THEORETICAL AND EXPERIMENTAL INVESTIGATIONS ON A NEW ADAPTIVE DUO SERVO DRUM BRAKE WITH HIGH AND CONSTANT BRAKE SHOE FACTOR

Zur Erlangung des akademischen Grades eines

DOKTORS DER INGENIEURWISSENSCHAFTEN (Dr.-Ing)

von der Fakultät für Maschinenbau

der Universität Paderborn

genehmigte

DISSERTATION

von

M*. Sc. Khaled R. M. Mahmoud*
aus El-Minia- Ägypten

ACKNOWLEDGEMENTS

This dissertation was elaborated during my stay as a member of Egyptian scholarships on the mechatronic and dynamic research group, university of Paderborn, Paderborn, Germany.

I would like to thank especially Prof. Dr-Ing. J. Wallaschek for his encouragement and guidance during my Ph. D. studies. He supported me far exceeded my expectations and contributed definitely to the success of this thesis. Also, he helped me to pursue this research.

I thank sincerely Prof. Dr-Ing. D. Zimmer, who nicely accepted to write the second expert's survey.

I want to thank sincerely to ass Prof. T. Sattel for his guidance for the theoretical part of the thesis and for his daily conversations and scientific discussions.

Special thanks to all members of mechatronic and dynamic research group for their help and assistance. I am particularly grateful to Dr-Ing H. Storck, Dipl-Ing R. Boer, Dipl-Ing M. Liekenbröcker and Dr-Ing T. Hemsel who were right downstairs when I had questions about my theoretical and experimental works.

I thank sincerely Prof. Dr-Ing. D. Zimmer and Dipl-Ing H. Kammler for their help and allowance to built my test rig in their laboratory during the experimental part of this study.

My wife and my children deserve an award for their patience and supporting during the writing of this thesis. I could not have done it without them. I am grateful for the supporting of my parents over the last decade or so as I finished my studies.

Paderborn, July 2005 Khaled Mahmoud

Aus dieser Arbeit wurden keine Teile bereits veröffentlicht.

Paderborn den 11.04.05

Khaled Mahmoud

To my wife

Contents

NOMENCLATURES

Symbol	***unit***	***meaning***
a	*m*	*Distance between drum centre point and floating link*
A	*N*	*Anchor pin side force*
b	*m*	*Distance between drum centre point and hydraulic cylinder*
c	*m*	*Distance between drum centre point and tangential force*
C^*		*Brake shoe factor*
C_1		*Brake shoe factor at primary shoe*
C_2		*Brake shoe factor at secondary shoe*
e	*N*	*Control error Force difference between modeled and actual forces*
E	*J*	*Brake energy*
F	*N*	*Applied force*
L	*m*	*Band length*
m	*kg*	*Vehicle mass*
N	*N*	*Normal force*
N_1	*N*	*Normal force at primary shoe*
N_2	*N*	*Normal force at secondary shoe*
R	*N*	*Brake force*
r	*m*	*Drum radius*
R_1	*N*	*Brake force at primary shoe*
R_2	*N*	*Brake force at secondary shoe*
R_{actual}	*N*	*Measured brake force*
R_{mod}	*N*	*Modeled brake force*
S	*N*	*Side force*

S_1	*N*	*Side force at floating link from primary shoe side*
S_2	*N*	*Side force at floating link from secondary shoe side*
α	*deg*	*Self amplification angle*
δ	*deg*	*Inclination of floating link to x-direction*
μ		*Coefficient of friction*

1 Introduction

Ever since the invention of the wheels thousands years ago, it was required to slow down or stop the wagon. Firstly, there were horse wagons and the animals were the only power source. They were not only used to accelerate but also to slow down and to completely stop the wagon. Later on, mechanical means were used to stop the vehicle such as a braking system consisting of a hand lever to push a wooden friction directly against the metal trade of the wheels.

There were many inventions to develop braking systems, for example the rope brakes developed around 100 AD or chariot braking systems depending on a wrapped chain around the protruding wheel hub with one end anchored to the chariot body to stop it.

The early years of automotive industry were an interesting period for design engineers. Braking system design was one of the important features that determined the limiting speed of the vehicle. Drum brakes were common in the first generation of the vehicle. The drum brake consisted of two shoes which could be expanded inside the drum.

In 1902 in England Dr. F. W. Lanchester introduced the first disc brake patent. Disc brakes were firstly produced for the Lanchester car in the period between 1906 to1914. At this time the drum brake still was more effective than the disc brake.

Drum and disc brakes both have advantages and disadvantages. Drum brakes are cheaper and less complex and their effectiveness is higher because of the self-amplification. Drum brakes have the disadvantage that they are more sensitive to brake fade because they are not so capable to dissipate the generated heat. Disc brakes have lower sensitivity to brake fade, but they are expensive and have lower brake effectiveness when compared with the drum brake.

Around 1960, environmental concerns lead to the replacement of asbestos based friction materials and the ABS (Anti-lock Braking System) which prevents or reduces the vehicle skidding was introduced.

The variation in the coefficient of friction is still generally the main problem for all brake types. This variation directly affects the brake force. Due to the self-amplification its effect is greater in the case of drum brakes. As a result of frequent and hard braking the contact temperature between the pad and the rotor increases which leads to a decrease of the coefficient of friction. This is called brake fade. In critical situations its effect results in a pronounced change of the brake effectiveness associated with a strange pedal feel.

In the last 30 years the brake performance was improved by applying mechatronics in the design and manufacturing of brake systems. Complex booster designs and servo-controlled hydraulic pressure systems as well as improved adaptive designs were developed. One very pronounced change was associated with the brake-by-wire in these systems where electromechanical or electro hydraulic brake technologies are being used. One advantage of brake-by-wire is that it can be used to compensate the increase or decrease on the brake force as a result of a changing coefficient of friction by controlling the applied (actuator) force. This needs to measure the brake force at the rotor.

Brake-by-wire however also it has disadvantages such as the complexity, the extra weight, extra power consumption and high costs. A major problem with the electromechanical brake with electric or hydraulic actuators is the high actuator force which has to be applied in order to achieve a sufficient brake effect especially when the coefficient of friction has low values. The need for high actuator force and the resulting large power demand of the actuator make it necessary to use large electric motors, which are heavy and expensive

There were many efforts to increase the brake effectiveness by changing the configuration of the brake. For example a wedge disc brake which is provided by a wedge to increase the brake force. In the last few years wedge-actuated disc brakes were presented but until now they are still in the research stage. For drum brakes an adaptive drum brake was presented to maintain the brake effectiveness at constant values. This design is still at an early development stage and requires further research before its full potential can be deployed.

The aim of the present work is to investigate how the self amplification of drum brakes can be automatically adapted such that the brake shoe factor will be kept at a high and constant value independently from the changes in the coefficient of friction. This investigation will be based on theoretical and experimental observations and will firstly show how the coefficient of friction affects the behavior of different types of drum brakes. Then, a novel design for an adaptive duo servo drum brake will be developed which has the advantage that the adaptation can be easily achieved using low power actuators. The experimental investigations are carried out on a prototype demonstrator and the experimental results are not only used to validate the design models but also to characterize the potential of the new design.

2 Background on automotive brake systems and state of the art

2.1 Friction brakes

The brake is a mechanism, which is used to absorb the kinetic energy of the vehicle with the aim to stop or retard the motion. Brakes transform kinetic energy into heat. Since the acceleration required during an emergency brake maneuver is much higher than the acceleration during normal operation, the brake power must be much higher than the motor power of the vehicle. Even for small vehicles a maximum brake power in the order of several hundred kilowatts is the rule rather than the exception.

The energy to be dissipated in braking from speed v on a slope of height h is

$$E = \frac{m\,v^2}{2} + m\,g\,h \qquad 2.1$$

where m is the vehicle mass. The first term of the equation is the kinetic energy while the second term of the equation evaluates the potential energy when moving downhill. The energies which must be dissipated are enormous and result in strong demands on the materials used in the friction contact which must withstand extremely high temperatures.

Brake linings can be classified into organic, metallic and carbon. Modern brake pads are composed of many different ingredients. These can be categorized as follows

Fibers: Metal-, carbon-, glass-, and/or kevlar fibers are added to the linings to give mechanical stability and provide wear resistance.

Matrix: The matrix is normally some kind of phenolic resin. It holds the ingredients and provides thermal resistance.

Filler: Fillers are used to make the material less expensive and/or improve manufacturability. Some fillers may also affect the friction characteristics of the materials.

Frictional additives: Frictional additives are added to control the coefficient of friction or to change the type of wear. Lubricants such as graphite are used to decrease the coefficient of friction. Abrasive particles are used to increase the coefficient of friction or to increase the brake wear with the purpose to remove iron oxides from the disc surface, thus providing more consistent braking properties [Eriksson et al, 2002].

The friction brakes are classified into disc brakes and drum brakes. The drum brakes are classified into simplex, duplex and duo servo drum brakes.

Automotive brakes include drum and disc brakes for passenger car, light truck, bus, heavy truck, and off-road vehicles. Drum brakes predominantly use internal expanding shoes with brake linings that load the majority (typically 50 to 70 %) of the drum-rubbing surface. Most automotive disc brakes use shoes that load a much smaller portion (from 7 to 25 %) of the disc-rubbing surface. Disc brakes offer faster cooling, with their larger exposed surface areas and better cooling geometry, but are damageable to either liquid or solid particulate contamination. Because of cooling, contamination and other basic issues, the combination of front disc brakes and rear drum brakes are commonly used. [Anderson, 1990]

Fig 2.1 shows the brake system with its power flow. The brake system has the pedal (foot) force from the driver as the input force and the brake or friction force as output force. The applied force is the result of the brake pedal force. Referring to [Leber et al, 1998], the pedal force is transformed with amplifying factor as well as a transmission efficiency. In the case of truck vehicles, the amplification is not enough to generate the desire brake force. Here and in many passenger cars, a pneumatic pressure booster is required which also has amplification factor and transmission efficiency. During the final step in the wheel brake system, the hydraulic force is converted to brake force with an amplification factor called the brake shoe factor C^*. In this example, typical values for the total amplification factor of the brake system are approximately 50 and the total transmission efficiency typically falls into the range 60%-65%.

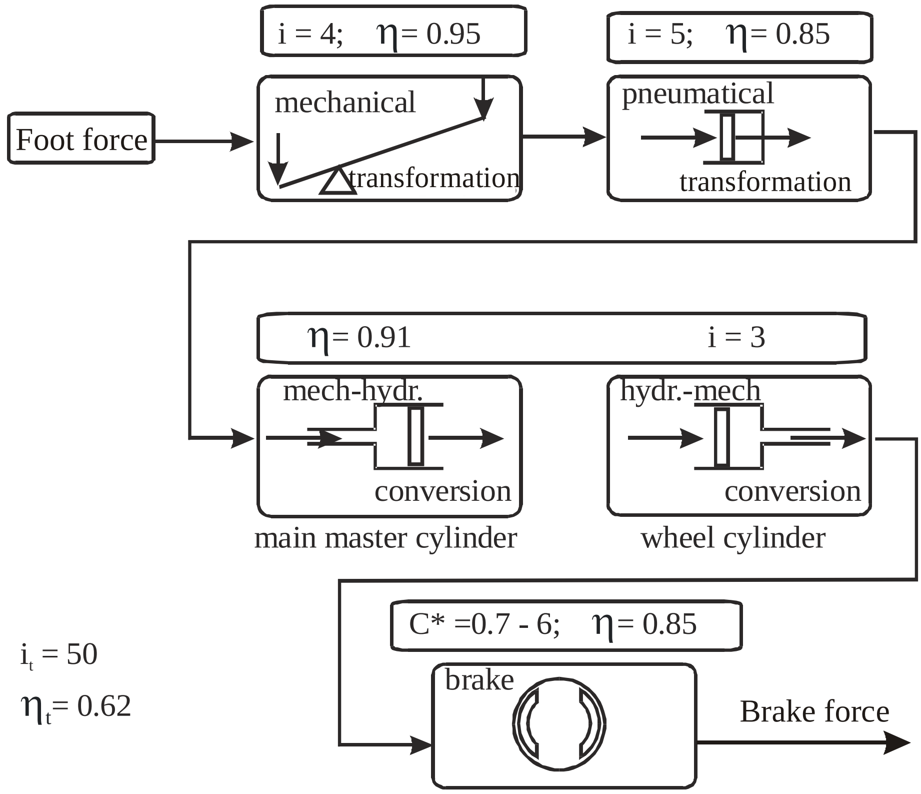
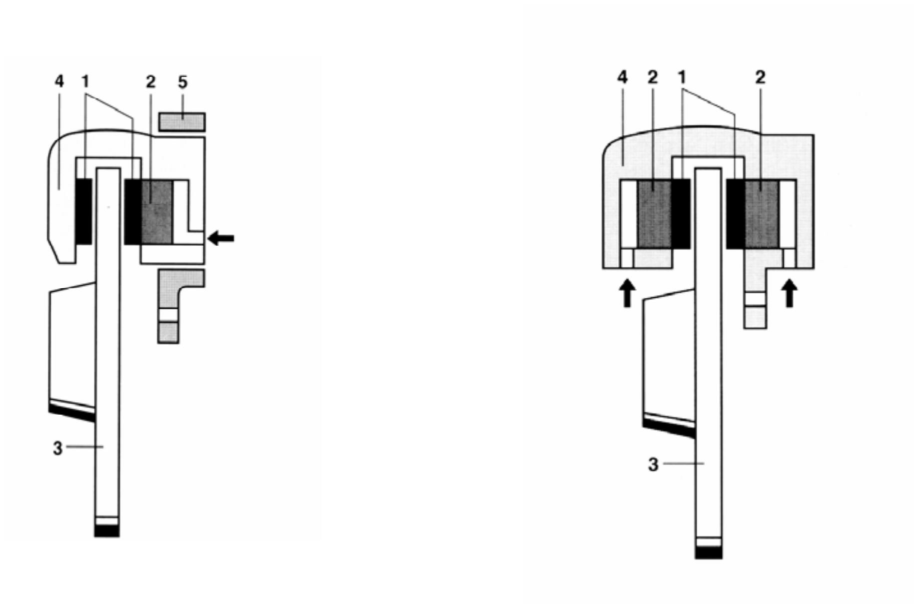


Fig 2.1 brake system with its power flow [according to Leber et al, 1998]

2.1.1 Disc brake

Disc brakes consist of a rotor (disc) and a caliper. The rotor turns with the wheel. Each side of the rotor is a friction surface. The caliper is attached to an anchor plate or mounting bracket on the vehicle suspension. The hydraulic piston or several pistons convert the hydraulic pressure into an applied force that presses the pad against the rotor. This generates the friction forces needed for the braking. *Fig 2.2* shows principal designs of typical disc brakes.

There are two types of disc brakes, floating caliper disc brakes and fixed caliper disc brakes. In floating caliper disc brakes the caliper is free to float laterally on its mounting pins. The hydraulic line is connected with one side of the caliper and the motion is transferred to the other side through the caliper. The fixed caliper disc brake is characterized by the caliper which has hydraulic lines connected with both sides of the caliper and the two pistons.



(a) (b)

1. Brake pad. 2. Piston 3. Disc (rotor) 4. Caliper 5. Bracket

Fig 2.2 Disc brakes (a- floating caliper b-fixed caliper) [according to BOSCH, 1994]

2.1.2 Drum brakes

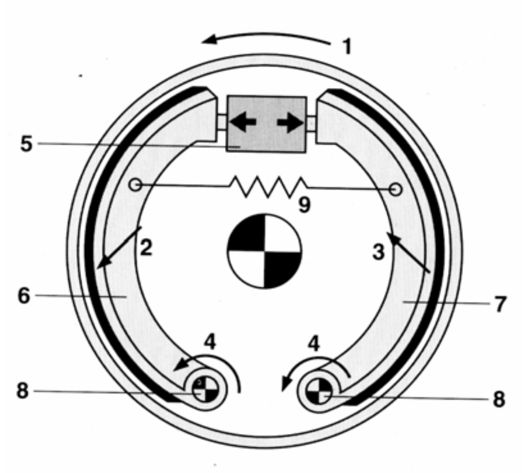
Drum brakes consist of a drum and a brake mechanism with two brake shoes that are curved to conform to the inside diameter of the drum. The drum brake has a steel or iron drum to which the wheel is bolted. The hydraulic pressure pushes the shoe-actuating pins out; hence the brake shoes are forced against the rotating drum. The resulting friction between the brake lining and the drum slows or stops the car.

Drum brakes may be different in appearance and construction, but functionally they are all the same. There are three types of drum brakes: Simplex, duplex and duo-servo drum brakes.

Early automotive brake systems used a drum design at all four wheels. They were called drum brakes because the components were housed in a round drum that rotated along with the wheel. The shoes were made of a heat-resistant friction material similar to that used on clutch plates.

2.1.2.1 Simplex drum brakes

Simplex drum brakes have diameters ranging from about 160 to 500 mm, with maximum torques ranging from 300 to 25000 Nm. The simplex drum brake has two shoes acting differently. According to the generated brake force, one is called the self-amplifying shoe and the other is the self-debilitating shoe. The source of the applied force is typically a hydraulic cylinder or in exceptional cases an electromechanical actuator. When the lining comes into frictional contact with the drum, the rotation of the drum will either pull the shoe along with it or push 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 amplification [Brake system parts manufacturers council, 1991]. *Fig 2.3* shows a construction of the simplex drum brake.

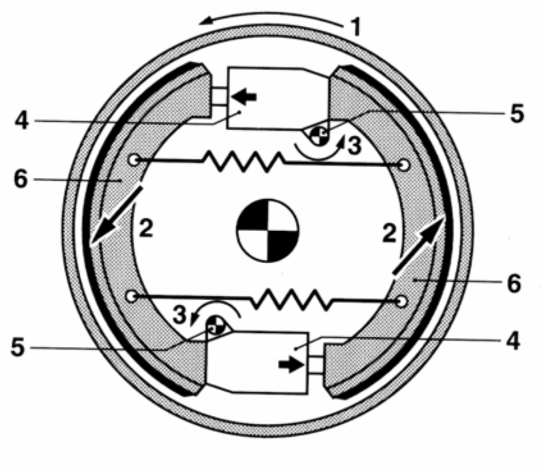


1. direction of motion 2.self-energition of the brake force 3.self-reduction of the brake force 4.torque 5. wheel cylinder 6. leading shoe 7. trailing shoe 8. pivoting point 9. return spring.

Fig 2.3 Simplex (leading trailing) drum brake [according to BOSCH, 1994] .

2.1.2.2 Duplex drum brakes

In duplex drum brakes both the primary and secondary shoe are self-amplified or leading shoes. The duplex (twin leading) drum brake has two separate single-piston wheel cylinders. Each brake shoe is applied by a cylinder. In the case of the inverse direction of rotation both shoes are self debilitating and this is a disadvantage of the duplex drum brake. *Fig 2.4* shows the typical design of the duplex drum brake.



1. direction of motion 2.self-energization of the brake force 3. torque 4. wheel cylinder 5. pivoting point 6. leading shoe

Fig 2.4 Duplex drum brake [according to BOSCH, 1994].

2.1.2.3 Duo servo drum brakes

In the duo-servo brake, both shoes are self-amplified and the self-amplifying force from one shoe is transferred to the other with the wheel rotating in either direction. With the forward rotation the primary shoe pivots around the adjusting screw. The drum, or course, prevents any outward movement and the shoe is wedged into the drum with a force greater than that applied by the wheel cylinder. Frictional forces from the primary shoe are transferred to the secondary shoe through the adjusting screw. This, plus the frictional forces building up in the secondary shoe, quickly overcome the wheel cylinder force and push the secondary shoe into contact with the anchor pin. The anchor pin prevents any further shoe movement. As the secondary shoe attempts to pivot outward around the anchor pin, it is pressed into the drum with even greater force. *Fig 2.5* illustrates the typical design of the duo servo drum brake.

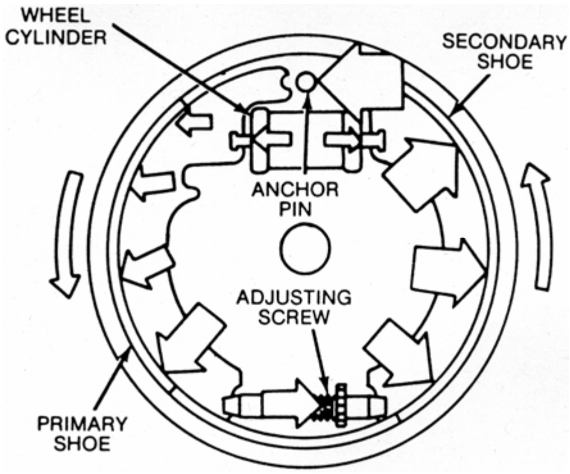


Fig 2.5 Duo-servo drum brake [Brake system parts manufacturers council, 1991]

2.2 Brake-by-wire

As an alternative to fully hydraulic brakes so called brake by wire systems have been discussed recently. In these brakes the applied force will be generated at the wheels by mechanical actuators.

The electromechanical brake (EMB) or also brake-by-wire is just like the electrical steering element an extremely safety-relevant system. The temptation is similar: Assembly simplification, combination of functionality, recycling improvement. The wear can be recognized over the way sensor of the brake and compensated accordinggly

The principle of brake by wire is based on separating the hydraulic connection between the brake pedal and the wheel brake and replacing it with electrical signals. The functions of the brake by wire are fully compatible with other electronic systems such as ABS. The driver desire to brake the vehicle is recognized by a sensor that delivers a signal according to the pedal stroke which is transmitted to the brake controller. The brake controller then generates signal to the wheel controller which addresses the electric control valves or solenoids. At the same time the hydraulic pump delivers the hydraulic fluid through the valves to the hydraulic actuators to generate the apply force at the disc or drum brakes.

[Wiehan, 1999] has summarized the positive aspects of brake-by-wire. A fundamental advantage is that the brake by wire is compatible with assistance electronic functions such as ABS (Antilock Braking System). The adapting will be easier with electrical interfaces instead of hydraulic interfaces. The absence of the mechanical links leads to reduce the whole weight of the brake system. The brake by wire is environment friendly because there is no hydraulic fluid in the EMB. [Smesch and Breuer, 2000] have investigated a control system that could adjust the clearance electronically.

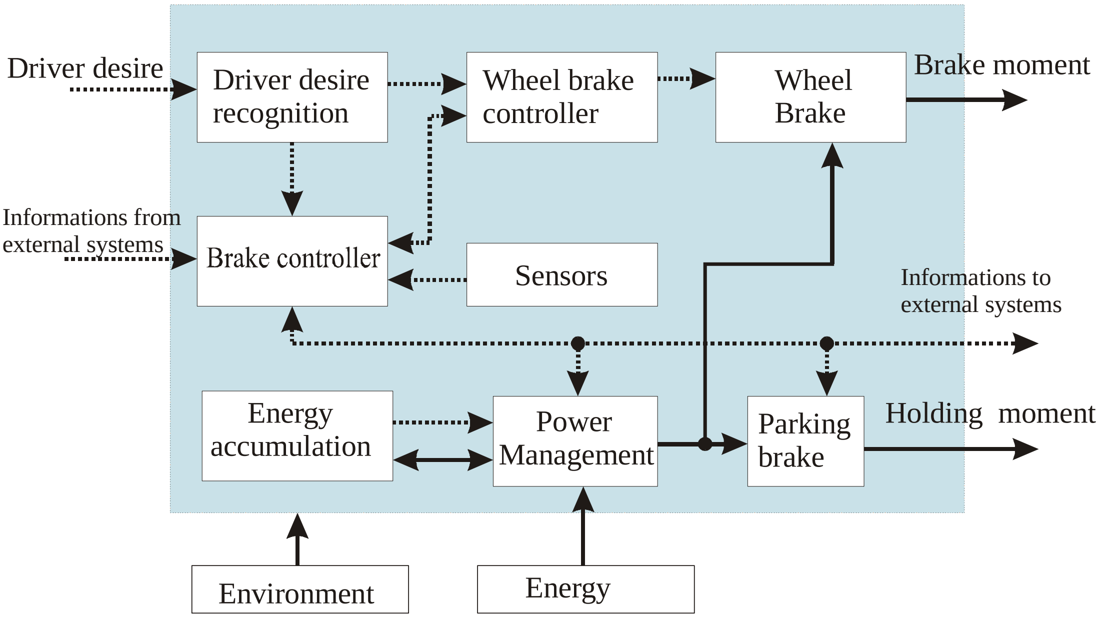


Fig 2.6 Electro mechanical brake system [Smesch and Breuer ,2000]

Modern vehicles are equipped with electronic systems such as Antilock Braking System, electronic brake force distribution, anti-slip control and vehicle stability system. These systems are sensed and actuated electronically. Fig 2.6 shows an electromechanical brake, which was presented by [Smesch and Breuer, 2000]. The information flow is shown as dotted lines and the power flow is shown as solid lines. When the driver wants to brake the vehicle, the desire is recognized by a sensor signals are sent to the brake controller. Then, the signals are forwarded to the power management unit which provides the brake actuator with the required energy. The brake force sensors provide the feedback of the system. The sensors signals go back to the brake controllers which adapt the generated brake force according to pedal force value. In the case of the Parking brake the energy from the power unit flows to the parking brake to hold the brake. [Smesch and Breuer, 2000] have investigated an electromechanical brake which used a set of valves, actuated by solenoids, to achieve a desired value of the braking force as well as to adjust the clearance between the disc and the shoes automatically. This system provides the brake system with a control unit. Inputs are the electricity and data flow for the controlling of the system

There are two types of wheel brakes for brake by wire: electro-hydraulic brake (EHB) and electromechanical brake (EMB).

