International Journal of Engineering Inventions
e-ISSN: 2278-7461, p-ISSN: 2319-6491
Volume 7, Issue 10 [October 2018] PP: 46-61

Effect of the sliding speed on the performance of conventional and modified disc brake at different initial operating temperatures

Ibrahim Ahmed[1], Khaled Abdelwahed[1], Yasser Fatouh[1] and Mostafa M. Makrahy[2]

1. *Automotive Technology Department, Faculty of Industrial Education, Helwan University, Cairo, Egypt.*
2. *Automotive Engineering Department, Faculty of Engineering, Minia University, Minia, Egypt.*

Corresponding Author: Ibrahim Ahmed

ABSTRACT: *This paper describes the drag-type brake dynamometer (test rig) which has been designed to study the effect of sliding speed on the performance of the conventional and modified disc brake at different initial operating temperatures. Several measuring instruments are used with this test rig, which are necessary for performing the experimental tests. A modification of the conventional disc brake has been done and a comparison between the performance of both conventional and modified disc brake then has been performed. After that, some experimental tests are conducted on the conventional and modified disc brake at different sliding speeds at constant brake oil pressure and at different initial operating temperatures. Finally, comparison between conventional and modified disc brake are performed. Experimental results showed that the sliding speed of the rotating disc has negative effect on the performance of the conventional and modified system. Also the test results indicated that the modified disc brake can amplify the mean brake force and mean friction coefficient of the conventional disc brake at the same conditions.*

Date of Submission: 24-01-2019 Date of acceptance: 08-02-2019

I. INTRODUCTION

One of the most important components in road vehicle is the braking system. Vehicle brake is classified in general as a mechanical device for generating the frictional resistance that retards or stops the motion of the vehicle. So, the brake system converts the kinetic energy of the vehicle into thermal energy by the process of friction, and this heat must be efficiently dissipating to the ambient air by the components of the brake system. There are two main types of vehicle brakes namely, drum and disc brakes. The main advantages of the drum brakes is that can apply more stopping power for a given amount of force applied to the brake pedal than disc brake. This is possible because the drum brake design offers a self- energizing action [1]. The disadvantages of the disc brakes are no self- energizing or servo action, brake noise, and poor parking brake performance [2]. Hence, there were many efforts to modify the disc brakes that could be self- amplified. Recently there were many efforts to develop the conventional disc brake with self-amplification phenomenon. The self-amplification action can be acquired for the conventional disc brake by two mechanisms. The first mechanism, the brake system has additional element, namely intensifying plugs which are connected to the brake pads and supported on the caliper. Due to the friction force, the brake pads move in the direction of brake disc rotation. The intensifying plugs prevent the brake pads movement generating an additional applied force; as a result the brake force increases [3]. The second mechanism, the actuator presses the brake lining in between the abutment and the brake disc with the motor force. An auxiliary force derived from the self-energizing effect is generated to build up the normal force. The braking force resulting from the contact between the brake disc and the brake lining acts in the same direction as the motor force. The self-energizing disc brake called wedge disc brake [4-5]. During the repetitive brake process in the vehicles, the temperature of the rotating disc can reach to (200-250℃) and 300℃ for the brake pad [6]. The high temperatures of the rotating disc and brake pads causing brake fad. Fade generally is a term used to indicate a loss in brake force at high temperatures, because of reduction in coefficient of friction [7]. The brake force and friction coefficient play an important role in brake system performance, and both of them are affected by different conditions such as the surface finishing, the type of friction material, temperature of the brake system components, rotational speed of the rotating disc, water and brake oil pressure. This leads to observation that the brake force and the coefficient of friction fluctuate with the braking time [8]. According to Severin and Dörsch [9], the increase of the number of brakes leads to increase the friction temperature as well as the friction coefficient decreases. Gao and Lin [10] stated that there was considerable evidence to show that the contact temperature is an integral factor reflecting the specific power friction influence of combined effect of normal force, sliding speed, the coefficient of friction, and thermo physical and durability properties of the materials of a frictional couple. The experiments showed that the coefficient of friction in general decreased with increasing of sliding speed, the normal force and disc temperature. Another experimental study was presented by Österle and Urban [11] and illustrated the variations of the friction coefficient against the time at continuous braking. The initial value of the friction coefficient is low approximately 0.13, but it increases rapidly up to 0.3 at 50 sec of braking time, however; beyond 200 sec a slight fading effect is evident. Mikael Eriksson [12] studied the effect of different friction-velocity behaviours. Number of different brake pads was evaluated with respect to friction behaviour with changing speed. The results showed that there is negative friction-velocity behaviour. Most pads showed a slightly higher coefficient of friction at low sliding speeds. It was also found that at braking with constant speed the effect slightly reduced. Blau [13] investigated experimentally the effect of the sliding speed on the friction coefficient of different materials. It was found that, the tendency of the friction coefficient to decrease as the sliding speed of the disc during braking increases. He explained the reason of decreasing the coefficient of friction with respect to the sliding speed of the disc, this reason is as a result of the effect of lubricating oxide form at the elevated temperatures, the shear strength of most materials decreases at high frictional temperatures, and if the surface frictionally melts, the molten liquid can lubricate the asperity contact, so the coefficient of friction decreases. Aviles et al. [14] have investigated experimentally the relation between the coefficient of friction and the sliding speed. They have used five different brakes. The coefficient of friction has no exact trend with sliding speed. Each brake has a trend that differs from the other, however, the coefficient of friction decreased with the sliding speed in all the brakes that were investigated in their study. The researches which use the experimental work are using the brake dynamometer in order to predict the performance of the brake system to investigate the effect of different parameters and operating conditions, to understand the characteristics of the brake system during the operation to confirm possible solutions that can improve the performance of the brake system. The brake dynamometer designs are classified into two types. The first type is the drag-type dynamometer. This type is used to examine the brake performance at constant speed [15-16]. The second type is an inertia-type brake dynamometer that has fly wheel attached to it and can be used to examine the brake performance at negative velocity slope [17-18]. In this paper, the effect of sliding speed at constant brake oil pressure and at different initial operating temperatures on the brake performance of the conventional and modified disc brake is experimentally investigated by using drag-type brake dynamometer.

II. TEST RIG DESCRIPTION

The brake test rig has two main objectives. The first objective is the ability to measure the generated brake power of the conventional and modified disc brake at all operating parameters. The second objective of the test rig is to generate the required kinetic energy that could be overcome by the braking system. The test rig is designed and constructed to achieve these requirements. Figure 1 shows the main components of the test rig which are: conventional and modified disc brake system assembly, components of generation the kinetic energy and the components of generation the normal force.

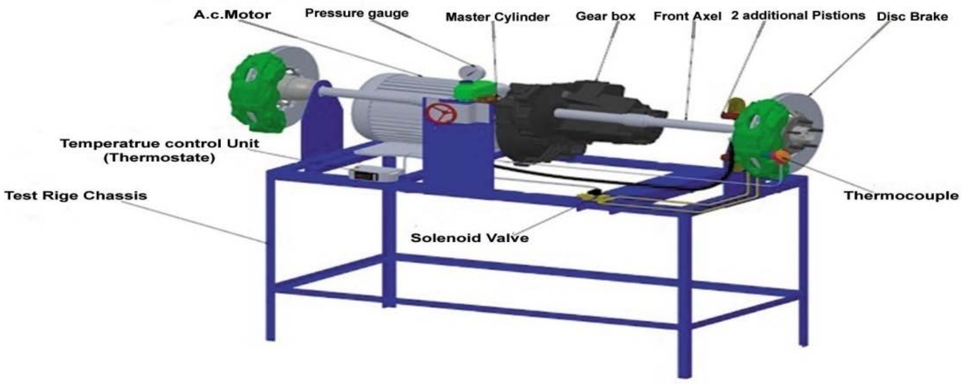

Figure (1) Main components of the test rig.

2.1 Conventional disc brake assembly

A disc brake of Hyundai Excel passenger car is used in the test rig. This braking system is a floating caliper disc brake. The main components of this system are shown in Figure 2. It consists of floating caliper with its slave cylinder which contains a hydraulic piston of diameter 5.3 cm, rotor disc, two brake pads, wheel bearing, finger and hub. The hydraulic pipe is connected between the master cylinder and the hydraulic piston.

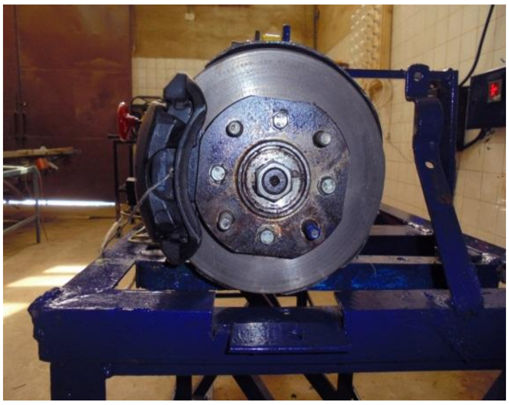

Figure (2) Conventional disc brake of Hyundai Excel.

