A STUDY ON THE EFFECT OF OUT-OF-ROUNDNESS OF DRUM BRAKE ROTOR ON THE BRAKING FORCE USING THE FINITE ELEMENT METHOD

by

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Thesis submitted in fulfilment of the requirements for the degree of Master of Science

JUNE 2007

ACKNOWLEDGEMENTS

I would like to express my truly gratitude and highest appreciation to my supervisor Assoc. Prof. Dr Zaidi Mohd Ripin for his precious guidance, support, training, advice and encouragement throughout my Master of Science study. My special acknowledgement is also dedicated to Universiti Sains Malaysia for providing me with the scholarship through the Graduate Assistant scheme during my study. I also would like to thank all the technicians for their valuable support and help.

I would like to express my gratefulness to my beloved family for their support especially to my parents who have done most excellent in providing me with education. I would also acknowledge my entire friends at School of Mechanical Engineering for their great and favorable support. A truly thankfulness are dedicated to all who involve in this project directly and indirectly. Thank you very much.

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NOMENCLATURES

a : distance between drum centre point and pivot point

b_1 : distance between drum centre point and cam contact point for leading shoe

b_2 : distance between drum centre point and cam contact point for trailing shoe

c : distance between drum centre point and tangential force

C^* : brake factor

C_1 : brake shoe factor (leading shoe)

C_2 : brake shoe factor (trailing shoe)

F : applied load force

F_f : friction force

F_b : braking force

h : distance between cam centre point and pivot point

K : Spring stiffness

M_a : applied moment

M_f : friction force moment

M_n : normal force moment

M_t : total moment

N : normal reaction force

P_i : pressure at node

P_{max} : maximum pressure (leading shoe)

P'_{max} : maximum pressure (trailing shoe)

P_n : normal pressure on small element

P_r : radial pressure on small element

R_1 : reaction force by friction (leading shoe)

R_2 : reaction force by friction (trailing shoe)

r : radius of brake lining

r_i : inner radius of drum rotor

r_l : length of the arm lever

r_o : outer radius of drum

S_1 : reaction force at cam or rolls point (leading shoe)

S_2 : reaction force at cam or rolls point (trailing shoe)

T : torque

T_{2L} : torque for double leading brake

w : width of brake lining

W : applied load

x_{cp} : centre of pressure

μ : friction coefficient

α : angle of brake lining

ϕ : angle of element on brake lining measured from pivot point

θ_1 : angle between pivot point and starting point of lining.

θ_2 : angle between pivot point and ending point of lining

LIST OF PUBLICATIONS & SEMINARS

KAJIAN KESAN KETIDAK-BULATAN PEMUTAR GELENDONG BREK TERHADAP DAYA BREK DENGAN MENGGUNAKAN KAEDAH UNSUR TERHINGGA

ABSTRAK

Rekabentuk kasut depan-kasut mengekor digunakan dengan meluas sebagai brek belakang untuk kereta penumpang dan lori ringan. Gelendong brek biasanya dianggap bulat sepenuhnya dan ini dilaporkan dalam mana-mana hasil kerja yang telah dilakukan oleh penyelidik. Sebelum ini, kesan sebenar terhadap kebulatan gelendong dengan daya brek belum lagi dikaji. Dalam kajian ini, gelendong brek motosikal diuji kebulatannya dan dijalankan ujian daya brek serta ujian mampatan bahan pelapiknya. Ujian kebulatan dilakukan dengan menggunakan alat ujian kebulatan Mitutoyo RA-100 untuk mendapatkan profil kebulatan. Manakala ujian daya brek dijalankan dengan menggunakan mesin ujian semesta. Ujian mampatan bahan pelapik juga dijalankan dengan menggunakan mesin ujian semesta untuk menentu-sahkan nilai modulus keanjalan Young. Analisis kaedah unsur juga digunakan dalam kajian ini. Hasil keputusan menunjukkan gelendong brek adalah tidak bulat sepenuhnya manakala keputusan ujian brek pula menunjukkan daya brek yang berubah-ubah. Hubungan antara hasil experimen dan kaedah unsur terhingga menunjukkan hubungan yang baik dengan nilai pekali Pearson 0.828. Hasil keputusan menunjukkan bahawa bentuk gelendong memberi kesan kepada taburan tekanan sentuhan, daya brek dan juga faktor brek. Daya brek dan faktor brek meningkat dengan daya yang dikenakan. Nilai brek faktor dikira dalam julat 0.5 ke 2.0 yang mana hampir kepada nilai teori faktor brek 2.0 bagi pekali geseran 0.3. Dengan ini boleh disimpulkan bahawa kebulatan mempengaruhi daya geseran brek yang terhasil yang mana akan mempengaruhi nilai pekali brek.

A STUDY ON THE EFFECT OF OUT OF ROUNDNESS OF DRUM BRAKE ROTOR ON THE BRAKING FORCE USING THE FINITE ELEMENT METHOD

ABSTRACT

The leading-trailing shoe design is used extensively as rear brake on passenger cars and light weight pickup trucks. In the drum brake, the drum is assumed to be perfectly round and this is the case as reported by simulation work done elsewhere. The true effect of roundness of the drum on the brake force has not been investigated. In this study, a brake drum of a motorcycle was subjected to the roundness test, the brake force test and the lining compression test. Firstly, the roundness test was carried out using the Mitutoyo Roundtest RA-100 to determine the roundness profile of drum. Then the brake force test was performed using the Universal Testing Machine (UTM). The lining compression test was also carried out by using Universal Testing Machine to verify the Modulus Young of lining. The finite element analysis was also performed to investigate the relationship between roundness and brake force. The results showed that the drum is not perfectly round and the brake force showed changes as the drum rotated. Correlation between experimental and finite element analysis of the brake forces show a good relationship with the Pearson coefficient of 0.828. The analysis and correlations showed that the shape of the brake drum affects contact pressure distribution, brake force and brake factor. The brake force and the brake factor increases with the applied load. The brake factor values were found to be in the range of 0.5 to 2.0 which is close to the theoretical brake factor value of 2.0 for $\mu=0.3$. It can be concluded that the roundness effect influenced the contact pressure distribution. The contact pressure distribution affected the brake friction force produced and this then affects the overall brake factor.

CHAPTER ONE
INTRODUCTION

1.0 Background

Drum brakes were the first types of brakes used on motor vehicles. Nowadays, over 100 years after the first usage, drum brakes are still used on the rear wheels of most vehicles. The drum brake is used widely as the rear brake particularly for small car and motorcycle. The leading-trailing shoe design is used extensively as rear brake on passenger cars and light weight pickup trucks. Most of the front-wheel-driven vehicles use rear leading-trailing shoe brakes. Such design provided low sensitivity to lining friction changes and has stable torque production (Limpert, 1999).

