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A Preliminary Experimental Investigation of a New Wedge Disc Brake

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ABSTRACT
This paper presents a preliminary study of a new wedge disc brake performance using a brake test rig that has been recently developed at the Automotive Laboratory, Minia University, Egypt. First, the brake test rig is designed and tested for conventional disc brake. Next, the mechanical structure of designed wedge disc brake is presented. After that, some performance tests are conducted at different operation conditions. Finally, comparison between conventional and novel disc brake are performed. Experimental results show that this novel wedge disc brake can multiply the brake force 3.7 times of conventional disc brake at the same conditions.
Keywords: *Brake test rig, brake performance**,** wedge disc brake*

I. Introduction

Nowadays, most vehicles use disc brakes as they dissipate heat better hence reducing fade when compared to drum brakes. The brake force and brake shoe factor are affected by several conditions such as the surface finishing, friction material, temperature, sliding speed, water, and normal force in order to these parameter effect on the friction coefficient. This leads to the observation that the friction coefficient is varies with the brake time. The brake system is a very important component to vehicles and machinery equipment in industries.

Research on brake performance has been conducted using theoretical, numerical and experimental approaches. The experimental approaches have been used to measure the brake performance for the system during braking event. However, the experiments are mostly expensive and time consuming [1]. Over the years, experimental approaches using brake dynamometers or on-road tests have been widely used to examine the brake performance, to investigate the effects of different parameters and operating conditions, to understand the characteristics of the brake system during braking event and to verify possible solutions that can improve the performance. Experimental methods are still play an important role for a number of reasons. Firstly, they offer more effective analysis tools than numerical or purely theoretical methods. Secondly, diagnosis of the cause of brake problems can often be found by experimental tests. Finally, the verification of solutions of simulation models can only be achieved through experimental means [2-3].

A brake dynamometer or an inertial dynamometer is used to test the performance of braking systems, such as wear of the pads and rotors and the amount of friction generated by the pads and rotors. It is also used to test problems that occur in braking systems. A brake dynamometer is designed to duplicate the deceleration of a vehicle in a controlled environment. It consists of a drive-train assembly and an absorption unit. The drive-train assembly is comprised of the motor, inertial disks and brake disk. The motor is accelerated to the desired speed and then disengaged to allow the assembly to run freely as a result of the inertia generated by the inertial disks. There are two designs for the brake dynamometer. The first design is an inertia-type brake dynamometer that has flywheel attached to it [4-6]. The second design is a drag-type brake dynamometer that can only test brake performance at a constant speed [7-12].

This paper presents a preliminary study of a new wedge disc brake performance using a brake test rig. The brake test rig is designed and constructed using different sub-system. Conventional disc brake is tested at different operation conditions. The structure of designed wedge disc brake is presented. The performance tests for the new wedge brake are conducted. Finally, a comparison between conventional and novel disc brake are performed.

II. Brake Test-Rig Description

The main objectives of the test rig are to measure the generated braking force at all the operating and the design parameters of the disc brake system. In addition to generate a kinetic energy that could be overcome by the braking system. To achieve these requirements the test rig is designed and constructed. The main units of the test rig which are shown in Fig.1 can be divided into: conventional brake system, wedge brake assembly, assembly of kinetic energy generation, assembly of applied force generation and water injecting system.

Fig.1 Main components of the test rig

2.1 Conventional Disc Brake Assembly

The main components of the brake system are: floating caliper, finger, hub, wheel bearing, rotor disc, and brake pad, as shown in Fig.2. The rotor had dimensions of 23.5cm diameter and 2 cm thickness. The brake pad consisted of two parts; friction material and back plate. Pad friction material dimensions are; 4.3 cm width, 10.5 cm length, and 1 cm thickness. Pad back plat dimensions are; 13cm length, and 6mm thickness. The hydraulic piston has a projected area of 16.0 cm^2 and the hydraulic line connected between the applied force mechanism and the hydraulic piston.

Fig.2 Conventional disc brake

2.2 Wedge Disc Brake Assembly

The conventional disc brake is modified to wedge disc brake in order to gain the self-amplification action and to improve its performance. The main components of the wedge assembly are: modified caliper, initial screw link, wedge angle variation link, main wedge angle plat, secondary wedge angle plat, rollers, and cotter, as shown in Fig.3. Fig.4 presents the brake caliper after modification. It illustrates that the hydraulic piston is built-in with the caliper, where the movement of the hydraulic piston affected the brake pad directly causing the normal force. The caliper is modified by cutting the piston and piston cylinder from the caliper. The caliper is fitted to fix the wedge assembly. The effect of normal force affects the brake pads by the cotter according to the hydraulic piston movement.

