 [bar] and the percentage of reduction in (DGD) is more than (GD) and (ND) of (FL) with respect to (FX) units and the percentage of reduction of (GD) is more than (ND) of (FL) with respect to (FX) units.

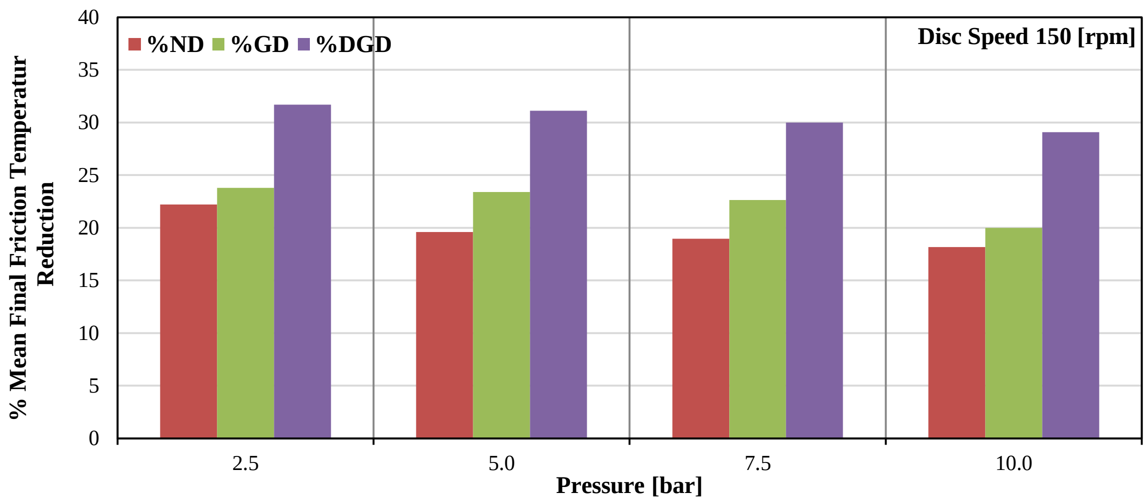

Fig. 16 Mean final friction temperature reduction percentage of (FL) rotors profiles w.r.t. corresponding (FX) rotors profiles for different brake oil pressure at disc speed 150 [rpm]

2. Comparison of (FL) and (FX) units based on varying rotating disc speed at constant brake oil pressure

The comparison between (FL) and (FX) units with three different profile designs of brake discs is carried out by varying disc speed at constant brake oil pressure.

A. Influence of rotating disc speed on brake force at constant brake oil pressure

The Comparison of (FL) and (FX) units mean brake force for disc speed 50, 100, 150 and 200 [rpm] and at brake oil pressure 10 [bar] is presented in Fig. 17-18. From Fig. 17, it is found that the mean brake force of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are decreased with increasing the rotating disc speed. The mean brake force of the floating caliper type is higher than the fixed caliper type at the same operation condition