2.2.1 Electromechanical brake EMB

The electromechanical brake even goes one step further than the electrohydraulic brakes and eliminates the brake cylinder, brake lines and hoses, as all these components are replaced by electric components. The use of electrics reduces maintenance expense, and also eliminates the expense of brake fluid disposal. EMB likewise measures the force of the driver's intention to stop the vehicle via sensors monitoring the system in the brake-pedal feel simulator. The electronic control unit ECU receives the signals, links them where appropriate to data from other sensors and control systems, and calculates for each wheel the force to be generated by the brake caliper when pressing the brake pads onto the brake disc.

The wheel brake modules essentially consist of an electric control unit, an electric motor and a transmission system. The electric motor with the transmission system generates the force which presses the pad against the disc. Power requirements for EMB are high and overload the capabilities of conventional 12 volt systems installed in today's vehicles. Therefore the electro-mechanical brake is designed for a working voltage of 42 volts. *Fig 2.7* shows the power flow of an electromechanical brake.

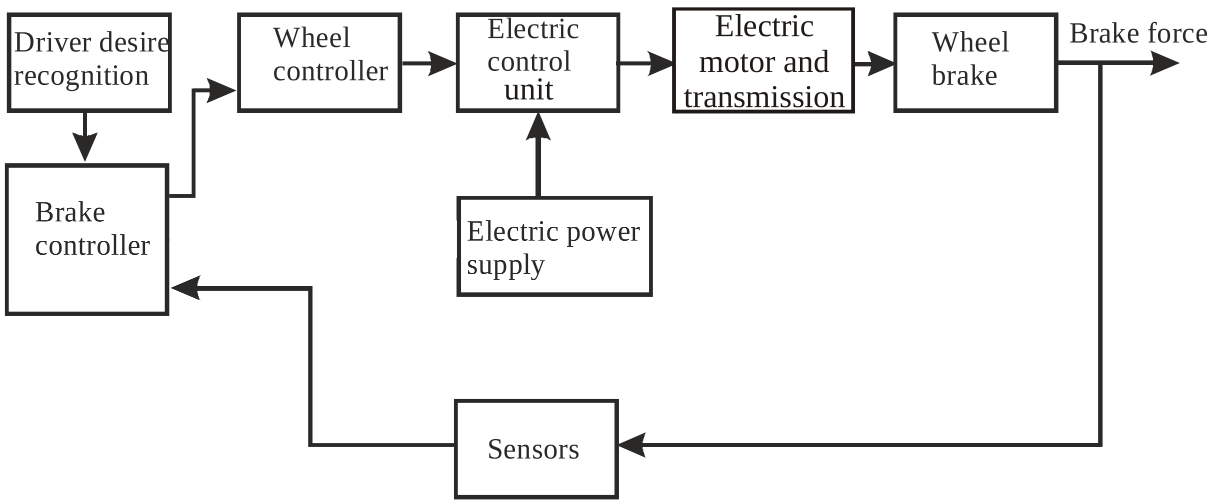


Fig 2.7 Electromechanical Brake (EMB)

When the driver wishes to brake the vehicle, his pedal force is converted to signals to the brake controller. The brake controller delivers the signals to all wheel controllers which are connected with the electronic control unit. The electronic control unit provides the electric motor with the required power. The electric motor with its transmission generates the brake force at the wheel brake. The feedback signals of the system through the force sensors go back to the brake controller to control the brake force according to driver pedal force.

2.3 Electro hydraulic brake EHB

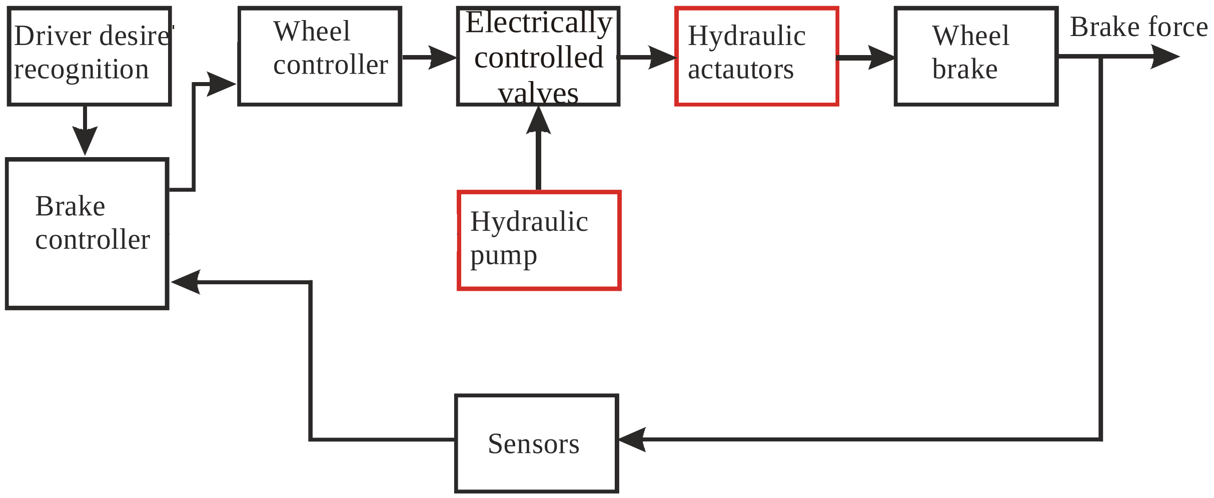


Fig 2.8 Electro-Hydraulic Brake EHB,

As shown in Fig 2.8, which illustrates an electrohydraulic brake, the main difference between the electromechanical and electro-hydraulic brakes is the presence of some hydraulic parts such as hydraulic pump instead of electric power supply with the electromechanical brake. Also, the electro-hydraulic brake has hydraulic actuators instead of electric motors with the electromechanical brake. The hydraulic actuators are connected with the hydraulic pump through valves which are electrically controlled. The electrohydraulic brake usually has a backup in the hydraulic system.

2.4 Design considerations for wheel brakes

The brake force is generated in a friction process. Hence, it is necessary to know the friction properties of pad and disc. The basic form of the friction is that presented by coulomb's law which postulates that the friction force is varied linearly with normal force. According to Coulomb' law, the coefficient of friction is defined as the ratio between the friction force and the normal force.

There were many efforts to determine accurately the coefficient of friction and its variation such as [Severin et al, 2001 and Blau, 1996] who presented a friction law depending on the normal force, sliding speed and contact temperature. In the following sections the parameters that affect the coefficient of friction will be discussed.

2.4.1 Coefficient of friction

The coefficient of friction is affected by several conditions such as the normal force, sliding speed and temperature. This leads to the observation that the coefficient of friction is varied with the brake time. The sensitivity to the coefficient of friction still is a problem for all brake types especially the drum brake.

[Kato, 1982] has investigated an experimental work aimed to obtain the relation between the sliding speed and the coefficient of friction under extreme loads. His results indicate that the sliding friction greatly increases with the decrease of the sliding velocity. [Jarvis et al,1963] have observed that the coefficient of friction decreased with the increase of the sliding speed

[Avilés and others, 1995] have experimentally investigated the relation between the coefficient of friction and the sliding speed. They have used five different brakes. The coefficient of friction has no exact trend with sliding speed. Each brake has a trend that differs form the others, however, the coefficient of friction decreased with the sliding speed in all the brakes that were investigated in their study. [Rhee, 1974] has carried out an experimental investigation aimed to study the frictional properties of a phenolic resin filled with iron and graphite. He investigated the influences of normal load, sliding speed and drum temperature on the coefficient of friction. His results illustrated that the coefficient of friction ranged from 0.3 to 0.4 and with increase both of the normal load and rotational speed the coefficient of friction decreased He modeled the dependence of the frictional force upon both the sliding speed and normal load as:

$$F = \mu N^{p_1} v^{p_2}$$

This equation is for constant temperature, where v is the sliding speed and p_1 and p_2 are constants.

An experimental study which is presented by [Severin, 2001] has illustrated the coefficient of friction variations with the no. of brake applies for a drum brake. *Fig 2.9* shows these variations which is limited by 0.25 to 0.47.

[Blau, 1996] has explained the decrease of the coefficient of friction with respect to the velocity as the effect of lubricating oxide form at elevated temperatures, and a corresponding decrease of the shear strength of the frictional material with high frictional temperature. The presence of molten material acts as lubricant.

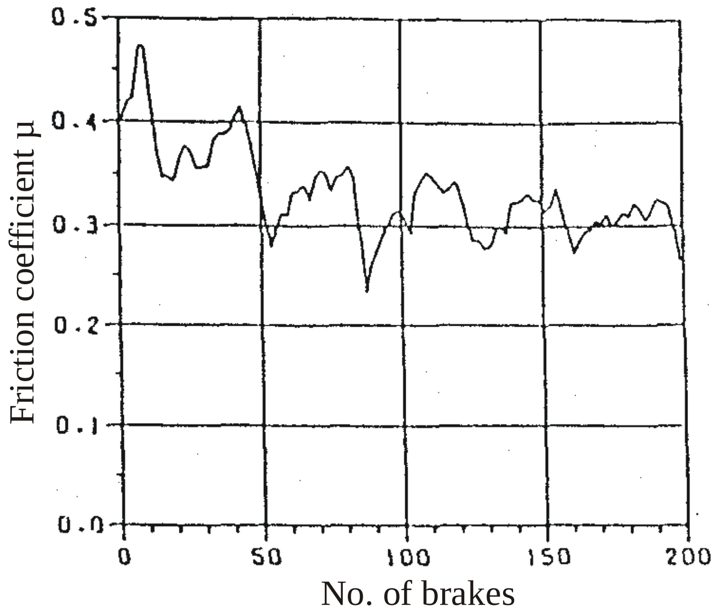


Fig 2.9 coefficient of friction variations with the no. of brakes with drum brake [Severin, 2001]

Fig 2.10 shows the effect of the normal force on the coefficient of friction which was presented by [Blau, 1996]. He divided the effect into three zones. When the normal force is low, the coefficient of friction decreases with the increase of the normal force because of the governing by oxides of metallic materials. At higher normal forces, the coefficient of friction raises as a result of film rupture and plowing the subsurface. At very high contact forces, the coefficient of friction decreases again with the increase of the normal force.

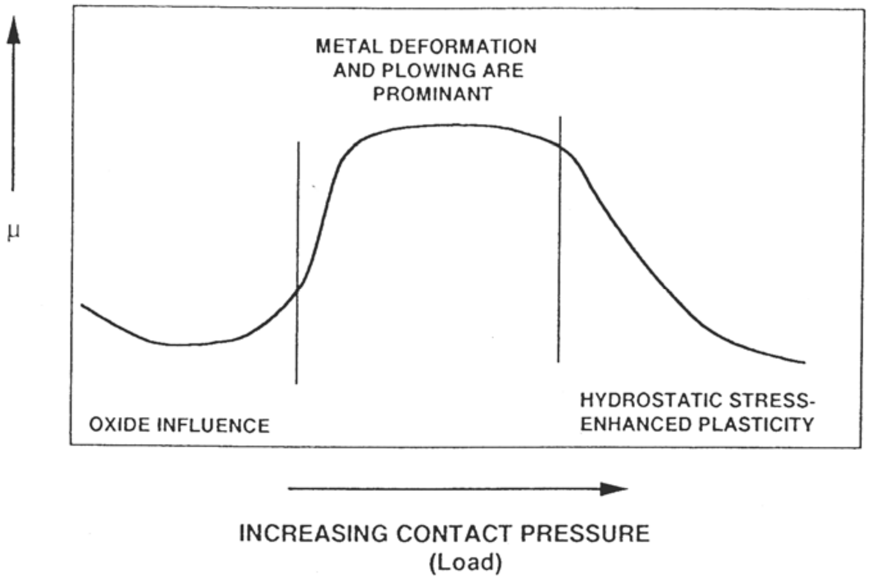


Fig 2.10 Effect of the contact pressure on the coefficient of friction[Blau, 1996]

[Blau, 1996] mentioned eight typical forms variations of coefficient of friction versus sliding time. He illustrated that the coefficient of friction versus sliding time is influenced by several conditions: contamination of the contact areas, boundary lubricated or unlubricated metals, cleanliness of the contact materials, and coatings of materials. These conditions affect the coefficient of friction as shown in Fig 2.11.

The vehicle's brakes create heat as a by product. The rate at which a wheel can be slowed depends on several factors including vehicle mass, brake geometry and total braking surface area. It also depends heavily on how quickly the frictional heat is removed from the brake components. This is where the difference between drum brakes and disc brakes is most pronounced.

The drum brake design proved capable under most circumstances, but it had one major flaw. Under severe braking conditions, like descending a steep hill with a heavy load or repeated high-speed slow downs, drum brakes would often fade and lose effectiveness. Usually this fading was the result of high temperature within the drum. For this reason, drum brakes can only operate as long as they can absorb and remove the heat generated by braking.

The disc brakes design is far superior to that of drum brakes with respect to thermal behavior. Instead of housing the major components within a metal drum, disc brakes use a slim rotor and small caliper.

In this way, the rotor is fully exposed to outside air. This exposure works to constantly cool the rotor, greatly reducing its tendency to overheat or cause fading. Not surprisingly, it was under racing circumstances that the strengths of disc brakes were first illustrated.

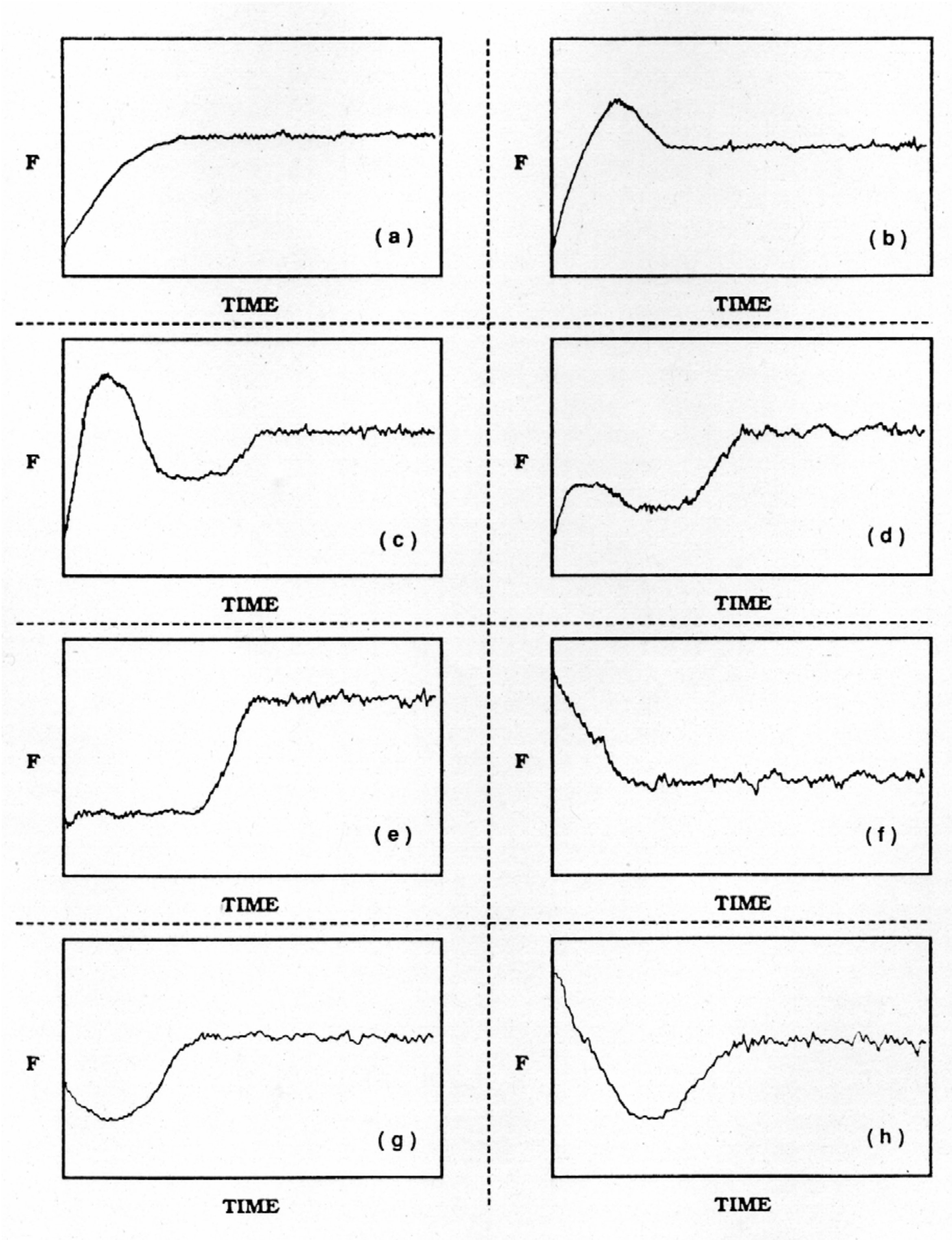


Fig 2.11 Eight forms of the coefficient of friction[Blau, 1996]

2.4.2 General aspects

2.4.2.1 Brake effectiveness

The brakes effectiveness is defined as the ratio of the total friction force on the shoe to the applied force at the tip of the shoe. It is also called the brake shoe factor. The brake force between the rotor and the pad is as a result of the friction process.

The shoe factor of course, depends on the value of the coefficient of friction. [Mitschke et al, 1995] has presented the variation on the shoe factor with the coefficient of friction. Fig 2.12 shows the shoe factor variations of different types of vehicle brakes. From this figure it can be seen that the duo servo drum brake has the highest shoe factor and that the disc brake has the lowest shoe factor.

Drum brakes make use of the self-amplification effect. As the lining touches the drum, the rotation of the drum will tend to either pull the shoe along with it or push the shoe away, depending on the position of the shoe anchor relative to the direction of rotation. These actions are related with the simplex drum brake and are referred to as energizing and de-energizing of the shoes. Self-energizing brakes are those where the frictional forces created when the brakes are applied tend to apply the brake even more. Most drum brakes are, at least in part, self-energizing. The drum brakes apply-force and the drum rotation are both in the same plane. When the apply force moves the brake shoe against the drum, the drum tries to pull the shoe along with it. The shoe pivot prevents this. However, frictional forces do pivot the shoe outward and force it against the drum even more. Thus the drum brake is self-energizing at the leading shoe. But for the trailing shoe, the friction force tries to reduce the braking force. There is an inverse proportionality between the effectiveness of brake systems which is defined as the ratio between the drum friction force to the apply force and the sensitivity to the coefficient of friction variations. Drum brakes especially the duo servo drum brake have high stopping power, but they are also highly sensitive to the coefficient of friction. From the other side the disc brake generates the lowest stopping power, but it has low sensitivity to the coefficient of friction. Drum brakes are characterized by the so-called self amplifying phenomena, the primary function of the brake drum assembly is to force the brake shoes against the rotating drum to provide the braking action.

The duplex drum brake has two leading shoes only with the forward motion.

The drum brake shoe factor doesn’t vary linearly with the coefficient of friction, see Fig 2.12:

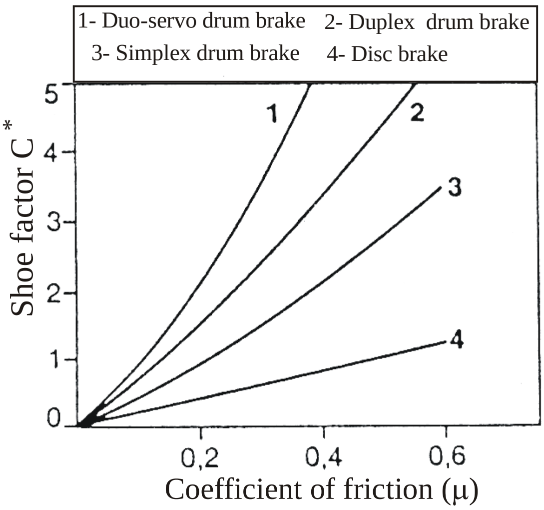


Fig 2.12 brake shoe factor variations with the coefficient of friction [Mitschke et al,1995]

The band brake is a very special design. It also uses the principle of self-amplification. It was analyzed by [Leber et al, 1998]. They classified the shoe factor variation as a function of the coefficient of friction and the inclination angle into three zones for the shoe factor according to its value:

- $C^* = \mu$ in the case of non self-amplified action
- $C^* > \mu$ when the brakes are self-amplifying
- $C^* < \mu$ when the brakes are self-debilitating

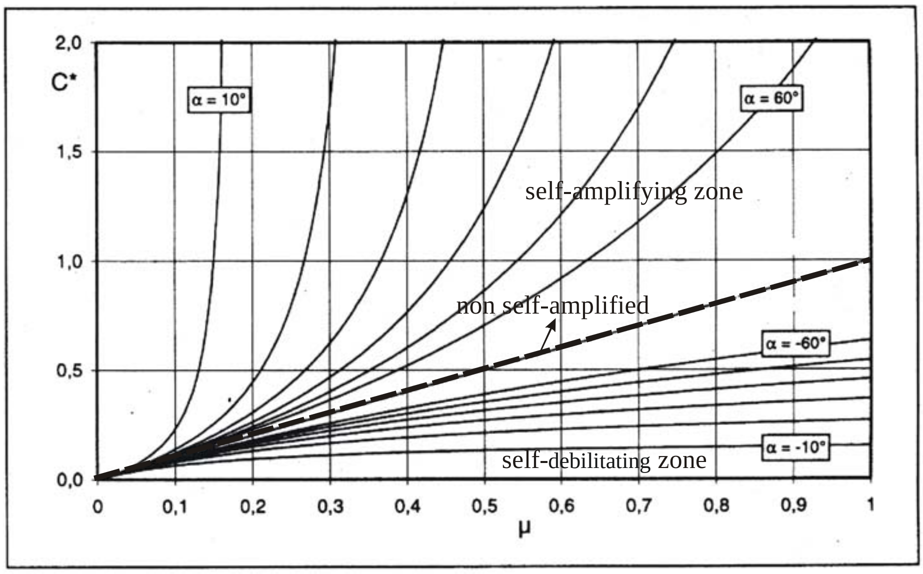


Fig 2.13 shoe factor with the coefficient of friction variations at different angle α with band brake [Leber et al, 1998].

The authors illustrated the relation between the shoe factor and the coefficient of friction at different angle α with the band brake. Fig 2.13 shows the three zones of the shoe factor with the coefficient of friction variations, the dotted line when the angle α equals 90° corresponds to the case of a non self-amplified zone and the shoe factor equals μ in this case. The zone when the angle α is greater than 0 and smaller than 90°, is the self-amplifying zone. The zone is when the angle α is less than 0 is the self-debilitating zone.

2.4.2.2 Brake fade

Brake fade is related to friction between the brake shoes and drum or pads and disc. It is often caused by heat build up due to repeated or prolonged brake application.

Brake fade can in extreme cases lead to a severe loss of brake force in critical situations. Brake fade is the number one high performance driving braking problem. It usually occurs gradually so that the driver is able to compensate the brake point by braking sooner, but sometimes brake fade happens suddenly and can have severe results. There are three kinds of fade commonly encountered in fast driving; pad fade, green fade and fluid fade. All friction materials have a coefficient of friction which changes with temperature. They have an optimal working temperature where the coefficient of friction is maximal. If the brake is used in such a way that the temperature rises over the point of maximum friction, the coefficient of friction will become smaller for rising temperatures.