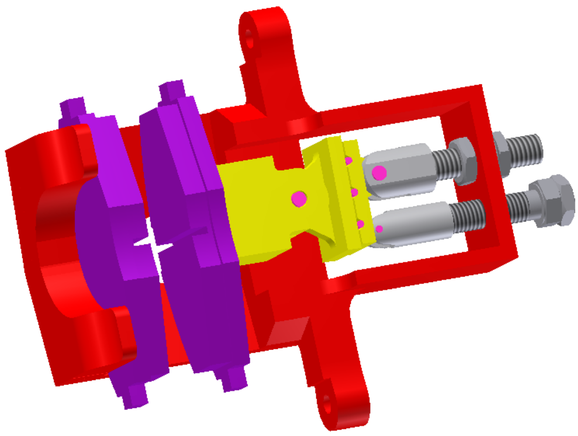

Fig.3 Modified (wedge) disc brake

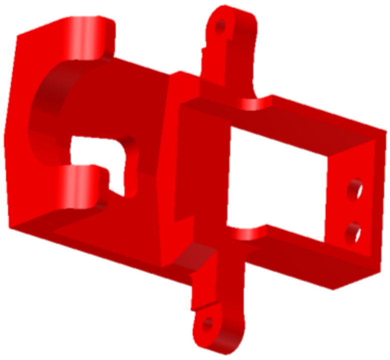

Fig.4 Sketch of Modified caliper of disc brake

The initial screw link is made of steel and used to regulate the clearance between the pad and the disc as well as it is connected to the main wedge plat as a pivot axis. The wedge angle variation link is made of steel and it is a screw link is used to vary the wedge angle according to operation condition. The wedge screw link varies the wedge angle between 0° and 45° where the complete revolution of the screw link gives 4° wedge inclination angle, as shown in Fig.5.

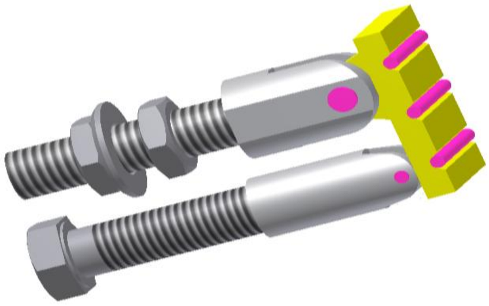

Fig.5 Sketch of initial screw link and wedge angle variation link

The main wedge angle plat is made of steel. It is connected to both the initial screw link and the wedge screw link to convert the displacement of the wedge screw link to wedge angle. The secondary wedge angle plat is connected to the cotter and contacted to the main wedge angle plat to shift the wedge angle to the cotter. Three rollers are placed between the main and the secondary wedge plates to reduce the friction and insure easy movements for the main and secondary wedge plates, as shown in Fig.6.

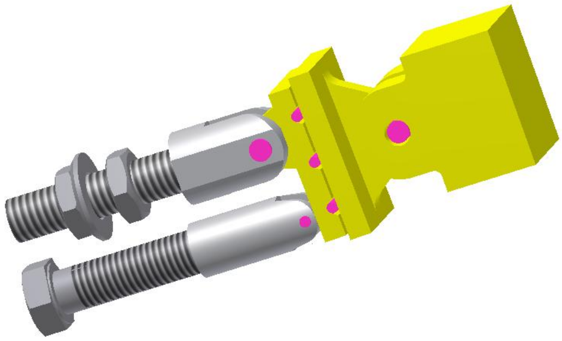

Fig.6 Sketch of the main and secondary wedge plates

The cotter is connected to the secondary wedge angle plat through the cotter pin and contacted to the pad back plat to transfer the normal force to the brake pad after the analysis of the applied force according to wedge angle. The applied force affects the cotter through the piston link which connected to the hydraulic piston, as shown in Fig.7.

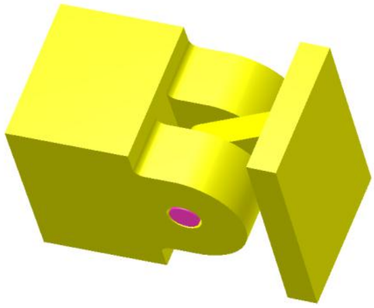

Fig.7 Sketch of the Cotter

2.3 Kinetic energy generation Assembly

The rotational speed plays a great role on friction coefficient variations and the brake shoe factor as a result. For this reason a 3-phase AC motor type; its maximum power is 18.65 kW (25 Hp) at 1500 rpm. In order to examine a various speeds, there are two gearboxes had different gears ratios are assembled to the motor and the braking system as well as increasing the output torque which inhabited to use severe operation conditions, as shown in Fig.8