2.2 Modified disc brake assembly

The conventional floating type disc brake system is modified in order to improve its performance at high temperatures or at any temperature. The idea of the modified disc brake depends on increasing the normal force which affects the brake pad. The increasing of the normal force which affects the brake pad is achieved by modifying the conventional disc brake be adding the following components: two additional hydraulic pistons of diameter 1.8 cm, temperature sensor (thermocouple), temperature control unit (thermostat) and solenoid valve. Figure 3 presents the modified disc brake of the used vehicle. It illustrates that the two additional hydraulic pistons squeeze the brake pad from its ends with the main hydraulic piston of the slave cylinder. So in modified disc brake the effect of the normal force affects the brake pad is by the main hydraulic piston of the slave cylinder and by the two additional pistons.

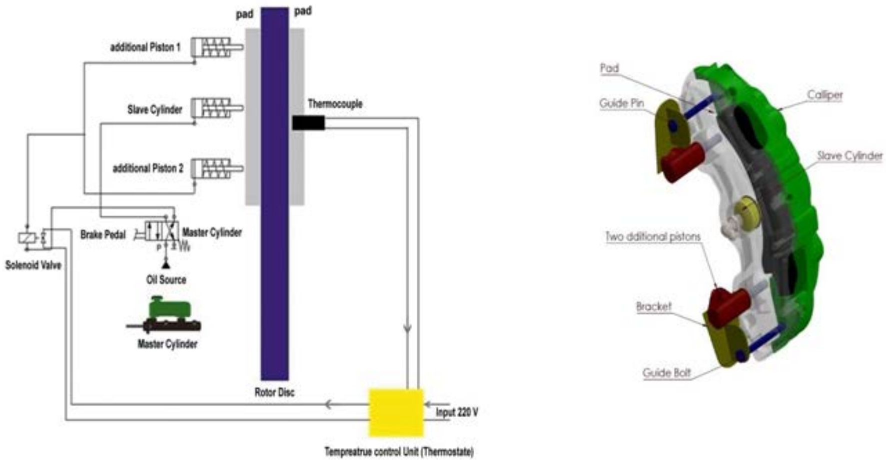

Figure (3) Modified disc brake.

Figure 4 shows the assembly of one of the two additional pistons with the caliper of the conventional disc brake by using steel bracket. One of the two brackets is connected to both of the first additional piston via two steel screws M10 and the caliper via the guide pin of the caliper. The other brackets also are connected to both of the second additional piston via two steel screws M10 and the caliper via the guide bolt of the caliper.

2.3 kinetic energy generation

An A.C electric motor is used in the test rig as shown in Figure 1. The electric motor is three phase type which has maximum power 10 Hp at 1500 r.p.m. In order to do the experiments at various speeds, a gear box with differential unit of a Hyundai Excel passenger car is installed between the electric motor and the brake system. This gear box and its differential unit have reduction ratios of 6.5, 3.9, 2.6, 1.9, 1.5 and a reverse reduction ratio of 6.8.

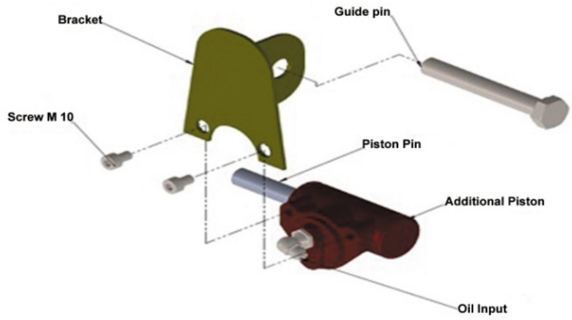

Figure (4) Assembly of an additional piston with the caliper.

2.4 Normal force generation

The braking force is depending on two main parameters. The first parameter is the normal force affecting the brake pad. The second parameter is the coefficient of friction between the brake pads and the rotor disc. So, the normal force is considering the main factor of generating the brake force. Hence its effect on the braking process has to be taken into consideration. The generated normal force must have constant values during the tests according to the operating conditions. A master cylinder of commercial passenger car model Hyundai Excel is used to generate the constant normal force. The master cylinder has two outlet of brake lines, each line is for two wheel slave cylinder. The hydraulic lines of the master cylinder are modified as shown in Figure 5. The front hydraulic line allows the hydraulic oil to pass directly to the hydraulic piston of the slave cylinder. The rear hydraulic line allows the hydraulic oil to pass directly to the solenoid valve which controls of passing the pressurized oil to the two additional pistons. The brake pedal and the power booster are replaced by a screw push link with a circular handle to give the required forces. The purpose of this mechanism is to supply oil pressure that squeezes the hydraulic piston of the slave cylinder and the two additional pistons to generate the normal force that affects the brake pads.

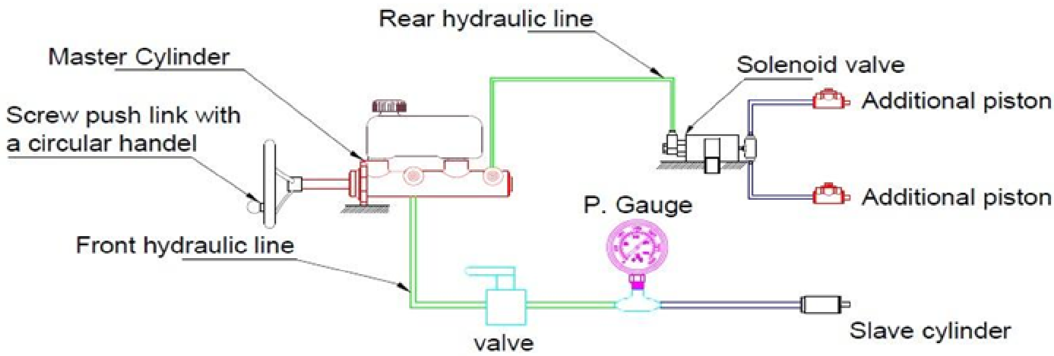

Figure (5) Normal force generation assembly.

III. MEASUREMENT INSTRUMENTATION

The measurement instrumentations systems which are used can be divided as following: Pressure measurement and normal force calculation, Speed measurement, Brake torque, brake force and coefficient of friction calculation and temperature measurement, as shown schematically in Figure 6.

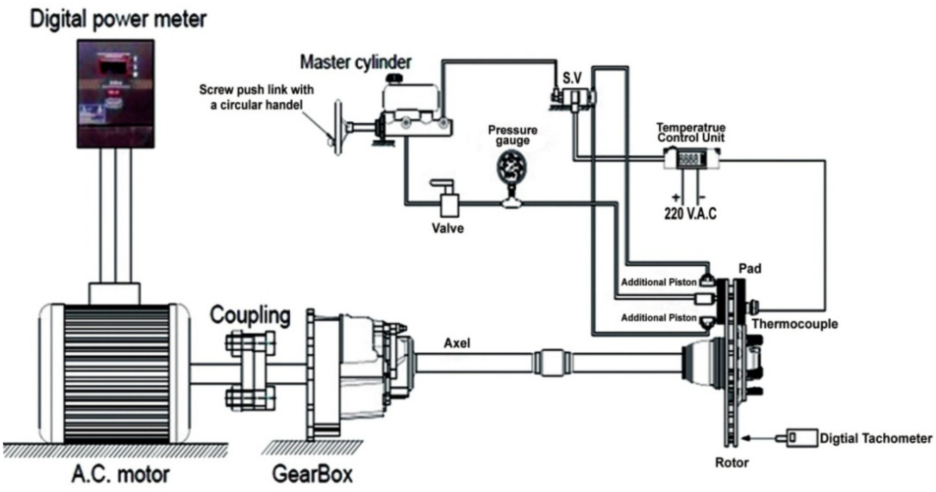

Figure (6) Schematic sketch of the test rig and measurement instrumentation.

3.1 Pressure measurement and normal force calculation

The value of the oil pressure in the brake system is measured by using an oil pressure gauge as shown in Figure 6. The pressure gauge is mounted in the hydraulic line between the master cylinder and the slave cylinder of the brake system. The normal force of the conventional system is calculated as the multiplication of the piston area of the slave cylinder and the magnitude of the oil pressure. Different values of the normal forces of the conventional system are determined according to the values of the oil pressure as shown in the equations below:

$$A_s = \frac{\pi}{4} D_s^2 \quad (1)$$

$$P = \frac{F_{n.c}}{A_s} \quad (2)$$

$$F_{n.c} = P * A_s \quad (3)$$

Where:

D_s The piston diameter of the slave cylinder equals 0.053 m.)