The mechanics of this decline in the coefficient of friction are manifold. At a certain temperature, certain elements of the pad can melt causing a lubrication effect; this is the classic glazed pad. Usually the organic binder resin starts to go first, and then even the metallic elements of the friction material can start to melt. At really high temperatures the friction material starts to vaporize and the pad can start to float on a boundary layer of vaporized metal and friction material which acts like a lubricant. Pad fade is felt as a car that still has a decent, non mushy feeling brake pedal that won’t stop even if you are pushing as hard as you can.

2.4.3 Special requirements for brake-by-wire

The x-by-wire is replacing the mechanical, hydraulic or pneumatic components by electromechanical devices. The x-by-wire could be applied in the vehicle in power train, steering and braking that are directly concerned with the driving of the vehicle. [Dorißen and Dürkopp, 2002] have presented a description of three applications of the x-by-wire, throttle-by-wire, brake-by-wire and steer-by-wire. They have distinguished the brake-by-wire as hydraulic brake-by-wire (wet) and electric brake-by-wire (dry). The wet brake-by-wire which is sometimes called electro-hydraulic brake is the improvement of the common braking systems.

Brake-by-wire is discussed by many researchers such as [Bertram et al, 1998] and [Leffler, 1999] who have evaluated the brake-by-wire as an application of mechatronics in the vehicle brake. They have mentioned that the next step of the evolution of mechatronics in the vehicle aim to:

- Increase safety
- Improve security

- Provide more comfort
- Achieve ecological friendliness
- Grow multimedia capabilities

The brake-by-wire application has many advantages. The main advantages of the electro-mechanical brake are summarized as follow:

- It allows to supply the required energy electrically instead of hydraulically.
- A connection with the other electronic systems such as Anti-lock Braking System, electronic brake force distribution, anti-slip control and vehicle stability system are possible.
- With the change in the clearance between the disc and the shoes due to wear, the apply force can be made constant or nearly constant.

The disadvantages of the electromechanical brake are summarized as follow:

- Solenoids have electromagnetic interference and also the electromagnetic solenoids are unidirectional, i.e. they work only in one direction as presented by [Pavlov et al, 2001].
- Expensive
- More brake system weight
- More complexity
- More system energy consumption

There are many patents that interested with the topic of electro-mechanical brake or electro-hydraulic brake or brake-by-wire such as [Harris and Martin, GB 2364355A, 2002], [Philips and Harris, EP0889817B1,1997] and [Sherriff, US0005661A1, 2002] which have the same main idea of providing the vehicle brake system with an accumulator, solenoids, electronic control units and sensors.

[Pavlov et al, 2001] have proposed an electromechanical brake using piezo-electric actuators to eliminate the discrimination of the electromagnetic interference as well as the unidirectionality of the solenoids. Their invention was to use piezoelectric actuators to generate a variable or controllable braking force. They mentioned that the piezoelectric actuator has the advantages that it needs little maintenance, eliminates electromagnetic interference and has lightweight. They invented three mechanisms; two for disc brake and one for drum brake. The first and the third were according to use a piezoelectric actuator linear motor with non-symmetrical driving force in the radial direction to generate the braking force in the axial direction. The first mechanism is designed for a disc brake while the third one is designed for a drum brake. The second mechanism was according to use a rotational piezoelectric actuator to generate braking toque, which is converted into braking force through gear trains and threads. But with regard to the price of each piezo electric actuator, the system becomes very expensive and more complex.

[Smesch and Breuer, 2000] have investigated the relation between the clearance between the disc and the shoes and the apply force. They found that with the increase of the clearance the apply force decreases. Therefore the brake system must be provided with a control system that could adjust the clearance electronically.

Most of the modern vehicles brakes are provided with some electronic systems such as Antilock Braking System, electronic brake force distribution, anti-slip control and vehicle stability system. These systems are sensed and actuated electronically.

2.5 Concepts for self-amplifying wheel brakes

2.5.1 Literature and patent survey

There are many concepts which deal with the vehicle braking. In the last few years, many inventors have presented very important patents in the field of vehicle brakes. For example, there are many concepts tried to apply the brake by wire in the automotive applications. The brake by wire has a widespread in the modern vehicles. The application of the brake by wire could improve the overall characteristics of the brake system by regulating the brake force that is maintained at the desired value indecently on the coefficient of friction variations. For example, the patents [EP 1138971A3] and [EP 1141573B1] have presented new prototypes of electromechanical vehicle brakes. There is a patent relates also to electromechanical brake was presented by [Curran et al, US 6325471B1, 2001] propose an electro-hydraulic brake where the brake controller is not only connected with the brake actuator but also with the Antilock-Braking-System. Another example, the patent presented by [Pavlov et al, GB 2369661A, 2001], relates to the use of piezoelectric actuator with electromechanical brakes. They summarized the disadvantages of the solenoids (magnetic interference, solenoids are unidirectional). They have a group of 3x4 piezoelectric actuators as actuator groups.

There are many patents related to electromechanical braking that aimed to minimize the applied or actuation forces which are needed for braking. Also, there are many efforts to increase the brake force. In other words they aimed to increase the brake shoe factor. These patents used to provide the brake with self-amplifying mechanisms with disc brakes and to improve the characteristic of the drum brakes by using electromechanical application.

In this section we will focus on the concepts which deal with the electromechanical brake with self-amplifying action. Firstly, all drum brakes are already characterized with the self-amplifying action whereas there is a difference in self-amplifying level with the drum brake type. The classical disc brake is not self-amplified but many inventors have tried to develop disc brake with self-amplifying characteristic.

2.5.2 Wedge disc brake

Another application of mechatronics is in a new disc brake, the so- called wedge disc brake. It was presented by [Roberts and others, 2002]. Their concept is based on the use of self-amplification action in the disc brake by using a mechanism of wedge which is shown in *Fig 2.14*. This mechanism is similar to the patent which is presented by [Dietrich et al, 2001]. In this concept the inventors tried to find the optimal operating point. The shoe factor is:

$$\mathbf{C}^{*} = \frac{\mathbf{2}\mu}{\mathbf{tan}\,\alpha - \mu} \tag{2.2}$$

They even considered to use the operating point when the shoe factor is infinity i.e. the term $\mathbf{tan}\,\alpha - \mu$ becomes zero.

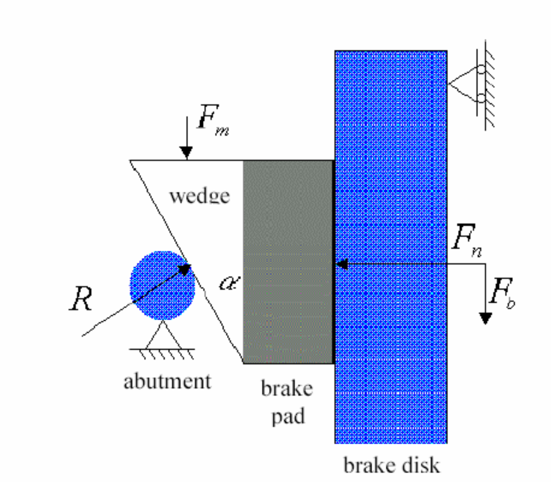


Fig 2.14 Wedge disc brake [Roberts 2002]

[Roberts and others, 2003] have presented another concept. They have proposed a self amplifying disc brake with two electrical actuators. As shown in Fig 2.15, the actuators generate the applied force in both forward and reverse directions. The actuators have a small size when compared with the previous. They have modeled the brake and they have compared the experimental and analytical results. Their results indicated that the power required to brake with self-amplifying disc brake is less than that without self-amplification.

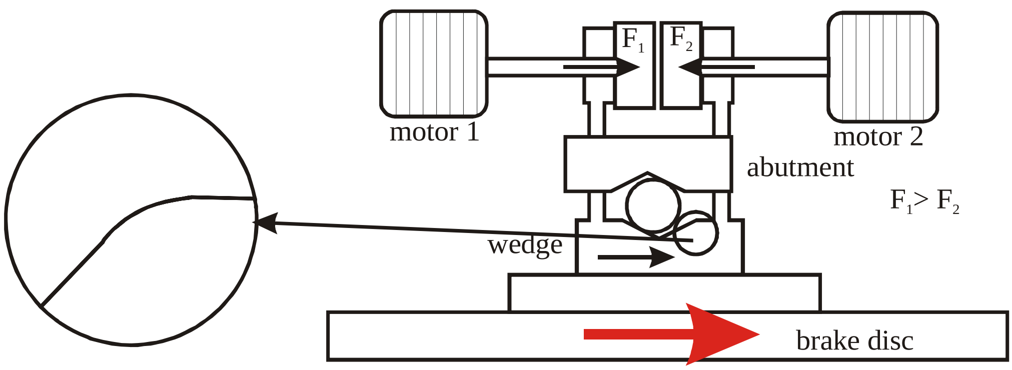


Fig 2.15 Wedge disc brake with self-amplification action [Roberts 2003]

[Dietrich and others, US 6318513B1 2001] have presented a patent aiming to provide the disc brake with an electromechanical system as well as to apply the self-amplification in the disc brake. They desired the shoe factor of the particular disc brake with self-amplification as:

$$C^{*}=\frac{\mu\tan\alpha}{\tan\alpha-\mu} \tag{2.3}$$

The difference between the forms of the shoe factor in equations (2.2) and (2.3) relates to the applied force direction. The applied force with the prototype of [Roberts and others, 2003] acts in the radial direction while the applied force with the prototype of [Dietrich and others, 2001] acts in the normal direction. The inventors have tried to gain the disc brake the advantage that it could be self-energized. Fig 2.16 shows a sketch of the electromechanical self-energized disc brake.

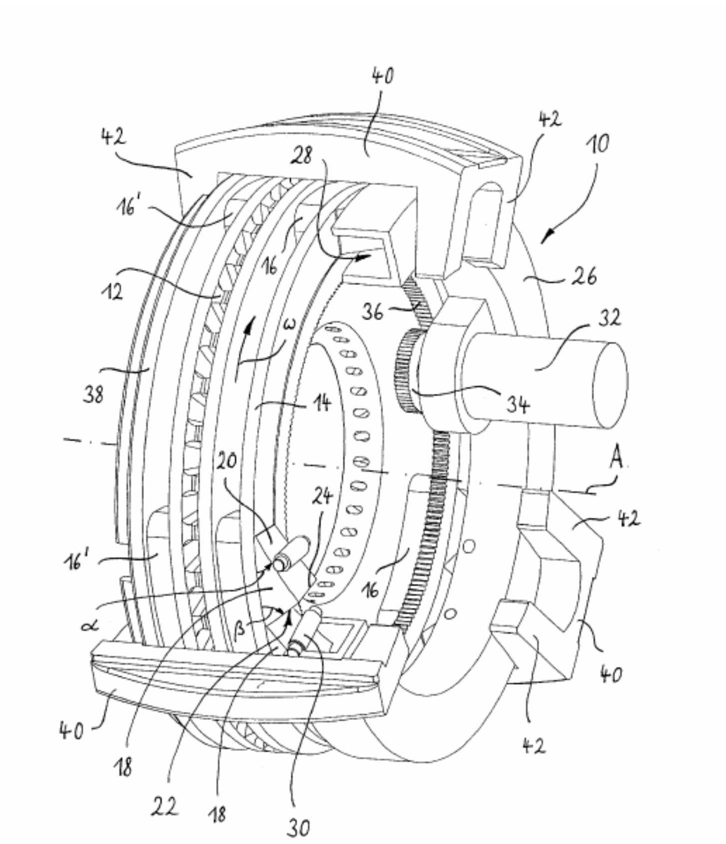


Fig 2.16 Electromechanical disc brake with self-amplification construction [Dietrich and others, 2001]

2.5.3 Drum brakes

There are some patents which deal with the drum brake like [Leber et al, 1998]. They aimed to provide the electromechanical sliding duplex drum brake with the possibility that the brake force would be kept at a constant value independently from the changes in the coefficient of friction. This is achieved not only by controlling the applied force but also by changing the angle α which is the angle between the resultant force of the brake and the normal force and the pivoting point. This patent aimed also to constant the brake shoe factor at a determined zone of the coefficient of friction variations. In this zone the shoe factor would be kept at a constant value by changing only the angle α . When the coefficient of friction is greater or less than these values of this zone limitations then the brake force will be kept at a constant value by changing the applied force. This prototype has the same problems like the classical duplex drum brake that the shoe factor has high values in forward motion while a huge reduction in the shoe factor value values in the reverse motion.

The patent which is presented by [Leber et al, 1998] is interested with the sliding duplex drum brake. They aimed to make a new adaptive drum brake. Their brake was a sliding duplex drum brake which was modified to an adaptive duplex drum brake. They have measured the friction force to calculate the value of the coefficient of friction μ. The control system extracts a suitable value of the inclination angle according the coefficient of friction value to maintain the shoe factor at constant value. They have used two actuators, one at each pivoting point side beside two mechanisms to generate the applied force at each shoe. This concept is limited by the coefficient of friction variations and the design limits for typical automotive brake i.e. the shoe factor is not constant at all times with the coefficient of friction variations. Also the duplex drum brake has a high shoe factor but still less than the duo servo one. There is also the disadvantage that in the reverse motion it has an enormous reduction of the shoe factor. It is like most electromechanical brakes relatively heavy and complex. Fig 2.17 shows schematic sketch of this brake concept. The friction force is maintained at constant value not only by controlling the inclination angle but also by controlling the applied force. In other words, the brake shoe factor in this model is not constant with variations of the coefficient of friction.

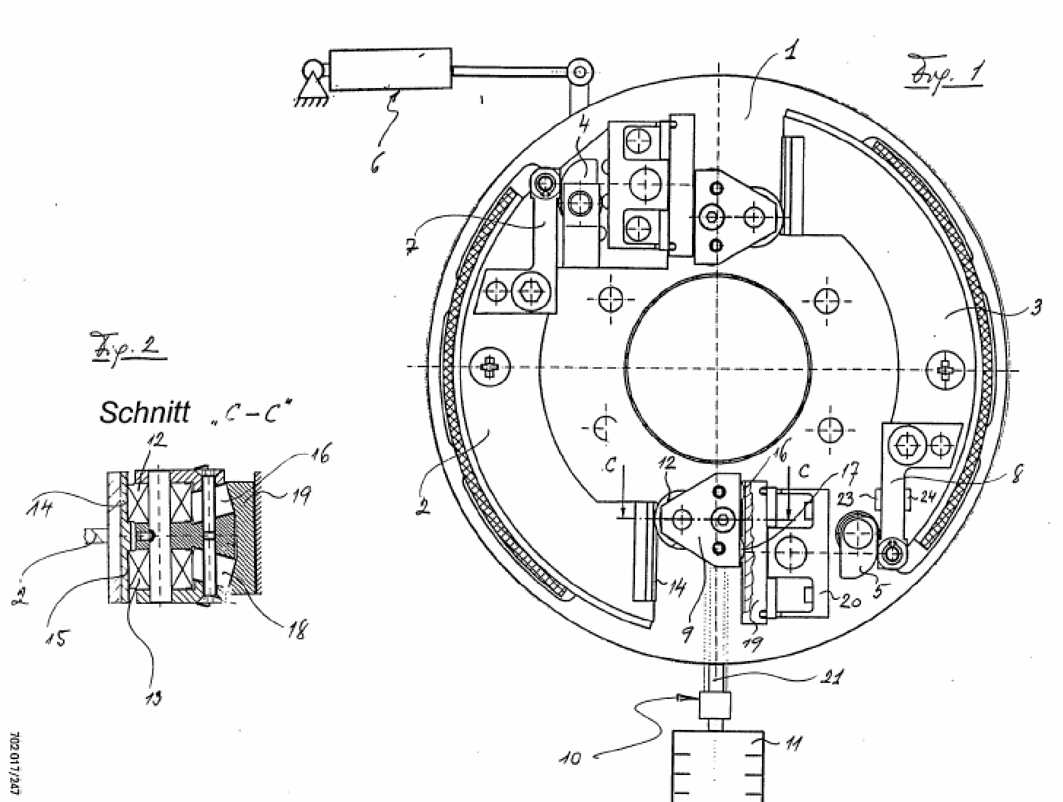


Fig 2.17 adaptive duplex drum brake concept [Leber et al, 1998]

3 Problem formulation

Several types of self-amplified brakes have been developed in the last few years. Initially, the drum brake was only self-amplified vehicle brake, but recently also the disc brake can also be self-amplified in combination with additional mechanisms like e.g. wedge. There are many efforts to develop a disc brake with self-amplification. The self amplified brakes have the possibility to generate a high brake force with low applied force. These mechanisms enable to use an actuator of small size and hence the consumed power can be reduced and overall brake system volume can be smaller.

3.1 Deficits of present day self-amplifying wheel brakes

The coefficient of friction is not constant with time or the number of brake applies. The coefficient of friction also depends on brake conditions such as temperature, applied force, and vehicle speed, which are always variable. The variations in the coefficient of friction cause a great variation in the brake force especially with drum brakes. The self amplified brakes are highly sensitive to variations of the coefficient of friction. In other words, at the same pedal force the generated brake force is not constant. Critical situation can occur at high temperature when the coefficient of friction is reduced. In this case, the brake force is enormously decreased. This phenomena is defined as brake fade.

The duo servo drum brake has the advantage that it can generate larger brake forces than any other brake type. At the same time it is highly sensitive to the coefficient of friction.

There are many problems of the self-amplifying wheel brake

- The self-amplifying brakes are sensitive to variations of the coefficient of friction variations.
- The generated brake force of self-amplifying brakes with the exception of the simplex drum brake and the duo servo type is not the same in both forward and reverse motions.
- The self-amplified brakes are more complex in design, there are many extra parts.

3.2 Proposed solution: adaptive self-amplification brake

As mentioned in the above section, the main disadvantage of the self-amplification brakes is the high sensitivity of the generated brake force to the coefficient of friction variations. To overcome this problem, there are two solutions:

- The first one is controlling the applied force that the brake force is maintained at constant value according to driver desire. The brake force can be maintained at constant value but it is then necessary to vary the applied force. Sometimes the required variations in the applied force are enormous especially when the coefficient of friction is reduced to minimum values, and large powerful actuators are needed.
- The other solution is to change the brake configuration during the brake process according to the variations of the coefficient of friction to maintain the brake force at constant values. The applied force will not be changed; hence the brake shoe factor will be maintained at constant values. The brake factor usually depends on the configuration of the brake system. The basic idea is to change the configuration during the brake process, in order to compensate for variations of the coefficient of friction.

As shown in Fig 3.1 and Fig 3.2, the brake force R of drum brake or wedge disc brake depends on the angle α and the coefficient of friction μ. The brake shoe factor is a function of the coefficient of friction and the inclination angle α .

$$C^{*} = f\,(\,\mu,\, a) \tag{3.1}$$

The brake force depends on the coefficient of friction as well as on the inclination angle. Variations in the coefficient of friction will be compensated by changing the inclination angle α .

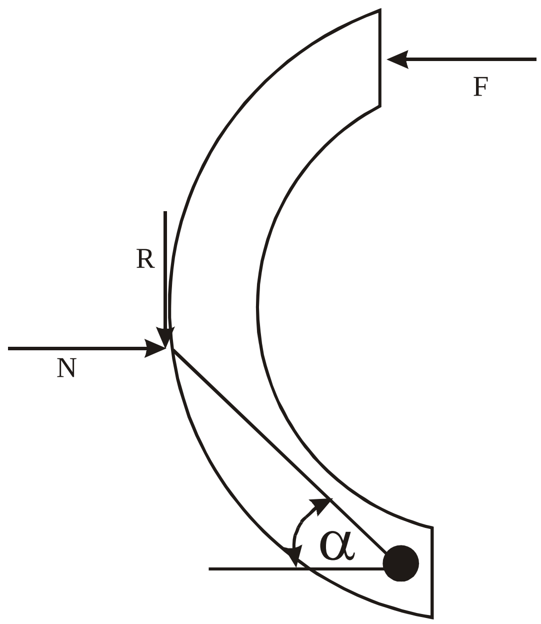


Fig 3.1 Drum brake primary shoe force analysis

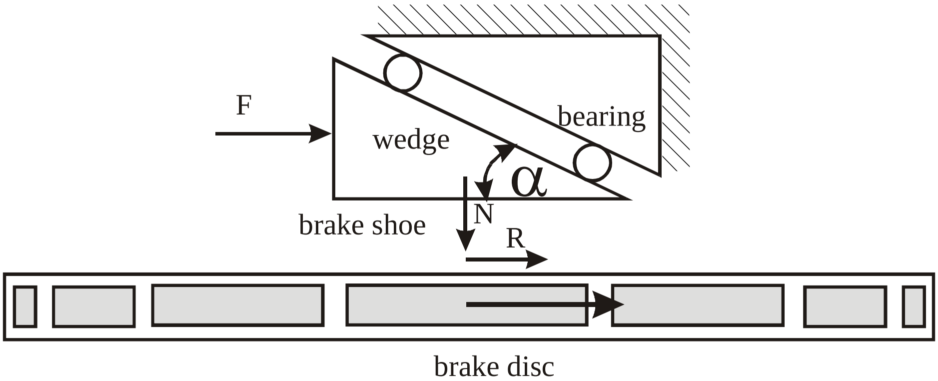


Fig 3.2 wedge disc brake force analysis

3.3 Aim of the study

The aim of the present study is therefore to:

1. Find a novel design for a duo servo drum brake which allows the adaptation of the angle α such that the brake shoe factor is constant even for varying coefficient of friction.
2. Derive a mathematical model which allows to calculate the quasistatic behavior of the system.
3. Design and build a prototype in order to validate the mathematical model and to perform experiments on a full scale test rig.
4. Derive a control algorithm for the adaptation of the angle α, implement and test it in the full scale test rig.
5. Evaluate the performance of the proposed design to provide a basis for feasibility studies concerning the applications of the concept in automotive braking systems.

4 Analysis of self-amplifying wheel brakes

4.1 Lever brake

A particular form of the self-amplification effect is shown in Fig 4.1. A probe is compressed against a moving band by an applied force F. When the band moves with a velocity $\boldsymbol{v}$, a tangential force R will be generated. The normal force $\boldsymbol{N}$ which accompanies the friction force R is larger than the applied force F. This phenomenon is known as self-amplification action.

Equilibrium of the moment of momentum about the pivot demands

$$F\ L\sin\alpha + R\ L\cos\alpha = N\ \ L\sin\alpha \tag{4.1}$$

Dividing by $L\sin\alpha$

$$F + \frac{R}{\tan\alpha} = N \tag{4.2}$$

With the coefficient of friction we obtain

$$\mu = \frac{R}{N}$$

$$F = \frac{R}{\mu} - \frac{R}{\tan\alpha} = R\left(\frac{\tan\alpha - \mu}{\mu\tan\alpha}\right) \tag{4.3}$$

The ratio between the friction force $\boldsymbol{R}$ and the applied force $\boldsymbol{F}$ is:

$$\frac{R}{F} = \frac{\mu\tan\alpha}{\tan\alpha - \mu} \tag{4.4}$$

The amplification factor depends on the coefficient of friction and on the inclination angle. If the motion is in the opposite direction, the generated friction force will be reduced. In this case the system will be called self-debilitated brake. Many researchers have studied this phenomenon such as [Severin et al, 2001 and Blau, 1996]. The form of the relation between the shoe factor and the tangential force with primary or leading shoe is:

$$\boldsymbol{C}^* = \frac{\mu\, tan\alpha}{tan\alpha - \mu} \tag{4.5}$$

Fig 4.2 shows the variations of the brake shoe factor C^* with the coefficient of friction at inclinations α of 10°, 30°, 50°, and 70°. With the increase of the coefficient of friction μ the brake shoe factor C^* also increases. Also, the brake shoe factor C^* increases with the decrease of the inclination α. When the coefficient of friction μ value equals the tangent of α, the brake shoe factor C^* goes to infinity i.e. the brake is blocked.