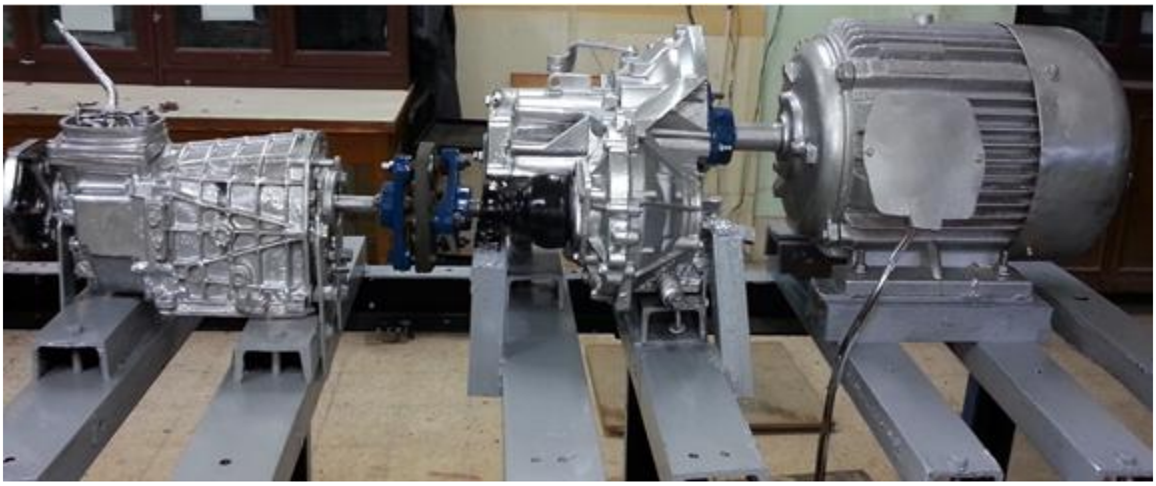

Figure (8) kinetic energy generation assembly

2.4 Applied force generation Assembly

The applied force is a main factor of generating the brake force. Hence it had to be taken into account its effect on the braking process. The generated applied force must have constant values during the test according to its conditions. A master cylinder is used to generate constant applied force. The function of the master cylinder is to converts mechanical force from the brake pedal, power booster and push rod into hydraulic pressure. This master cylinder is modified as shown in Fig.9. Three lines of the hydraulic oil are closed and the hydraulic oil passed through one line to the hydraulic piston. The brake pedal and the power booster are replaced by screw push link with Handel. All of the master cylinder and the hydraulic piston had been fixed to the test rig. The piston force is being transmitted to the wedge cotter by the piston link.

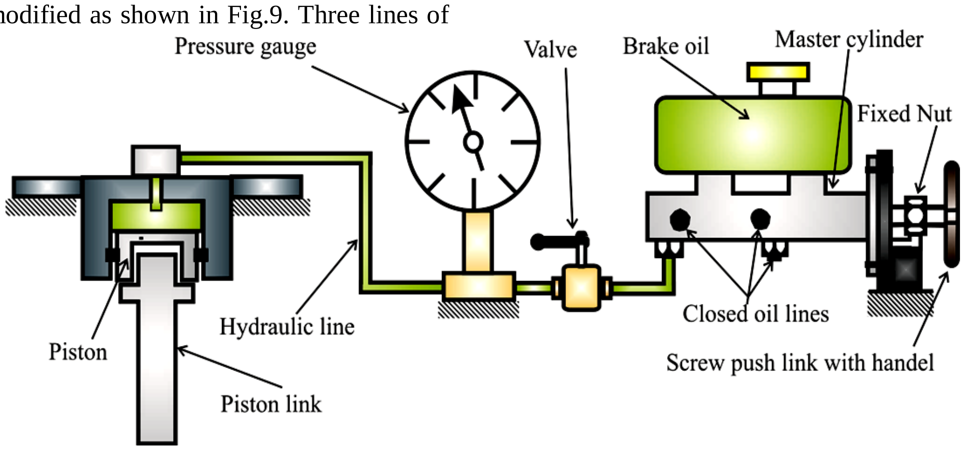

Fig.9 Applied force generation assembly

2.5 Water spraying system

The effect of water spraying on wedge disc brake performance is one of the operation parameters which are investigated at different quantities of water. The time period for spraying water is fixed to all tests. The water system is designed and constructed to insure these conditions, as shown in Fig.10. The water system consists of water hose, water mechanical valve, and two nozzles. The water timing unit had been used to control the injecting time at three different phases: the injecting time period, the time of the beginning of injection, and the end of the injection time. This unit consists of; solenoid valve, timer, two-port electric switch, and wires. The mechanical water valve connected to the water hose to controls the quantity of water that comes out of the nozzles to the disc and the pads contact area. The solenoid valve connected to the timer via the two-port switch and wiring to controls the water time injection. The brake performance has been investigated at dry condition and at 80, 160, 240 mm^3 of water. The overall timing of test is 60 seconds and the injecting spray began at the 30^{th} second and ended at the 40^{th} second.