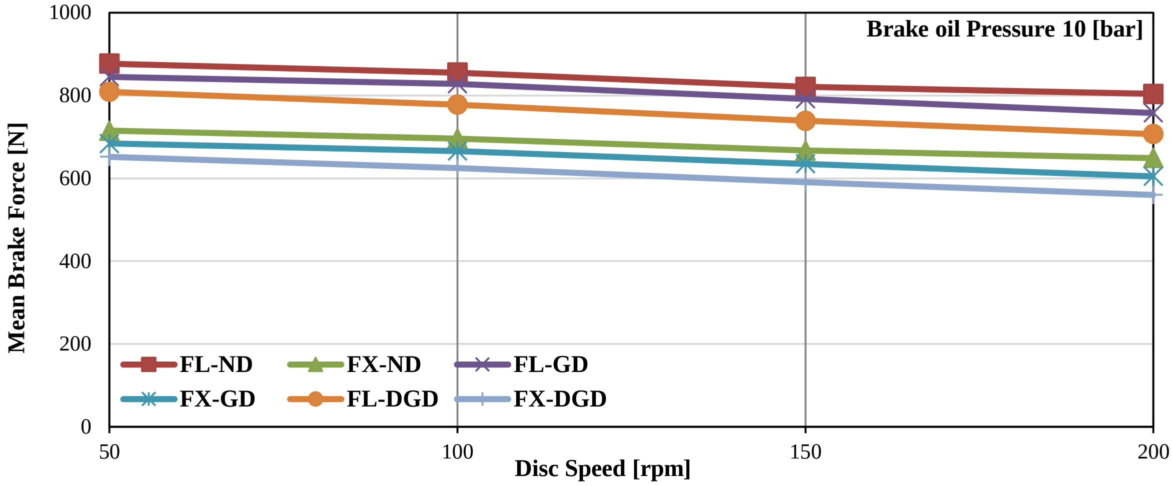

Fig. 17 (FL) and (FX) mean brake forces vs. rotating disc speed for different discs profiles at brake oil pressure 10 [bar]

Fig. 18 shows the percentage of improvement of mean brake force of (FL) with respect to (FX) for different disc brake rotors profiles. The percentage of improvement of (FL) mean brake force is increased from 22.66% to 23.88% for (ND), from 23.54% to 25.29% for (GD) and from 24.08%, to 26.25% for (DGD) relative to corresponding profile in (FX) for disc speed from 50 [rpm] to 200 [rpm] at constant brake oil pressure 10 [bar]. It is noticed that the percentage of improvement at high disc speed 200 [rpm] is slightly more than at low disc speed 50 [rpm]. Also, the percentage of improvement in (ND) is slightly less than (GD) and (DGD) of (FL) with respect to (FX).

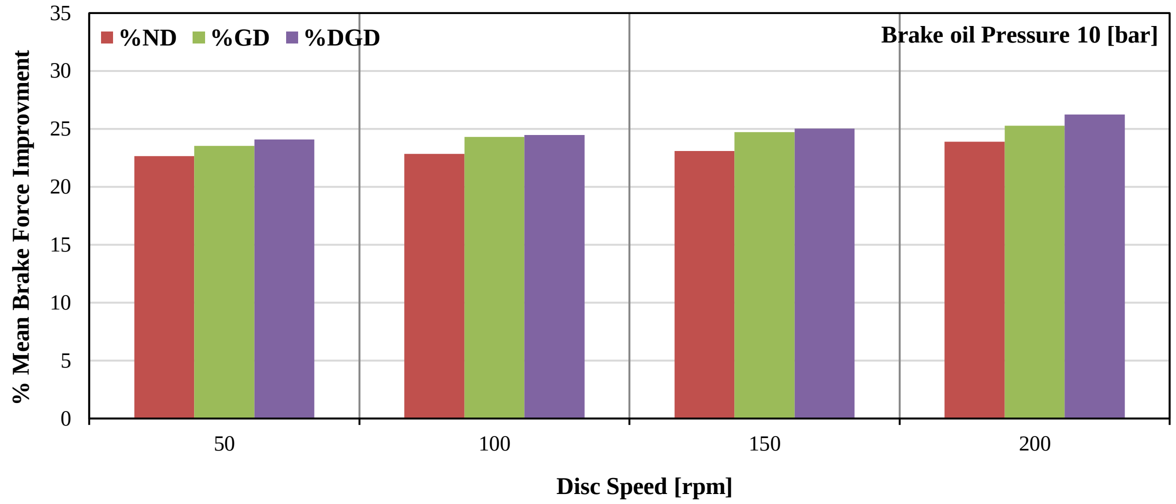

Fig. 18 Mean brake force improvement percentage of (FL) rotors profiles w.r.t. corresponding (FX) rotors profiles for different rotating disc speed at brake oil pressure 10 [bar]

B. Influence of rotating disc speed on friction coefficient at constant brake oil pressure

The comparison of mean friction coefficients of (FL) and (FX) units for disc speed 50, 100, 150 and 200 [rpm] and at oil pressure 10 [bar] is presented in Fig. 19-20. From Fig. 19, it is found that the mean friction coefficient of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are decreased with increasing the rotating disc speed. The increase of the rotating disc speed from 50 to 200 [rpm] causes a decrease in the mean friction coefficient from 0.447 to 0.410 for (FL-ND), from 0.431 to 0.386 for (FL-GD), from 0.412 to 0.360 for (FL-DGD), from 0.372 to 0.337 for (FX-ND), from 0.356 to 0.314 for (FX-GD) and from 0.339 to 0.291 for (FX-DGD). The results also indicated that the mean friction coefficient of the floating type is higher than the fixed type at the same operation condition.