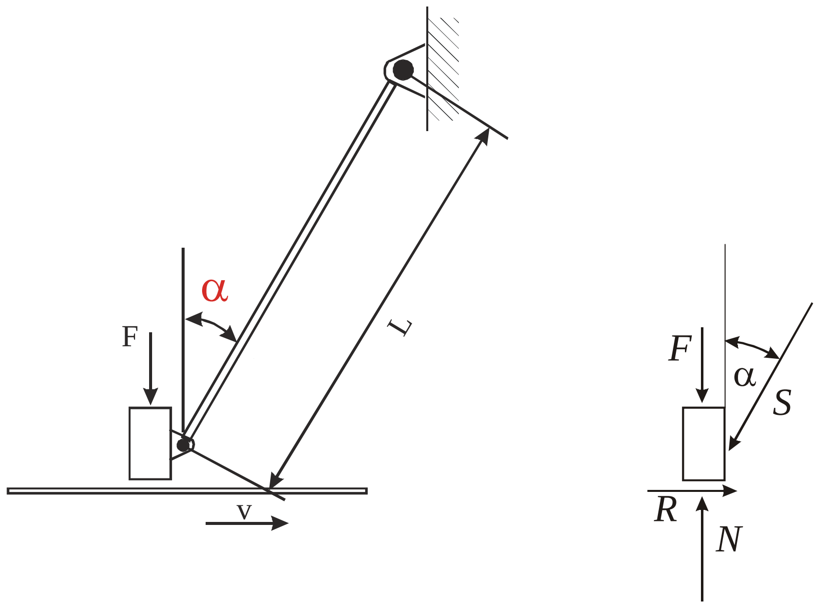


Fig 4.1 Lever brake force analysis

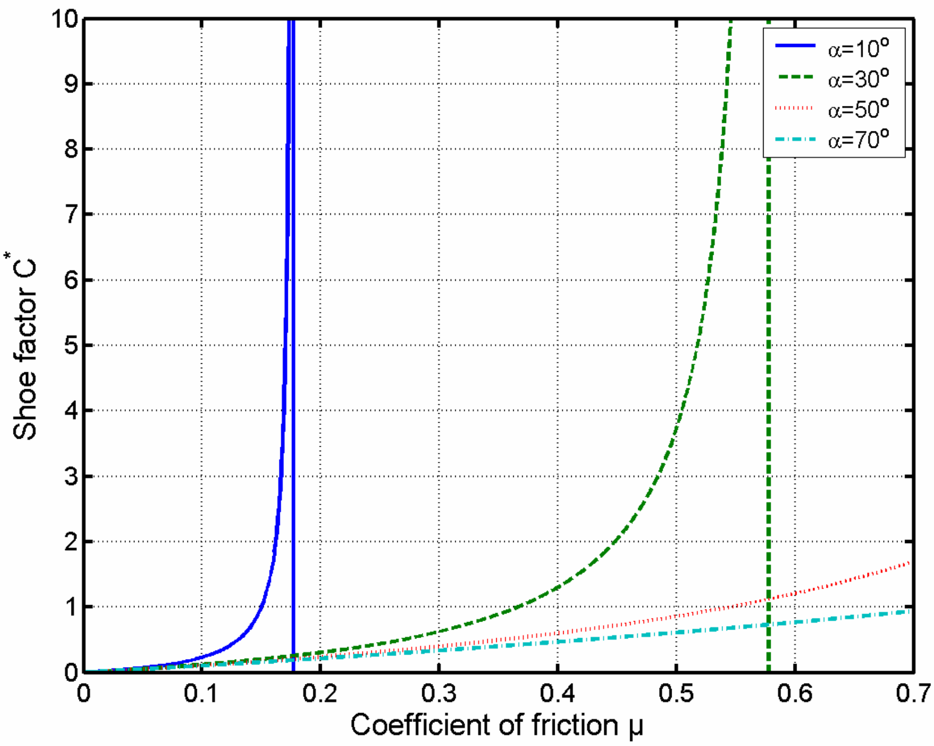


Fig 4.2 Effect of inclination angle α on the shoe factor of the lever brake

4.2 Wedge brake

The wedge disc brake also has a self-amplifying effect. In the last few years there were many efforts to further develop this particular type of the disc brake with the aim to reduce the needed applied force. Fig 4.3 shows the wedge brake design principle. The applied force F acts at the wedge to generate the brake force R between the brake pad and the brake disc. As a result of the friction, the force R is self-amplified according to the following relations.

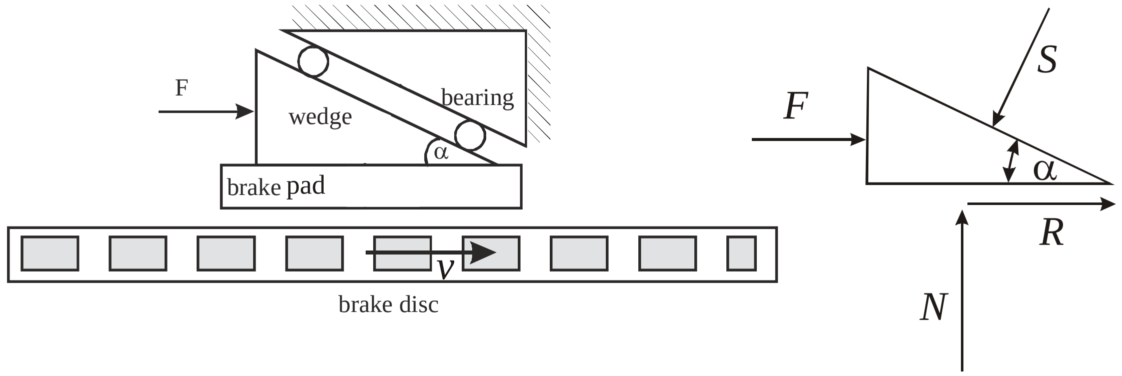


Fig 4.3 Wedge disc brake

$$F + R = S \sin\alpha \tag{4.6}$$

$$N = S \cos\alpha \tag{4.7}$$

Using the coefficient of friction to express the normal force as $N = \dfrac{R}{\mu}$ we obtain

$$R = \mu\, S \cos\alpha \tag{4.8}$$

then, by substituting in (4.6) we obtain

$$F = \frac{R}{\mu}\,\frac{\sin\alpha}{\cos\alpha} - R$$

and the shoe factor for the wedge brake is:

$$C^* = \frac{R}{F} = \frac{\mu}{\tan\alpha - \mu} \tag{4.9}$$

It should be noticed that this holds for a single wedge. For the floating type we obtain

$$C^* = \frac{R}{F} = \frac{2\mu}{\tan\alpha - \mu} \tag{4.10}$$

The effect of the coefficient of friction μ on the shoe factor of the wedge disc brake (floating type) is alike that with the lever brake. Fig 4.4 shows the variations of the brake shoe factor C^* of the wedge disc brake (floating type) with the coefficient of friction at inclinations α of 10°, 30°, 50°, and 70°.

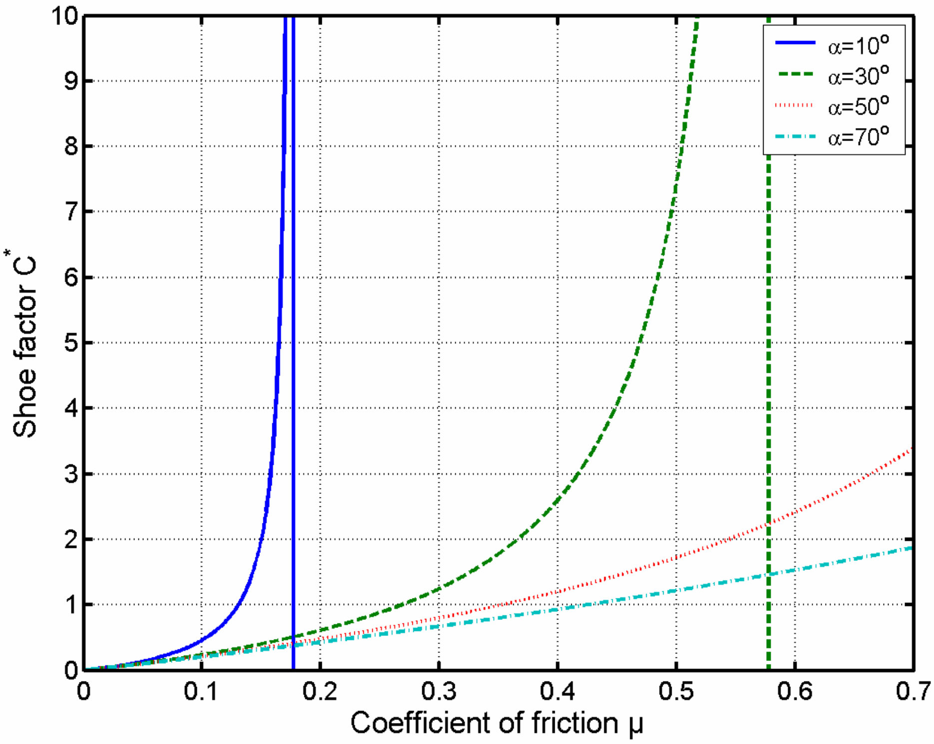


Fig 4.4 Effect of inclination angle α on the shoe factor of the wedge disc brake

4.3 Drum brakes

All drum brakes are self-amplifying brakes. According to the brake shoe and its pivoting point arrangement, the brake shoe is called leading or trailing. In the following sections, the different types of drum brakes will be discussed in detail.

4.3.1 Simplex drum brake

According to the shoe configuration, there are various types of drum brakes. The simplex drum brake has pivoted shoes. In this type of brake the applied force is produced either by a hydraulic cylinder or by an electromechanical actuator. The effect of the coefficient of friction on the shoe factor and the effect of different pivoting point positions will be investigated. The simplex drum brake has two different shoes, the primary (leading) and the secondary (trailing) shoe. Fig 4.5 illustrates the forces acting on the simplex drum brake primary shoe. By evaluating the moment of momentum for point B we have:

$$F(a+b)+R_1\,c-N_1\,a=0 \tag{4.11}$$

$$N_1=\frac{R_1}{\mu} \tag{4.12}$$

$$F(a+b)=\frac{R_1}{\mu}a-R_1\,c \tag{4.13}$$

$$F(a+b)=\frac{R_1}{\mu}(a-c\,\mu) \tag{4.14}$$

The shoe factor of the primary shoe C_1 is the ratio between the brake force R_1 and the applied force F.

$$C_1 = \frac{R_1}{F} = \frac{\mu(a+b)}{a - c\,\mu} \tag{4.15}$$

An equivalent angle of inclination can be defined as:

$$\alpha = \tan^{-1}\left(\frac{a}{c}\right) \tag{4.16}$$

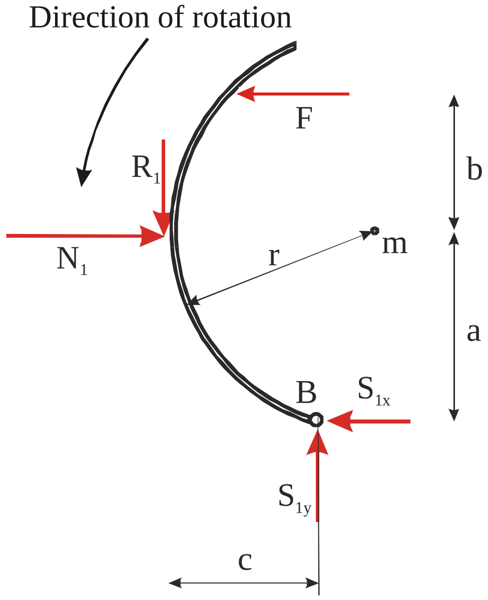


Fig 4.5 Simplex drum brake primary shoe force analysis

Fig 4.6 shows the force analysis for the simplex drum brake secondary shoe. Similarly, by evaluating moment of momentum for the pivoting point C, the form of the shoe factor C_2 is as follows:

$$C_2 = \frac{R_2}{F} = \frac{\mu(a+b)}{a + c\,\mu} \tag{4.17}$$

The overall shoe factor of the simplex drum brake is the summation of the shoe factor of the primary shoe and the shoe factor of the secondary shoe:

$$C^* = C_1 + C_2 = \frac{\mu(a+b)}{a - c\,\mu} + \frac{\mu(a+b)}{a + c\,\mu} \tag{4.18}$$

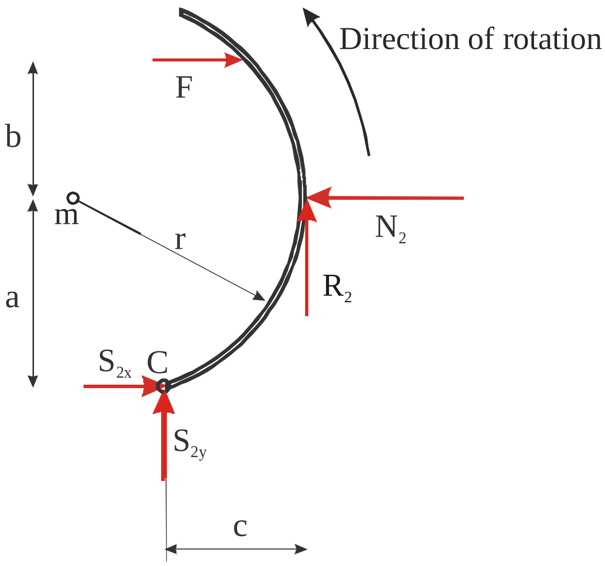


Fig 4.6 Simplex drum brake secondary shoe force analysis

The generated brake force is self-amplified due to the friction process at the primary shoe. The brake force at the secondary shoe is self-debilitated. Overall the self-amplification dominates, see Fig 4.7.

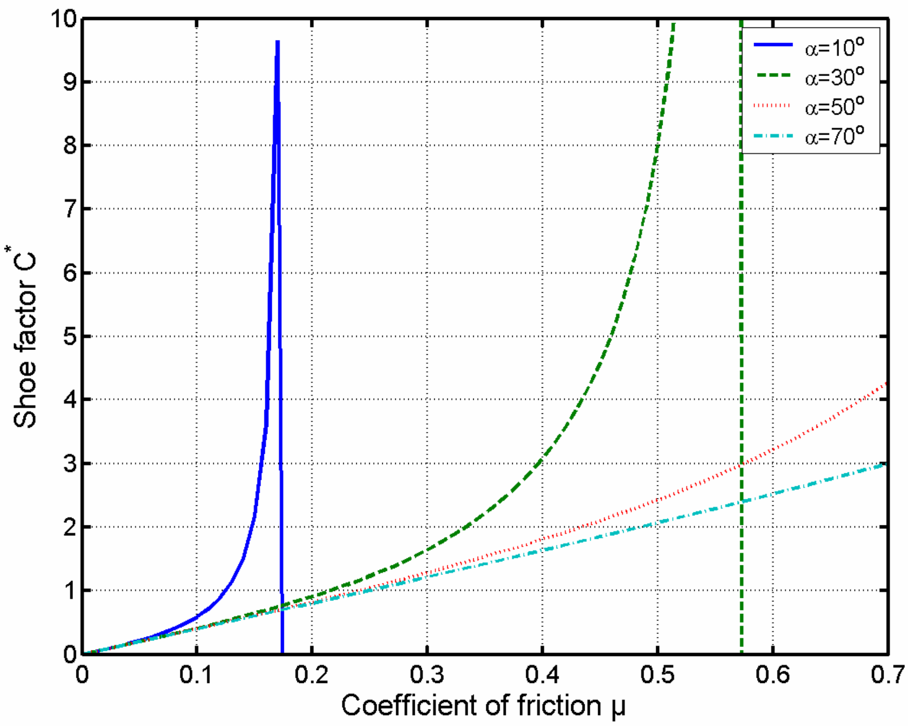


Fig 4.7 Effect of inclination angle α *on the shoe factor of the simplex drum brake* $(\tan\alpha = \frac{a}{c})$

4.3.2 Duplex drum brake

The duplex drum brake consists of two similar shoes. It is designed such that both of the brake shoes are self-amplified in the forward motion. It has the disadvantage that it is self-debilitated in the reverse motion. Thus, the generated brake force equals two times that of the generated brake force with the simplex primary shoe in the forward motion and twice that of the generated brake force with the simplex secondary shoe in backward motion. The shoe factor for forward motion is:

$$C^* = \frac{2\mu(a+b)}{a - c\,\mu} \tag{4.19}$$

And that for reverse motion is:

$$C^* = \frac{2\mu(a+b)}{a + c\,\mu} \tag{4.20}$$

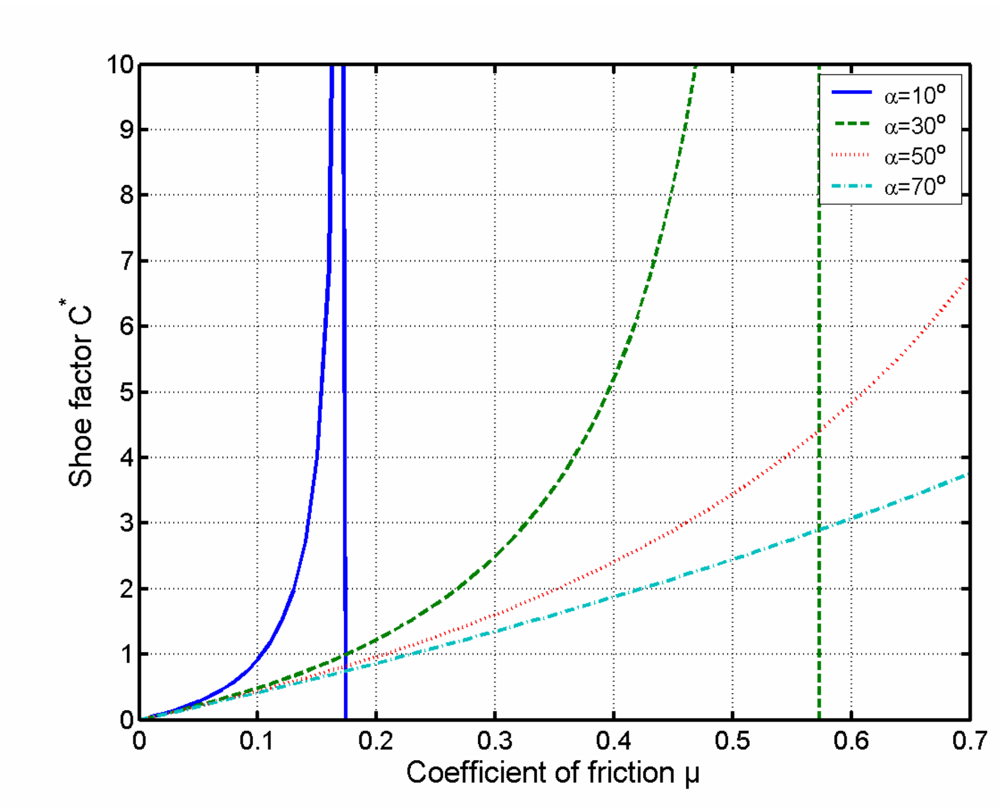


Fig 4.8 Effect of inclination angle α on the shoe factor of the duplex drum brake in forward motion ($\tan\alpha = \frac{a}{c}$)

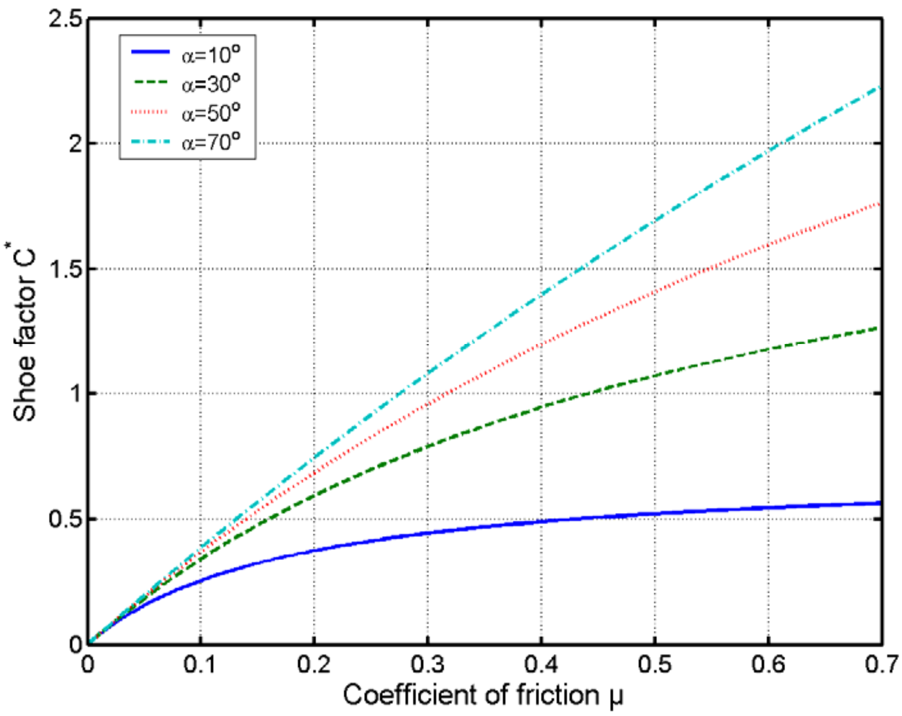


Fig 4.9 Effect of inclination angle α *on the shoe factor of the duplex drum brake in reverse motion* ($\tan\alpha = \frac{a}{c}$)

4.3.3 Duo servo drum brake

The duo servo drum brake has high self-amplification in forward and in reverse motion. But at the same time it is highly sensitive to variations of the coefficient of friction. The duo servo drum brake gets its name from the fact that in both directions of wheel rotation, both brake shoes are self-energized. The secondary shoe is not only activated by the applied force but also by the reaction force of the primary shoe. This is the main advantage of the duo servo drum brake. The duo servo has two types according to the construction of the link between the primary and the secondary shoes. The first one is the duo servo drum brake with floating link. The other one is duo servo drum brake with guided link.

In the following sections the force analysis for both types will be discussed.

4.3.3.1 Duo servo drum brake with floating link

The duo servo drum brake with floating link is widely used in the automotive industry. It is characterized by a link between the primary and the secondary shoe which can float freely. The link only transmits force in its longitudinal direction. Fig 4.10 shows the force analysis of the duo servo drum brake with floating link. Also, Fig 4.11 shows the force distribution at the primary and the secondary shoe of the duo servo drum brake with floating link.

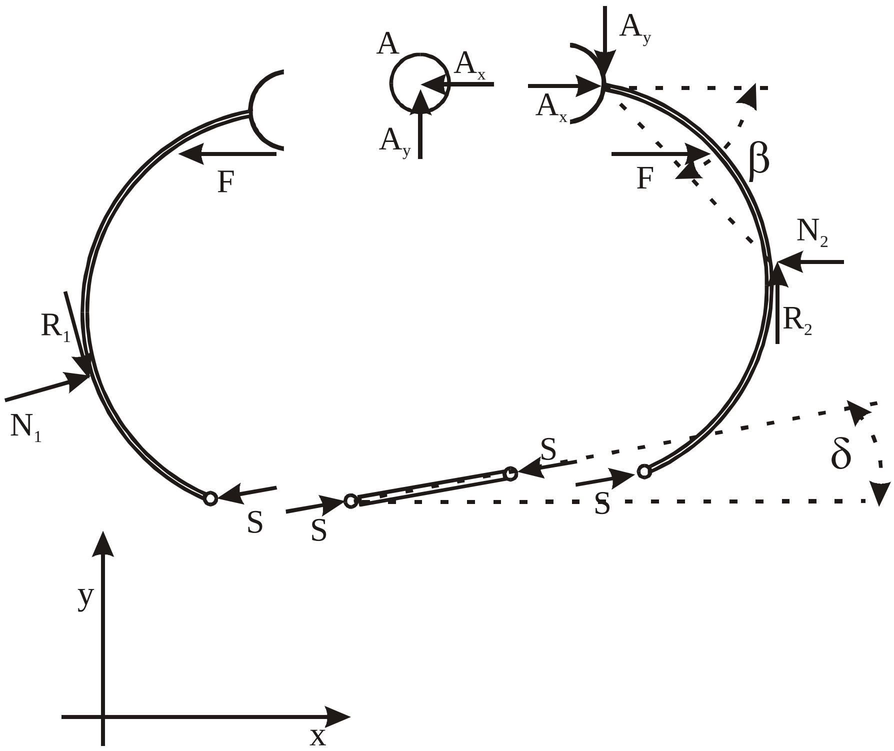


Fig 4.10 Duo servo drum brake with floating adjusting lever force analysis

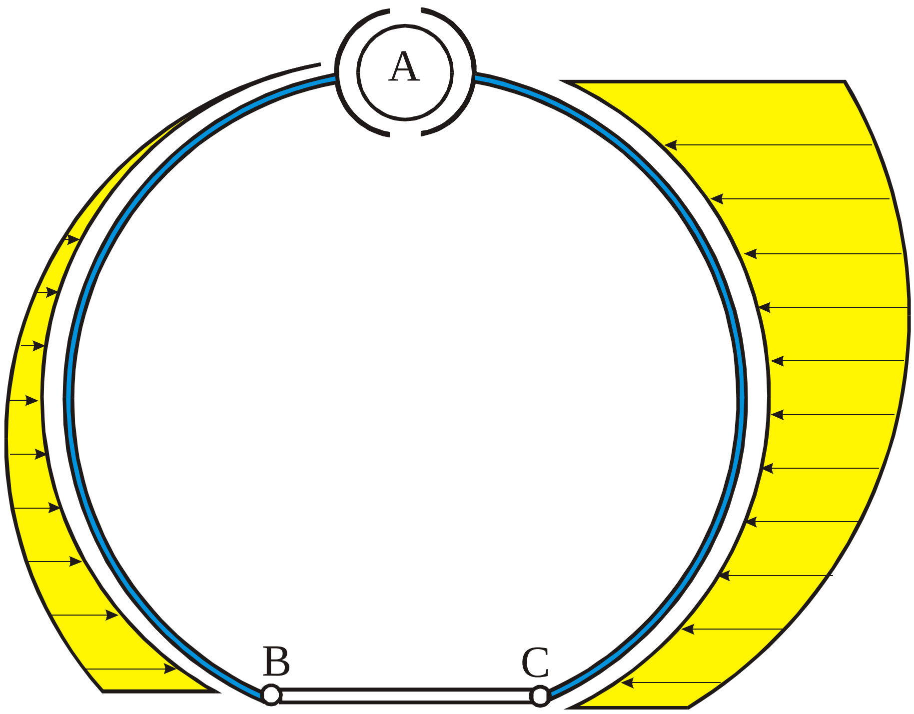


Fig 4.11 Duo servo drum brake force distribution

4.3.3.1.1 Primary shoe

The primary shoe is always applied by the force F . The resultant of both friction and normal forces are R_1 and N_1 respectively.