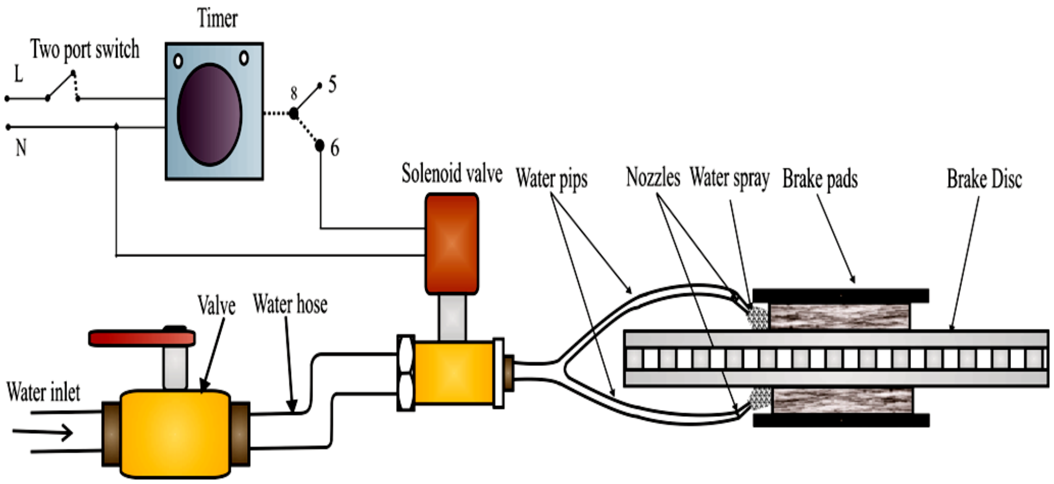

Fig.10 Schematic sketch of water injecting system

III. Measurement Instrumentation

The measuring system can be divided as the following: applied force measurement, speed measurement, brake force measurement, brake shoe factor and Friction coefficient calculation, and temperature measurement, as shown in Fig.11.

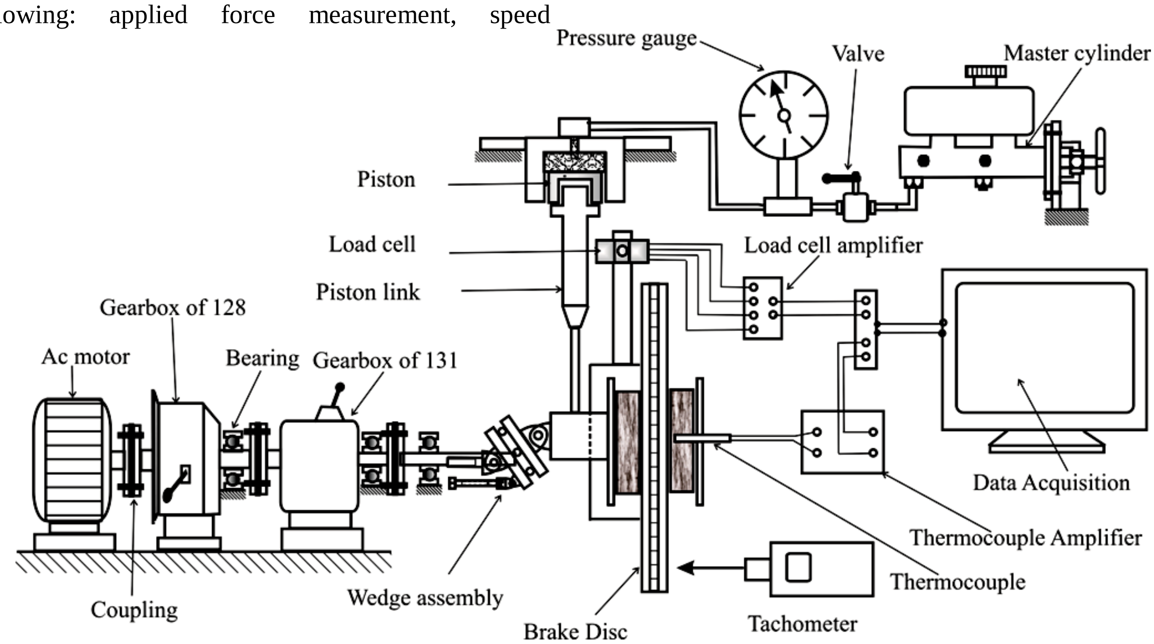

Fig.11 Schematic sketch of the test rig and measurement instrumentation

3.1 Applied force measurement

The brake oil pressure will be measured using an oil pressure gauge, as shown in Fig.12. The pressure gauge is inserted into the hydraulic line between the master cylinder and the hydraulic piston of the brake system. The applied force is calculated by using the projected area of the brake hydraulic piston and measuring the magnitudes of the braking oil pressure. The different values of the applied force are determined according to the values of the braking oil pressure as shown in the equations (1:3). There are four oil pressure values are selected to test. These four pressure values are 2.5 bar, 5 bar, 7.5 bar, and 10 bar. According to equation (3), these values of pressure equals applied forces of 400 N, 800 N, 1200 N, and 1600 N respectively. To confirm a constant applied force during the test, an additional valve is used. The valve is inserted into the hydraulic line between the master cylinder and the pressure gauge.

$$A = \frac{\pi}{4} D^2 \tag{1}$$

$$P = \frac{F_{app}}{A} \tag{2}$$

$$F_{app} = P * A \tag{3}$$

Where:

“ D ”The piston diameter equals 0.45 m

“ A “The piston area equals 0.16*10^{-3} m^2

“ P ”The hydraulic pressure

“ F_{app} ”The applied force

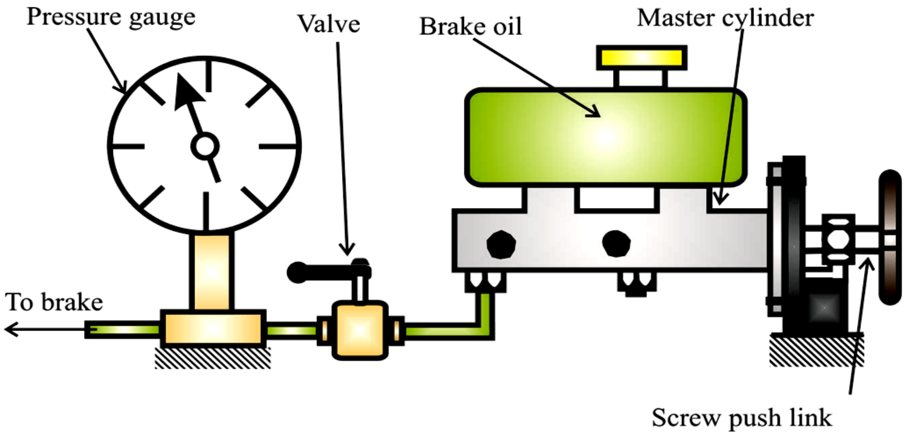

Fig.12 Schematic sketch of the applied force measurement assembly

3.2 Speed measurement

The rotational speed of the brake rotor is a result of different stages of the AC motor rotational speed. The rotational speed of the braking system is measured by a digital photo tachometer type $DT\,2234A$ which has range from 5 to $100{,}000$ rpm The aim of measuring the brake rotational speed is to calculate the vehicle speed which corresponding to the rotational speed of the braking system. The vehicle speed is calculated according to equation [4]. Four rotational speeds are selected to investigate their effects on the brake performance. These four rotational speeds are 54 rpm, 105 rpm, 205 rpm, and 329 rpm. From equation (4), the investigated vehicle speeds equal 6 km/hr., 11.6 km/hr., 22.6 km/hr., and 36.3 km/hr. respectively.

$$v = 2\,\pi\, r_d\, n_r = 0.38\, n_r\, r_d \qquad (4)$$

Where

"v" The vehicle speed [km/h]

"n_r" The rotational speed [rpm]

"r_d" The tire dynamic radius of the vehicle [m]

3.3 Brake force measurements and calculations

In this work the brake force is measured by load cell. As shown in Fig.13 and Fig.14, the load cell is connected between the brake system and the test rig to measure the brake force. To investigate the variation of brake force trend and the mean brake force at different operation and design parameters as function of time. The output volt of the load cell is linked to Data Acquisition System via amplifier to amplify the load cell volt. This Data Acquisition System is a computer electronic card with four input channels. The load cell, amplifier, and data acquisition are calibrated experimentally into the test rig to obtain the best results. The actual brake force is calculated according to the equation below.

$$F_{Br} = R_{total} - R_{static} \qquad (5)$$

"F_{Br}" the actual brake force

"R_{total}" the total brake force measured by the load cell during each test conditions

"S_{static}" the brake force measured by the load cell for each test conditions static condition.

3.4 Brake shoe factor and friction coefficient calculations

The brake shoe factor "C^*" is the important factor that evaluates the brake performance. The increase of brake force "F_{Br}" at constant low applied force "F_{app}" leads to the increase of brake shoe factor which indicates the increase of wedge self amplification action. Friction coefficient "μ" is the main parameter of generating the brake force. The variation of brake force and the brake shoe factor is a result of the friction coefficient variation, so the friction coefficient is calculated.

By measuring the applied force" F_{app} ", brake force" F_{Br} ", and determination of wedge inclination angle "α"; brake shoe factor and friction coefficient "μ" could be calculated as follows:

$$C^* = \frac{F_{Br}}{F_{app}} \qquad (6)$$

$$C^* = \frac{2\mu}{\tan\alpha - \mu} \qquad (7)$$

$$\mu = \frac{C^* \tan\alpha}{2 + C^*} \qquad (8)$$

3.5 Temperature measurement

In this section, the effect of temperature is considered to investigate its effect on brake performance at the different operation and design parameters. A thermocouple had been used to measure temperature. This thermocouple is fixed in the brake pad to measure the friction temperature at the contact area between the brake pad and brake disc. The output volt of this thermocouple is small millivolts, so an amplifier is designed and constructed in the laboratory to increase the magnitude of the thermocouple output volt.

The output volt of the thermocouple amplifier is connected to the Data Acquisition system as show in Fig.13 and Fig.14. This thermocouple with the amplifier and the Data Acquisition card had been calibrated to insure the best results.

