Study the effect of changing the lining form of the duplex drum brake on the brake performance

Ibrahim Ahmed[1], Sayed S. Mohamed[2], Khaled Abdel Wahed[3] and Yasser Fatouh[4]

[1]Professor at Automotive and Tractors Technology Department, Helwan University, Cairo, Egypt
[2]Researcher at Department of Automotive Technology, Helwan University, Cairo, Egypt
[3]Assistant Professor at Automotive and Tractors Technology Department, Helwan University, Cairo, Egypt
[4]Assistant Professor at Automotive and Tractors Technology Department, Helwan University, Cairo, Egypt

---***---

Abstract – *Brakes are the most important parts of the car. The safety of passengers in the car depends mainly on the efficiency of the brake system, and that is very important to its role in preserving individuals and construction. So, very severe requirements are imposed on the brake system in terms of reliability, durability and workmanship quality. The brake lining temperature can reach 300 °C while frequent brakes are used in vehicles. The experimental results in this paper showed that, the final temperature of the drum brake system without slots and with slots are increased with increasing the brake oil pressure. At each constant pressure and at sliding speed of 100 r.p.m decreases the final temperature of the drum brake system with one slot by 3.3%, 4% and 3.6% respectively. The final temperature of the drum brake system with three slots decreases by 11%, 8% and 11.4% respectively compared to the drum brake without slots.*

Key Words: Drum Brake, Lining, Duplex, Simplex, leading and trailing.

1.INTRODUCTION

The 4-wheels hydraulic brakes were introduced to the market in 1918 and was invented by Malcolm Lugid [1]. This kind of brake system replaced mechanical brakes that were previously used in vehicles [1]. Automotive companies and research centers seek to manufacture and amend parts of the brake system due to their importance in vehicles to maintain the safety of individuals and facilities. So that, very severe requirements are imposed on the brake system in terms of reliability, durability and workmanship quality. The brake lining temperature can reach 300 °C while frequent brakes are used in vehicles [2]. The friction brake in the car works by converting the kinetic energy into thermal energy, which produces heat, due to the friction in the interface between the rotor (disc or drum) and the stator (pads or shoes). This heat must be absorbed to maintain brake performance by the stator and rotor. Sufficient cooling of these components is necessary to achieve the required performance of the braking system [3]. **Figure 1** shows the simplex drum brake system that has diameters generally ranging from 170 to 550 mm [4]. The duplex drum brake type has two shoes effective differently depending on the generated brake force. The first shoe is called the leading shoe however; the other shoe called the trailing shoe. The exporter of the applied force is usually a hydraulic cylinder, when the lining arrives to frictional contact for the drum. The rotation of the drum will drag the shoe over with it or press the shoe away depending on the position of the shoe pivoting point relative to the direction of rotation. This action provides a form of self-energizing as clear in **figure (1)** that shows a construction of the simplex drum brake [4].

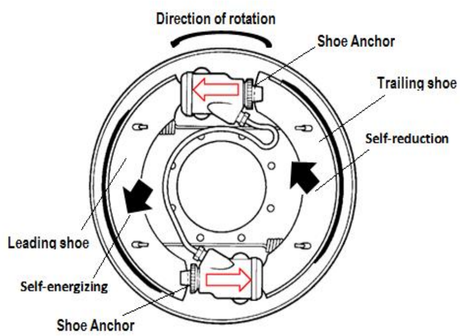

Fig – 1: Simplex (leading trailing) drum brake.

The duplex drum brake is designed to be self-energizing brake that let the shoe to mechanically increase the force applied by the hydraulic system and assist in the braking action. The increase is achieved by mounting the shoes to the backing plate in such a way that take advantage of frictional force that occurs when the brake lining contacts the rotating drum [5]. During drum rotation, the upper tips of the shoes are pushed apart by the expander force and a normal inward reaction force provided by the drum. So that, the drum slide over the shoe linings and a tangential frictional force is produced between each pair of rubbing surfaces. The friction force (leading shoe) tends to move in the same direction as the shoe tip force which produces it. as, this helps to drag the shoe onto the drum. This increase in shoe tip force above the input expander force is termed as positive servo, and shoes that provide this self-energizing are known as leading shoes or self-energizing shoes [6].