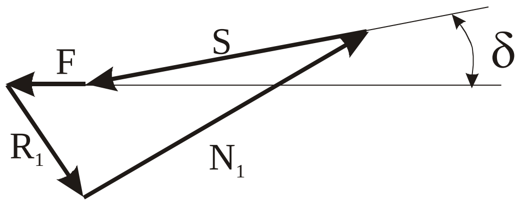


Fig 4.12 Duo servo primary shoe force polygon

The primary shoe force polygon which is shown in Fig 4.12 illustrates the forces that act on the primary shoe. The applied force F always acts in x-direction. The friction force R_1 is perpendicular to the normal force N_1. The equivalent resultant forces must lie at the same adjusting lever inclination angle δ because the lever can only transmit forces in normal direction. The angle δ is limited by the clearance between the drum wheel and the brake shoes, and the adjusting lever length. It is normally so small that it can be neglected. It is therefore possible to consider the force S acting in the x-direction. From the force polygon geometry shown in Fig 4.13 the force S can be calculated as follow:

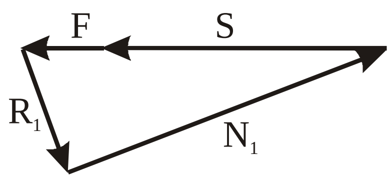


Fig 4.13 Duo servo primary shoe force polygon [considering $\delta = 0$]

$$S = \sqrt{R_1^2 + N_1^2} - F \tag{4.21}$$

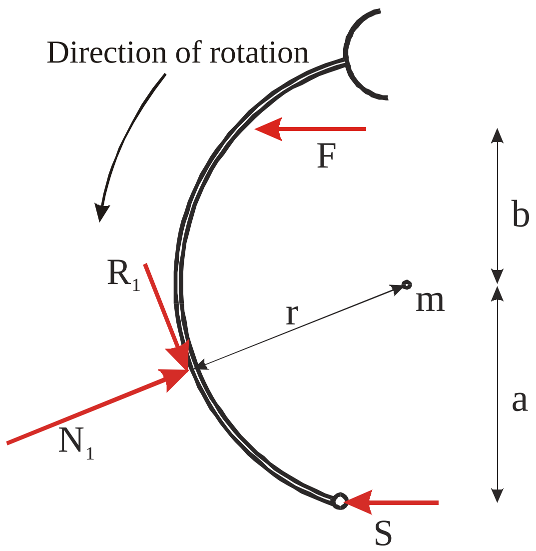


Fig 4.14 the forces at the primary shoe

According to the forces that act at the primary shoe, shown in Fig 4.14, the moment of momentum around point m yields:

$$F\ b + R_1\ r\ = S\ \ a \tag{4.22}$$

With $N_1 = \dfrac{R_1}{\mu}$ we obtain from (4.21)

$$S = \frac{R_1}{\mu}\sqrt{1+\mu^2} - F \qquad (4.23)$$

And by substituting in equation (4.22):

$$F(a+b) = R_1\left[\frac{a\sqrt{1+\mu^2} - \mu\, r}{\mu}\right] \qquad (4.24)$$

And the shoe factor for the primary shoe becomes

$$C_1 = \frac{R_1}{F} = \frac{\mu(a+b)}{a\sqrt{1+\mu^2} - \mu\, r} \qquad (4.25)$$

4.3.3.1.2 Secondary shoe

The secondary shoe is applied by two forces, the applied force F and the reaction force S_1. The anchor pin is the pivoting point of the secondary shoe. By equilibrium of the moment of momentum for the drum centre point m,taking into account that the effect of the force S_{2y} can be neglected, we obtain

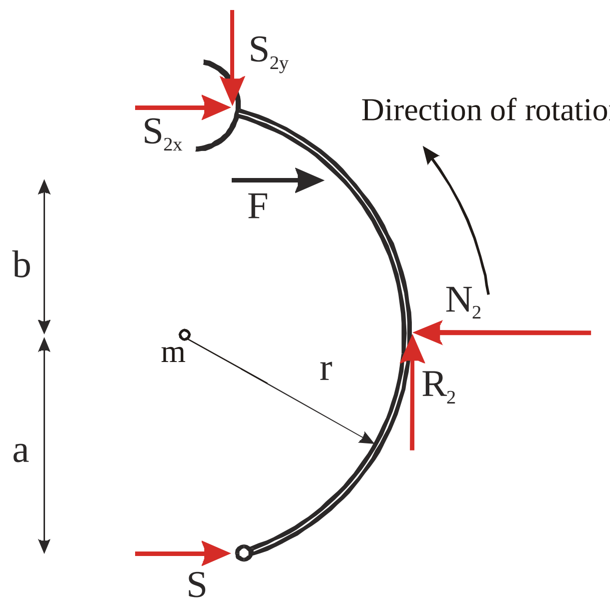


Fig 4.15 the forces at the secondary shoe

$$S\ a + R_2\ r - F\ b = S_{2x}\ r \qquad (4.26)$$

The reaction force at the anchor pin S_2 can be calculated as follow:

The summation of the forces in x-direction yields

$$\sum F_x = 0, \qquad N_2 - S - F - S_{2x} = 0 \qquad (4.27)$$

and we obtain

$$S_{2x} = N_2 - S - F \qquad (4.28)$$

and by substituting in (4.26) we obtain

$$S\,a + R_2\ r - F\ b = \left(N_2 - S - F\right)r \tag{4.29}$$

with $N_2 = \dfrac{R_2}{\mu}$ we obtain

$$S(a+r) - F(b-r) = R_2\,\frac{r(1-\mu)}{\mu} \tag{4.30}$$

using $S = R_1\,\dfrac{\sqrt{1+\mu^2}}{\mu} - F$ we obtain

$$\left(R_1\ \frac{\sqrt{1+\mu^2}}{\mu} - F\right)(a+r) - F\,(b-r) = R_2\ \frac{r(1-\mu)}{\mu} \tag{4.31}$$

$$\left(R_1\ \frac{\sqrt{1+\mu^2}}{\mu}\right)(a+r) - F(a+b) = R_2\,\frac{r(1-\mu)}{\mu} \tag{4.32}$$

dividing by $\dfrac{r(1-\mu)}{\mu}$ we obtain

$$R_1\,\frac{a+r}{r}\,\frac{\sqrt{1+\mu^2}}{(1-\mu)} - F\,\frac{(a+b)}{r}\,\frac{\mu}{1-\mu} = R_2 \tag{4.33}$$

Dividing by F we obtain

$$\frac{R_1}{F}\,\frac{a+r}{r}\,\frac{\sqrt{1+\mu^2}}{(1-\mu)} - \frac{(a+b)}{r}\,\frac{\mu}{1-\mu} = \frac{R_2}{F} \tag{4.34}$$

and with $C_1 = \dfrac{R_1}{F}$ and $C_2 = \dfrac{R_2}{F}$ we obtain

$$C_2\ = C_1\left(\frac{a+r}{r}\,\frac{\sqrt{1+\mu^2}}{(1-\mu)}\right) - \frac{(a+b)}{r}\,\frac{\mu}{1-\mu} \tag{4.35}$$

And by considering $k_1 = \left(\dfrac{a+r}{r}\,\dfrac{\sqrt{1+\mu^2}}{(1-\mu)}\right)$ and $k_2 = -\dfrac{(a+b)}{r}\,\dfrac{\mu}{1-\mu}$ we finally obtain:

$$C_2 = k_1\ C_1\ + k_2$$

The overall duo servo drum brake shoe factor is the sum of the shoe factors of the primary and the secondary shoe:

$$C^* = C_1\ + C_2 \tag{4.36}$$

$$C^* = C_1\,(1 + k_1) + k_2 \tag{4.37}$$

It is difficult to plot the brake shoe factor C^* as a function of the inclination angle α for the duo servo drum brake with floating link because the resultant brake force of the primary shoe R_1 does not lie at the centre of the brake shoe. Whereas the inclination angle α depends on the floating link position a, it can also be plotted as a function of the position a. We have substituted the main dimensions of an actual drum brake of VW Golf II at rear axle. Table shows the main dimensions of the drum brake of VW Golf II.

Table 4.1Main dimensions of the drum brake of VW Golf II

Parameter	Dimensions (mm)
Drum radius.	90
Floating link position a from the drum centre.	55
Distance between hydraulic cylinder and drum centre b	45

Fig 4.16 shows the variations of the brake shoe factor C^* of the duo servo drum brake with floating link with the coefficient of friction μ. The brake shoe factor C^* is plotted at floating link position of 40, 50, and 60 mm. An increase of the coefficient of friction leads to a corresponding increase of the brake shoe factor while a decrease of the floating link position increases the brake shoe factor.

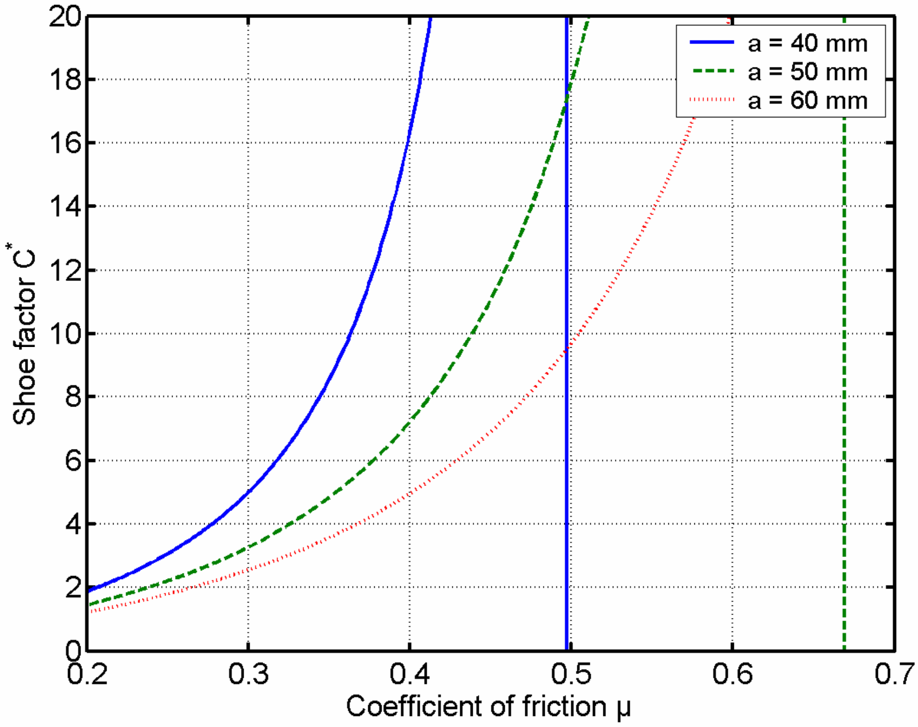


Fig 4.16 The effect of floating link position on the duo servo drum brake shoe factor as a function of the coefficient of friction.

4.3.3.2 Duo servo drum brake with guided link

In the case of the duo servo drum brake with guided link, the link is guided with a bearing that allows the only motions in the axial direction. Fig 4.17 shows the force analysis for the duo servo drum brake with guided link.

4.3.3.2.1 Primary shoe

The balance of the moment of momentum around the pivot S yields:

$$F(a+b)+R_1\,c=N_1\,a \tag{4.38}$$

With $N_1=\dfrac{R_1}{\mu}$ we obtain

$$F(a+b)=R_1\left(\frac{a-\mu c}{\mu}\right) \tag{4.39}$$

The shoe factor becomes:

$$C_1=\frac{R_1}{F}=\frac{\mu(a+b)}{a-\mu c} \tag{4.40}$$

4.3.3.2.2 Secondary shoe

The moment of momentum around the anchor pin yields:

$$F\ (r-b)+R_2\,r+S_x\,(r+a)-N_2\,r=0 \tag{4.41}$$

With $S_x=N_1-F$ we obtain

$$F\ (r-b)+R_2\,r+(N_1-F)(r+a)=N_2\,r \tag{4.42}$$

And with $N_1=\dfrac{R_1}{\mu}$ and $N_2=\dfrac{R_2}{\mu}$ it becomes

$$F\ (r-b-r-a)+R_2\,r+\frac{R_1}{\mu}(r+a)=\frac{R_2}{\mu}\,r \tag{4.43}$$

$$-F\ (a+b)+\frac{R_1}{\mu}(r+a)=R_2\,r\,\frac{(1-\mu)}{\mu} \tag{4.44}$$

And dividing by $r\dfrac{(1-\mu)}{\mu}$

$$-F\ \frac{(a+b)}{r}\frac{\mu}{(1-\mu)}+R_1\frac{(r+a)}{r(1-\mu)}=R_2 \tag{4.45}$$

Dividing by F and considering $C_1=\dfrac{R_1}{F}$ and $C_2=\dfrac{R_2}{F}$

$$C_2=C_1\frac{(r+a)}{r(1-\mu)}-\frac{(a+b)}{r}\frac{\mu}{(1-\mu)} \tag{4.46}$$

And by considering $k_1=\dfrac{(r+a)}{r(1-\mu)}$ and $k_2=-\dfrac{(a+b)}{r}\dfrac{\mu}{(1-\mu)}$ we obtain:

$$C_2=k_1\,C_1+k_2$$

The overall duo servo drum brake shoe factor is the sum of the shoe factors of the primary and the secondary shoe.

$$C^{*} = C_1 + C_2 \tag{4.47}$$

$$C^{*} = C_1(1+k_1)+k_2 \tag{4.48}$$

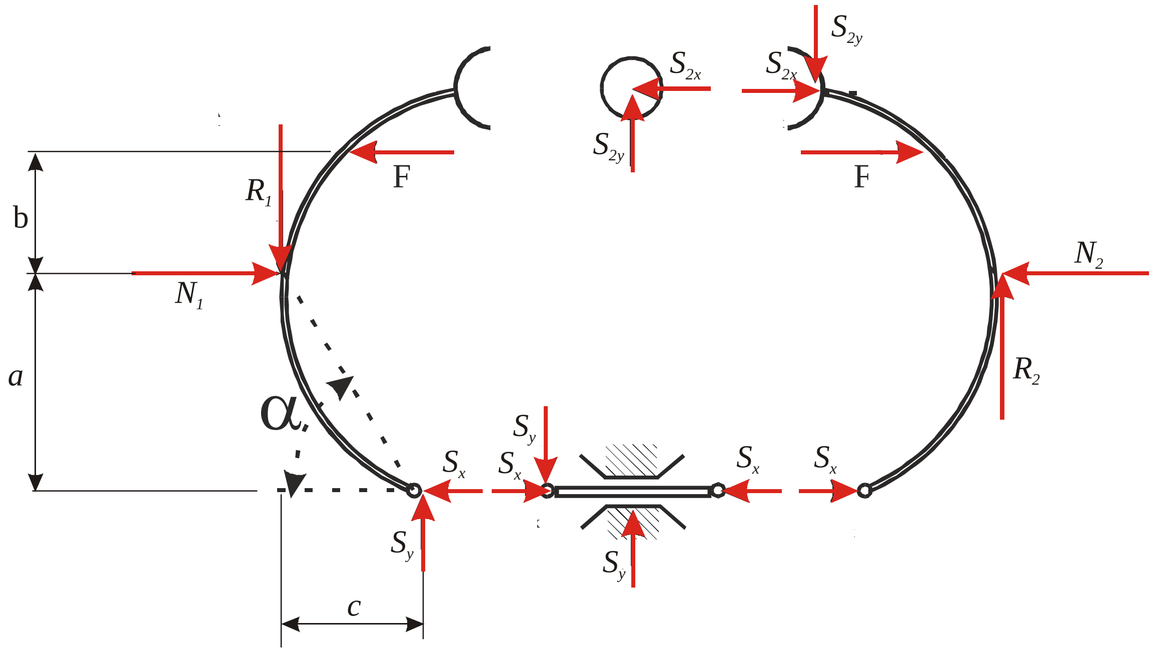


Fig 4.17 Free body diagram of the duo servo drum brake with guided link

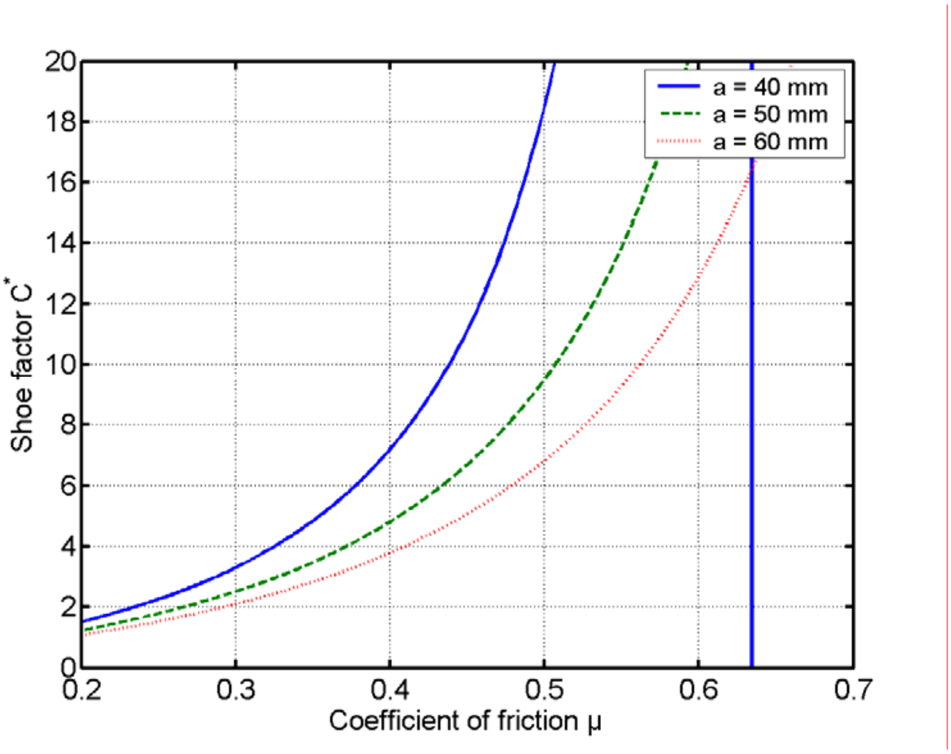


Fig 4.18 The effect of adjusting link position on the shoe factor of the duo servo drum with guided link brake as a function of the coefficient of friction.

Fig 4.18 shows the shoe factor of the duo servo drum brake with guided link as a function of the coefficient of friction. The brake shoe factor is plotted for link positions of 40, 50, and 60 mm.

4.3.3.3 Comparison between guided and floating link duo servo drum brake

In order to compare between the duo servo drum brakes with floating and guided link, it is necessary to plot the brake shoe factor as a function of the link position a at the same brake system. The dimensions of the VW Golf II are also used in this comparison. Fig 4.19 shows the brake shoe factor C^* of the duo servo drum brake with floating and guided link as a function of the coefficient of friction μ for link positions of 40 mm and 60 mm. From this figure it can be seen that at the same link position a, the brake shoe factor of the duo servo drum brake with floating link is higher than that with guided link. The difference increases with the increase of the coefficient of friction.

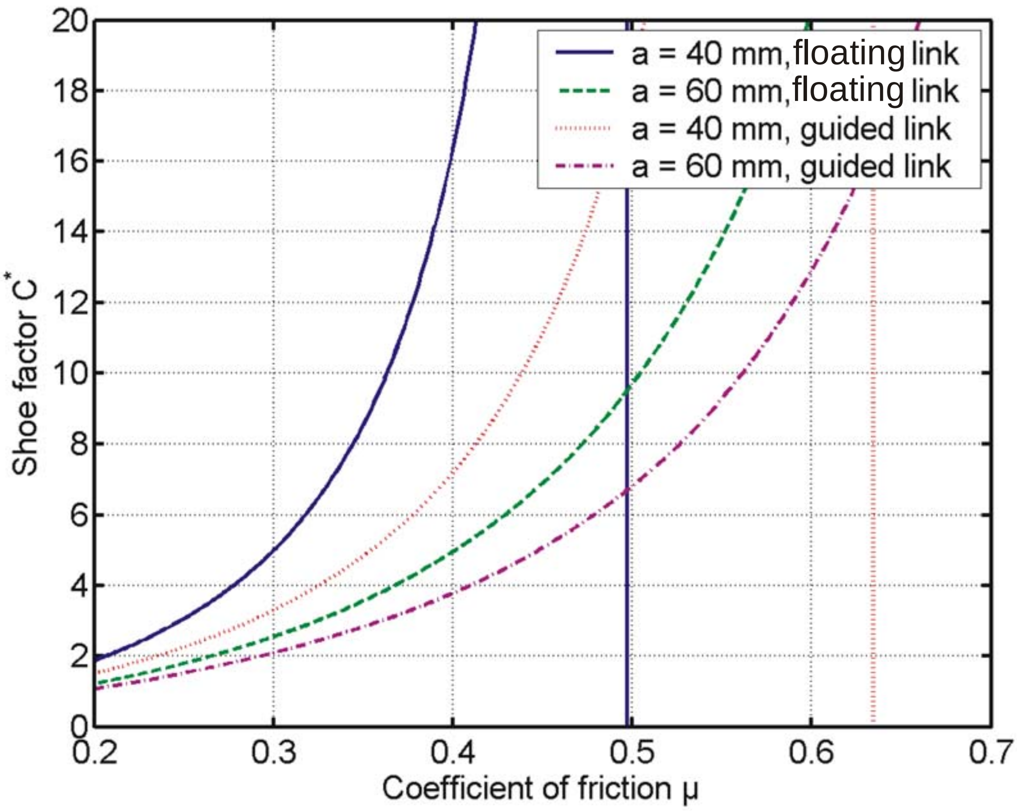


Fig 4.19 Comparison between the duo servo drum brake with floating and guided links as a function of the coefficient of friction.

4.4 Comparison of the concepts and selection of the most promising solution approach

From the investigations of the force analysis for the self-amplification braking systems, the duo servo drum brake has the highest level of self amplification. The duplex drum brake also has a high level of self amplification but it has only a low level of self-amplification at reverse motion. The simplex drum brake and wedge disc brake have still lower levels of self-amplification than the duo servo or duplex drum brake.

The duo servo drum brake with floating link was chosen for our investigation because it offers the highest level of self-amplification and because it has equal performance for both forward and reverse braking maneuvers.