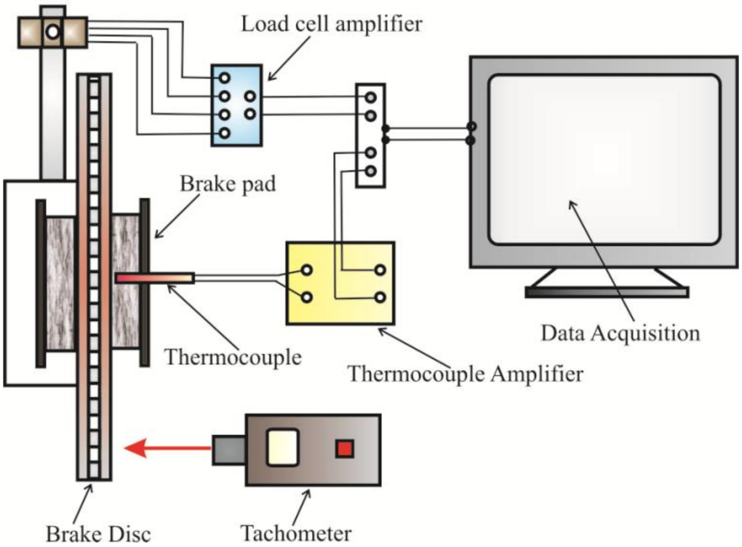

Fig.13Schematic sketch of Brake force, Temperature and Speed measurement instrumentations

Fig.14Brake force and Temperature measurement instrumentations

IV. Force Analysis and Calculations

The wedge disc brake is a self-amplified disc brake. Its principle based on the use of self-amplification action in the disc brake using a mechanism of wedge. The applied force moves the wedge cotter down toward the brake pad and the wedge assembly causing the wedge (cotter) force " S ". The wedge force varies according to the applied force and the wedge inclination angle “α”. The wedge force affecting the wedge cotter analyzes into two components; one component acts on the horizontal direction and the other acts on the vertical direction. The wedge force component in the horizontal plane equals the normal force “N”affects the brake pad. Multiplying the normal force with the friction coefficient “μ”, equals the brake force “F_{Br}”. The wedge forces analyses shown in Fig.15.

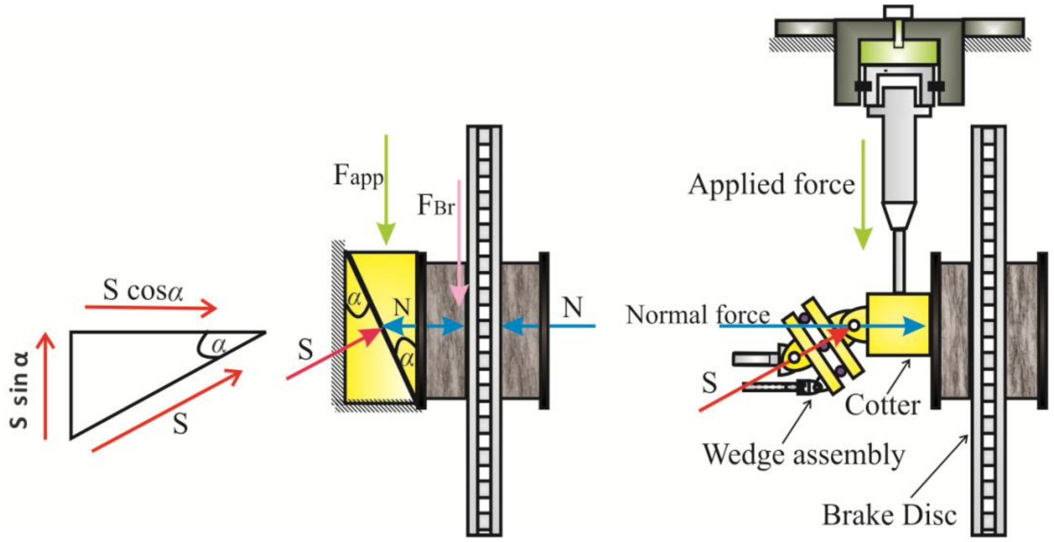

Fig.15 Schematic sketch of wedge forces analysis

$$\sum F_X = 0\,,\ N = S\cos\alpha \tag{9}$$

$$\sum F_Y = 0\,,\ F_{Br} + F_{app} = S\sin\alpha \tag{10}$$

$$S = \frac{N}{\cos\alpha} \tag{11}$$

According to Coulomb's law of friction,

$$N = \frac{F_{Br}}{\mu} \tag{12}$$

$$F_{app} + F_{Br} = \frac{N\sin\alpha}{\cos\alpha} \tag{13}$$

$$F_{app} + F_{Br} = \frac{F_{Br}\sin\alpha}{\mu\cos\alpha} \tag{14}$$

$$F_{app} + F_{Br} = \frac{F_{Br}\tan\alpha}{\mu} \tag{15}$$

$$F_{app} = \frac{F_{Br}\tan\alpha}{\mu} - F_{Br} \tag{16}$$

$$F_{app} = F_{Br}\left(\frac{\tan\alpha - \mu}{\mu}\right) \tag{17}$$

$$\frac{F_{Br}}{F_{app}} = \frac{\mu}{\tan\alpha - \mu} \tag{18}$$

$$C^* = \frac{F_{Br}}{F_{app}} \tag{19}$$

$$C^* = \frac{\mu}{\tan\alpha - \mu} \tag{20}$$

Hence, the brake shoe factor "C^*"of wedge disc brake with floating caliper is:

$$C^* = 2\frac{F_{Br}}{F_{app}} \tag{21}$$

$$C^* = \frac{2\mu}{\tan\alpha - \mu} \tag{22}$$

V. Results and Discussion

The experimental results showed that the brake force variation with the applied force of conventional disc brake and wedge disc brake with inclination of 45° brake at vehicle speed of 22.6 km/hr. and dry condition is shown in Fig.16. It can be seen that, the increase of the applied force causes increase in the brake force of both conventional and wedge disc brake. The increase with the wedge disc brake is greater than with the conventional disc brake which could be attributed to the self-amplification effect of wedge disc brake. For both conventional and wedge disc brake there are fluctuations of the brake force that have no identical trend at each constant applied force. This fluctuation in the brake force is caused due to the variation of friction coefficient with the braking time where the friction coefficient depends on many parameters such as contact pressure and disc/pad interface temperature. The increase of normal force increases the disc/pad interface temperature which leads to the decrease of the friction coefficient. At low contact pressure the ambient oxides on the metal surface will control friction but very high contact pressure tend to reduce the friction coefficient. From Fig.17, it can be seen that, at applied force 400 N the mean brake force increases from 214.4 N of conventional disc brake to 794 N of wedge disc brake i.e. 3.7 times, at applied force 800 N the mean brake force increases from 429.9 N of conventional disc brake to 1592.2 N of wedge disc brake i.e. 271 %, at applied force 1200 N the mean brake force increases from 641.8 N of conventional disc brake to 2376.9 N of wedge disc brake i.e. 270%, At applied force 1600 N the mean brake force increases from 849.1 N of conventional disc brake to 3144.8 N of wedge disc brake i.e. 3.69 times.

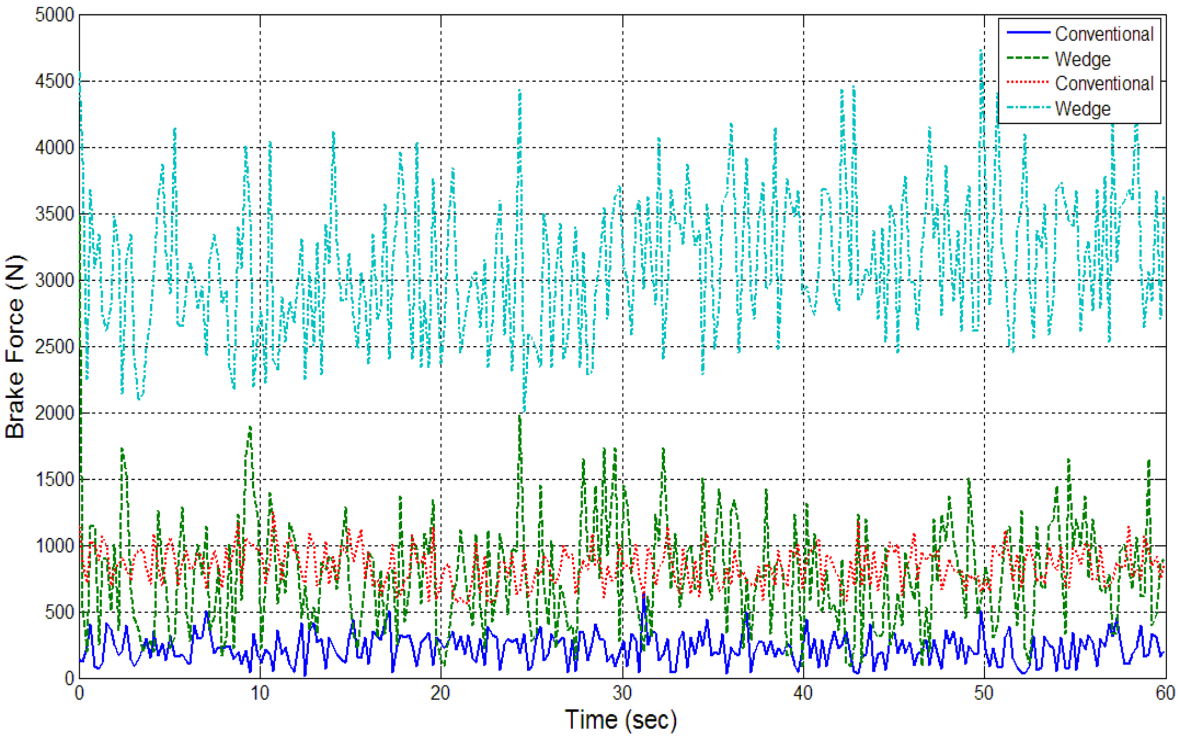

Fig.16 Variation of brake force with time for conventional and wedge disc brake at different applied force