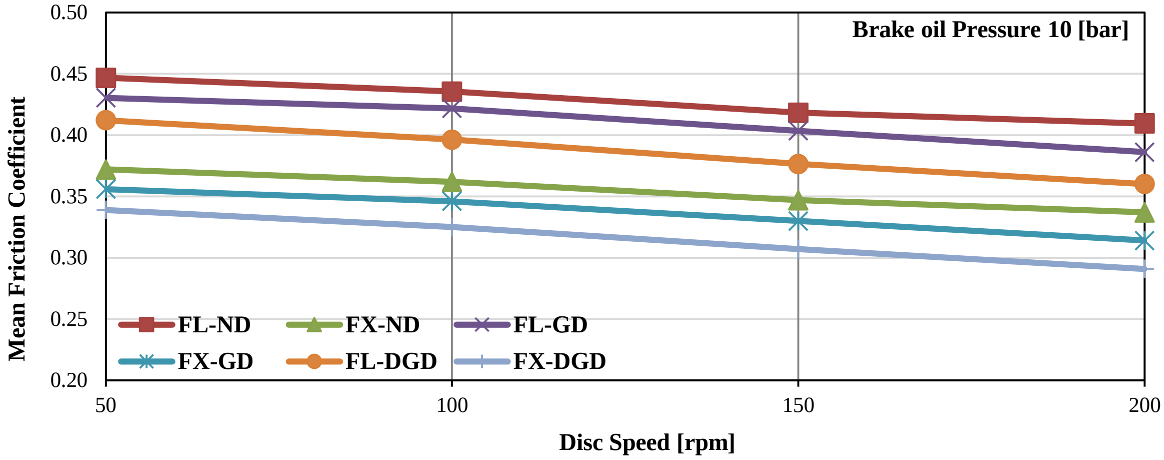

Fig. 19 (FL) and (FX) mean friction coefficients vs. disc speed for different discs profiles at brake oil pressure 10 [bar]

Fig. 20 shows the percentage of improvement of mean friction coefficient of (FL) with respect to (FX) for different disc brake rotors profiles. The percentage of improvement of (FL) mean friction coefficient is increased from 20.10% to 21.54% for (ND), from 20.92% to 22.98% for (GD) and from 21.57%, to 23.77% for (DGD) relative to corresponding profile in (FX) for disc speed from 50 [rpm] to 200 [rpm] at constant brake oil pressure 10 [bar]. It is noticed that the percentage of improvement of mean friction coefficient of (FL) with respect to (FX) for different disc brake rotors profiles approximately the same range from 20% to 23.7% for disc speed in the range from 50 [rpm] to 200 [rpm] for brake oil pressure 10 [bar].