2. EXPERIMENTAL SETUP

The main parts of the brake test rig that is constructed in vehicle laboratory, Automotive and tractor department, Faculty of technology and Industrial Education, Helwan University are shown in **figure (2).**

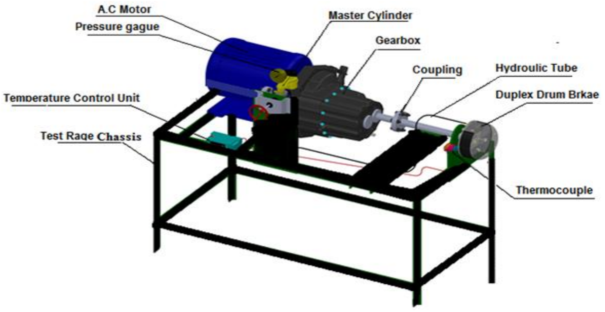

Fig – 2: Main components of the test rig

A drum brake of Suzuki Swift is used in the test rig. This braking system is a duplex drum brake. The main components of the brake system are shown in **figure (3)**. It consists of leading shoe and trailing shoe. The duplex drum brake system also contains two-wheel cylinders and each contains a hydraulic piston, braking plate, return springs and the brake pads as clear in **figure 3**. The specifications of the used duplex drum brake system are shown in **Table1**.

In the laboratory experiments, three different brake shoes are used in this work. The first brake shoe of the used Duplex drum brake is the plain shoe without any modifications as shown in **figure 4**. The second used brake shoe is modified to be a shoe with one slot with dimensions (4 mm wide and 4 mm thick) as clear in **figure 5** as the friction area decreased by 2%. However; the last brake shoe used in the experiments is modified to be with three slots also with dimensions (width 4 mm thickness 4 mm) as shown in **figure 6** where less friction area by 6%.

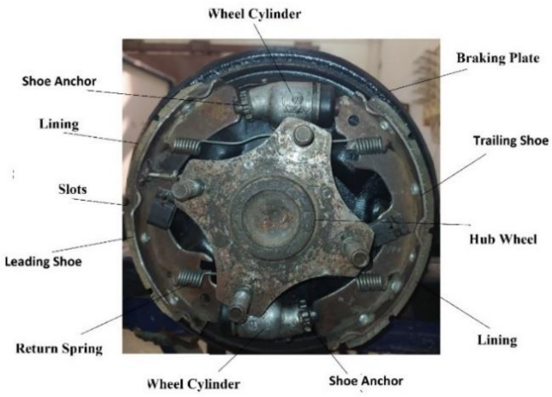

Fig – 3: Duplex drum brake of Suzuki swift.

Table-1: Specifications of the main components of the duplex drum brake system.

Component	Value
Shoe Length	246 mm
Shoe Width	41 mm
Shoe Thickness	3 mm
Shoe Arc	142°
Drum Inner Diameter	210 mm
Drum Outer Diameter	250 mm
Piston Diameter	25 mm
Brake Lining Length	225 mm
Brake Lining Width	40 mm
Lining Arc	122°

Fig – 4: Plain brake shoe (original without modification).

Fig – 5: Modified brake shoe with one slot.

Fig – 6: Modified brake shoe with three slots.

The braking torque is the most important factor upon which the results in this study were based. In this work, a power meter is used in the test rig that shown in **figure 2** to measure the power consumption of the electric motor during the braking process and the normal force affects the braking system. The sliding speed was also measured to determine the brake torque. The power of the electric motor was measured using a Schneider PM 1200 digital power meter that ranges from 20 watts to 300 kW and has a 1% reading accuracy for energy. It also measures the non-load force, which is the energy that is consumed by the electric motor to overcome the forces of inertia of the elements of the test platform. In this study, the brake strength is determined as follows:

$$P_b = P_L - P_{no} \quad (1)$$

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 drum (sliding speed) is also a very significant parameter in the braking process due to its affection on the friction of the brake system units. It was measured by a digital tachometer of type (DT6234 B) and it has a speed range from 5 to 100000 r.p.m with accuracy of 0.5 %.

By calculating the brake power of the braking system during the braking process as mentioned in equation (1) and the angular speed of the rotating disc, the brake torque was calculated as follow:

$$T_b = \frac{P_b}{\omega} \quad (2)$$

$$\omega = \frac{2\pi n}{60} \quad (3)$$

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)

Analysis of the duplex drum brake by Mahmoud [7] as shown in **figure (7)** for each leading shoe and trailing shoe. In the duplex drum brake, the applied force is produced through a hydraulic cylinder.

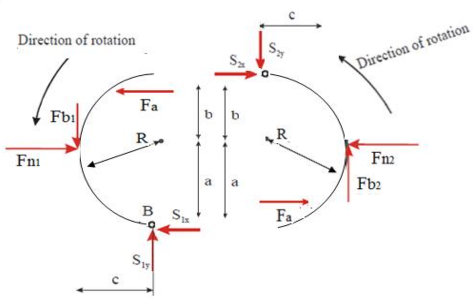

Fig – 7: Force analysis of the duplex drum brake.