A_s The piston area of the slave cylinder equals 2.2×10^{-3} m^2.

$F_{n.c}$ The normal force of the conventional system which affects the brake pad .

Four oil pressure values of 2.5, 5, 7.5 and 10 bar are selected during theses tests. According to equation (3) these values of pressure equal normal forces of 550, 1100, 1650 and 2200 N respectively for the conventional system. To insure that the normal force is constant during the tests, a control valve was used to achieve this aim. The valve was mounted into the hydraulic line between the master cylinder and the slave cylinder. This valve is opened to identify the required pressure and it is closed during the test to insure that the pressure is constant as well as constant normal force. The normal force of the modified system is calculated as shown in the equations below:

$$F_{n.m} = p \times (A_s + 2A_d) \quad (4)$$

Where:

$F_{n.m}$ The normal force of the modified system that affects the brake pad.

A_d The cross section area of one of the two additional pistons equals 2.5×10^{-4} $m^2 = \frac{\pi}{4} d^2$ (5)

d The diameter length of the additional piston, equals 1.8 cm.

According to equation (4) the values of the oil pressure of 2.5, 5, 7.5 and 10 bar produce normal force of values 675, 1350, 2025 and 2700 N respectively for the modified system.

3.2 Brake torque calculation and speed measurement

In this work the brake power is measured by using digital power meter as shown in Figure 6. The type of the digital power meter is Schneider PM 1200 which has range from 20 watt to 300 kw and has accuracy of 1% for reading of power and gives 60 readings per minute. The power meter measured the power of the electric motor during the braking process as the normal force affected the brake pad. It also measured the no load power as there was no normal force affected the brake pad. The no load power is the power which consumed from the electric motor to overcome the inertia forces of the test rig elements. In this study the brake power was determined as follow:

$$P_b = P_L - P_{no} \tag{6}$$

Where :

P_b The brake power (watt)
P_L The electric motor power during the braking process (watt)
P_{no} The electric motor power during the operation at no braking load (watt)

The rotational speed of the rotor disc (sliding speed) is also a very significant parameter in the braking process. The sliding speed of the braking system was measured by a digital tachometer which its type is (DT2234) and it has range from 5 to 100000 r.p.m with accuracy of 0.5 %. The first aim of measuring the sliding speed of the braking system was to calculate the angular speed of the rotating disc which was used with brake power to calculate the brake torque. The second aim was to know the behavior of the brake system with different sliding speeds.

By calculating the brake power of the braking system during the braking process as mentioned in equation (6) and the angular speed of the rotating disc, the brake torque was calculated as follow:

$$T_b = \frac{P_b}{\omega} \tag{7}$$

$$\omega = \frac{2\pi n}{60} \tag{8}$$

Where :

T_b The braking torque (N.m)
ω The angular speed of the rotating disc (rad/sec.)
n The sliding speed of the rotating disc (r.p.m)

Four sliding speeds of the rotating disc are selected during theses tests. These values are 50, 100, 150 and 200 r.p.m respectively.

3.3 Brake force and friction coefficient calculations

The brake force and friction coefficient are most important parameters indicate the performance of both conventional and modified disc brake at high temperatures in this work. For conventional and modified disc brake, by calculating the brake torque as mentioned in equation (7) the braking force of the conventional and modified system can be calculated as follow:

$$T_b = F_b \, . \, r_{eff} \tag{9}$$

For a disc brake system there is a pair of brake pads, thus the total brake torque is:

$$T_b = 2\, F_b \, r_{eff} \tag{10}$$

$$r_{eff} = \frac{r_o + r_i}{2} \tag{11}$$

Where:

F_b The brake force generated at the contact interface (N)
r_{eff} The effective radius of the brake pad, equals 0.089 m
r_o The outer radius of the brake pad (m)
r_i The inner radius of the brake pad(m)

From equation (10) the brake force of the conventional and modified system can be calculated as follow:

$$F_{b.c} = \frac{T_{b.c}}{2\, r_{eff}} \quad (12)$$

$$F_{b.m} = \frac{T_{b.m}}{2\, r_{eff}} \quad (13)$$

Where :

$F_{b.c}$ The brake force of the conventional system (N)
$F_{b.m}$ The brake force of the modified system (N)
$T_{b.c}$ The brake torque of the conventional system (N.m)
$T_{b.m}$ The brake torque of the modified system (N.m)

However the braking force is dependent upon the normal force and the friction coefficient, which is derived as below:

$$F_b = \mu\, F_n \quad (14)$$

The generated normal force of the conventional and modified system were determined based on the brake-line pressure (P) as mentioned in equations (3) and (4), then by substituting equations (3) and (4) into equation (14), the coefficient of friction of the conventional and modified system can be calculated as follow:

$$F_{b.c} = \mu_c\, P\, A_s \quad (15)$$

$$F_{b.m} = \mu_m\, P\, (A_s + 2A_d) \quad (16)$$

$$\mu_c = \frac{F_{b.c}}{P\, A_s} \quad (17)$$

$$\mu_m = \frac{F_{b.m}}{P(A_s + 2A_d)} \quad (18)$$

Where:

μ_c The friction coefficient of the conventional system.
μ_m The friction coefficient of the modified system.

3.4 Temperature measurement

The effect of the initial operating temperature is considered during this work to investigate its effect on the performance of the conventional and modified system. A thermocouple of J-type was selected and is fixed in the brake pad to measure the friction temperature at the contact area between the brake disc and the brake pad. The output signal of the thermocouple was sent to the temperature control unit (thermostat). The temperature control unit is adjusted at a certain temperature. As the brake pad temperature reaches to the adjusted temperature of the control unit, in this case the temperature control unit sends an electrical signal to the coil of the solenoid valve. So the pressurized brake oil passes through the solenoid valve to the two additional pistons. Four initial operating temperatures are selected during the tests. These values are 50, 100,150 and 200°C. Figure 7 shows the temperature measurement instrumentation.

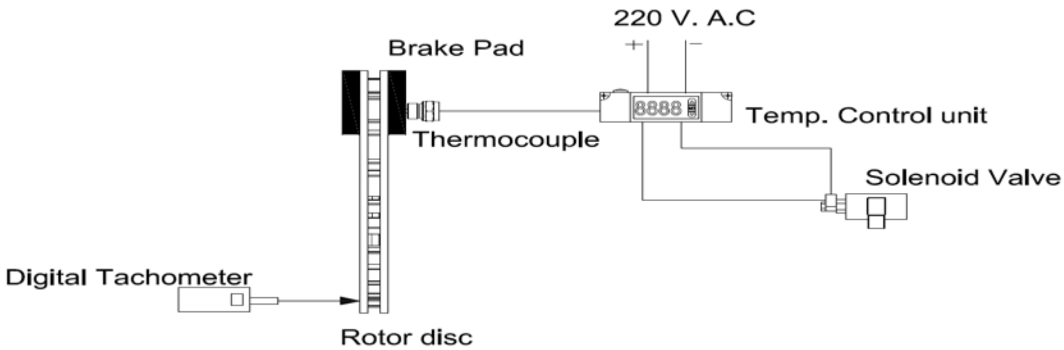

Figure (7) Temperature measurement instrumentation.

IV. RESULTS AND DISCUSSION

The experimental work is carried out to investigate the effect of sliding speed at constant brake oil pressure and at different initial operating temperatures on the brake force and friction coefficient of the conventional and modified disc brake. All experimental tests are conducted in the same conditions 60 seconds of braking. The brake power was measured every second by the digital power meter. The sliding speed, the brake oil pressure and the initial operating temperature were measured during each test for the conventional and modified disc brake. The brake force and friction coefficient of the conventional and modified disc brake were calculated every second and plotted with the brake time during each test.

4.1 Effect of sliding speed at brake oil pressure 10 bar and initial temperature 50 ℃

The effect of sliding speed of the rotating disc on the brake forces of both conventional and modified system at pressure 10 bar and initial temperature 50 ℃ is presented in Figure 8 and Figure 9. The results showed that, the brake forces of the conventional and modified system fluctuate with no identical trend at each constant sliding speed with the braking time. The fluctuation of the brake force is due to the variation of the friction coefficient with the braking time. Also it can be seen that, the brake forces of both conventional and modified system are decreased with increasing the sliding speed. The results presented in Figure 10 show the variation of the mean brake force of the conventional system and modified system at different sliding speeds. From the results, it can be seen that the increase of the sliding speed of the rotating disc cause a decrease of the mean brake force of the conventional and modified system. The mean brake forces of the conventional system are 850, 830, 811 and 793 N, and the mean brake forces of the modified system are 1046, 1025, 1004 and 987 N at sliding speeds 50, 100, 150 and 200 r.p.m respectively. The increase of the sliding speed from 50 r.p.m to 200 r.p.m causes a decrease on the mean brake force of the conventional system from 850 N to 793 N which is nearly 7%. Also the increase of the sliding speed from 50 r.p.m to 200 r.p.m causes a decrease on the mean brake force of the modified system from 1046 N to 987 N which is nearly 6%. Also Figure 10 shows that, at each constant speed, the mean brake force of the modified system increases approximately by 23%, 23%, 23% and 24% respectively above the mean brake force of the conventional system, this is because the normal force affecting the brake pad of the modified system is greater than the normal force affecting the brake pad of the conventional system at the same pressure 10 bar.