5 Design of a prototype of an adaptive duo servo drum brake

5.1 General concept

The aim of the work is to investigate an adaptive duo servo drum brake where the inclination angle α can be varied to maintain the brake shoe factor C^* at high and constant values. To achieve these requirements it was decided to design a duo servo drum brake where the adjusting lever can be moved along the shoes to change the inclination angle α. A linear actuator for example can be used to move the adjusting lever. The floating lever has the role to transfer the primary shoe reaction force to the secondary shoe. It is also an element of adjusting the clearance between the drum and the shoes. Two roller bearings are used at the ends of the floating link. The shoes have two grooves to enable the adjusting lever with its bearings to move up and down. The floating link must be able to move in the axial direction for applying the secondary shoe with the reaction force of the primary shoe. To enable this movement, a linear guide is constructed between the adjusting lever and the linear actuator. Fig 5.1 shows the concept for the adaptive duo servo drum brake.

The applied force presses the brake shoes into contact with the drum. Then, the friction force rotates the primary shoe about the floating link pivoting point. The reaction force at the floating link is then transmitted to the secondary shoe. The secondary shoe is applied by two forces; apply force of the wheel cylinder and the floating link reaction force.

As mentioned earlier, the coefficient of friction varies during braking and depends on many parameters such as normal force, temperature, speed and others. The generated force is highly sensitive to variations in the coefficient of friction. The main idea of the adaptive duo servo drum brake is that when the coefficient of friction is known, the floating link position can be adjusted to maintain the brake shoe factor at a constant and high value.

The coefficient of friction can be calculated by measuring the brake forces or by using an observer (how to construct the observer will be explained in detail in the next chapters). An adapter is programmed to calculate the required linear motion of the actuator that adjusts the floating link position.

The actuator job is to move the adjusting lever along the shoe in the grooves. The actuator is perpendicular to the floating link. The grooves are arranged in the brake case parallel to each other and at the same time parallel to the actuator arm. Fig 5.2 shows a Plexiglas model of the adaptive duo servo drum brake model.

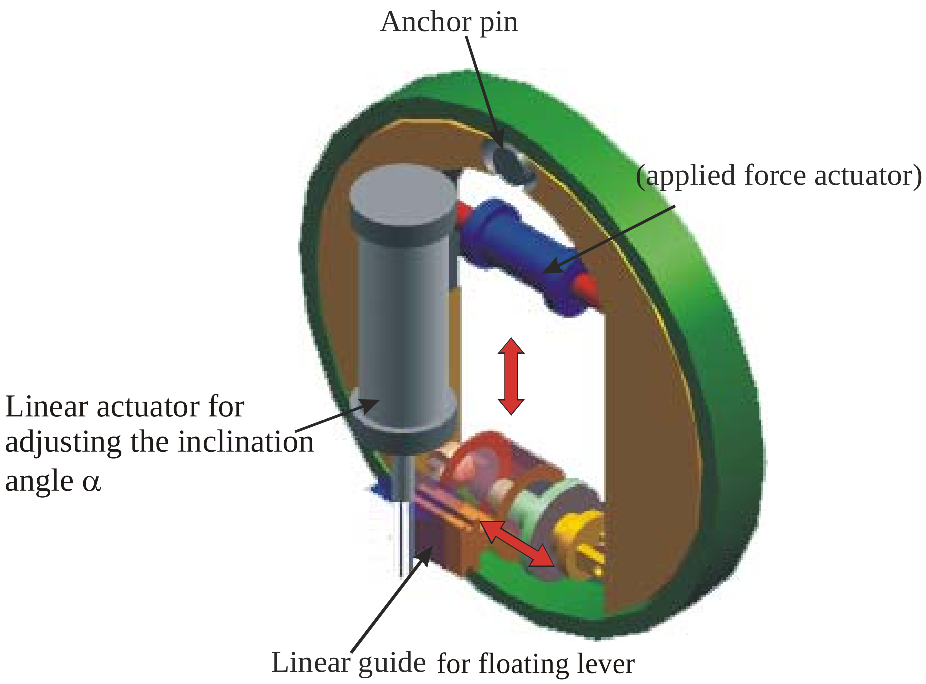


Fig 5.1 adaptive duo servo drum brake concept

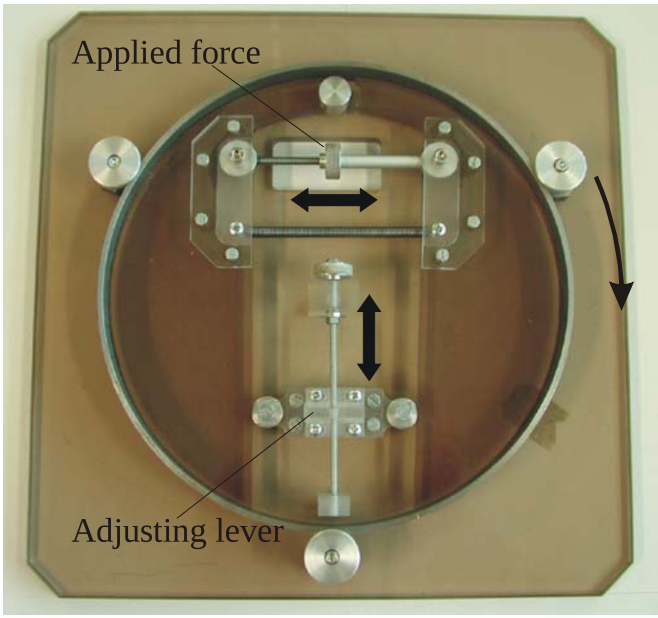


Fig 5.2 Plexiglas model for the adaptive duo servo drum brake concept

From the equations which were derived in the last chapter it can be seen that, the shoe factor (primary or secondary) depends on the friction coefficient and the inclination angle α. The generated friction force at the secondary shoe is greater than that at the primary shoe because the applied forces at the secondary shoe are greater.

The shoe factor C^* depends on the inclination angle α. To maintain the shoe factor at high and constant values thus the angle must be changed.

5.2 Control concepts

5.2.1 Brake control system

The block diagram of the adaptive duo servo drum brake is shown in Fig 5.3. The applied force acts on the duo servo drum brake to generate the brake force. The brake force R_{actual} is estimated by measuring the torque at the drum using a torque meter. At the same time the applied force acts on a brake system model that has an assumed coefficient of friction value and generates a modeled brake force R_{model}. To calculate the coefficient of friction between the shoes and the drum, an observer is built in the system by using the difference or error between the actual and modeled forces as the input value.

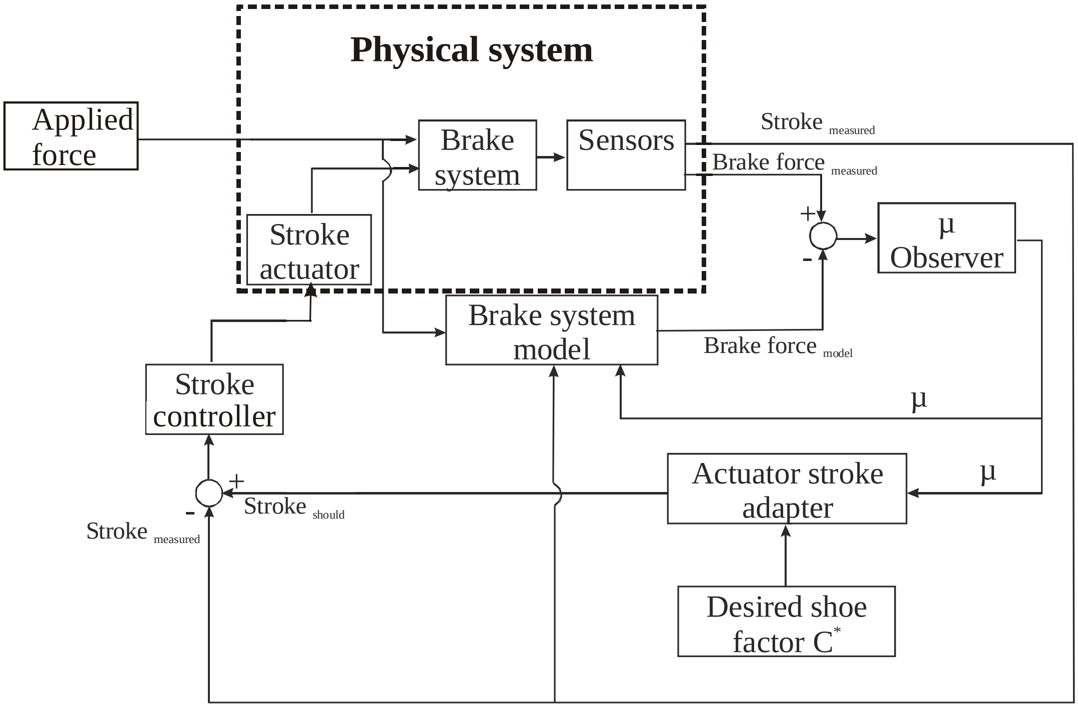


Fig 5.3 Adaptive duo servo drum brake block diagram

5.2.2 Friction observer

The coefficient of friction is defined as the ratio between the friction force and the normal force. In a first approach, both the friction and normal force can be measured by force sensors and the coefficient of friction can be estimated from these measurement signals. However, using many sensors is not economic and the mechanical and electrical integration of force sensors is difficult in the present case. The use of a friction observer can save one sensor by measuring only the brake force and comparing the actual friction force value with the assumed friction force. The suitable observer in this case is a PID observer that has good performance with the non-linear variation of the friction coefficient. The input value to the observer is the difference between the actual value of the brake force R_{act} and the model value R_{model} which is calculated according to an assumed valued of the coefficient of friction μ.

The observer estimates the coefficient of friction and sends its value back to the brake model. To obtain a constant shoe factor, the suitable stroke must then be determined according to the actual value of the coefficient of friction. To this end, the stroke adapter converts the µ signals according to the relation between α and µ at constant shoe factor. The actuator stroke is measured using potentiometer. Then, the error of the difference between the actual and the target actuator stroke is observed by a PID observer to eliminate the accumulated errors caused by internal friction.

5.2.3 Adaptation strategies

5.2.3.1 Floating link position adapter

The job of the floating position adapter is to convert the coefficient of friction μ signals to suitable signals of the floating link position a in such a way that that the overall brake shoe factor C^* is maintained at a high and constant level. As presented, the brake shoe factor C^* is a function of the coefficient of friction μ and the inclination angle α (or floating link position a). The overall shoe factor C^* for duo servo drum brake with floating link was expressed as:

$$C^* = C_1(1+k_1)+k_2$$

where

$$C_1 = \frac{\mu(a+b)}{a\sqrt{1+\mu^2} - \mu r}, \quad k_1 = \frac{a+r}{r}\frac{\sqrt{1+\mu^2}}{(1-\mu)}, \text{ and } k_2 = -\frac{a+b}{r}\frac{\mu}{(1-\mu)}$$

The floating link position a can be expressed as a function of the coefficient of friction µ and the overall brake shoe C^* by solving this equation. The solution can be expressed using a MATHEMATICA computer program as follow:

$$a = \frac{\mu\left(rC^*\mu - b - rC^* - b\sqrt{1+\mu^2}\right)}{\mu + \left(C^*\mu + \mu - C^*\right)\sqrt{1+\mu^2}} \tag{5.1}$$

Considering the distance between the cylinder and the drum centre b = 45 mm and the drum radius r = 90 mm, the graph of Fig 5.4 was obtained.

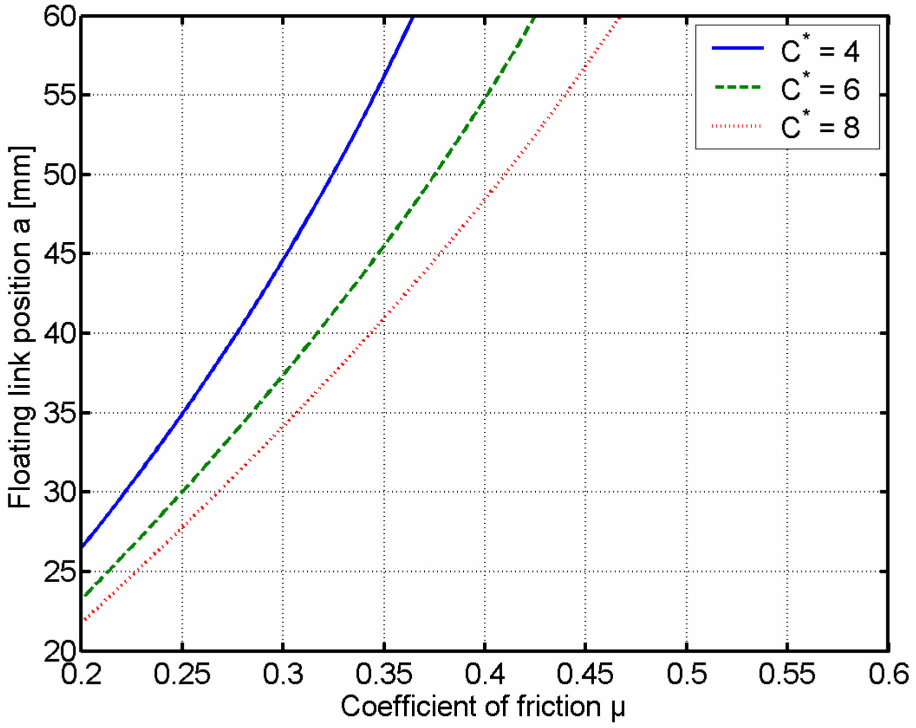


Fig 5.4 Floating link position as a function of coefficient of friction at constant brake shoe factors

6 Experimental setup and results

6.1 Description of the test-rig prototype

The objective of the test equipment is twofold: Firstly it serves to validate the models described earlier. Secondly it is used as a demonstrator to prove that it is indeed possible to control the brake force at a high and constant shoe factor by adapting the angle α. To achieve these requirements, the test rig was designed and constructed. The following section describes the construction of the test equipment.

The sliding simplex drum brake system of a VW Golf II, which is shown in Fig 6.1, was first redesigned to a duo servo drum brake by extracting an anchor pin. The duo servo drum brake is characterized by the presence of a lever that connects the primary shoe with the secondary shoe. In our adaptive drum brake, the adjusting lever was redesigned in such a way that it can be moved in the axial direction along the brake shoes by an actuator. As shown in Fig 6.2, two ball bearings were constructed at each end of the adjusting lever moving along two grooves. These grooves are fixed on the brake shoes. During braking, the primary shoe moves toward the secondary shoe to transfer the reaction force to the secondary shoe. Therefore, the adjusting lever is fixed on a linear guide to permit this movement.

Fig 6.1 Sliding simplex drum brake [VW-GOLF II]

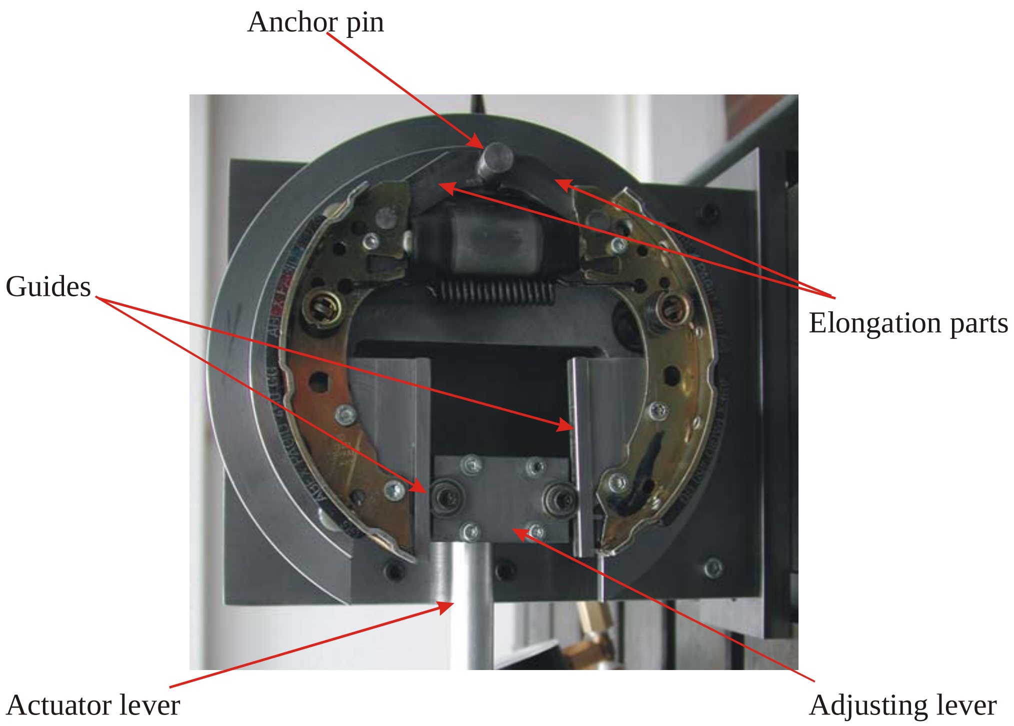


Fig 6.2 Prototype of the adaptive duo servo drum brake

6.2 Description of the test-rig and measurement equipment

6.2.1 Applied force generation

In this prototype the applied force is generated hydraulically, the brake force amplification is controlled by varying the adjusting lever position. Hence, the hydraulic cylinder of the drum brake VW-Golf II is used to generate the applied force. A hydraulic cylinder with its pump (hand drive) type ENERPAC is used to generate the desired hydraulic pressure. This cylinder however has the disadvantage that it leaks at low pressure. To solve this problem, a valve and a manometer were implemented into the hydraulic line between the hydraulic cylinder (pressure source) and the hydraulic brake cylinder. When the hydraulic pressure reaches the desired value, the valve is closed.

6.2.2 Linear actuator construction

The brake force is controlled by varying the position of the floating link. To achieve the required movement, a linear actuator shown in Fig 6.3, (electromechanical linear actuator type RORIGUEZ S24-09A04-04-LA1). The actuator has 100 mm stroke at 24 V DC and has 2.8 A current when fully loaded. It can generate 110 N dynamically and has a maximum velocity of 80 mm/s. The floating link with its roller bearings is fixed with the movable part of a linear guide (carriage). The fixed part of the linear guide is perpendicularly connected with another linear guide through a plate. The actuator is fixed from one side at the plate and from the other side on a stand which is fixed on the rig.

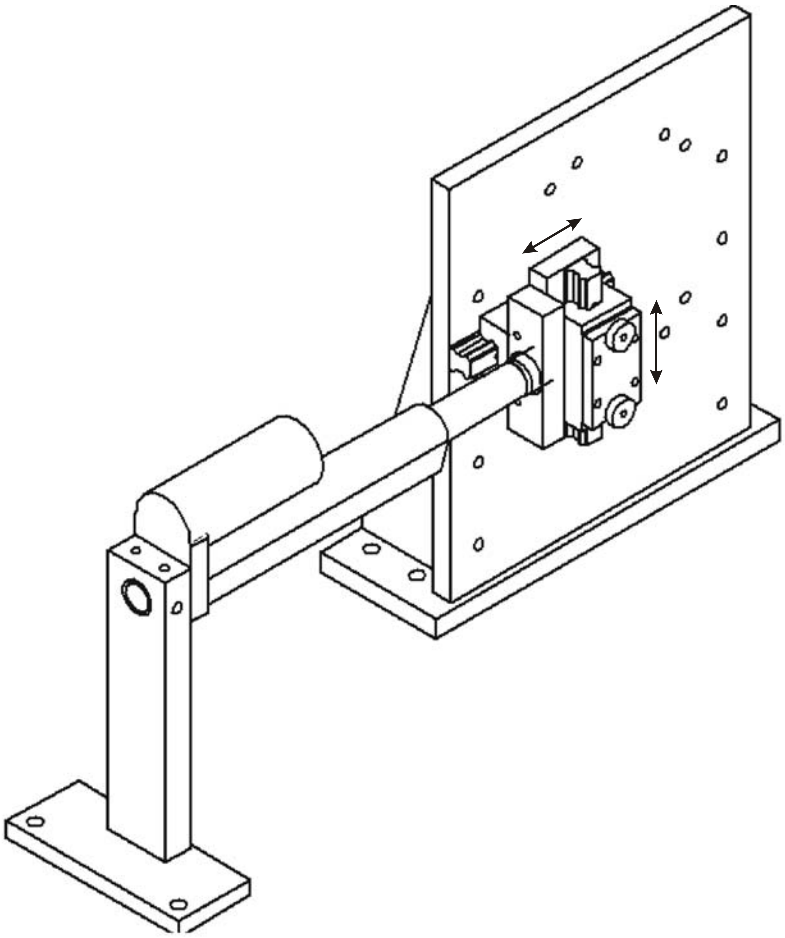


Fig 6.3 Linear actuator construction

6.2.3 Stroke sensor measurement

The stroke of the floating link position is measured using a linear variable differential transformer stroke sensor type LUCAS 1002 XS-D MC. Its measurement range is between -25 and 25 mm with analogue voltage output of -10 to 10 V. The carrier frequency for this sensor is about 3500 Hz. The sensor evaluates the change in the inductance during the relative movement between the movable and fixed parts.

As shown in Fig 6.4, the fixed part of the sensor is fixed with the actuator stand, while the movable part is fixed with the plate which is connected together with the movable part of the actuator. The signals are amplified using an amplifier type ALTHEN MC-KS-24E-B10 and then received to the dSPACE box.

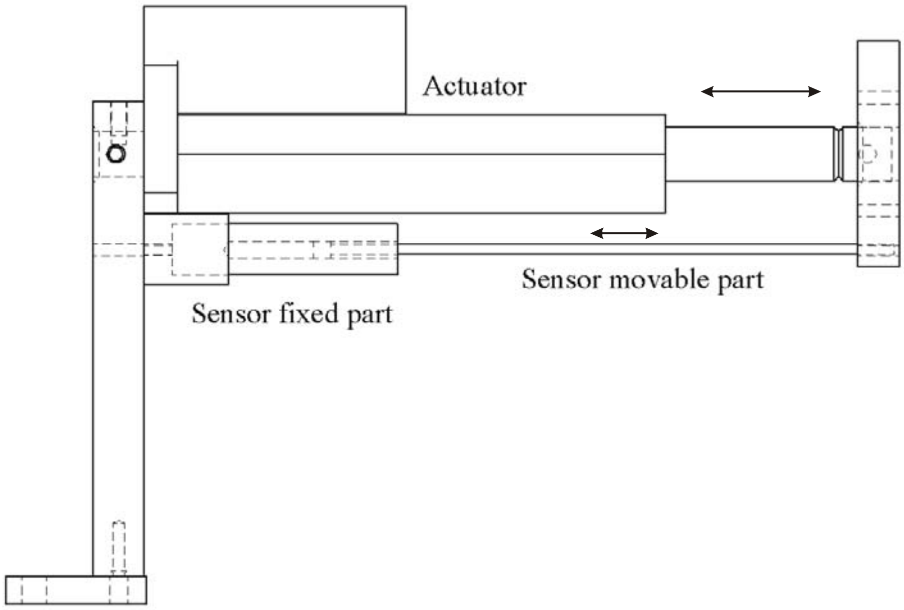


Fig 6.4 stroke sensor measurement technique

6.2.4 Torque and speed measurements

The brake torque is measured using a torquemeter type Dr Staiger Mohilo 0130-AE. Its measurement range is between 5 and 5000 Nm and its accuracy class is 0.1. The output signal is ± 10 V direct current. The torquemeter is provided with a sensor which can also measure the rotational speed up to 8000 rpm. The signals are amplified by an amplifier and then they are calibrated.

To eliminate the axial and the radial loads at the torquemeter two bearings and a flexible coupling are assembled between the braking system and the torquemeter.

6.2.5 Motor and gearbox

There are many methods to generate the kinetic energy such as using a fly wheel with suitable mass moment of inertia or to use an electric motor. The flywheel has the advantage that it can generate a high kinetic energy by increasing the rotational speed but it has also the disadvantage that the simulated speed is not constant i. e. the simulated speed varies with the brake time. Using an electric motor has the advantage that the simulated speed is constant. This however requires enormous power at high speed and torque and generates a great amount of heat energy that leads to increase the contact temperature.

In our investigation, the power source of the test rig is an AC electric motor type Lentze with 19,3 kW maximum output power at 2650 rpm rotational speed. A frequency converter is used to control the motor revolution. To generate a high output torque at low rotational speed a gearbox is assembled between the motor and the torquemeter. Fig 6.5 and Fig 6.6 illustrate the construction of the electric motor with the gearbox. The gearing ratios of the gearbox are 6.19, 3.41, 2.105 and 1.2. The used gearing ratio is 6.19.