The brake drum of a motorcycle is usually made from cast aluminum and essentially a cylinder sandwiched between the wheel rim and the wheel hub. Within the drum are brake shoes lines with friction material. The brake shoes are pressed against the inside of the drum surface by a cam or actuators inside the wheel cylinders. The inside surface of the drum is acted upon by the linings of the brake shoes. When the brakes are applied, the brake shoes are forced into contact with the inside surface of the brake drums to slow the rotation of the wheels.

The drum may appear to be perfectly round to the eye and it appears that they have a constant diameter when measured with a vernier or micrometer but it is not perfectly circular. The drum may have lobes and bores which makes it not really round and can affect the function due to the manufacturing defects and errors. The component would function well for a short time but wear at some point and may result in noise and vibration.

In this study a typical motorcycle drum brake with a diameter of 110mm is selected for analysis. A flat cam mechanism is used to provide force on one end of the shoes while the other end of the shoes is pivoted.

1.1 Problem statement

Brakes are machine elements that absorb kinetic energy in the process of slowing down or stopping a moving part. Brake capacity depends upon the unit pressure between the braking surfaces, the coefficient of friction, and the ability of the brake to dissipate heat equivalent to the energy being absorbed. In braking system, drum brake is used mostly for automotive application. During the braking process, the forces and pressures in a drum brake are difficult to determine because of the manner in which the shoe contacts the drum. Since the shoe is long the contact pressure will vary along the mating surface, contact pressures are distributed along the mating surface will influence the stability of the brake system and lead to squeal noise problem. Many researchers found that pressure distribution is difficult to obtain directly from experiment (Day et al, 1984 & 1991). Recently the finite element method is used to tackle the problem. In the drum brake, the drum is assured to be perfectly round and this is the case as reported by simulation work done elsewhere. The true effect of roundness of the drum on the brake force has not been investigated.

1.2 Research scope and objective

The scope of study is mainly about determining roundness of the brake drum rotor and its effect on the brake force. The roundness test is to determine the roundness of the brake drum rotor using the Mitutoyo Roundtest RA-100. The brake force test and the compression test are carried out using the Universal Testing Machine (UTM).

This project is to investigate the relationship of the brake force and the roundness of the brake drum. The objectives of this research can be broken into four:

i. To obtain the brake force data from experiments.
ii. To develop a finite element model of drum brake system.
iii. To measure the roundness for the brake drum rotor
iv. To study the effect and relationship between brake force and roundness by experimental and finite element method.

A motorcycle brake drum is chosen for this analysis because the geometry is relatively simple and produces a relatively low braking torque which requires less torque to be supplied in the laboratory work. The purpose of this study is to determine the contact pressure distribution and braking force affected by out of roundness on brake drum rotor using finite element method and experimental.

1.3 Thesis outline

The thesis is presented in five chapters including introduction, literature review, methodology, results and discussion and finally conclusion.

Chapter One consists of introduction and background of the analysis. It includes research objectives and thesis outline. Chapter Two consists of the literature review. The previous analysis regarding to the brake system particularly on drum brake are reviewed and discussed. Chapter Three describes the methodology used in this analysis including experimental and finite element modeling. Chapter Four shows the results and also the discussions on the relationship between brake force and roundness. Chapter Five presents the conclusion and recommendation of the present analysis.

CHAPTER TWO
LITERATURE REVIEW

2.0 General overview

Automotive friction brakes are classified into drum and disc brake. The drum brakes are further classified into simplex, duplex and duo servo drum brakes (Halderman and Mitchell, 2004). Drum brakes use brake shoes that expand the brake linings to the drum in radial direction. Drum brakes are cheaper and less complex and their effectiveness is higher because of the self-amplification system. This is because the drum brake design offers a self-energizing action that pushes brake linings tightly against the drum. Drum brakes also have disadvantages that they are more sensitive to brake fade because they are not capable to dissipate the generated heat. Meanwhile, disc brakes have lower sensitivity to brake fade but they are more expensive and have lower brake effectiveness when compared with the drum brake (Mahmoud, 2005). At the same applied force, drum brake will provide higher braking force compared to disc brake because of its self-energizing design. Automotive friction brakes use either or both drum and disc brakes. Drum brakes mostly use internal expanding shoes with brake linings surface that contact with drum-rubbing surface. Meanwhile disc brakes use shoes that contact to small portion of disc-rubbing surface. Disc brakes provide faster cooling because of their larger surface exposed to the outside air.

Drum and disc rotor are fabricated as round and circular element for brake system. Farago and Curtis (1994) have defined perfectly round as having all points of its perimeter equidistant from the axis. In the terms of a issued standard by ANSI, circularity (roundness) is a condition of a surface of revolution where for a cylinder or cone, all points of the surface intersected by any plane perpendicular to a common axis are equidistant from that axis and for a sphere, all points on the surface intersected by any plane passing through a common center are equidistant from that center (Farago and Curtis, 1994).

2.1 Friction material

Brake lining friction materials can be classified into organic, metallic and carbon. Asbestos linings were called organic which are composed of many ingredients such as asbestos fiber, phenolic resin, organic friction modifiers, inorganic friction modifiers, abrasive particle and carbon (Halderman, 2004). General classification of brake friction materials used in brake industries are metallic (predominantly metallic such as steel fibers), semi-metallic (mixture of metallic and organic ingredients) and non-asbestos organic (predominantly organic such as mineral fibers) (Chan, 2004). The gradual phasing-out of asbestos lining in automotive friction material has introduced new friction material such as semi-metallic, non-asbestos, carbon fiber and ceramic friction material. The modern brake linings composed of many different ingredients including binder, structural materials, fillers and frictional additives (Eriksson et al, 2002).

Semi-metallic linings have some disadvantages (Gilles, 2005):

- As semi-metallic wear, a dark brown dust develops on the front wheel.
- They have a tendency to make noise.
- They tend to wear rotors more quickly.

All types of friction materials give different and various coefficients. The sensitivity to the coefficient of friction is still a problem for all brakes type especially the drum brake.

An experimental study by Severin and Dorsch (2001) has showed that the coefficient of friction changes with the number of brake applies for drum brake which the average is about around 0.3. Figure 2.1 shows these variations which are limited by 0.25 to 0.47.

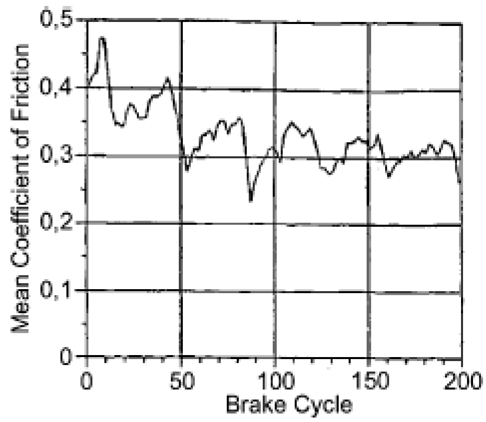


Figure 2.1: Coefficient of friction variations against the number of brakes with the drum brake (Severin and Dorsch, 2001)

According to Halderman and Mitchell (2004), the best coefficient of friction between brake lining and cast iron drum brake is 0.4 for static coefficient and 0.3 for kinetic coefficient of friction.