The results presented in Figure 11 show the variation of the mean friction coefficient of the conventional and modified system at pressure 10 bar and initial temperature 50℃ at different sliding speeds 50, 100, 150 and 200 r.p.m. The results indicated that, the increase of the sliding speed of the rotating disc cause a decrease of the mean friction coefficient of the conventional and modified system. The increase of sliding speed from 50 r.p.m to 200 r.p.m causes a decrease on the mean friction coefficient from 0.386 to 0.36 for conventional system and from 0.388 to 0.365 for modified system. Also the mean friction coefficient of the modified system is higher than the mean friction coefficient of the conventional system at each constant speed, this is because the normal force affecting the brake pad of the modified system is greater than the normal force affecting the brake pad of the conventional system at the same pressure of 10 bar.

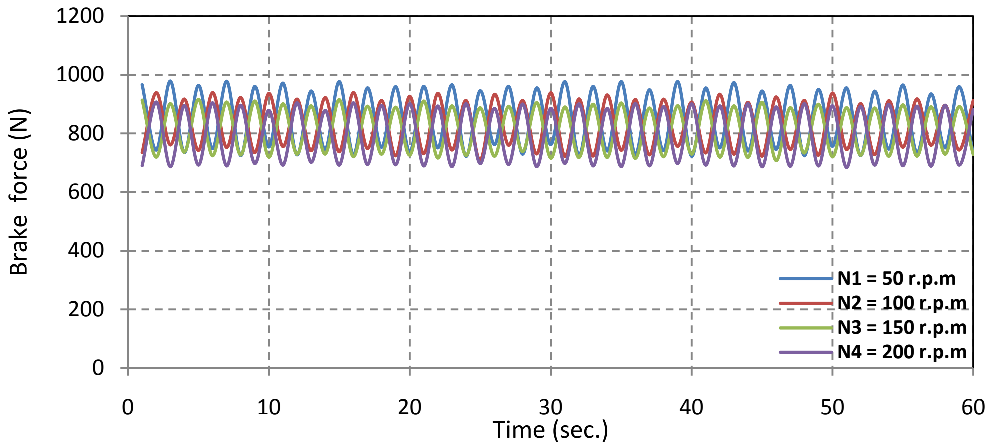

Figure (8) Variation of brake force against time at pressure of 10 bar and initial temperature of 50℃ and at different sliding speeds for the conventional system.

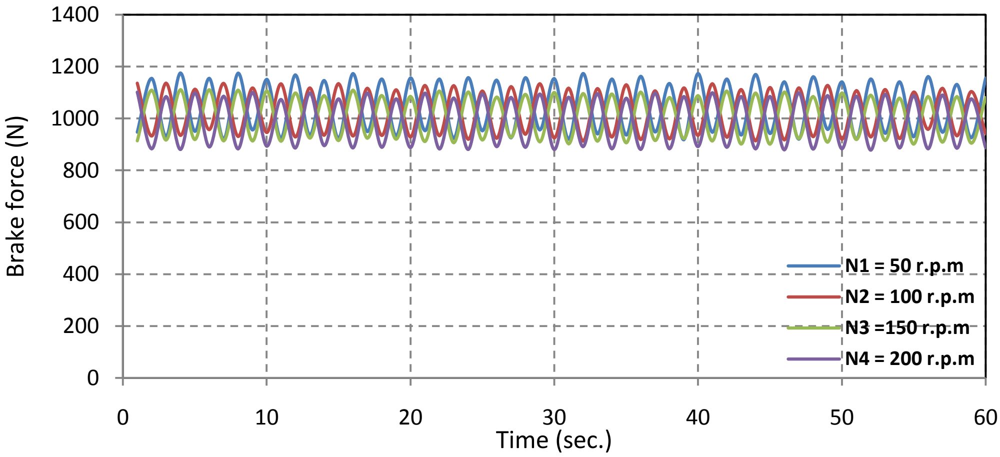

Figure (9) Variation of brake force against time at pressure of 10 bar and initial temperature of 50°C at different sliding speeds for the modified system.

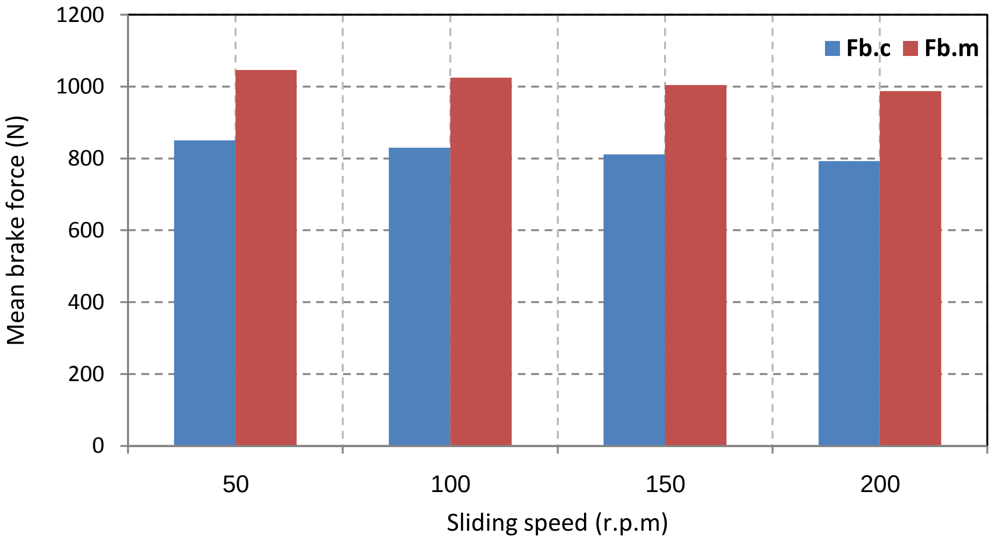

Figure (10) Effect of sliding speed on the mean brake force at pressure of 10 bar and initial temperature of 50°C for conventional and modified brake system.

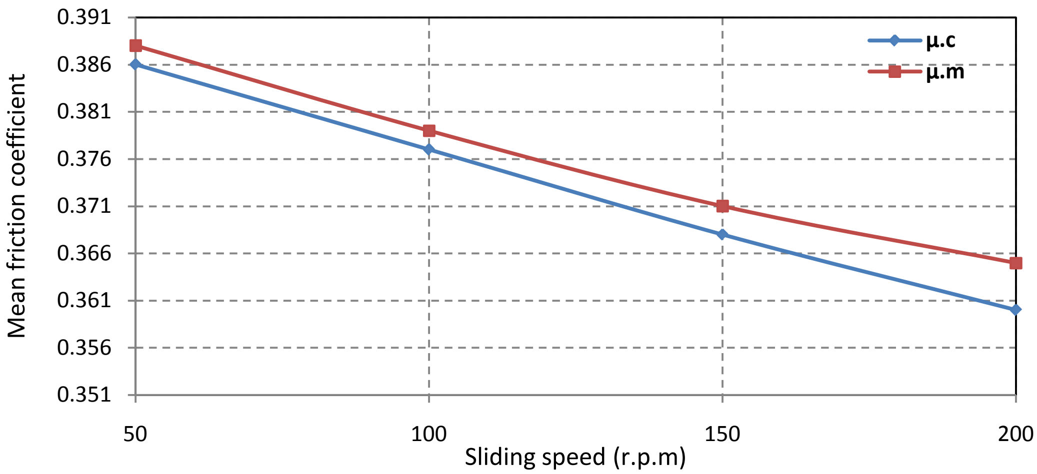

Figure (11) Effect of the sliding speed on the mean friction coefficient at pressure of 10 bar and initial temperature of 50 °C for conventional and modified brake system.