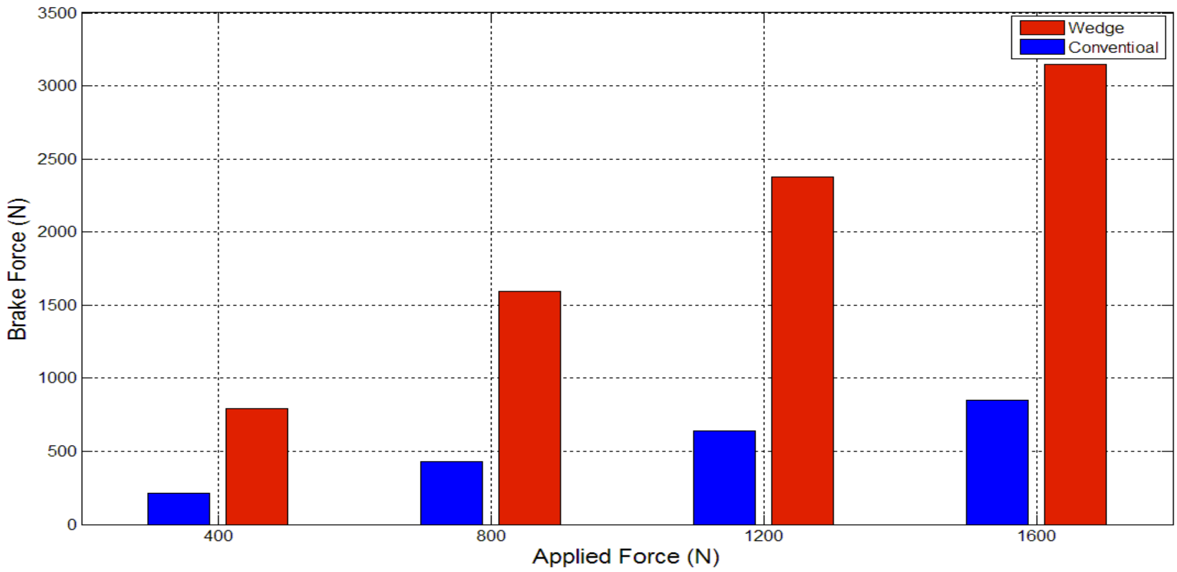

Fig.17 Effect of applied force on the mean brake force for conventional and wedge disc brake

VI. Conclusion

The experimental study of a new wedge brake is carried out using a designed brake dynamometer. Various operating conditions are applied to the brake assembly and several tests are recorded. The comparison between conventional and novel disc brake are performed. It is found that this novel wedge disc brake can multiply the brake force 3.7 times of conventional disc brake at the same conditions. The future work of this research is to investigate the brake performance of the new wedge disc brake using statistical methods in order to reduce the high hardware cost of experiments and save the time of tests.

REFERENCE

[1] Ouyang, H., Nack, W. V., Yuan, Y. and Chen, F. "Numerical analysis of automotive disc brake squeal: a review", *Int. J Vehicle Noise and Vibrations*, Vol. 1, Nos. 3-4, pp. 207-230, 2005.

[2] Nouby M., Abdo J., Mathivanan D. and Srinivasan K. "Evaluation of Disc Brake

Materials for Squeal Reduction' *Tribology Transactions, 54:* 644-656, 2011.

[3] Nouby M., and Srinivasan K., Simulation of structural modifications of a disc brake system to reduce brake squeal, *Proc. IMechE, Part D: J. Automobile Engineering*, Vol. 225, No. 5, 2011, 653–672.

[4] Trichés, M. J., Samir, N. Y. and Jordan, R. "Reduction of squeal noise from disc brake systems using constrained layer damping*", J. of the Brazilian Society of Mechanical Science and Engineering*, Vol. 26, pp. 340-348, 2004.

[5] Chen T. F., (2005), "*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.

[6] Amr M. M. Rabia, Nouby M. Ghazaly, M. M. M. Salem, Ali M. Abd-El-Tawwab. 2013 "An Experimental Study of Automotive Disc Brake Vibrations" *The International Journal of Engineering and Science (IJES),* Vol.2, Issue 01, PP. 194-200.

[7] Dunlap, K. B., Riehle, M. A. and Longhouse, R. E. "An investigative overview of automotive disc brake noise", *SAE Paper* 1999-01-0142.

[8] Bergman, F., Eriksson, M. and Jacobson, S. "Influence of disc topography on generation of brake squeal", *Wear*, pp. 225-229, 1999.

[9] Cunefare, K. A. and Graf, A. J. "Experimental active control of automotive disc brake rotor squeal using dither", *Journal of Sound and Vibration*, Vol. 250, No. 4, pp. 575-590, 2002.

[10] James, S. "*An experimental study of disc brake squeal*", PhD Thesis, Department of Engineering, University of Liverpool, May 2003.

[11] Nouby M. Ghazaly *"Study on Automotive Disc Brake Squeal Using Finite Element Analysis and Design of Experiments*" PhD. Thesis, Department of Mechanical Engineering, Anna University, India, 2011.

[12] Fieldhouse, J. D., Steel, W. P., Talbot, C. J. and Siddiqui, M. A. "Brake noise reduction using rotor asymmetric", *Proc. of IMechE International Conference Braking 2004, Professional Engineering Publishing Ltd*, pp. 209-222, 2004.