By evaluating the moment about the point B in figure 7, it will give the following equation:

$$Fa\,(a + b) + Fb_1 . c - FN_1 . a = 0 \quad (3)$$

The braking force or the frictional force which produced by the leading shoe can be calculated from the following equation:

$$Fb_1 = \mu . FN_1 \quad (4)$$

In substitution for equation (4) in equation (3), the following equation will be presented:

$$Fa\,(a + b) + \mu . FN_1 . c - FN_1 . a = 0 \quad (5)$$

By arranging the previous equation will give the following equation:

$Fa\,(a + b) = FN_1 . a - \mu . FN_1 . c$

$Fa\,(a + b) = FN_1 . (a - \mu c)$,

so the normal reaction force (leading shoe) is:

$$FN_1 = \frac{Fa\,(a+b)}{a - \mu . c} \quad (6)$$

By substituting from equation (6) in equation (5), so the braking force which produced by the leading shoe will be:

$$Fb_1 = \frac{\mu . Fa\,(a+b)}{a - \mu\, c} \quad (7)$$

Similarly, for trailing shoe, by evaluating the moment about the point C will give the following equation:

$$Fa\,(a + b) + Fb_2 . c - FN_2 . a = 0 \quad (8)$$

The braking force which produced by the trailing shoe can be calculated from the following equation:

$$Fb_2 = \mu.\ FN_1 \quad (9)$$

By substituting from equation (9) in equation (8) will give the following equation:

$$Fa\ (a + b) + \mu.\ FN_2\ .c - FN_2.a = 0 \quad (10)$$

By arranging the previous equation will give the following equation:

$$Fa\ (a + b) = FN_2.a - \mu.\ FN_2.\ C \quad (11)$$

Since the value $Fa\ (a + b) = FN_2\ (a - \mu\ c)$, so the normal reaction force (trailing shoe) is:

$$FN_2 = \frac{Fa\ (a+b)}{a - \mu\ .c} \quad (12)$$

By substituting from equation (12) in equation (9), so the braking force which produced by the trailing shoe will be:

$$Fb2 = \frac{\mu.Fa\ (a+b)}{a - \mu\ c} \quad (12)$$

3. RESULTS AND DISCUSSION

The Friction coefficient is a main factor determining the brake shoe factor which usually used to evaluate the brake performance. In this research, the experimental work is carried out to investigate the effect of different operating parameters such as sliding speed, initial friction temperature and brake oil pressure at the contact area between the brake drum and lining of duplex drum brake on the brake force and friction coefficient. All experimental work was performed in the same conditions for 60 seconds of braking. Three brake oil pressures were selected during cylinder brake tests with or without slots. These pressures were selected as 5, 10, and 15 bar respectively. Four sliding speeds were also chosen during laboratory work as 50,100,150 and 200 rpm respectively. Brake force and friction coefficient of cylinder brakes are calculated from previous equations every second and drawn with brake time during tests. A summary of the study results is presented as follows.

3.1 Effect of brake oil pressure on the brake force at constant sliding speed.

The effect of changing the brake oil pressure on the brake force of the drum brake system with plain, one-slotted and three slotted shoes are shown in **figures 8, 9 and 10** respectively at constant sliding speed of 50 rpm. The results show that as the oil pressure increase, the brake force increase at any constant parameter. The brake force fluctuates due to the change in the friction coefficient with time. **Figure (11)** shows the increase in the average braking force due to the increase in the brake oil pressure and also the results shows that the mean brake force with the slotted-shoes is less than mean brake force of duplex drum brake with the plain shoe at all pressure values and this is because of decreasing the friction area in the brake lining. The normal force that affects the brake lining of the drum brake system with slotted shoes are less than that of the plain shoe at the same pressure and this increases the braking force of the drum brake system without slots at all pressures.

The mean brake forces of the drum brake system with plain shoe are 1815, 3835 and 4582 N respectively at the three different used pressures of 5, 10 and 15 bar. However; the brake forces of the drum brake system with one-slotted shoe are 1454, 3675 and 4289 N respectively and 1314, 3046 and 3907 N respectively for the three-slotted shoe at also the same oil pressure of 5, 10 and 15 bar respectively. It is realized that at each pressure, the mean brake force of the duplex drum brake system with one-slotted shoe decreases by 20%, 5%, 7% respectively. And 28 %, 21%, 15% respectively for the duplex drum brake system with three-slotted shoe depending to the value of the mean brake force of the duplex drum brake shoe without slots.