4.2 Effect of sliding speed at brake oil pressure 10 bar and initial temperature 100 ℃

Figure 12 and Figure 13 explain the effect of the sliding speed of the rotating disc on the brake forces of the conventional and modified system at brake oil pressure 10 bar and initial temperature 100℃. The experimental results showed that, the increase of sliding speed of the brake disc leads to decrease the brake force of both conventional and modified systems. Also the brake forces of the conventional and modified system fluctuate with no identical trend with the brake time at each constant speed. The fluctuation of the brake force is due to the variation of the friction coefficient with the braking time. The effect of the sliding speed on the mean brake force of the conventional and modified system is shown in Figure 14. From the results, it can be seen that the increase of the sliding speed of the rotating disc cause a decrease of the mean brake force of the conventional and modified system. The mean brake forces of the conventional system are 835, 816, 798, 781 N and the mean brake forces of the modified system are 1030, 1011, 993, 976 N at sliding speeds 50, 100, 150 and 200 r.p.m respectively. The increase of the sliding speed from 50 r.pm to 200 r.p.m decreases the mean brake force of the conventional system from 835 N to 781 N which is nearly 7 % and decreases the mean brake force of the modified system from 1030 N to 976 N which is nearly 6 % . Furthermore, at each constant speed, the value of the mean brake force of the modified system increases approximately by 23%,23%,24%,24% respectively over the value of the mean brake force of the conventional system, this is because the normal force affecting the brake pad of the modified system is greater than the normal force affecting the brake pad of the conventional system at the same pressure 10 bar.

Figure 15 illustrate the effect of the sliding speed of the rotating disc on the mean friction coefficient of the conventional and modified system at brake oil pressure of 10 bar and initial temperature of 100 ℃. The results indicated that, the increase of the sliding speed of the rotating disc cause a decrease of the mean friction coefficient of the conventional and modified system. The increase of sliding speed from 50 r.p.m to 200 r.p.m causes a decrease on the mean friction coefficient from 0.379 to 0.355 for conventional system and from 0.381 to 0.361 for modified system. Furthermore, the mean friction coefficient of the modified system is higher than the mean friction coefficient of the conventional system at each constant speed; this is because the generated normal force from the modified system is greater than the generated normal force from the conventional system at the same pressure of 10 bar.

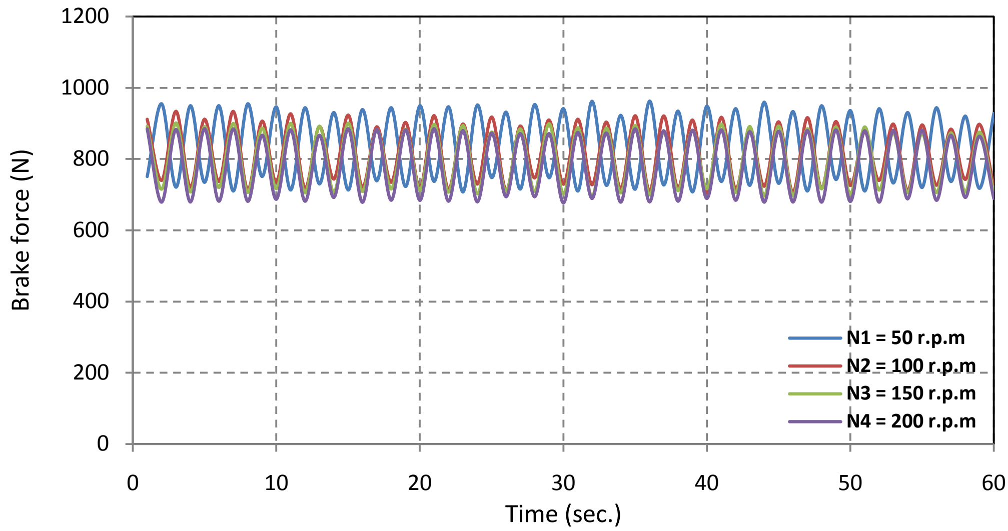

Figure (12) Variation of brake force against time at pressure of 10 bar and initial temperature of 100 ℃ at different sliding speeds for conventional brake system.

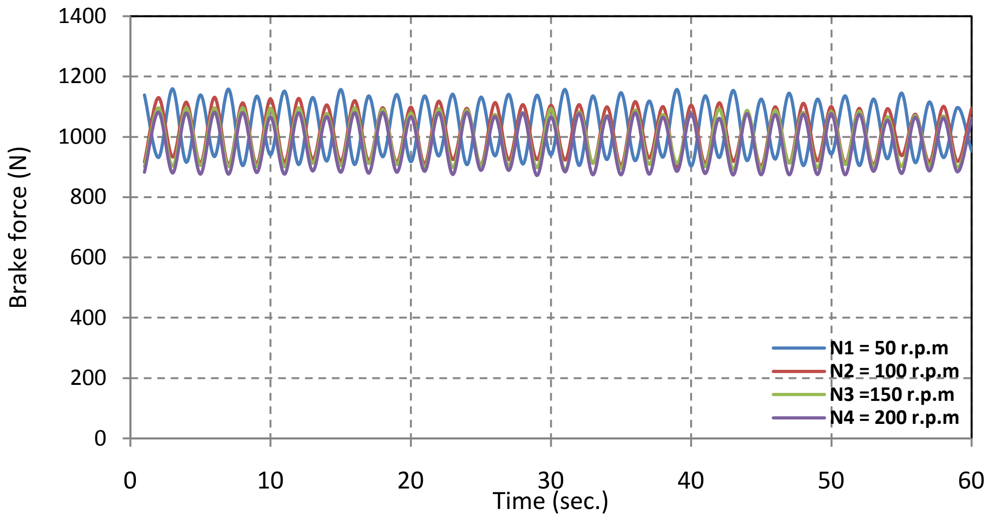

Figure (13) Variation of brake force against time at pressure of 10 bar and initial temperature of 100 °C at different sliding speeds for modified brake system.

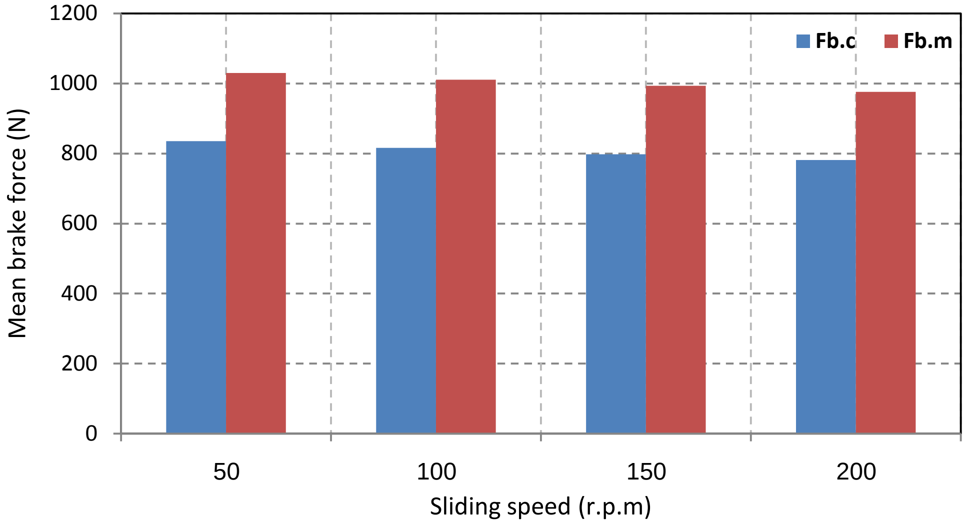

Figure (14) Effect of sliding speed on the mean brake force at pressure of 10 bar and initial temperature of 100°C for conventional and modified brake system.

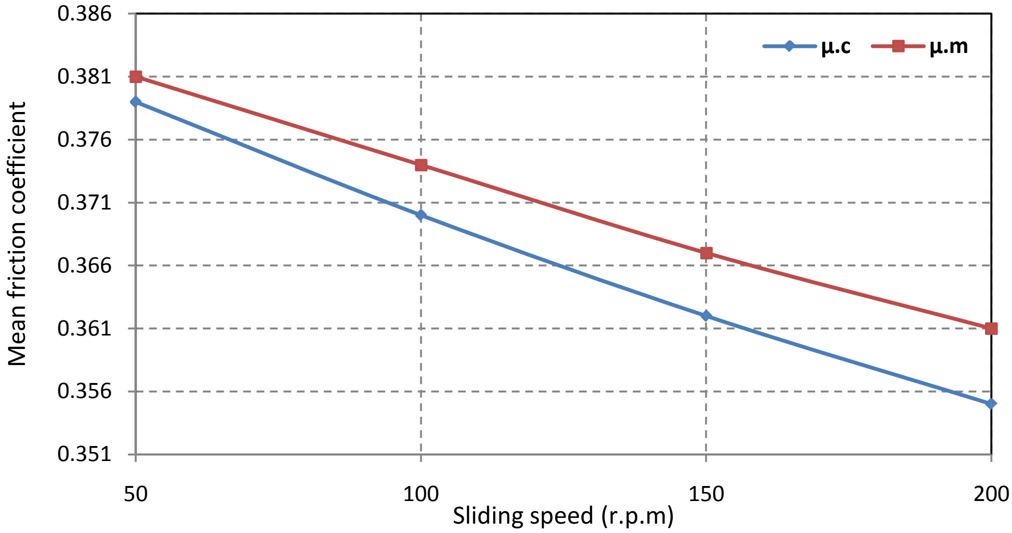

Figure (15) Effect of the sliding speed on mean friction coefficient at pressure of 10 bar and initial temperature of 100 °C for conventional and modified brake system.