2.2 Drum brake construction

The main advantages of drum brakes is that they can apply more stopping power for a given amount of force applied to brake pedal compared to disc brake system. This is because the drum brake design offers a self-energizing action that pushes brake linings tightly against the drum. Moreover, some new drum brake designs use a method called servo action that enables one brake shoe to help the other for increased stopping power (Gilles, 2005).

Generally, the brake drum is a heavy flat-topped cylinder usually made from cast iron. The brake drum is sandwiched between the wheel rim and the wheel hub. The inside surface of the drum is acted upon by the linings of the brake shoes. When the brakes are applied, the brake shoes are forced into contact with the inside surface of the brake drums to slow the rotation of the wheels.

Drum brakes can be divided into external band and internal shoe brakes. Typical internal shoe brakes can be divided into leading-trailing (simplex), two-leading (duplex) or duo servo brakes, according to the shoe arrangement (Orthwein, 2004). Internal shoe brakes can be further divided according to the shoe abutment or anchorage into shoes supported by parallel or inclined sliding abutment or pivoted shoes. The brake shoe actuation may be grouped into hydraulic wheel cylinder, wedge, cam, screw and mechanical linkage actuation.

For the leading-trailing and the two-leading shoe brakes, each shoe has its own anchorage to the backing plate. The basic components include shoes, wheel cylinder, automatic adjuster and parking brake mechanism of a leading-trailing shoe brake. The leading-trailing shoe design is used extensively as rear brake on passenger cars and light weight pickup trucks. With a few exceptions, front-wheel-driven vehicles use rear leading-trailing shoe brakes. The advantage of this arrangement is a low sensitivity to lining friction changes and stable torque production (Halderman and Mitchell, 2004).

The simplex drum brake has two shoes performing differently. Based on the energy generated, one is called the self amplifying shoe or leading shoe and the other is the self-debilitating shoe or trailing shoe. As the forward or leading shoe contacts the drum, the drum attempts to rotate the shoe along with it but the shoe cannot rotate because of it is fixed in by an anchor or pivoted. So the drum rotation energizes the shoe by forcing it outward and wedging it tightly against the brake drum. This action produces a form of self amplification system brake. The drum also attempts to rotate the trailing shoe when it contacts the drum which is the far end of the shoe is not solidly anchored so drum rotation will de-energizes the shoe by forcing it inward away from the brake drum. A leading shoe is always energized by drum rotation meanwhile a trailing shoe is always de-energized by drum rotation (Halderman and Mitchell, 2004). Figure 2.2 shows the construction of simplex or leading-trailing brake.

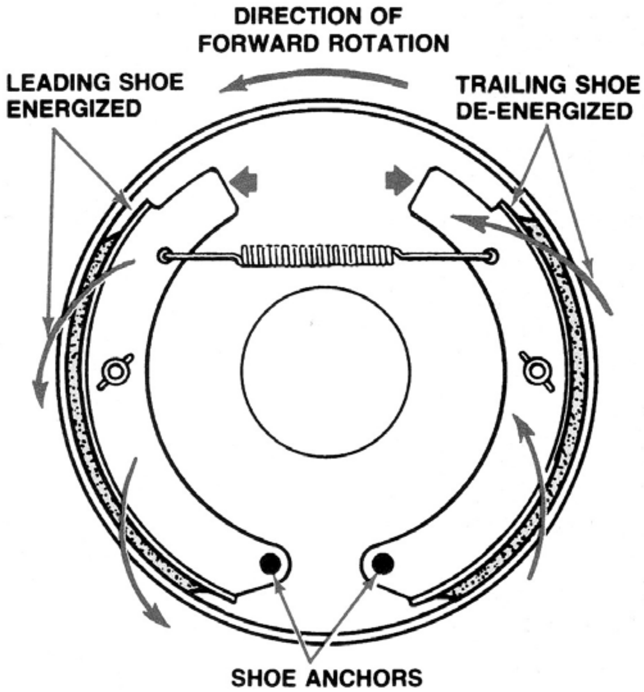


Figure 2.2: A design of simplex or leading-trailing drum brake (Halderman and Mitchell, 2004).

The duplex drum brake has both shoes as leading shoes which are self-amplified. The duplex or double leading brake uses two separate single-piston cylinder to actuate the shoes. The constructions of the duplex drum brake as shown in Figure 2.3. When in the case of inverse direction of rotation, both shoes become self-debilitating and act as double trailing shoes and this is one of the disadvantages of duplex drum brake. An advantage of duplex drum brake is the practically equal brake lining wear on both shoes and the significantly higher internal transmission ratio in comparison to simplex drum brake. Using two leading shoes, brake factor of $C^* \approx 3.0$ are achieved, although these figure cannot be held constant throughout a long period of braking due to brake's susceptibility to fading (Bosch, 1993).

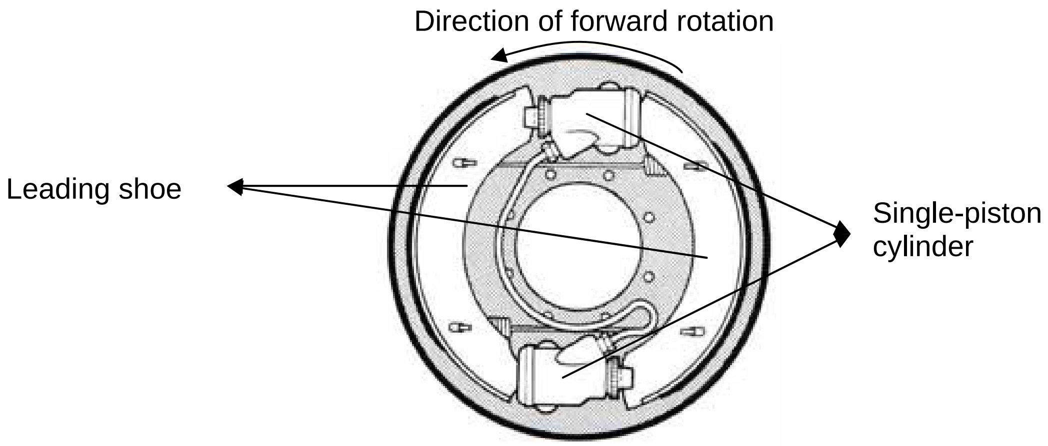


Figure 2.3: Duplex drum brake with two single-piston wheel cylinder (Limpert, 1999)