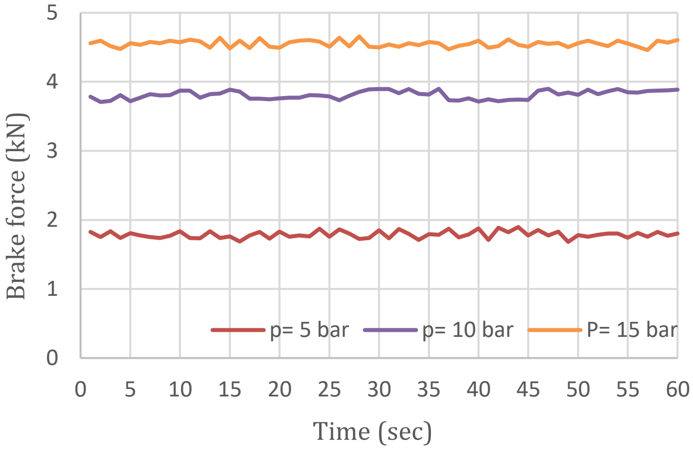

Fig – 8: Brake force against time for the duplex drum brake with plain shoe at sliding speed of 50 r.p.m and different brake oil pressure.

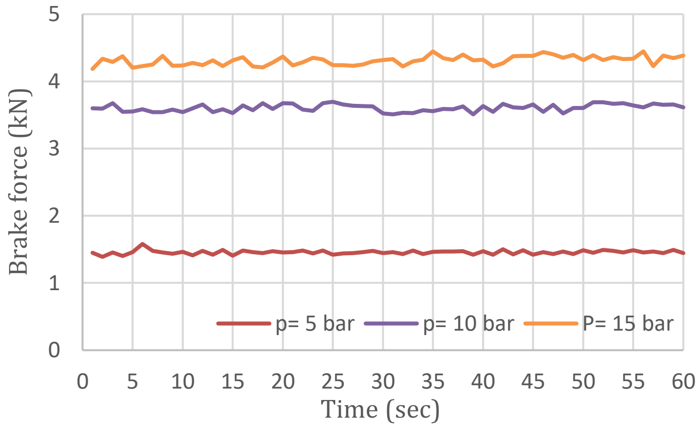

Fig – 9: Brake force against time for the duplex drum brake with one-slotted shoe at sliding speed of 50 r.p.m and different brake oil pressure.

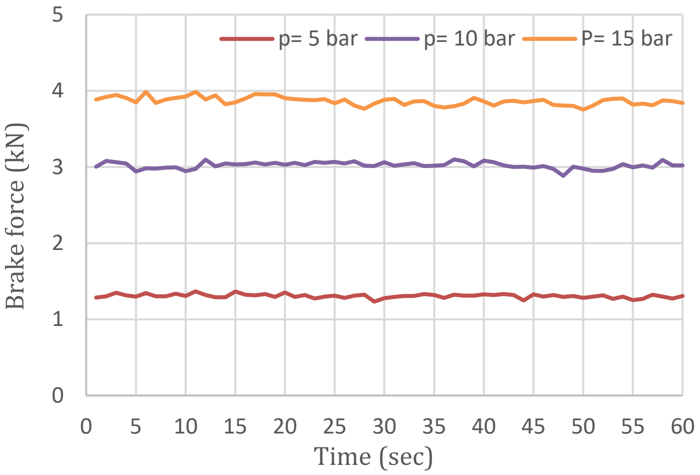

Fig – 10: Brake force against time for the duplex drum brake with three-slotted shoe at sliding speed of 50 r.p.m and different brake oil pressure.

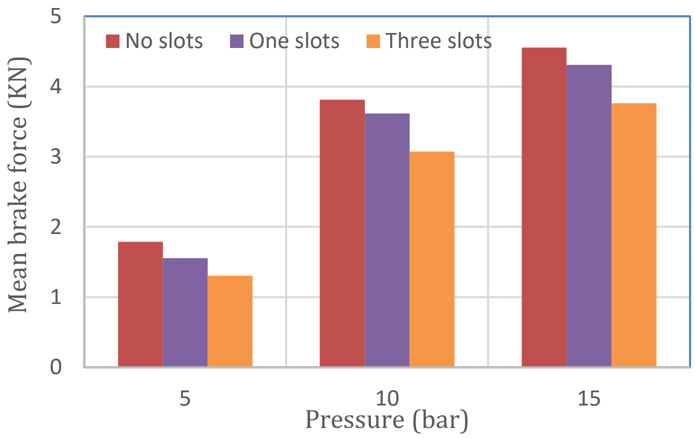

Fig – 11: Effect of brake oil pressure on the mean brake force of duplex drum brake with plain shoe, one-slotted and three-slotted shoes at sliding speed of 50 r.p.m.