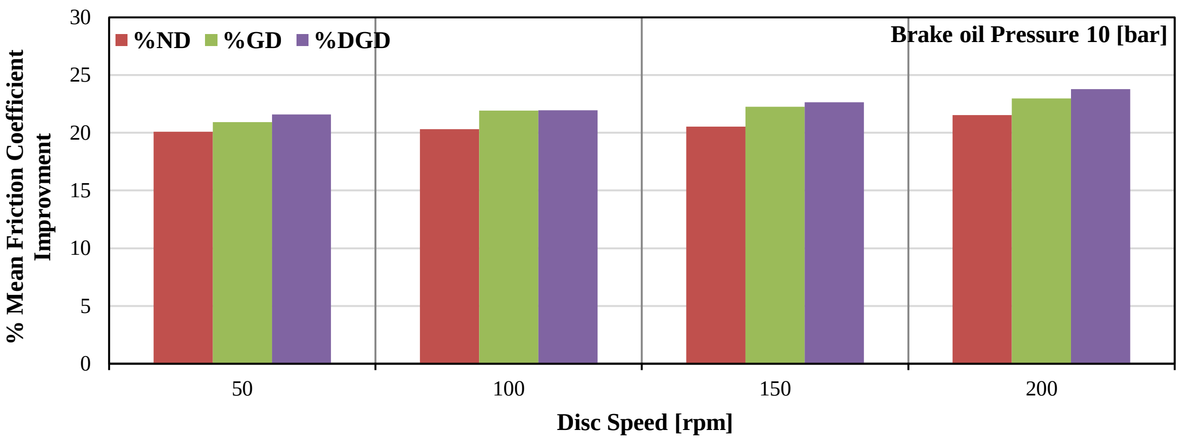

Fig. 20 Mean friction coefficients improvement percentage of (FL) rotors profiles w.r.t. corresponding (FX) rotors profiles for different rotating disc speed at brake oil pressure 10 [bar].

C. Influence of rotating disc speed on final friction temperature at constant brake oil pressure

The comparison of mean final friction temperature of (FL) and (FX) units for disc speed 50, 100, 150 and 200 [rpm] and at oil pressure 10 [bar] is presented in figures (21-22). From Fig. 21 it is found that the mean final friction temperature of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are increased with increasing the disc speed. The mean final friction temperature of the fixed caliper type is higher than the floating caliper type at the same operation condition

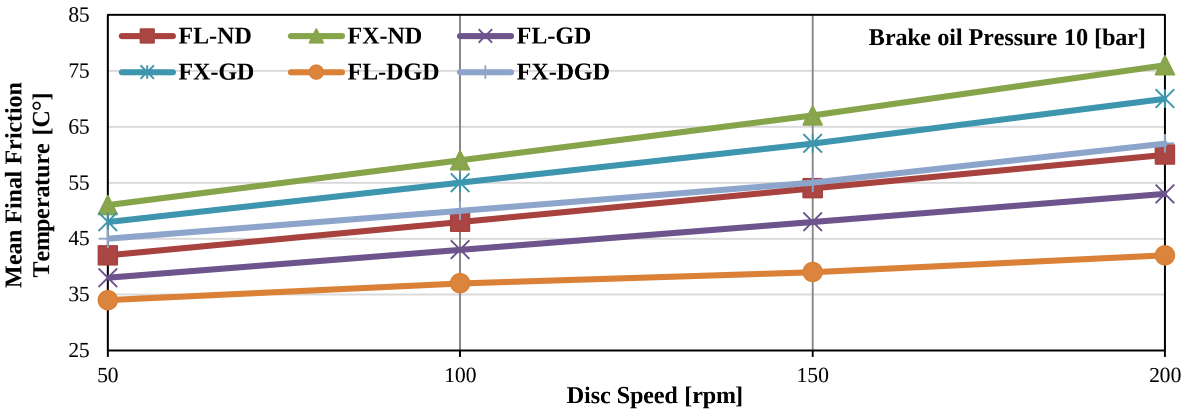

Fig. 21 (FL) and (FX) mean final friction temperature vs. different disc speed for different discs profiles at brake oil pressure 10 [bar].

Fig. 22 shows the percentage of reduction of mean final friction temperature of (FL) with respect to (FX) for different disc brake rotors profiles. The percentage of reduction of (FL) mean final friction temperature is reduced from 17.65% to 21.05% for (ND), from 20.83% to 24.29% for (GD) and from 24.44%, to 32.26% for (DGD) relative to corresponding profile in (FX) for disc speed from 50 [rpm] to 200 [rpm] respectively at constant brake oil pressure 10 [bar]. It is noticed that the percentage of reduction of mean final friction temperature at high disc speed 200 [rpm] is more than at low disc speed 50 [rpm]. Also, the percentage of reduction in (DGD) is more than (GD) and (ND) of (FL) with respect to (FX) and the percentage of reduction of (GD) is more than (ND) of (FL) with respect to (FX).