4.3 Effect of sliding speed at brake oil pressure 10 bar and initial temperature 150℃

At pressure 10 bar and initial temperature 150 ℃, the effect of sliding speed of the rotating disc on the brake forces of the conventional and modified system are plotted in Figure 16 and Figure 17. The results indicate that, the brake forces of the conventional and modified system decrease as the sliding speed of the rotating disc increases. Also with the increase of the braking time at initial operating temperature 150℃, the brake forces of the conventional and modified system tend to decrease especially at high speed. Due to the increase of the braking time, the friction temperature increases and become over 150℃. Hence, the brake force tends to decrease as a result of friction coefficient decrease. The effect of the sliding speed on the mean brake force of the conventional and modified system is shown in Figure 18. From the results, it can be seen that the increase of the sliding speed of the rotating disc cause a decrease of the mean brake force of the conventional and modified system. The mean brake forces of the conventional system are 646, 627, 608 and 590 N and the mean brake forces of the modified system are 841, 821, 803 and 784 N at sliding speeds 50, 100, 150 and 200 r.p.m respectively. The increase of the sliding speed from 50 r.pm to 200 r.p.m decreases the mean brake force of the conventional system from 646 N to 590 N which is nearly 9 %. Also the mean brake force of the modified system decreases from 841 N to 784 N which is nearly 7 % . Also Figure 18 illustrated that, at each constant speed, the mean brake force of the modified system increases approximately by 30%,30%,32%,32% respectively over the value of the mean brake force of the conventional system, this is because the normal force affecting the brake pad of the modified system is greater than the normal force affecting the brake pad of the conventional system at the same pressure 10 bar.

The effect of sliding speed of the rotating disc on the mean friction coefficient of the conventional and modified system at brake oil pressure of 10 bar and initial temperature of 150℃ is showed in Figure 19. The results indicated that, the increase of the sliding speed of the rotating disc cause a decrease of the mean friction coefficient of the conventional and modified system. The increase of sliding speed from 50 r.p.m to 200 r.p.m causes a decrease of the mean friction coefficient of the conventional system from 0.293 to 0.268 and causes a decrease of the mean friction coefficient of the modified system from 0.311 to 0.29. Also the mean friction coefficient of the modified system is greater than the mean friction coefficient of the conventional system at each constant speed, this is because the normal force affecting the brake pad of the modified system is greater than the normal force affecting the brake pad of the conventional system at the same pressure of 10 bar.

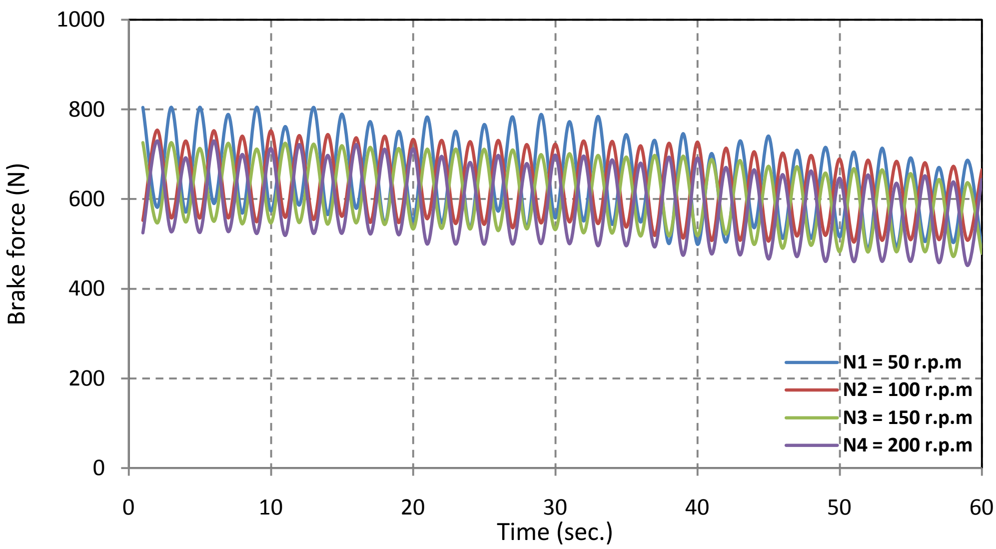

Figure (16) Variation of brake force against time at pressure of 10 bar and initial temperature of 150℃ at different sliding speeds for conventional brake system.

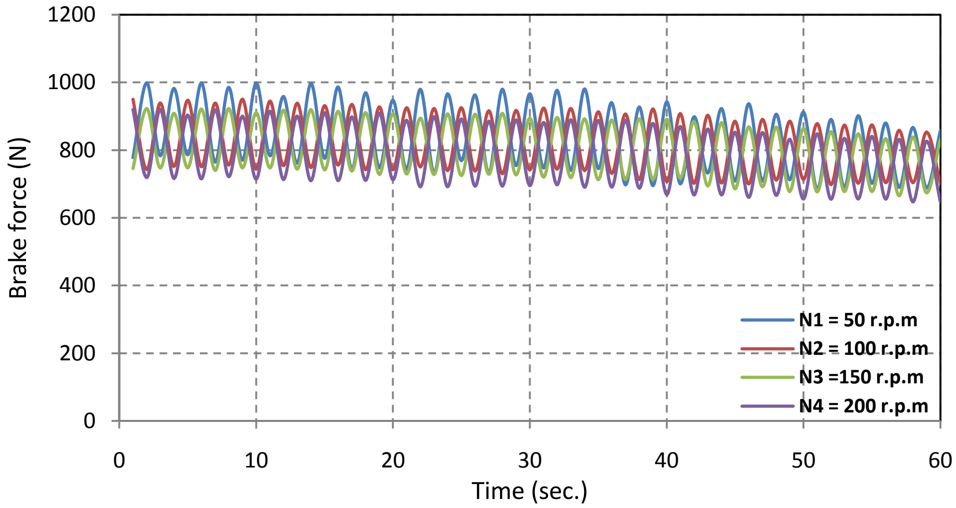

Figure (17) Variation of brake force against time at pressure of 10 bar and initial temperature of 150°C at different sliding speeds for modified brake system.

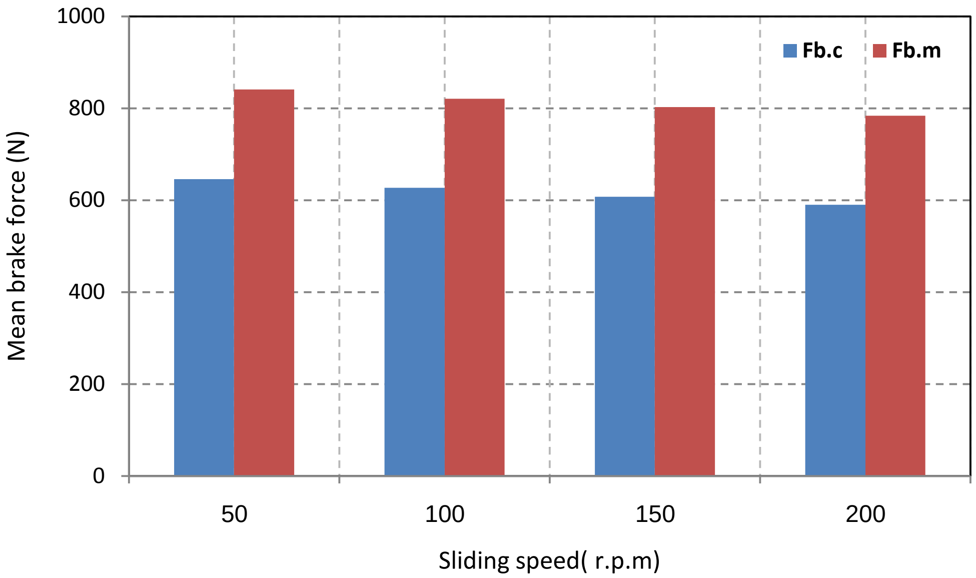

Figure (18) Effect of sliding speed on the mean brake force at pressure of 10 bar and initial temperature of 150°C for conventional and modified brake system.