In **Figure (12)**, the results show the effect of the brake oil pressure on the mean friction coefficient of the duplex drum brake with plain shoe, one-slotted and three-slotted shoes respectively at sliding speed of 50 r.p.m. The results show that as the brake pressure increases, the average friction coefficient increases. It is shown that the average friction coefficient of the duplex drum brake with slotted-shoes are less than the average friction coefficient of the duplex drum brake without slots at each pressure and this is due to the reduction of the friction area in the friction lining. The increase of the oil pressure from 5 to 15 bar causes an increase on the mean friction coefficient from 0.2607 to 0.5927 for duplex drum brake without slots, from 0.2538 to 0.5794 for duplex drum brake with one slot and from 0.2244 to 0.5455 for drum brake with three slots.

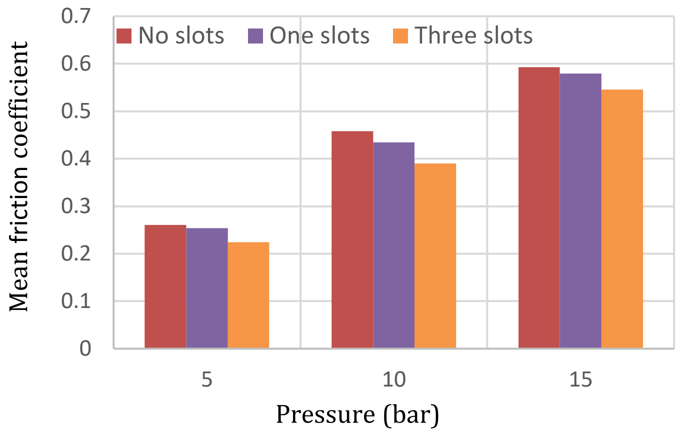

Fig – 12: Effect of brake pressure on the mean friction coefficient of duplex drum brake with plain, one-slotted and three-slotted shoes at sliding speed of 50 r.p.m..

3.2 Effect of brake oil pressure on operating temperature at different sliding speed.

Figure (13) shows the effect of brake oil pressure on the final temperature of the duplex drum brake system with slots and without slots at sliding speed 50 r.p.m. The results show that, as the brake oil pressure increases, the final temperature of the duplex drum brake system increases with slots and without slots, also. The final temperature of the duplex drum brake system with slotted shoes are less than the final temperature of the duplex drum brake system with plain shoe at all pressures. This is due to the reduced friction area in the brake lining. The final temperature of the duplex drum brake system without slots is 27,40 and 60 °C at the brake oil pressure of 5, 10 and 15 bar respectively. However; the final temperature of the duplex drum brake system with one-slotted shoe are 25,39 and 58 °C respectively. The final temperature of the duplex drum brake system with three-slotted shoe are 24,37 and 53 °C respectively for the same different used oil pressure of 5, 10 and 15 bar.

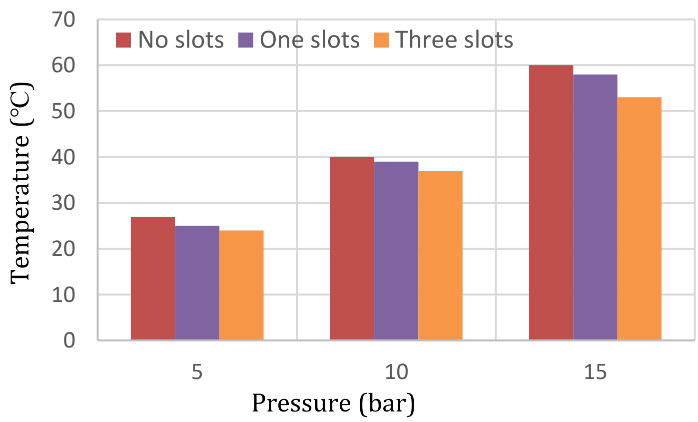

Fig – 13: Effect of friction area on the final temperature of the drum brake system at sliding speed 50 r.p.m and different brake oil pressure.