In the duo-servo drum brake, both shoes are self-amplified same as duplex drum brake but it uses only one anchor and one double-piston wheel cylinder to actuate both shoes. The self amplifying force from one shoe (primary shoe) is transferred to the other shoe (secondary shoe) through the adjusting screw or it can be said that one shoe serves the other to increase application force. As the primary shoe makes contact with the drum, it rotates with the drum because its end is not directly anchored and forces the adjusting link and the secondary shoe until the secondary shoe touched firmly against the anchor. The anchor pin prevents any further shoe movements. Although the wheel cylinder attempts to push the secondary shoe outward the total force from primary shoe and the frictional force developed by secondary shoe are much greater than application force by hydraulic pressure. When the duo-servo drum brake is applied with the vehicle moving in reverse, the primary and secondary shoes switch roles (Halderman and Mitchell, 2004). The main advantage of the duo-servo brake is its high brake torque or brake factor for a given input force from the wheel cylinder pushing the shoes apart. The major disadvantage of the duo-servo brake design is its high variation in brake torque for small changes in lining friction coefficient. This drastic unintended increase in rear brake torque may cause premature rear brake lockup and loss of vehicle stability during braking. Figure 2.4 illustrates the design of duo-servo drum brake. The primary shoe on the left exerts a force on the secondary shoe on the right.

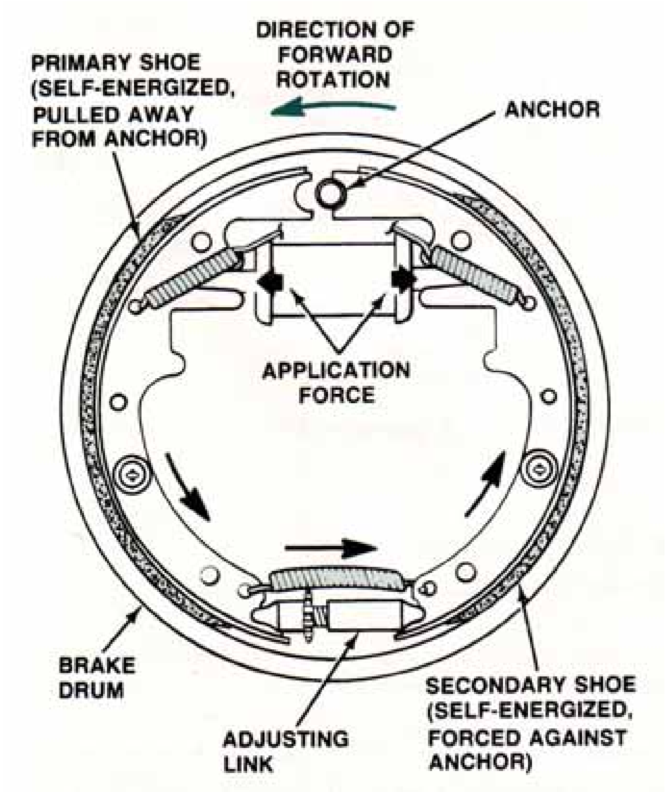


Figure 2.4: Schematic diagram of duo-servo drum brake operation (Halderman and Mitchell, 2004).

The S-cam and flat cam brake uses the leading-trailing shoe design. The shoes are applied mechanically by rotation of a cam shaped in an S-form or flat cam. The main part of this design are the leading and trailing shoes, S-cam, automatic slack adjuster and air brake chamber. Rotation of the cam pushes the roller and tips of the shoes apart. Due to the cam geometry, the application force against the leading shoe will have a smaller lever arm relative to the pivot anchor of the leading shoe than that of the trailing shoe, resulting in nearly uniform wear of both the leading and trailing shoe and therefore, long lining life. This is also the reason that the standard leading-trailing shoe brake torque must be modified for S-cam brakes. S-cam brakes are simple and rugged. It is strong and designed to be used in difficult conditions. Therefore it can be inspected and maintained easily. Their major disadvantage is in stop fade, a limited brake factor and the need for tight adjustment (Bosch, 1993).

Wedge brakes use either the leading-trailing or the two-leading shoe design. In the wedge brake, a wedge is forced between the tips of the shoe, forcing the linings against the drums. The leading-trailing shoe brake uses one brake chamber, the two-leading shoe brake uses two. One benefit of wedge brakes is the integral automatic adjuster which ensures optimum drum-to-lining clearance. Another advantage over S-cam brakes is the higher brake force and more compact size and lower weight (Bosch, 1993).

For cooling purposes, the drums are usually covered with fins on their outer surfaces. Unlike disc brakes, they cannot be cooled internally because water will enter through the air vent cooling holes. If this happens, then braking force would be greatly reduced (Gilles, 2005).

2.3 Brake effectiveness

Brake effectiveness is defined based on the ratio of the total friction force or moment on the shoes to the applied force or moment at the tip of the shoes. It is also called as brake shoe factor or brake factor and symbol as C^* (Mahmoud, 2005).

Schematic of internally expanding shoe brake of the simplex drum brake is shown in Figure 2.5 with one shoe to derive the equation for calculating the moment based on Mubeen (1995). The shoe is anchored at A while a force F is applied at B. The vertical distance between A and B is b while the anchor pin A is located at a radial distance a from the centre of the drum. The drum is rotating clockwise. The lining on the shoe subtends an angle θ at the centre and its nearest edge is at an angle θ_1 from the anchor A. As the force F tends to close the shoe in the inside of the drum, a force of reaction is applied by the drum on the shoe causing the shoe to rotate about its anchor *A*. This force is perpendicular to any line joining *A* with the element of the shoe considered.

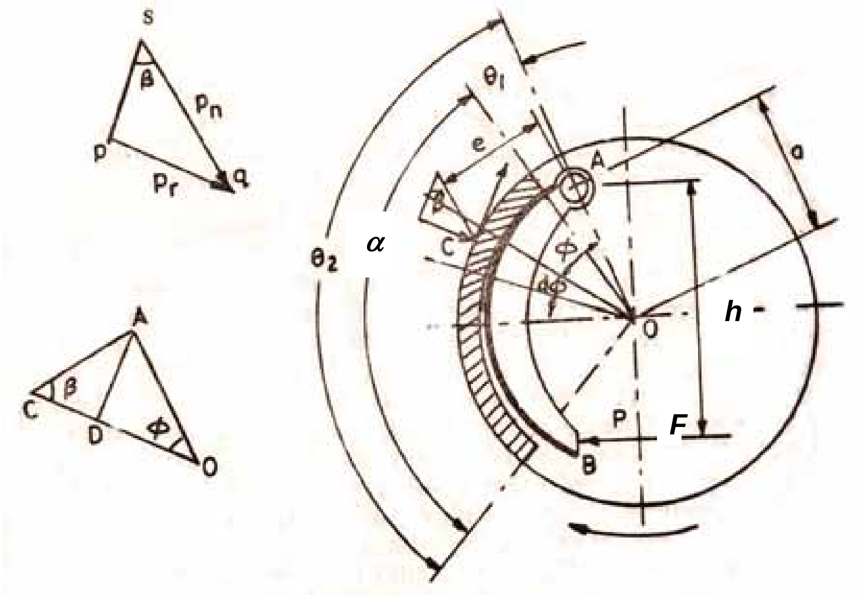


Figure 2.5: Schematic of an internally expanding shoe brake (Mubeen, 1995)