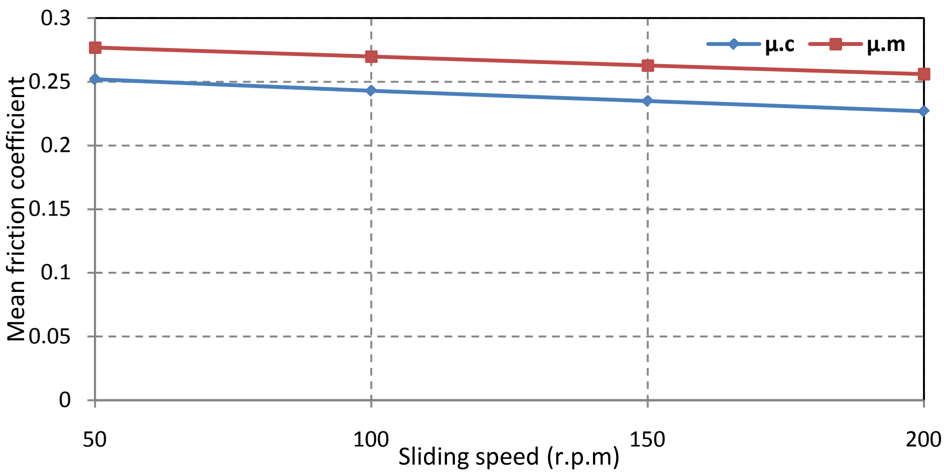

Figure (19) Effect of the sliding speed on the mean friction coefficient at Pressure of 10 bar and initial temperature of 150 °C for conventional and modified brake system..

4.4 Effect of sliding speed at brake oil pressure 10 bar and initial temperature 200℃

The effect of the sliding speed of the rotating disc on the brake force of the conventional and modified system at pressure 10 bar and initial temperature 200 ℃ is presented in Figure 20 and Figure 21. From the shown figures, it can be seen that, the increase of the sliding speed causes a decrease of the brake forces of the conventional and modified system. Also the results illustrate that the brake forces of the two systems tend to decrease at initial operating temperature 200℃ as the braking time increases especially at the high speed. Due to the increase of the braking time, the friction temperature increases and become over 200 ℃. Hence, the friction coefficient decrease; therefore, the brake force tend to decrease. The results presented in Figure 22 show the effect of sliding speed of the rotating disc on the mean brake force of the conventional and modified system. From the results, it can be seen that the increase of the sliding speed of the rotating disc cause a decrease of the mean brake force of the conventional and modified system. The mean brake forces of the conventional system are 555, 536, 518 and 500 N and the mean brake forces of the modified system are 750, 730, 712 and 693 N at sliding speeds 50, 100, 150 and 200 r.p.m respectively. The increase of the sliding speed from 50 r.pm to 200 r.p.m decreases the mean brake force of the conventional system from 555 N to 500 N which is nearly 10 %. Also the mean brake force of the modified system decreases from 750 N to 693 N which is nearly 8 %**.** Also Figure (22) shows that, at each constant speed, the value of the mean brake force of the modified system increases approximately by 35%, 36%, 37% and 38% respectively over the value of the mean brake force of the conventional system, this is because the normal force affecting the brake pad of the modified system is greater than the normal force affecting the brake pad of the conventional system at the same pressure 10 bar.

Figure 23 shows the effect of sliding speed of the rotating disc on the mean friction coefficient of the conventional and modified system at brake oil pressure of 10 bar and initial temperature of 200 ℃. The results indicated that, the increase of the sliding speed of the rotating disc cause a decrease of the mean friction coefficient of the conventional and modified system. The increase of sliding speed from 50 r.p.m to 200 r.p.m causes a decrease on the mean friction coefficient from 0.252 to 0.227 for the conventional system and from 0.277 to 0.256 for the modified system. Also Figure 23 shows that, the mean friction coefficient of the modified system is higher than the mean friction coefficient of the conventional system at each constant speed, this is because the generated normal force from the modified system is greater than the generated normal force from the conventional system at the same pressure of 10 bar.

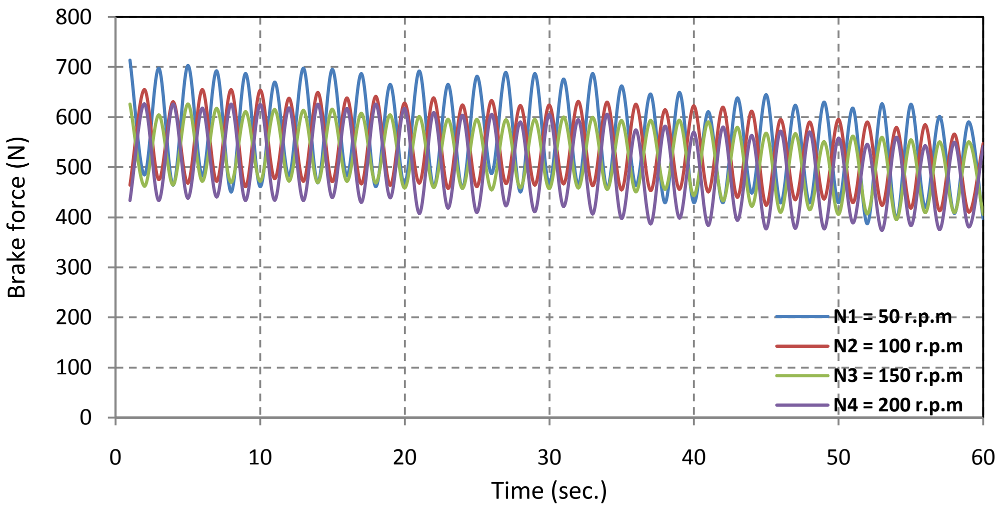

Figure (20) Variation of brake force against time at pressure of 10 bar and initial temperature of 200℃ at different sliding speeds for conventional brake system.

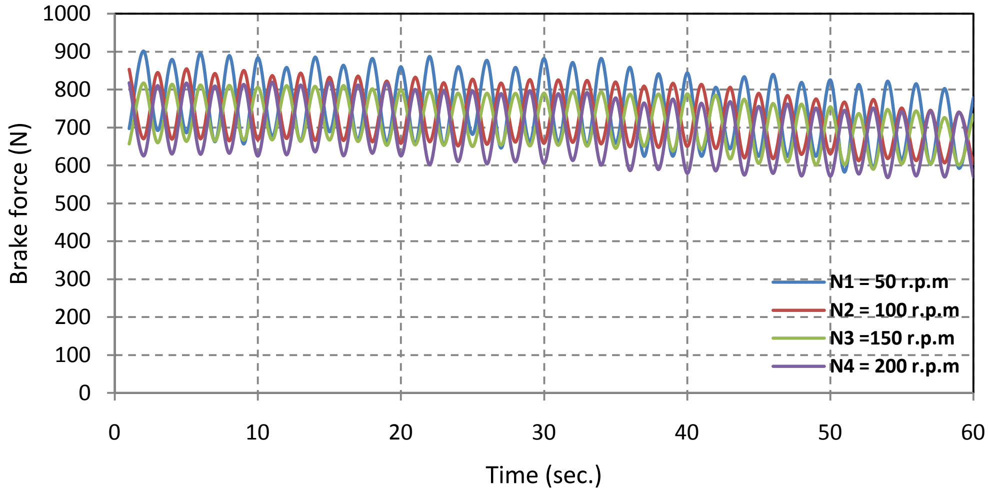

Figure (21) Variation of brake force against time at pressure of 10 bar and initial temperature of 200°C at different sliding speeds for modified brake system.

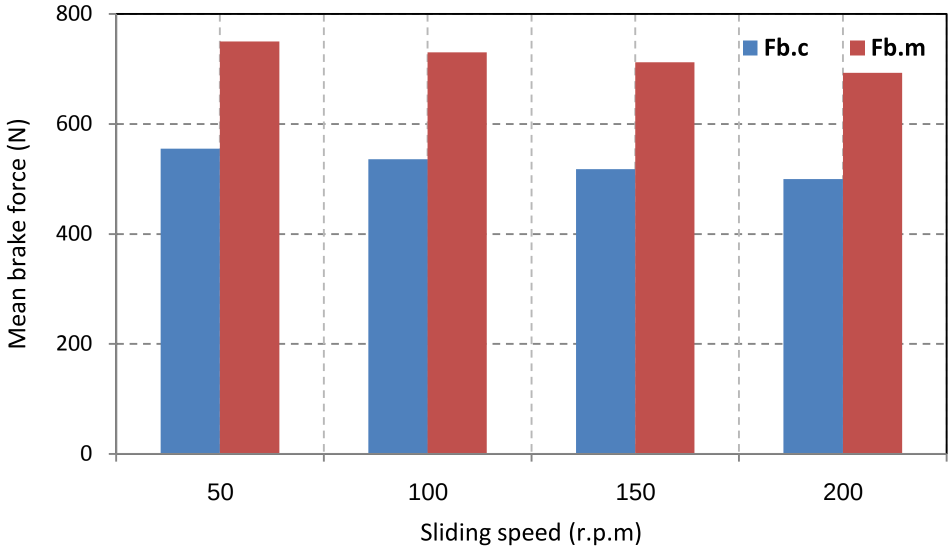

Figure (22) Effect of sliding speed on the mean brake force at pressure of 10 bar and initial temperature of 200°C for conventional and modified brake system.