International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 07 Issue: 07 | July 2020 www.irjet.net p-ISSN: 2395-0072

The final temperature of the drum brake with plain shoe indicate a decrease in its value by 8%, 3.2% and 3.4% respectively compared to the final temperature of the drum brake with one-slotted shoe and 11.2%, 8.5% and 11.7% respectively compared to the drum brake with three-slotted shoe.

3.3 Effect of sliding speed on the mean friction coefficient.

Figure 14, 15 and 16 show the relationship between the brake force and the tine at constant brake pressure of 5 bar and different sliding speeds of 50, 100, 150 and 200 rpm respectively for the plain, one-slotted and three-slotted shoes. These figures show that the brake forces of the duplex drum brake system with slotted and non-slotted shoes 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.

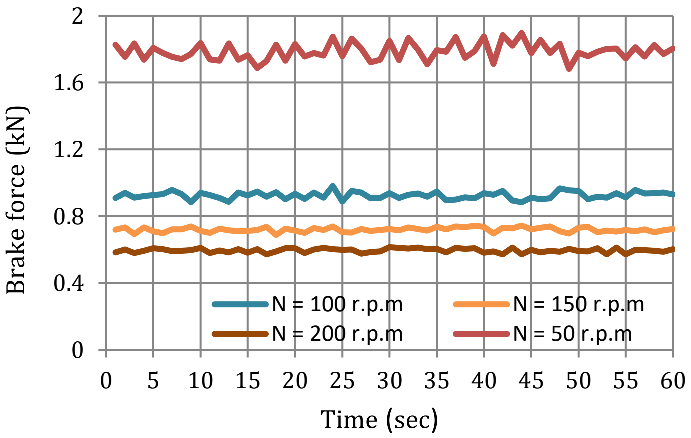

Fig – 14: Brake force against time for the duplex drum brake with plain shoe at brake oil pressure of 5 bar and different sliding speeds.

Figure (17) shows the variation of the mean brake force of the duplex drum brake system with slots and without slots at different sliding speeds. It shows that, the increase of the sliding speed of the rotating drum cause a decrease of the mean brake force of the duplex drum brake system with slots and without slots. This was due to a decrease of the friction coefficient. It can be concluded that the friction coefficient decreases with increasing sliding speed may be due to lubricating oxides form at elevated temperatures, and if the surface frictionally melts, the molten liquid can lubricate the asperity contacts. The mean brake forces of the duplex drum brake system without slots are 1815, 920, 721 and 592 N respectively and the mean brake forces for the duplex drum brake system with one-slotted shoe are 1446, 830, 651 and 546 N respectively. However; the mean brake forces of the duplex drum brake system with three slots are 1295, 723, 518 and 427 N respectively at sliding speeds of 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 duplex drum brake system without slots from 1815 N to 592 N. And from 1446 N to 546 for duplex drum brake system with one slot and from 1295 N to 427 N for the duplex drum brake system with three slots. **Figure (17)** also show that, at each constant speed, the value of the mean brake force of the drum brake system with one slot decrease approximately by 21%, 10.8%, 9.8% and 7.8% respectively. However; for the drum brake with three-slotted shoe the decrease is approximately by 28.7%, 22.5%, 28.2% and 27.9% respectively according the value of the mean brake force of the drum brake system without slots. This is due to the reduction in the friction area in the brake lining.

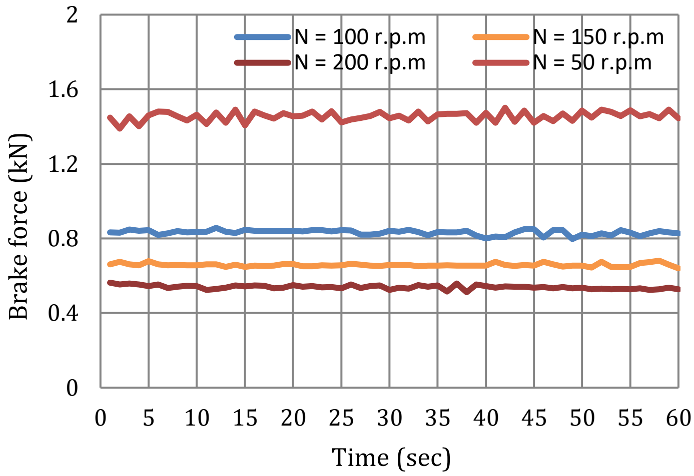

Fig – 15: Brake force against time for the duplex drum brake with one-slotted shoe at brake oil pressure of 5 bar and different sliding speeds.