For this purpose a small element of shoe lining subtending at an angle $d\phi$ at the centre and at angle of ϕ from the radial line of anchor *A* is chosen. The pressure on this element at point *C* due to rotation about anchor *A* is perpendicular to the radius from *A* and proportional to the radius *AC*=*e*. Let this pressure be P_n. The radial pressure on the small element $rd\phi$ is denoted by P_r. P_n and P_r are shown in Figure 2.5 by triangle *pqs* and angle *psq* is denoted by β.

$$p_r = p_n \sin\beta$$

but $\quad p_n = Ke \quad$ (*K* being a constant)

$$\therefore \quad p_r = Ke\sin\beta \qquad \text{… (2.1)}$$

With *O* as centre of the drum, *OC* is the radius of the drum and *OA* is distance between *O* and the shoe anchor *A*, denoted by *a*. Triangle *OAC* in figure 2.5 presents geometrical orientation of $OA=a$, $OC=r$ and $AC=e$. AD is the perpendicular on OC from A. Then,

$$AD = e \sin \beta = a \sin \phi \qquad \text{… (2.2)}$$

Using eq. (2.2) in eq. (2.1), $p_r = Ka \sin \phi$

If the width of the shoe lining is w, the area of element

$$dA = wrd\phi$$

Thus the normal force on elemental area

$$dN = p_r dA = Kwar \sin \phi d\phi$$

The friction moment that is applied on the brake is calculated as integration of μdNr .

$$M_f = \mu \int_{\theta_1}^{\theta_2} Kwar^2 \sin \phi d\phi$$

$$M_f = \mu Kwar^2 \left(\cos \theta_1 - \cos \theta_2 \right) \qquad \text{… (2.3)}$$

Remembering that pressure lining made of any material will have a maximum permissible pressure, p_{max}. From (ii) it can be seen that p_r will become maximum when $\sin \phi = 1$ or $\phi = \pi/2$

$$\therefore \qquad p_{\max} = Ka$$

$$K = \frac{p_{\max}}{a}$$

So the friction moment;

$$M_f = p_{\max} \mu wr^2 \left(\cos \theta_1 - \cos \theta_2 \right). \qquad \text{… (2.4)}$$

In an actual brake there are two shoes as shown in Figure 2.6 for simplex drum brake. If the same force acts on both the shoes as the free end, the maximum pressure developed in trailing shoe is less than the leading shoe. This results in a smaller friction torque on trailing shoe. In case of leading shoe the friction moment M_f and moment due to actuating force P_b are in the same direction about anchor pin *A*. Contrarily, in case of trailing shoe the friction force moment M'_f and normal force moment M'_n are in the same direction and appose the moment due to actuating force about A. M_n, M_f, M'_n and M'_f can be calculated by usual consideration of small shoe element upon which the normal force *dN* is acting (Mubeen, 1995).

For leading shoe;

$$M_n = p_{\max} war\left(\frac{\theta_2 - \theta_1}{2} - \frac{\sin 2\theta_2}{4} + \frac{\sin 2\theta_1}{4}\right) \quad \text{… (2.5)}$$

$$M_f = p_{\max} \mu wr\left[r - \cos\theta_2 + r\cos\theta_1 - \frac{a}{2}\left(\cos^2\theta_2 - \cos^2\theta_1\right)\right] \quad \text{… (2.6)}$$

$$M = M_n + M_f \quad \text{… (2.7)}$$

And for trailing shoe;

$$M'_n = p'_{\max} war\left(\frac{\theta_2 - \theta_1}{2} - \frac{\sin 2\theta_2}{4} + \frac{\sin 2\theta_1}{4}\right) \quad \text{… (2.8)}$$

$$M'_f = p'_{\max} \mu wr\left[- \cos\theta_2 + r\cos\theta_1 - \frac{a}{2}\left(\cos^2\theta_2 - \cos^2\theta_1\right)\right] \quad \text{… (2.9)}$$

$$M' = M'_n + M'_f \quad \text{… (2.10)}$$

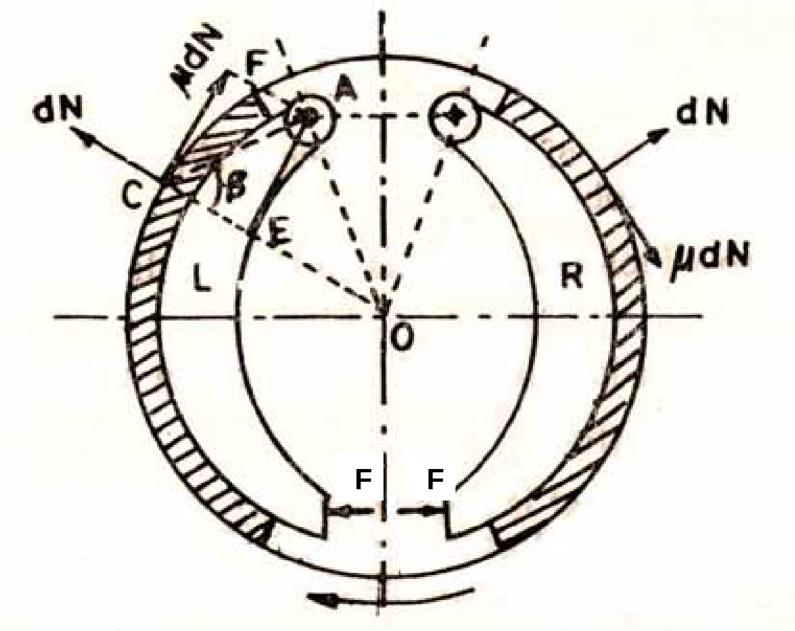


Figure 2.6: Two shoes on simplex drum brake (Mubeen, 1995).