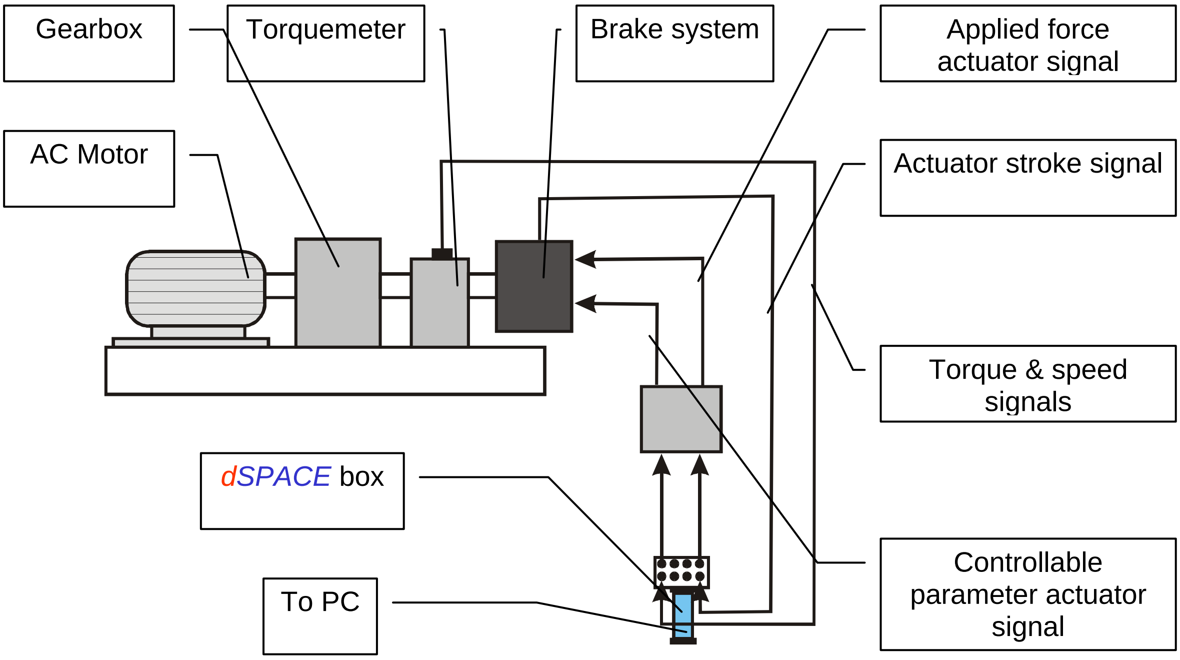


Fig 6.5 Schematic sketch of the test rig

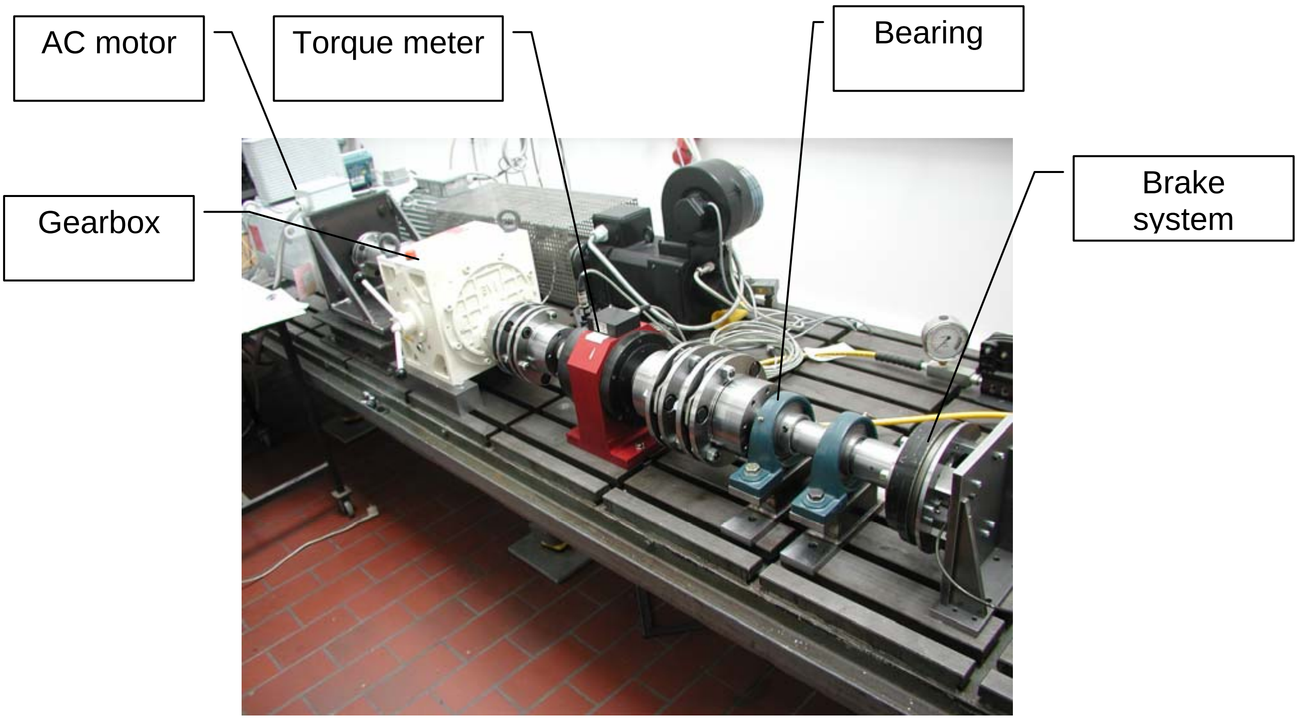


Fig 6.6 photo of the test rig

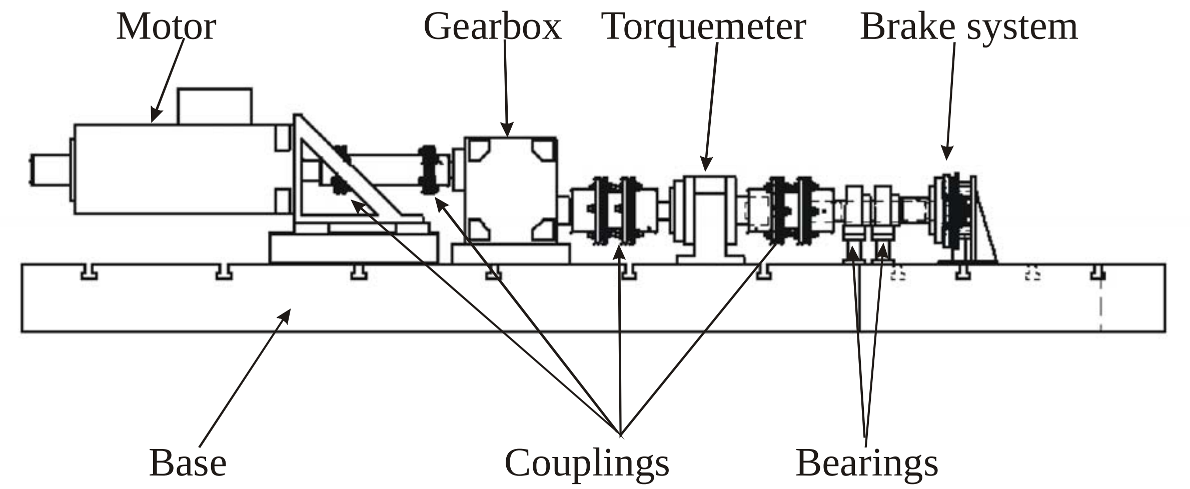


Fig 6.7 technical drawing of the test rig

6.2.6 Experimental investigations of the brake system

The sliding simplex drum brake of VW Golf II was modified to a duo servo drum brake by adding an anchor pin, enlarging the brake shoes and constructing a lever between the primary and the secondary shoe. The floating link is designed in such a way that it has the ability to move along the brake shoes to change the inclination angle α.

The maximum brake torque for a single wheel is about 180 Nm for the brake investigated here. As shown in Table 6.1, the braking torque was approximately set to 25, 50, 75, and 100% of the maximum braking torque. The nominal shoe factor was set to $C^{*}=5$ and varied during the experiments according to the variations of the floating link position. The floating link position is 55 mm away from the drum centre and then is decreased gradually to 50, 45, and 40 mm. The simulated vehicle speed is 2.63 km/h and is increased gradually to 5.3, 13.2, and 21.4 km/h.

Table 6.1 Conditions investigated during the experiments

Variation	Conditions	No of tests	
Applied force F	190 N 366 N 570 N 700 N	• Simulated vehicle speed $v_s = 2.63 km/h$ • Floating link position $a = 72 mm$	4
Simulted vehicle speed v_s	2.63 km/h 5.3 km/h 13.2 km/h 21.4 km/h	• Applied force $F = 190 N$ • Floating link position $a = 72 mm$	4
Floating link position a	55 mm 50 mm 45 mm 40 mm	• Simulated vehicle speed $v_s = 2.63 km/h$ • Applied force $F = 190 N$	4
Applied force F (repeatability effect)	190 N 366 N 570 N 700 N	• Simulated vehicle speed $v_s = 2.63 km/h$ • Floating link position $a = 72 mm$	4
Total no of tests			16 tests

6.3 Experimental results

The experimental work was carried out in two stages; the first part was to investigate the effect of the working parameters such as normal force, drum speed and floating link position. In the investigation of each parameter, the parameter value was gradually increased or decreased to investigate its influence on the brake force or brake shoe factor. The second stage of the experimental results is the control of the brake force as well as the brake shoe factor at constant values by varying the floating link position. In each experiment, applied force, drum brake moment, drum speed and floating link position will be measured. Then, the coefficient of friction μ is calculated using the μ-observer. The brake force is calculated by division of the drum torque to the drum radius and the brake shoe factor is evaluated as the ratio between the brake force and the applied force.

6.3.1 Effect of the working parameters

6.3.1.1 The effect of the applied force

The generated normal force is proportional to the applied force. In this section, the effect of the applied force will be investigated. As mentioned in the background the coefficient of friction is affected by the normal force. *Fig 6.8* shows the brake force variations with 190, 366, 510 and 700 N applied force at 7 km/h vehicle speed and 55 mm actuator stroke. From this figure it can be seen that, with the increase of the applied force there is an increase in the brake force. At constant applied force, the brake force is not constant with the time especially when the applied force is at high values. The difference between the maximum and minimum values reaches to 40% approximately. The difference in the brake force is cause by the variation in the coefficient of friction.

The variations in the brake forces with the change of applied forces could be caused by many factors. For example [Blau, 1996] has established that with low contact pressure, the chemistry of the contact areas has a large role in the variations of the coefficient of friction. While under high or extreme pressure and high temperature conditions, the coefficient of friction tends to decrease again as hydrostatic stress produce increased plasticity.

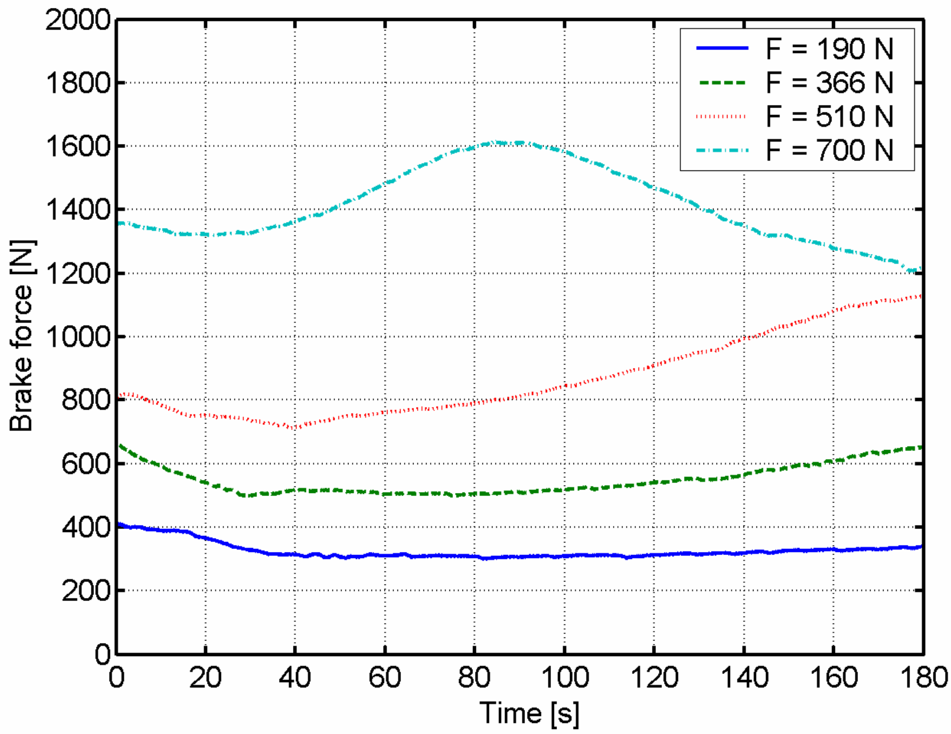


Fig 6.8 Brake force variations with the applied forces (v=2.63 km/h and a= 72mm)

The brake shoe factor C* is replotted in as a function of brake time by dividing the brake force to the applied force. *Fig 6.9* shows the effect of the applied force on the shoe factor variations with the brake time.

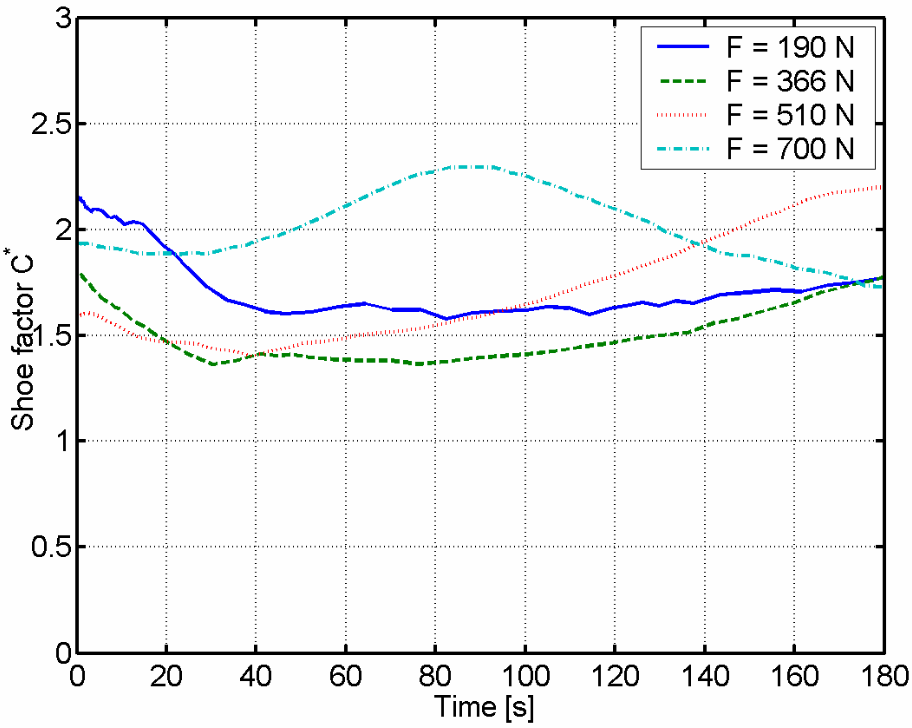


Fig 6.9 Shoe factor variations with the applied forces (v=2.63 km/h and a= 72mm)

The coefficient of friction μ is calculated using a MATLAB Simulink model for the brake system. *Fig 6.10* shows the effect of the applied force on the coefficient of friction. From this figure it can be seen that there are variations in the coefficient of friction with the time. These variations depend on the applied force value. The difference between the minimum and maximum values increases with the increase of the applied force. With the increase of the applied force the generated temperature also increase which leads to reduce the coefficient of friction μ.

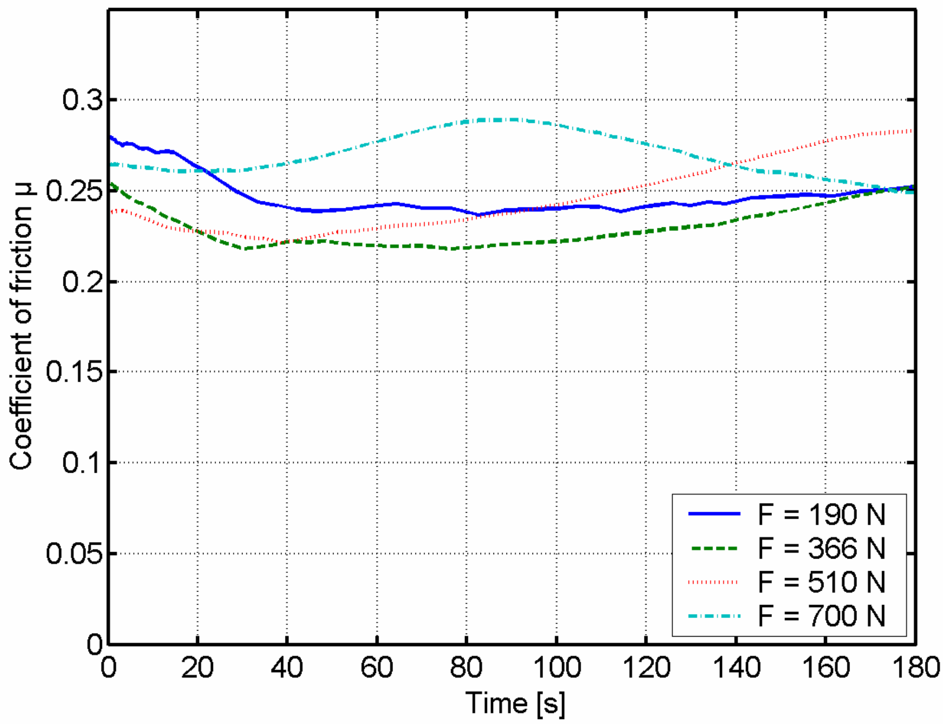


Fig 6.10 the coefficient of friction variations with the applied forces (v=2.63 km/h and a= 72mm)

The nature of friction can be affected by many parameters such as sliding speed, contact force, cleanliness of the contact, temperature, atmospheric pressure and relative humidity. The high normal force tends to reduce the coefficient of friction to more than 20 %. Increasing the normal force is accompanied with three zones. The first zone is oxide influence that the coefficient of friction decreases to the minimum value and then increases. The second zone is metal deformation and plowing are prominent that the coefficient of friction still to increase to the maximum value and still at constant values. The last zone is hydrostatic stress-enhanced plasticity that the coefficient of friction reduces with the increase of the normal force [Blau, 1996].

6.3.1.2 The effect of vehicle speed

The coefficient of friction μ, the brake force as well as the brake shoe factor are affected by the relative speed between the rotor (drum) and the brake shoes. Fig 6.11 and Fig 6.12 illustrate the effect of the simulated vehicle speed on the brake force and brake shoe factor respectively at 191 N applied force and 72 mm floating link stroke. Also, Fig 6.13 shows the effect of vehicle speed on the coefficient of friction at the same conditions. From these figures it can be seen that at low vehicle speed (2.63 km/h) there is no great difference in the brake force or in brake shoe factor. There is no fixed trend of the variations in the coefficient of friction μ with the vehicle speed. For example, there are only small peaks in the coefficient of friction when the vehicle speed is low (2.63 km/h) but it is nearly constant. The variations of the brake force increase with the time at 5.3 vehicle speed from approximately 570 N to 825 N i.e. 44 % increase in the brake force as well as brake shoe factor in 300 seconds. The coefficient of friction increases with the time from 0.33 to 0.38 which is only an increase of 15% at 5.3 km/h vehicle speed. At 13.2 km/h vehicle speed the brake force and the brake shoe factor increase from 620 N to 1000 N i.e. 63% in the first hundred seconds and then decrease to 600 N approximately, while the coefficient of friction increases with the time from 0.35 to 0.42 at 100 seconds and then it decreases. At 21.1 km/h vehicle speed the brake force has a sharp decrease approximately from 1250 N to 575 N with the time, i.e. 117% decrease in the brake force in 120 seconds. The coefficient of friction decreases with the time from 0.45 to 0.33. This means that 36 % decrease in the coefficient of friction meets 117% decrease in the brake force and the brake shoe factor. This last measurement took only 120 seconds because the brake becomes too hot.

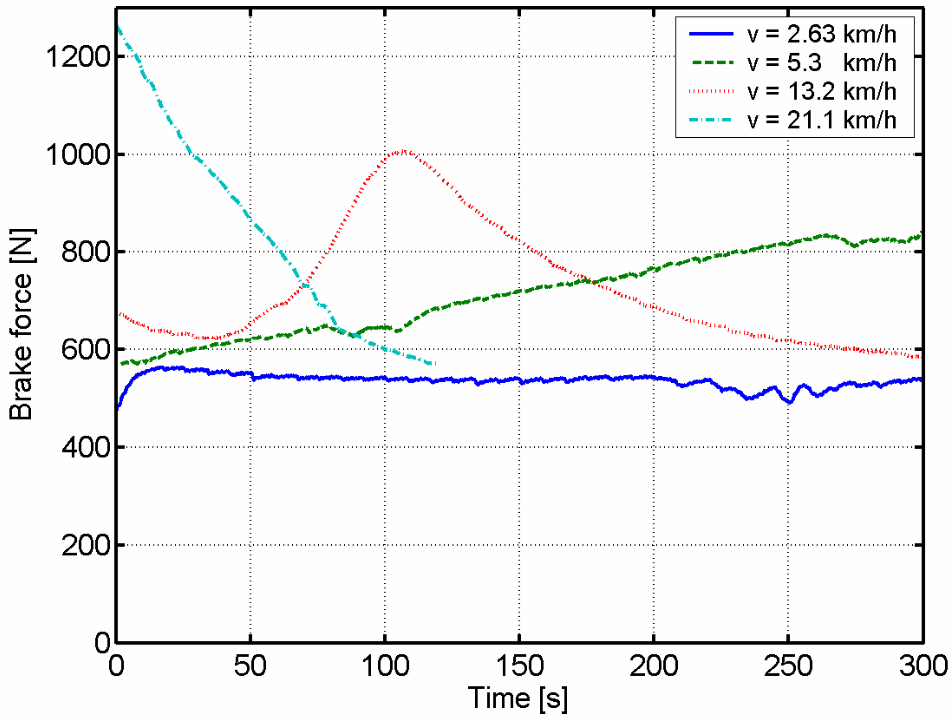


Fig 6.11 Brake force variations with the vehicle speed (F=191 N and a= 72mm)

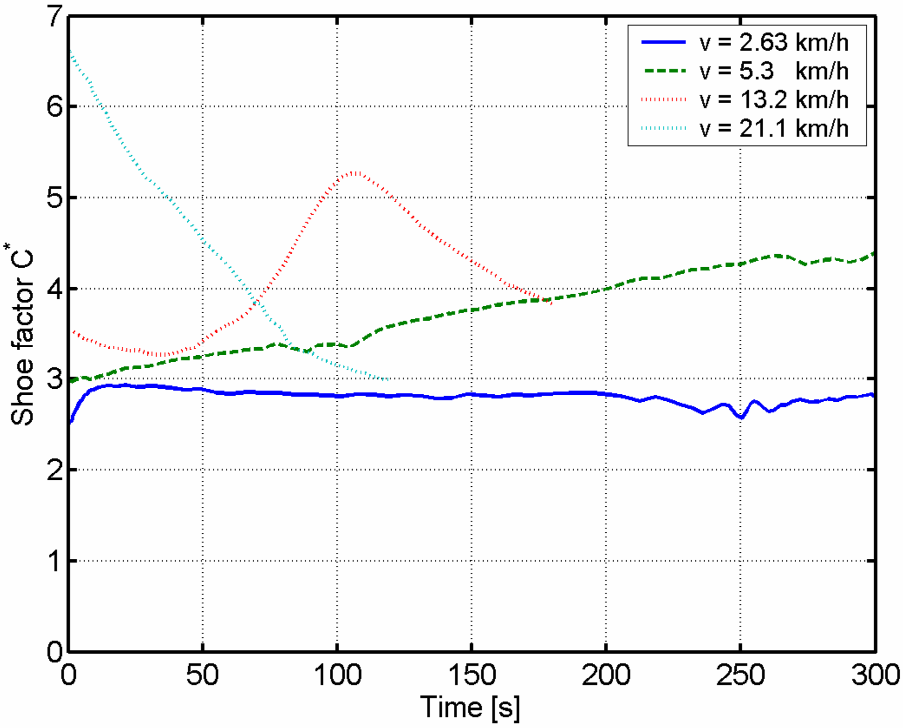


Fig 6.12 Brake shoe factor variations with the vehicle speed (F=191 N and a= 72mm)

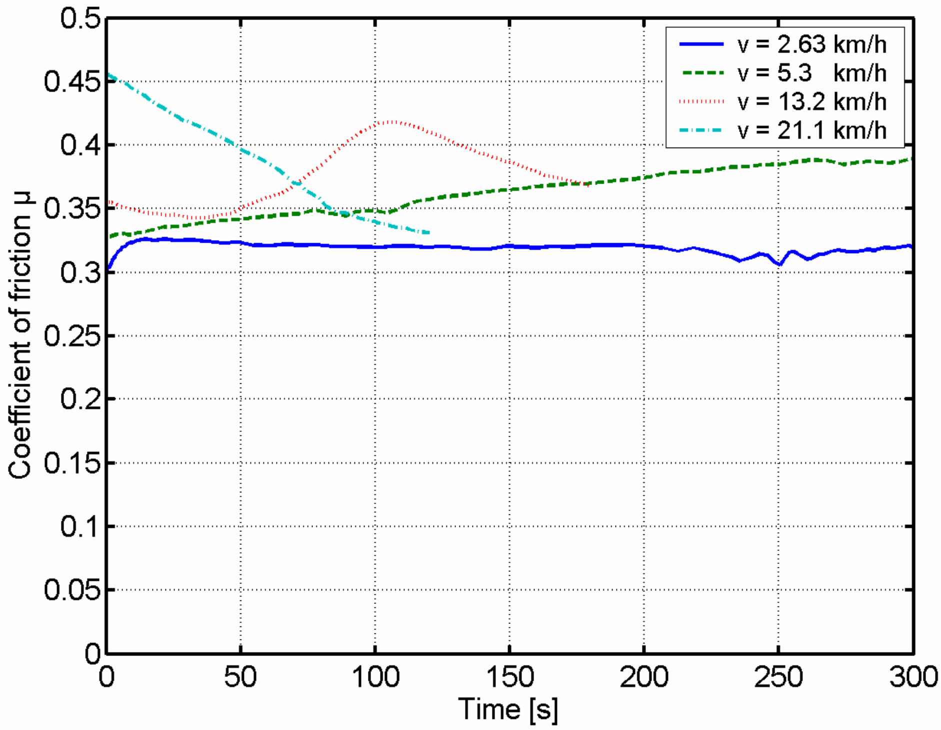


Fig 6.13 Coefficient of friction variations as a function of vehicle speed (F=191 N and a= 72 mm)

With high sliding speed, friction is affected by many parameters such as surface softening by friction heating, increased plowing in the softer materials, enhanced transfer and oxide film formation. It is difficult to identify which parameter does play the main role on the coefficient of friction variations.