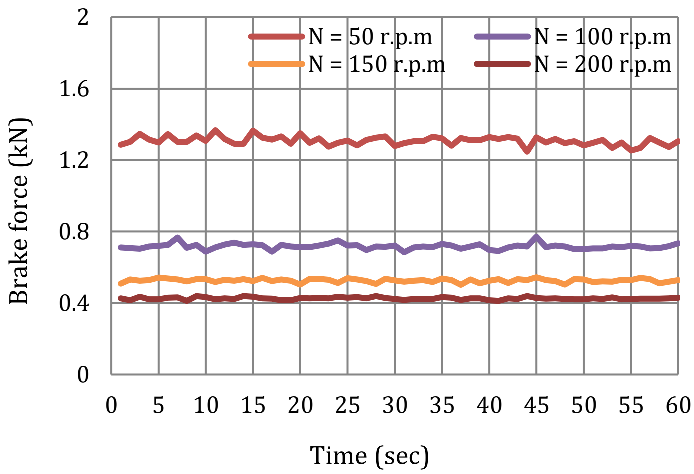

Fig – 16: Brake force against time for the duplex drum brake with three-slotted shoe at brake oil pressure of 5 bar and different sliding speeds.

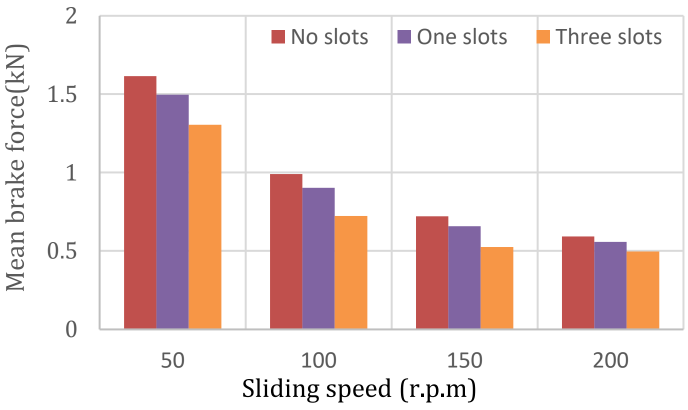

Fig – 17: Effect of sliding speed on the mean brake force of the drum brake system without slots and with slots at Pressure 5 bar.

3.4 Effect of sliding speed on operating temperature at different brake oil pressure

Figure (18) shows the effects of sliding speed of the rotating drum on the final temperature of the drum brake system without slots and with slots at pressure of 5 bar. It indicates that the variation of the final temperature of the drum brake system with slots and without slots at different sliding speeds. the results explain that, with the increase of the sliding speed of the rotating drum this leads to an increase of the final temperature of the drum brake system with slots and without slots. The final temperature of the duplex drum brake system without slots are 27, 37, 42 and 53 ℃ respectively. However; for the duplex drum brake system with one slot are 25, 36, 40, 51 ℃ and for the duplex drum brake system with three slots are 24, 33, 37, 48 ℃ at sliding speeds of 50, 100, 150 and 200 r.p.m respectively. **Figure 18** also shows that, at each constant speed, the value of the final temperature of the drum brake system with one slot decrease approximately by 7.5%, 2.8%, 4.8% and 3.8% respectively, and for the drum brake with three slots decrease approximately by 11.2%, 8.9%, 12% and 9.5% respectively compared to the final temperature of the drum brake system without slots. This is due to the decreasing of the friction area in the brake lining.

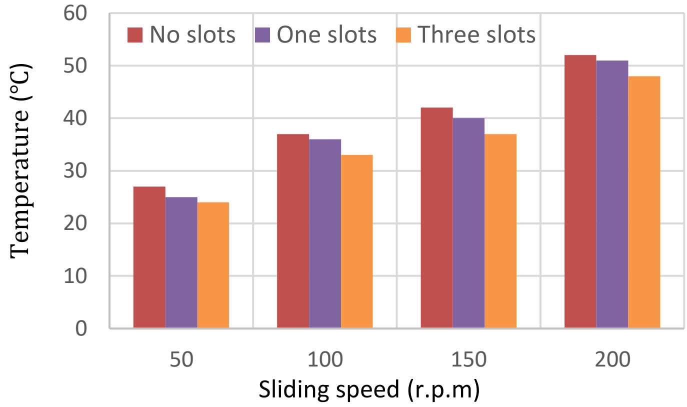

Fig – 18: Effect of friction area on the final temperature of the drum brake system at pressure 5 bar and different sliding speed.