Hence total moment of the brake;

$$M_t = M + M^{'}$$
$$M_t = \mu w r^2 (\cos\theta_1 - \cos\theta_2)(p_{\max} + p^{'}_{\max}) \quad \text{… (2.11)}$$

Regarding to Orthwein (2004), drum brake efficiency may be measured in terms of the ratio of the torque produced by the brake itself to the torque required to activate the brake, also known as the shoe factor; namely,

$$\frac{T}{M_a} = \frac{T}{M_n \pm M_f} \quad \text{… (2.12)}$$

An interesting design modification of brake was achieved by making both shoes as leading shoe as shown in Figure 2.7. If equal actuating forces each equal to P are applied at the free ends of shoes, the normal reaction N and friction force μN will act upon each shoe. Clearly the torque capacity of such brake will be twice that of left and right brake. This torque will be higher than that in case of one leading and one trailing shoe with equal actuating forces.

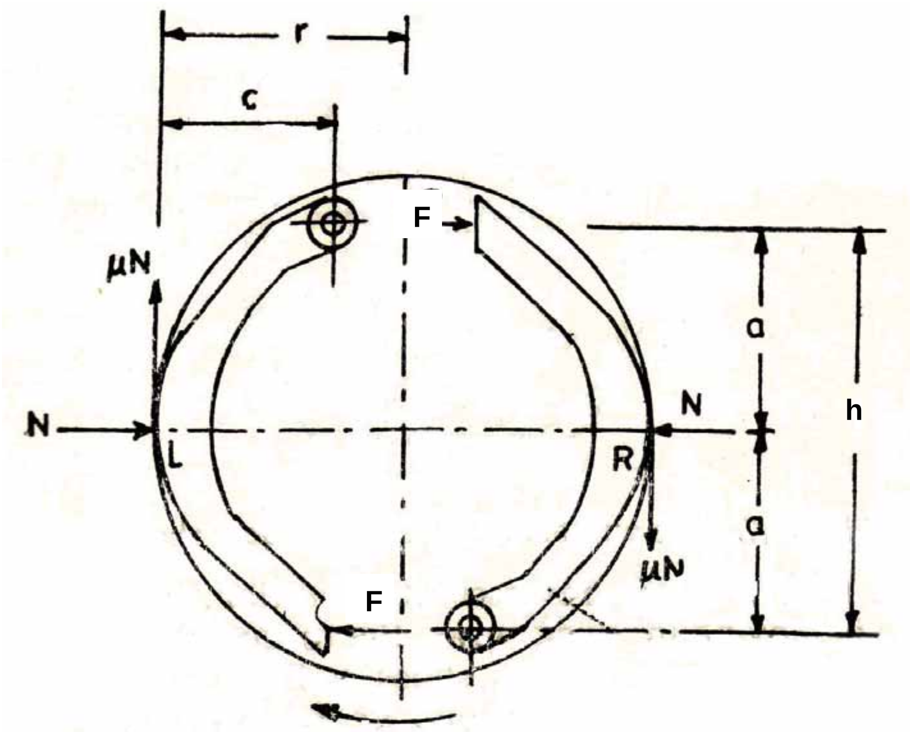


Figure 2.7: A brake with two leading shoes (Mubeen, 1995).

The torque capacity of brake for two leading shoes;

$$T = 2\mu Nr$$
$$T = \frac{2\mu Fhr}{a - \mu b} \qquad \text{… (2.13)}$$

This may be compared with the torque capacity of brake having one leading and other trailing shoe. That torque capacity of such double leading brake is;

$$T_{2L} = \frac{2\mu Fhra}{a^2 - \mu^2 b^2} \qquad \text{… (2.14)}$$

The brake factor C* as an assessment criterion for brake performance indicates the ratio of braking force to actuating force. This value takes into account the influence of the internal transmission ratio of the brake as well as the friction coefficient, which in turn is mainly dependent on the parameters such as speed, brake pressure and temperature. Figure 2.8 shows the value of brake factor for different types of brakes as function to the friction coefficient. From this figure it can be said that duo-servo brake has the highest brake factor meanwhile disc brake has the lowest brake factor. Drum brakes have higher brake factor rather than disc brake because of the self-amplification effect (Bosch, 1993). Drum brakes especially duo-servo drum brakes have high stopping power but they are also more sensitive to the friction coefficient. Although the disc brake produces lowest brake factor but it has lowest sensitivity to the friction coefficient. The drum brake shoe factor does not change linearly with the friction coefficient (Bosch, 1993).

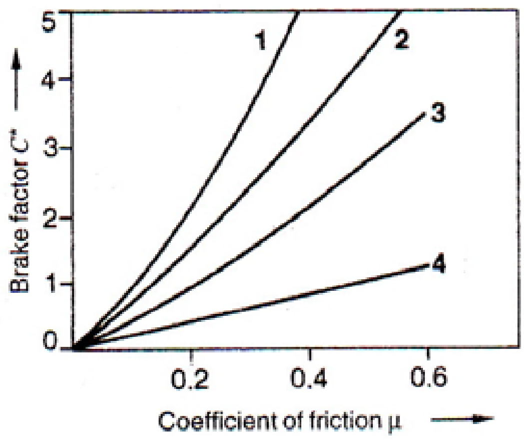


Figure 2.8: Brake factor C* as a function of the coefficient of friction for 1. Duo-servo drum brake 2. Duplex drum brake 3. Simplex drum brake 4. Disc brake (Bosch, 1993)

Hohmann et al. (1999) have done contact analysis for simplex S-cam drum brake as shown in Figure 2.9 to get the contact pressure distribution and brake parameter or brake factor *C**. Forces S_1 and S_2 act upon the rolls as applied forces meanwhile R_1 and R_2 are reaction force developed from friction process on each shoes. The solution of the integrals results in a relationship between applied force and the braking moment and the brake factor *C** is:

$$C^{*} = \frac{R_1 + R_2}{S_1 + S_2} = 2\frac{f(\alpha)\frac{a}{r}}{\left(f(\alpha)\frac{a}{r}\right)^2 - \mu^2}\frac{h}{r}\mu \qquad \text{… (2.15)}$$

With $$f(\alpha) = \frac{\alpha + \sin(\alpha)}{4\sin\left(\frac{1}{2}\alpha\right)}$$