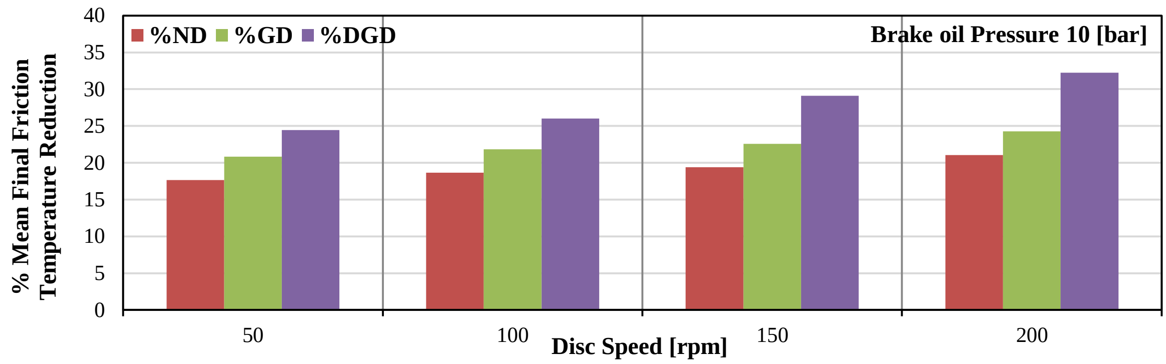

Fig. 22 Mean final friction temperature reduction percentage of (FL) rotors profiles w.r.t.corresponding (FX) rotors profiles for different disc speed at brake oil pressure 10 [bar].

VII. CONCLUSION

The conclusions from the previous comparisons presented in this study is summarized as follows:

1. The mean brake force of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are increased with increasing the brake oil pressure and decreased with increasing the rotating disc speed. The mean brake force of (FL) is higher than (FX) at the same operation condition.
2. The percentage of improvement of mean brake force at low brake oil pressure 2.5 [bar] is more than at high brake oil pressure 10 [bar] and the percentage of improvement in (ND) is more than (GD) and (DGD) of (FL) with respect to (FX) and the percentage of improvement in (DGD) is more than (GD) of (FL) with respect to (FX).
3. The better percentage of improvement of mean brake force for (FL-ND) relative to (FX-ND) is 30.67% at low brake oil pressure 2.5 [bar] and constant disc speed 150 [rpm].
4. The mean friction coefficient of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are increased with increasing the brake oil pressure and are decreased with increasing rotating disc speed. The mean friction coefficient of (FL) is higher than (FX) at the same operation condition.

5. The percentage of improvement of mean friction coefficient at low brake oil pressure 2.5 [bar] is more than at high brake oil pressure 10 [bar] and the percentage of improvement in (ND) is more than (GD) and (DGD) of (FL) with respect to (FX) and the percentage of improvement in (DGD) is more than (GD) of (FL) with respect to (FX).
6. The better increase of mean friction coefficient of (FL-ND) from 0.399 to 0.418 for the brake oil pressure from 2.5 to 10 [bar] at constant disc speed 150 [rpm] and from 0.447 to 0.410 for rotation disc speed from 50 to 200 [rpm] at constant brake oil pressure 10 [bar].
7. The better percentage of improvement of mean friction coefficient for (FL- ND) relative to (FX- ND) is 27.94% at low brake oil pressure 2.5 [bar] and constant disc speed 150 [rpm].
8. The mean final friction temperature of all selected different designs of brake discs rotors with floating or fixed calipers disc brakes are increased with increasing the brake oil pressure and the disc speed. The mean final friction temperature of (FX) is higher than (FL) at the same operation condition.
9. The percentage of reduction of mean final friction temperature at high brake oil pressure 10 [bar] is more than at low brake oil pressure 2.5 [bar] at constant disc speed 150 [rpm] and at high disc speed 200 [rpm] is more than low disc speed 50 [rpm] at brake oil pressure 10 [bar] also, the percentage of reduction in (DGD) is more than (GD) and (ND) of (FL) with respect to (FX) and the percentage of reduction of (GD) is more than (ND) of (FL) with respect to (FX).
10. The better percentage of reduction of mean final friction temperature for (FL- DGD) relative to (FX- ND) is 31.71% at low brake oil pressure 2.5 [bar] and constant disc speed 150 [rpm] and is 32.26% for disc speed 200 [rpm] at constant brake oil pressure 10 [bar]

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