4. CONCLUSIONS

The following points are a summary of the conclusions from this study:

1. Friction coefficient decreases with increasing brake time at a sliding speed of 150 rpm and 200 rpm, which leads to brake wear and tear, especially with increasing temperatures.
2. Increasing of the brake oil pressure at different sliding speed of 50, 100, 150, 200 r.p.m decreases the mean friction coefficient of the drum brake system without slots and with slots.
3. Increasing the brake oil pressure increases the average brake force of the duplex drum braking system without slots and slots where one slot was made in the brake lining which led to a decrease in the friction area by 2%, and three slots was made in the brake lining which led to a decrease in the friction area by 6%.
4. Increasing the brake oil pressure at sliding speed of 50 r.p.m decreases the mean brake force of the drum brake system with one slot by 20%, 5%, 7% respectively. And 28 %, 21%, 15% respectively for the duplex drum brake system with three slots compared to the brake force of the duplex drum brake without slots.
5. The increase of the brake oil pressure of values 5, 10, and 15 bars respectively at sliding speed of 100 r.p.m decreases the mean brake force of the drum brake system with one slot by 10 %, 7%, 5% and 24%, 17%, 12% respectively for the duplex drum brake system with three slots compared to the average brake force of the drum brake without slots.
6. Increasing of the brake oil pressure at constant sliding speed of 150 r.p.m decreases the mean brake force of the drum brake system with one slot by 6.5%, 5.4%, and 6.2%. and 18.2%, 19.7% and 12.7% respectively for the duplex drum brake system with three-slotted shoe compared to the average brake force of the drum brake with plain shoe.

5. REFERENCES

1- James D. Halderman. J. D., Mitchell, C. D. Jr. 'Automotive Brake Systems', NJ, USA. pp. 1-60, (2015).
2- Yehia Talkhan : Studying the Frictional behavior of disc brake system at different operating conditions and their effects on clarifying the importance of the practical demonstration with the automotive teachers : A thesis master, 2017.
3- Arthur Stephens.: Aerodynamic cooling of Automotive Disc Brakes: M.Sc. thesis, School of Aerospace, Mechanical & Manufacturing Engineering, RMIT University, March 2006.
4- Halderman, J. D.: Automotive Technology: principles, diagnoses, and service: Pearson Education, Inc., 2009.
5- Bosch : Bremsanlagen für Kraftfahrzeuge: Robert Bosch GmbH, 1994.
6- Khaled A. K.: Improving the disc brake performance at high temperatures through modifying the brake system and designing an instructional unit of the theory of vehicle course for the third-year students at the faculty of industrial education according to the research results: PhD thesis, Faculty of Industrial Education, Helwan University, 2018.
7- Mahmoud, K. R. M.: theoretical and experimental investigations on a new adaptive duo servo drum brake with high and constant brake shoe factor: PhD thesis, Heinz Nixdorf Institute, University of Paderborn, 2005.

BIOGRAPHIES

Ibrahim Ahmed, is a Professor of Vehicle Dynamic and Control at Helwan University in Egypt. He is currently the Head of Production Technology Department. He obtained his B.Sc. (1990) and M.Sc. (1995) of Automotive Engineering from Helwan University in Cairo, Egypt followed by another M.Sc. from Eindhoven University 1997. He obtained also the PhD (2002) from Newcastle Upon Tyne, UK. He has about 50 papers in the field of Vehicle Dynamics and Tribology. He has many contributions in the field of Noise, Vibration and Harshness (NVH).
E-mail address:
ilmahmed1968@yahoo.co.uk

Sayed S. Mohamed, is a researcher (M.Sc. student) at Helwan University. He obtained the B.Sc. in Industrial Education, Automotive and tractors Technology Department from Helwan University in 2015. He is currently working as a demonstrator at Automotive and Tractors Technology Department, Faculty of Industrial Education, Helwan University in Egypt. He was born in 1992 in Giza, Egypt.
E-mail address:
engineer.sayedsaad@gmail.com

Khaled Abdel Wahed, is an Assistant Professor of Vehicle Dynamic and Control at Helwan University in Egypt. He obtained his B.Sc. (1992) and M.Sc. (1997) in Mechanical Design Engineering and PhD (2007) from Helwan University in Cairo, Egypt. He has many contributions in the field of Noise, Vibration and Harshness (NVH).

Yasser Fatouh, is an Assistant Professor of Vehicle Dynamic and Control at Helwan University in Egypt. He obtained his B.Sc. (1992) and M.Sc. (1997) in Mechanical Design Engineering and PhD (2007) from Helwan University in Cairo, Egypt. He has many contributions in the field of Noise, Vibration and Harshness (NVH).