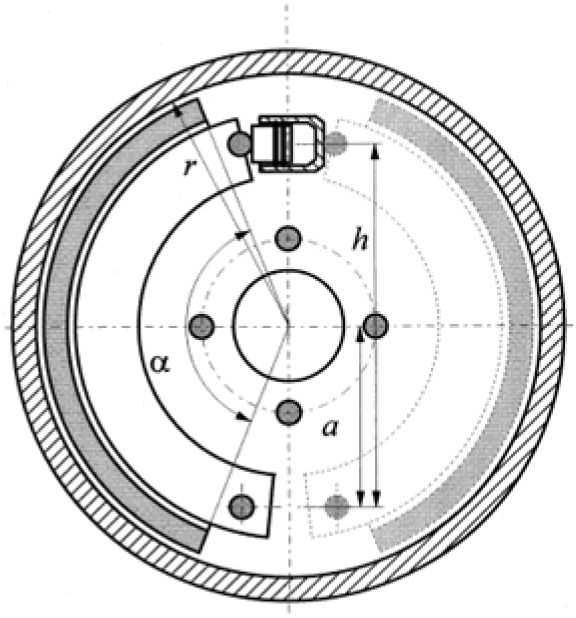
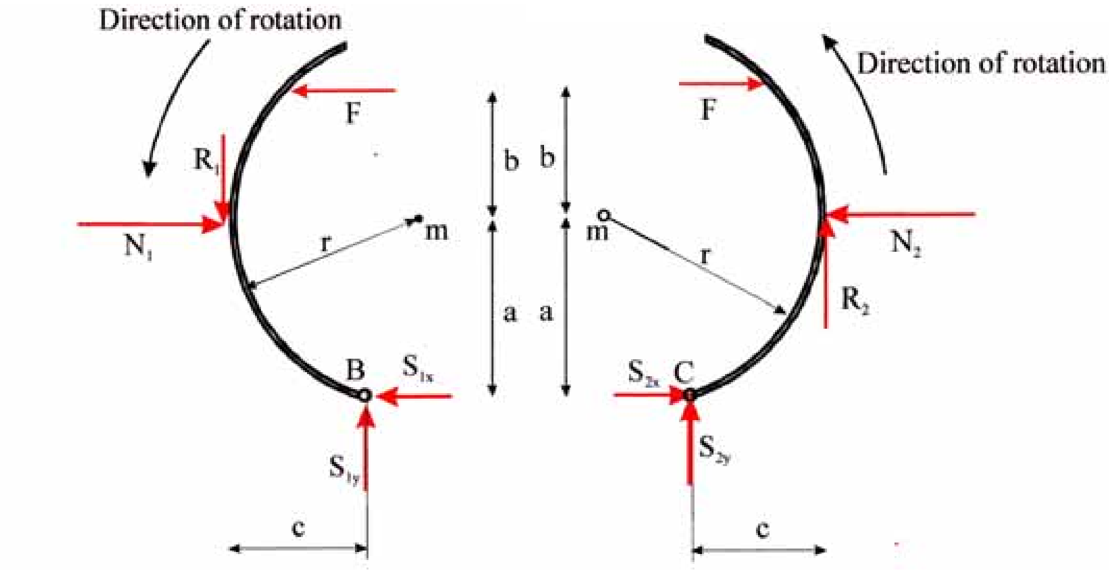


Figure 2.9: Schematic of simplex S-cam drum brake used by Hohmann et al (1999) in analysis to formulate brake parameter.

According to Hohmann et al. (1999), the average brake parameter *C** for the simplex drum brakes ≈ 2.0, for duplex drum brakes ≈ 3.0 and for duo-servo drum brakes ≈ 5.0.

Analysis by Mahmoud (2005) about brake factor for simplex drum brakes are simplified as shown in Figure 2.10 for each leading and trailing shoe. In this simplex drum brake, the applied force is produced either by hydraulic cylinder, electromechanical actuator or by cam.



(a) Leading shoe (b) trailing shoe

Figure 2.10: Simplified Free Body Diagram for simplex drum brake by Mahmoud (2005).

The shoe factor of the leading shoe C_1 is the ratio between brake force R_1 and the applied force F by evaluating the moment of the point B;

$$C_1 = \frac{R_1}{F} = \frac{\mu(a+b)}{a-c\mu} \qquad \text{… (2.16)}$$

Similarly for trailing shoe by evaluating the moment about point C will give the shoe factor C_2 as follows;

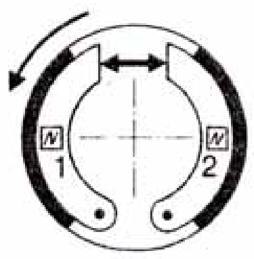
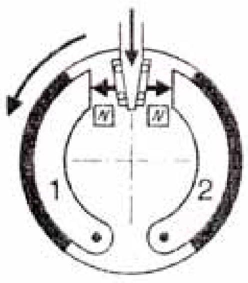
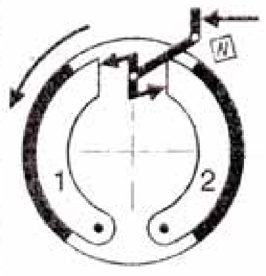
$$C_2 = \frac{R_2}{F} = \frac{\mu(a+b)}{a+c\mu} \qquad \text{… (2.17)}$$

An equivalent angle of inclination can be defined as:

$$\alpha = \tan^{-1}\left(\frac{a}{c}\right) \qquad \text{… (2.18)}$$

According to Bosch (1993), the brake factor for simplex drum brakes is combination between shoe factors for leading and trailing shoes based on the types of actuator whether by cam, wedge or hydraulic cylinder as shown in Table 2.1.

Table 2.1: Total brake factors for different designs of simplex drum brake (Bosch, 1993).

Design	Rotating shoe	Wedge	S-cam
Operating principle			
Brake factor	$C^* = C_1 + C_2$		$C^* = 4/(1/C_1 + 1/C_2)$
Brake shoes	1 Leading shoe, 2 Trailing shoe		

The duplex drum brake consists of two similar shoes act as leading shoes in forward rotation and as trailing in reverse rotation. Thus in forward motion, the generated brake force equals two times of the generated brake force for the leading shoe in simplex drum brake and twice that of generated brake force for trailing simplex drum brake when in backward motion (Mahmoud, 2005).

The shoe factor in forward motion;

$$C^* = \frac{2\mu(a+b)}{a-c\mu} \qquad \text{… (2.19)}$$

And the shoe factor when in backward motion;

$$C^* = \frac{2\mu(a+b)}{a+c\mu} \qquad \text{… (2.20)}$$

2.4 Contact analysis

Day and Newcomb (1984) have analyzed dissipation of frictional energy from the contact interface for disc brake. They showed that the generation of frictional energy at the friction material surface is proportional to the rate of work done as defined by local interface pressure, sliding velocity and coefficient of friction and therefore corresponds to the form of the interface pressure distribution. Under dynamic braking conditions interface pressure is seldom uniform and varies with time. Friction material and mating surface temperatures are not necessarily equal at adjacent positions because of the influence of contact resistance upon the transfer of frictional heat across the interface.