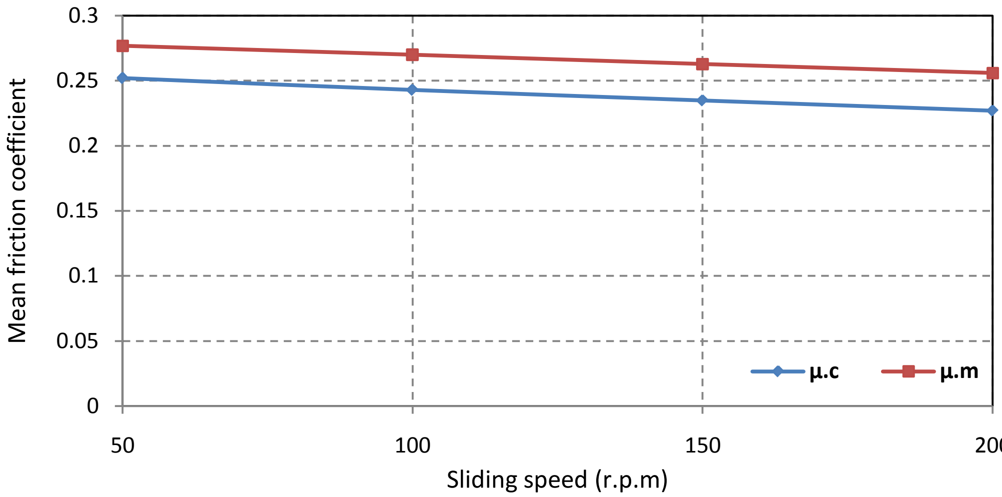

Figure (23) Effect of the sliding speed on mean friction coefficient at Pressure of 10 bar and initial temperature of 200 °C for conventional and modified brake system..

CONCLUSION

The main conclusions from the present study can be summarized in the following points:

1- The brake force of the conventional and modified system varies and fluctuates with no certain trend with the brake time due to the variation of the friction coefficient with the brake time. It tend to decrease with the brake time at the initial operating temperature 150℃ and 200℃ which gives an indication of the brake fade with increase of temperature.

2- Increasing the sliding speed decreases the mean brake force of the conventional and modified brake system.

3- The increase of the sliding speed from 50 r.p.m to 200 r.p.m at initial temperature of 50℃ and brake oil pressure of 10 bar decreases the mean brake force of the conventional and modified brake system by 7% and 6% respectively. Also increasing the sliding speed from 50 rpm to 200 r.p.m at initial temperature of 100℃ and brake oil pressure of 10 bar decreases the mean brake force of the conventional and modified brake system by 7% and 6% respectively.

4- Increasing the sliding speed from 50 rpm to 200 rpm at initial temperature of 150℃ and brake oil pressure of 10 bar decreases the mean brake force of the conventional and modified brake system by 9% and 7% respectively. However; increasing the sliding speed from 50 rpm to 200 rpm at initial temperature of 200℃ and brake oil pressure decreases the mean brake force of the conventional and modified brake system by 10% and 8% respectively.

5- Increasing the sliding speed at constant pressure of 10 bar and at different initial operating temperature of 50, 100, 150 and 200℃ decreasing the mean friction coefficient of the conventional and modified brake system. But at each speed the mean friction coefficient of the modified system was greater than the mean friction coefficient of the conventional system.

6- At sliding speeds 50, 100, 150 and 200 r.p.m and at brake oil pressure of 10 bar the mean brake force of the modified system was greater than the mean brake force of the conventional system by 23% and 24% at initial operating temperatures of 50℃ and 100℃ respectively.

7- At sliding speeds of 50, 100, 150 and 200 r.p.m and at brake oil pressure of 10 bar the mean brake force of the modified system was greater than the mean brake force of the conventional system by 30% and 32% at initial operating temperatures of 150℃.

8- At sliding speeds 50, 100, 150, 200 r.p.m and at brake oil pressure 10 bar the mean brake force of the modified system was greater than the mean brake force of the conventional system by (35%:38%) at the initial operating temperatures 200℃.

9- The increase of the initial operating temperature decreases the mean brake force of the conventional and modified system.

10- Increasing the initial operating temperature from 50℃ to 100℃ at different sliding speeds (50 r.p.m:200 r.p.m) at brake oil pressure 10 bar decreases the mean brake force of the conventional and modified system by (2%: 3%). But the increase of the initial operating temperature from 50℃ to 150℃ at the same condition decreases the mean brake force of the conventional and modified system by (24% : 26%) and (20% : 21%) respectively.

11- Increasing the initial operating temperature from 50℃ to 200℃ at different sliding speeds from 50 r.p.m to 200 r.p.m at brake oil pressure of 10 bar decreases the mean brake force of the conventional and modified system by (35%, 37%,) and (29% and 30%) respectively.

REFERENCES

[1]. Mostafa Mohamed Makrahy "Optimization and Experimental Investigation of a New Wedge Disc Brake", PhD. Thesis, Department of Automotive and Tractors Engineering, Minia University, Egypt, pp 6-7, 2014.

[2]. Halderman, J. D. "Automotive Technology, principles, diagnosis, and service", Pearson Education, Inc., 2009.

[3]. Dzmitry Tretsiak. "Experimental investigation of the brake system's efficiency for commercial vehicles equipped with disc brakes", Proc IMechE Part D, J Automobile Engineering, 226(6) 725–739, 2012.

[4]. C. Jo, S. Lee, H. Song, Y. Cho, I. Kim, D. Hyun and H. Kim. "Development of a Novel Electro-Wedge Brake System", KSAE, pp. 1321-1326, November, 2008.

[5]. C. Jo, S. Hwang and H. Kim. " Clamping-Force Control for Electromechanical Brake", IEEE, Vol. 59, pp. 3205-3212, Number 7, 2010.

[6]. Mikael Eriksson,Filip Bergman,Staffan Jacobson, "On the nature of tribological contact in automotive brake", Wear 252 (2002), pp.26-36.

[7]. J. Bijwe,Nidhi,N. Majumdar and B.K.Satapathy, "Influence of modified phenolic resins of the fade and recovery behavior of friction materials", Wear 259 (2005), pp.1068-1078.
[8]. Mostafa M. Makrahy, Nouby M. Ghazaly, K. A. Abd El-Gwwad, K. R. Mahmoud and Ali M. Abd-El-Tawwab "A Preliminary Experimental Investigation of a New Wedge Disc Brake", Int. Journal of Engineering Research and Application, ISSN: 2248-9622,vol.3, Issue 6, Nov-Dec 2013, pp.735-744.
[9]. Severin, D. and, Dörsch, S. "Friction mechanism in industrial brakes", Wear 249,771-779, 2001.
[10]. C.H. Gao, X.Z. Lin "Transient temperature field analysis of a brake in a non-axisymmetric three dimensional model ", I. Materials processing Technology, vol. 129, No. 1-3, PP.513-517,2002.
[11]. Österle, W. and Urban, I. "Friction layers and friction films on PMC brake pads", Wear 257, 215-226, 2004.
[12]. Mikael Eriksson "Friction and contact phenomena of disc brakes related to squeal", PhD thesis, Faculty of Science and Technology, Uppsala University, 2000.
[13]. Blau, P. J. "Friction science and technology" , Marcel Dekker, Inc., 1996.
[14]. Aviles, R., Hennequent, G., Hernandez, and A., Licrente, J., I. "Low frequency vibrations in disc brakes at high car speed", Part one, experimental approach, Inderscience Enterpprises Ltd, 1995.
[15]. Dunlap, K. B., Riehle, M. A. and Longhouse, R. E. "An investigation overview of automotive disc brake noise", SAE paper 1999-01-0142.
[16]. Bergman, F., Eriksson, M. and Jacobson, S. "Influence of disc topography on generation of brake squeal", Wear, pp.225-229, 1999.
[17]. Riches, M. J., Samir, N. Y., and Jordan, R. "Reduction of squeal noise from the disc brake system using constrained layer damping", J. of the Brazilian Society of Mechanical Science and Engineering, vol. 26, pp. 340-348, 2004.
[18]. Chen T. F., "Relationship between Formulation and Noise of phenolic Resin Matrix Friction Lining Tested In Acoustic Chamber on Automotive Brake Dynamometer", Master of Science Thesis, Southern Illinois University, 2005.

Ibrahim Ahmed" Effect of the sliding speed on the performance of conventional and modified disc brake at different initial operating temperatures" International Journal Of Engineering Inventions, Vol. 07, No. 10, 2018, pp. 46-61