The distribution of contact pressure at the brake lining and drum interface is an important characteristic to consider in the design of drum brake to improve the efficiency and effectiveness. Day (1991) investigated the drum brake interface pressure distributions. The lining and drum interface pressure distribution is continually changed by lining wear and will always tends towards a uniform pressure distribution during a given brake application. However, because of flexure and deformation of the brake shoe, lining and drum, different brake applications will induce different wear distributions. In general, sliding abutment brakes where shoe movement is less controlled show a greater tendency towards a U-shaped pressure distribution pattern than pivoted shoe drum brakes as shown in Figure 2.11. However, pivoted shoe brakes are sensitive to the type of lining and drum pressure distributions especially at low actuation forces. Contact at each end of the lining arc or heel and toe contact is known both by conventional brake analysis and from experimental evidence which is to increase the brake shoe factor. Similarly contact over the central region of the lining arc or crown contact decreases the brake shoe factor.

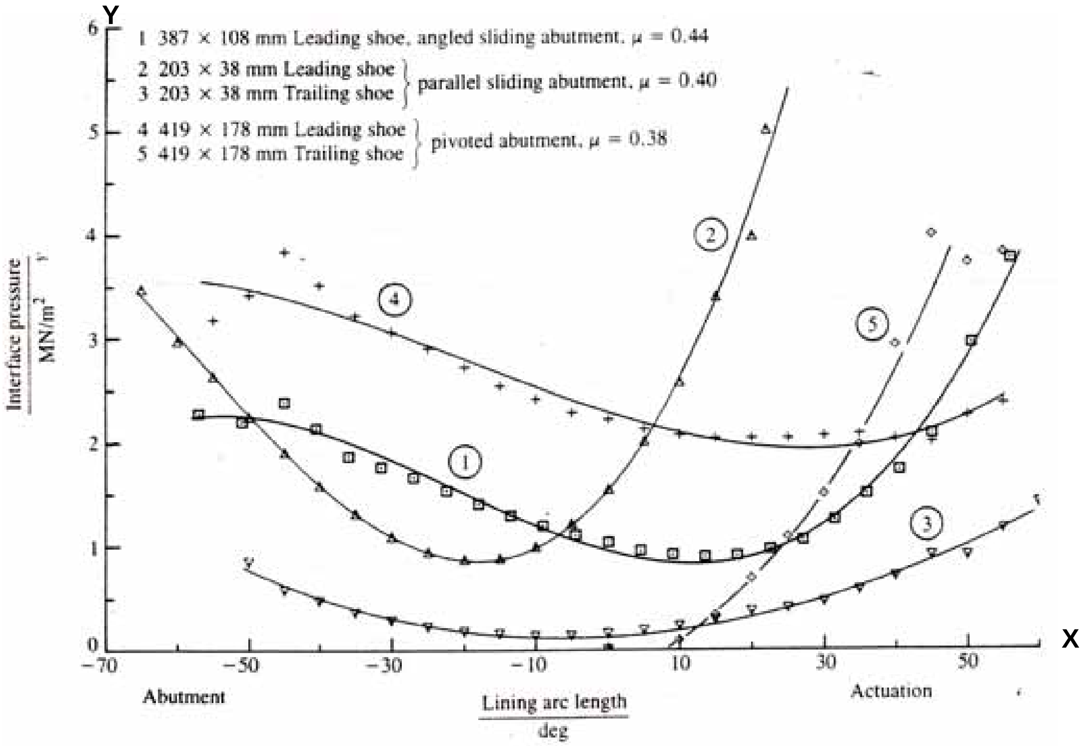


Figure 2.11: Drum brake pressure distributions (Day, 1991)

The importance of the pressure distribution at the brake friction interface in thermally related braking problems was discussed by Day et al. (1991). In both drum and disc brakes, the heat bands and their rate of displacement are mostly affected by the pressure distribution and heat deformation of the contacting elements during braking. High interface pressure can be generated locally on the brake friction surfaces even though the general level of braking duty is low. It is important to minimize this effect by aiming for as uniform pressure distribution as possible. This may be achieved by the careful design of robust brake components and by careful preparation of the mating surfaces to ensure geometric conformability. This shows that brake effectiveness and brake thermal also are influenced by contact pressure distribution pattern which is very important consideration during brake design.

Hohmann et al. (1999) has investigated the contact pressure on both the disc and drum brake using finite element method. The contact pressure distribution was found to be biased towards the leading edge of both S-cam brake shoes with the trailing shoe having peak contact pressure at the roll (cam side) and the leading shoe having peak contact pressure at the quarter end of the shoe. The contact forces have a non-uniform distribution which differs from the assumed distribution of the analytic solution. High peaks are apparent at the top end of the lining especially on the trailing shoe close to the roll. The distribution of the contact pressure tends to that of the old lining for analytic solutions. Nonetheless, the results were not verified with any experimental results.

Huang and Shyr (2002) analyzed a simplex drum brake using boundary element method. The results illustrated that the pressure distribution is continuous and biased towards the supporting point, however the pressure distribution curves did not differentiate between the leading and trailing shoes. If the deflections of the metal shoe and the lining plate and the thermal effect are neglected, the pressure distribution is more likely sinusoidal curve.

Huang and Shyr (2002) have summarized that in order to get the best design of drum brake with a more uniform pressure distribution:

- The location of the lining plate should be close to the supporting point.
- The lining plate should be thicker.
- The friction coefficient should be smaller.
- The modulus Young for brake shoe and lining should be smaller.
- The angle of actuation force should be larger.

Although the material with low friction coefficient can provide more uniform pressure but it should not be too small because it will lower the braking effect. The friction coefficient of material about 0.2 is an optimal selection for application.

Contact analysis has also been done by Rumold and Swift (2002) and Abu Bakar et al (2003) on disc brake to investigate the effect of the non-uniform pad loading. In order to examine pad loading characteristic, a multibody dynamics model has been utilized by Rumold and Swift (2002) to perform an investigation into caliper geometry changes that could be effectively improve pad loading uniformity. They found that by shifting the position of the fingers or the pistons to the trailing side, a rather uniform interface pressure distribution were established. In all cases, improvements to the overall inner and outer pad load distributions could be achieved. Meanwhile Abu Bakar et al (2003) has focused around the piston pad. They found that by making right connections between the piston head ring and the back plate in axial direction of the piston line pressure, the contact area can be increased and the contact pressure distribution can be improved. From that, the flexible multibody dynamic model may be selectively introduced to further refine the pressure distribution prediction for drum and disc brake.

A non-linear contact analysis of a leading-trailing shoe drum brake in conjunction with a complex eigenvalue using the finite element method is presented by Ioannidis et al. (2003) to predict the noise and vibrations behaviour affected by contact pressure. The FE model accurately captures both the static and pseudo-dynamic behaviour at the friction interface. Statically, contact takes place between the toe and crown area of both leading and trailing lining. As the brake starts to generate torque, the leading shoe centre of pressure moves towards the abutment while the respective trailing centre of pressure moves towards the actuation side, both following therefore the direction of the drum rotation. The obtained results suggested that the distribution of contact pressure was sensitive to the initial conditions.