6.3.1.3 The effect of the floating link position

In chapter 5, the role of the floating link position on the brake force as well as brake shoe factor has been investigated theoretically. The final form of the brake shoe factor was as follow:

$$C^* = C_1^*(1+k_1)+k_2 \tag{6.1}$$

where $k_1 = \left(\frac{a+r}{r}\,\frac{\sqrt{1+\mu^2}}{(1-\mu)}\right)$, $k_2 = -\frac{(a+b)}{r}\,\frac{\mu}{(1-\mu)}$

and $C_1^* = \frac{R_1}{F} = \frac{\mu(a+b)}{a\sqrt{1+\mu^2}-\mu\cdot r}$

In order to investigate the effect of the floating link position on the brake force. Fig 6.14 and Fig 6.15 show the effect of the floating link position on the brake force and brake shoe factor respectively at 3.63 km/h simulated vehicle speed and 191 N applied force. As shown these measurement took 120 seconds because of fast heating. The floating link position a was gradually decreased from 55 mm to 40 mm with 5 mm pitch. The coefficient of friction for these measurements are calculated and plotted in Fig 6.16. From these figures it can be seen that, as the floating link position decreases, there is an increase in the brake force and brake shoe factor. For example, in the beginning, as the stroke decreases from 55 mm to 40 mm, the brake force increases approximately from 1350 N to 3200 N. Then, the brake forces are decreased with the time for all floating link position values, especially at 40 mm. At the 60th second, as the floating link position decreases from 55 mm to 40 mm the brake force increases approximately from 1500 N to 1900 N. It is noticed that, with the decrease of the floating link position, there is an increase in the brake force as well as the brake shoe factor. The increase of the brake force tends to increase the brake energy which converts to heat energy. Increasing the temperature leads to material softening phenomena which leads to decrease the coefficient of friction μ and hence decreasing the brake force.

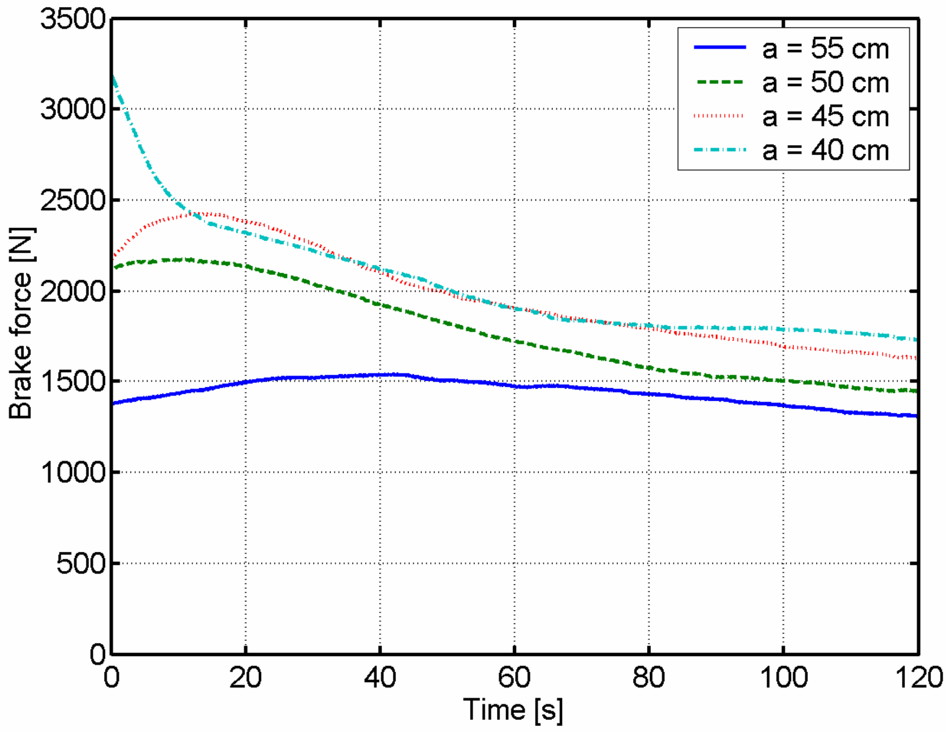


Fig 6.14 Effect of the floating link position on the brake force (F=191 N and v= 2.63 km/h)

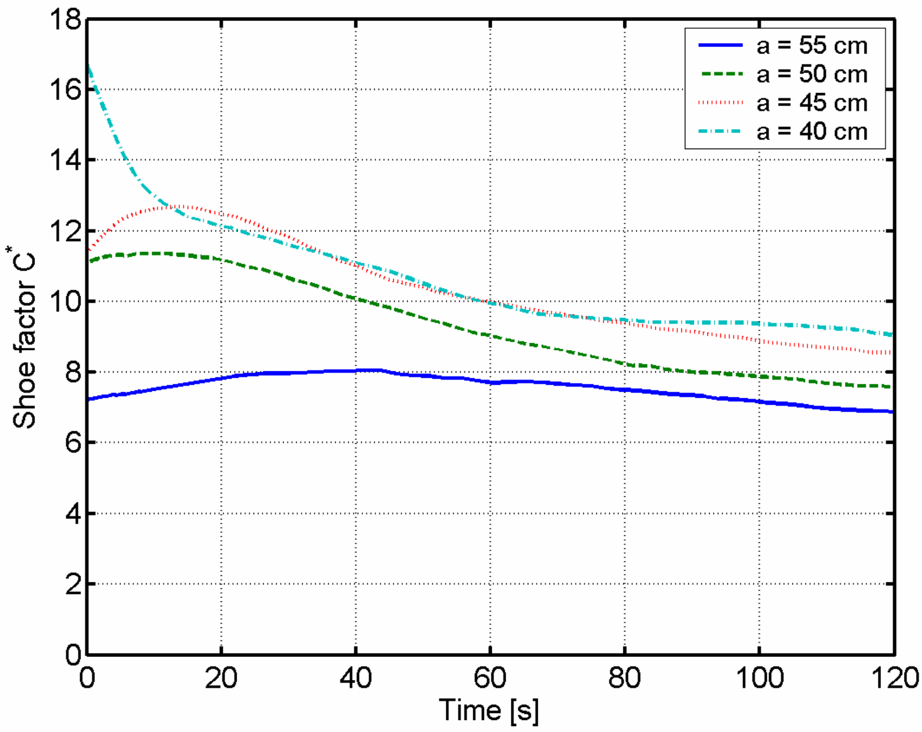


Fig 6.15 Effect of the floating link position on the brake shoe factor (F=191 N and v= 2.63 km/h)

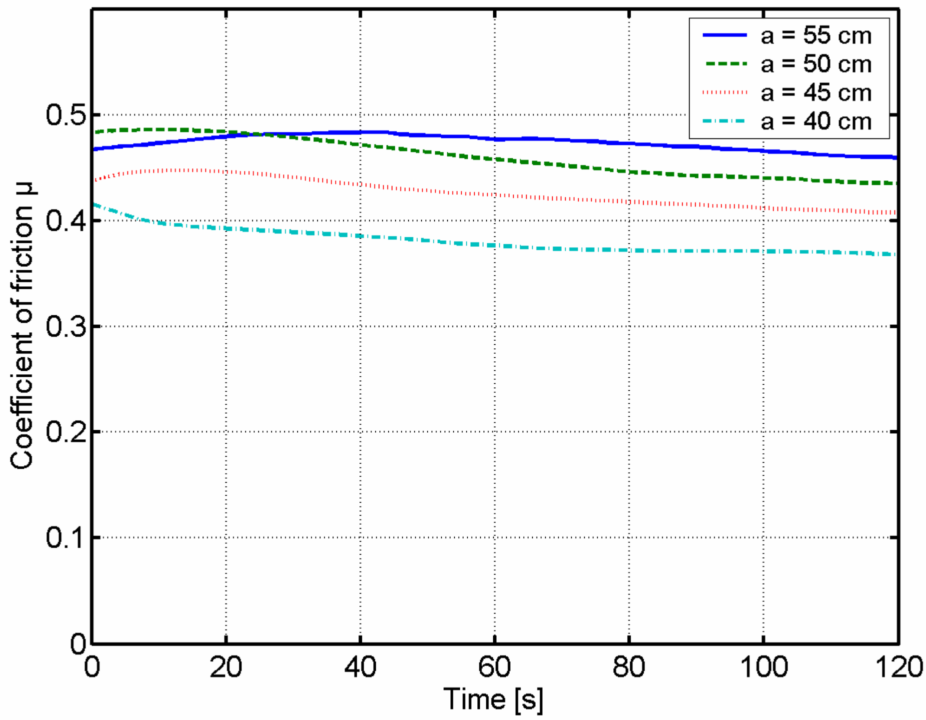


Fig 6.16 Effect of the floating link position on the coefficient of friction variations (F=191 N and v= 2.63 km/h)

6.3.1.4 Measurement repeatability

The investigation of brake systems and other systems with friction have the problem that the experiments are often not repeatable. In order to study the repeatability of the results the first measurement (variation of the applied force) was repeated. In this measurement, it is investigated the same parameter with the same conditions. *Fig 6.17* and show the effect of the applied force on the brake force and brake shoe factor respectively. The coefficient of friction µ is calculated and plotted in *Fig 6.19*. From these figures it can be seen that there is a small difference in the coefficient of friction between the two measurements at the same conditions. The coefficient of friction of the new measurements is higher than that of the old measurements. In the old measurements the coefficient of friction is ranged between 0.22 and 0.29 while it is ranged between 0.31 and 0.42 with the new measurements. There is also a difference in brake force as well as in the brake shoe factor. The brake force is ranged about between 320 N and 1600 N with the old measurements while the brake force is ranged about between 850 N to 3300 N with the new measurements. The main reason is the variation of the coefficient of friction µ. These variations are affected by contact area and number of brake applies.

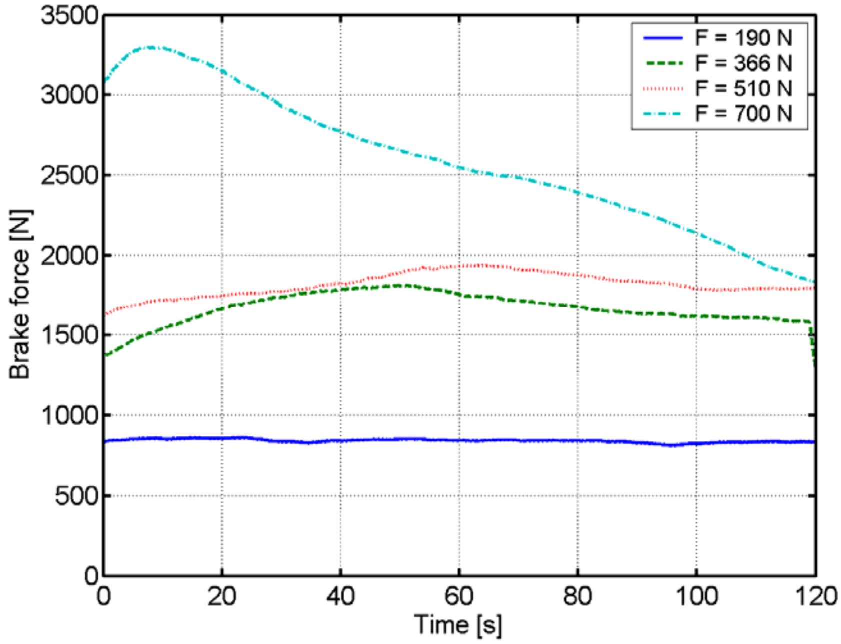


Fig 6.17 Effect of the applied force on the force (a =72 m and v=2.63 km/h) *[repeat.]*

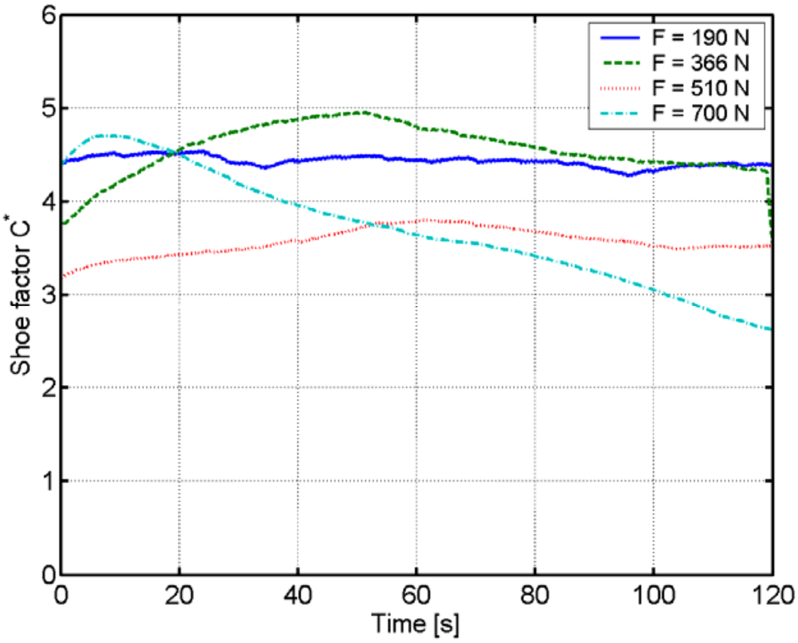


Fig 6.18 Effect of the applied force on the brake shoe factor (a =72 m and v=2.63 km/h) [repeat.]

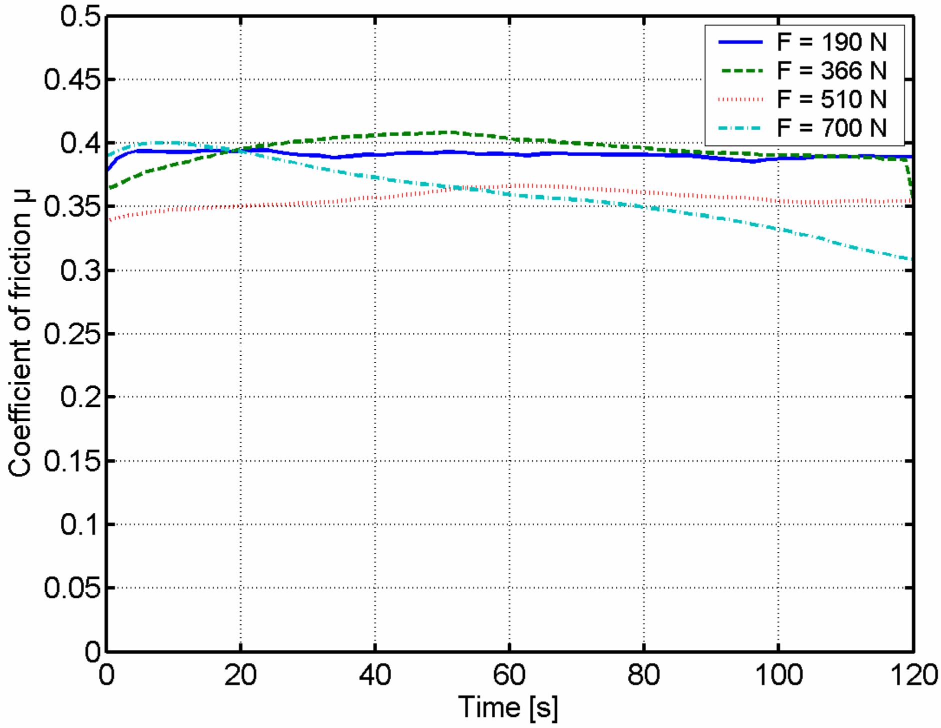


Fig 6.19 Effect of the applied force on the coefficient of friction variations (a =72 m and v=2.63 km/h) [repeat]

6.3.2 Control of the brake shoe factor

The second step of the experimental work was to control the brake shoe factor at constant and high values with the coefficient of friction variations. It was tried to maintain the brake shoe factor at 4, 5 and 6. Fig 6.20 shows the experimental measurement with a desired brake shoe factor of 4. In this figure the brake force, brake shoe factor, coefficient of friction and floating link stroke variations are depicted as a function of the time. It is tried to keep the brake force as well as the brake shoe factor at constant values. The brake force is observed by human eyes and then, a suitable signal is given to the actuator to change the floating link position to keep the brake force at constant values with the coefficient of friction variations. The applied force is maintained at 287 N and the brake drum runs with 8.5 km/h simulated vehicle speed. From these figures, it can be seen that the brake force and brake shoe factor are successfully controlled to be maintained at constant values with the variations of the coefficient of friction. But there are differences between the desired and actual values. These differences are ranged between -16% and 10 %. The coefficient of friction is approximately varied between 0.33 and 0.47. To maintain the brake force at constant value, the floating link was moved in a range between 70 mm and 51mm.

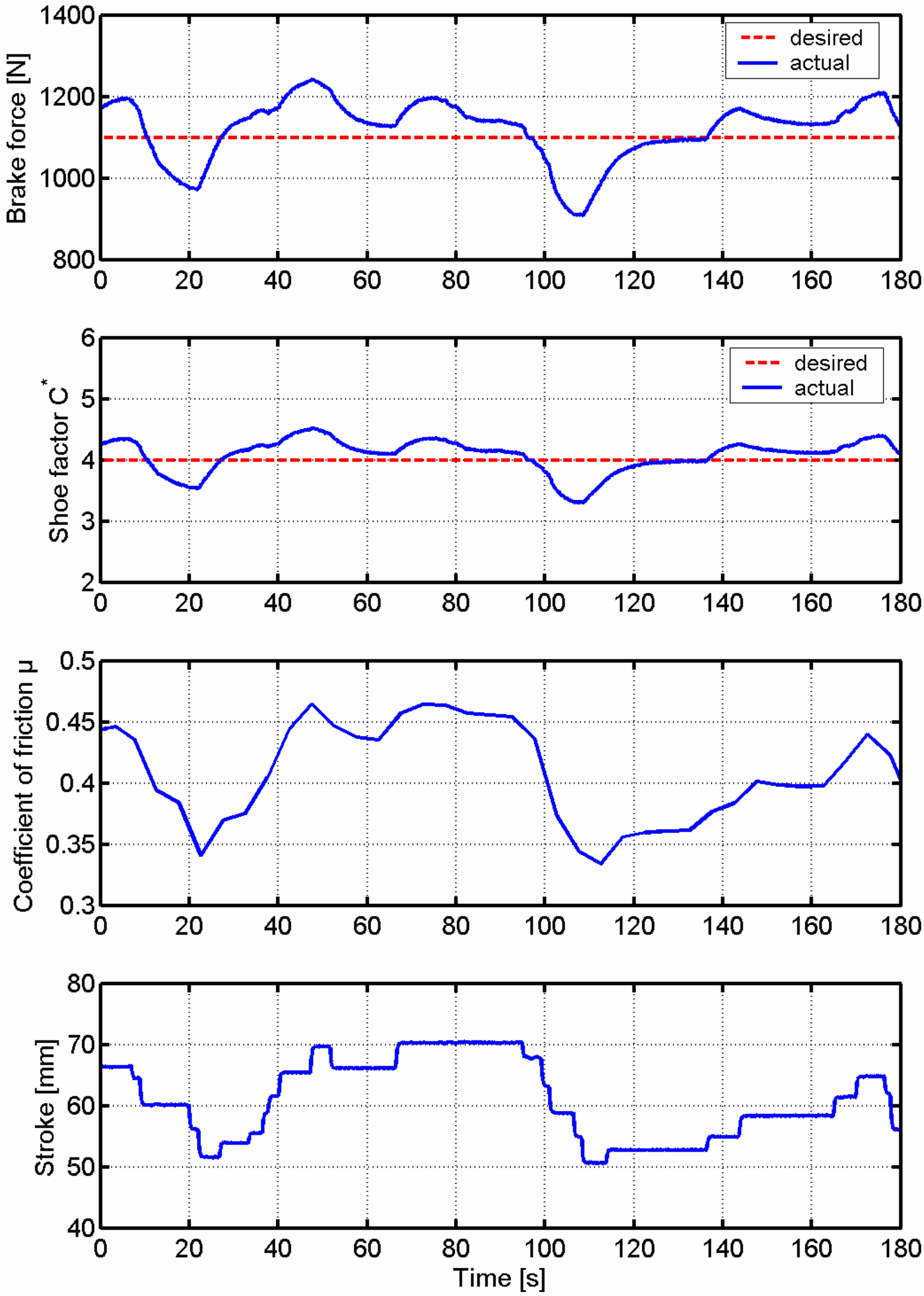


Fig 6.20 Brake force, shoe factor, coefficient of friction and floating link stroke variations when the desired brake shoe factor=4 (applied force =287 N and simulated vehicle speed= 8.5 km/h)

The experimental measurements with the desired shoe factor of 5 are shown in *Fig 6.21*. The measured force with these measurements is not filtered. The applied force is maintained at 240 N and the brake drum runs with 8.5 km/h simulated vehicle speed. From these figures, it can be seen that the brake force and brake shoe factor are successfully controlled to be maintained at constant values with the variations of the coefficient of friction. The differences between the desired and actual values are not great. These differences are ranged between -5 % and 8 % approximately. The coefficient of friction is varied between 0.37 and 0.48 approximately. To maintain the brake force at constant value, the floating link was moved in a range between 65 mm and 48 mm.

Similarly, Fig 6.22 illustrates the brake force, brake shoe factor, coefficient of friction and floating link position stroke variations with the time at a desired brake shoe factor of 6. The applied force is maintained at 191 N and the drum runs also with 8.5 km/h simulated vehicle speed. The difference between the actual and the desired brake forces is ranged between -12% and 5% approximately. The coefficient of friction is reduced from 0.44 to 0.28 as a result of the brake running time for 180 seconds. Also, to maintain the brake force as well as the brake shoe factor at the desired value with the reduction in the coefficient of friction, it is required to change the floating link position between 54 mm and 36 mm.

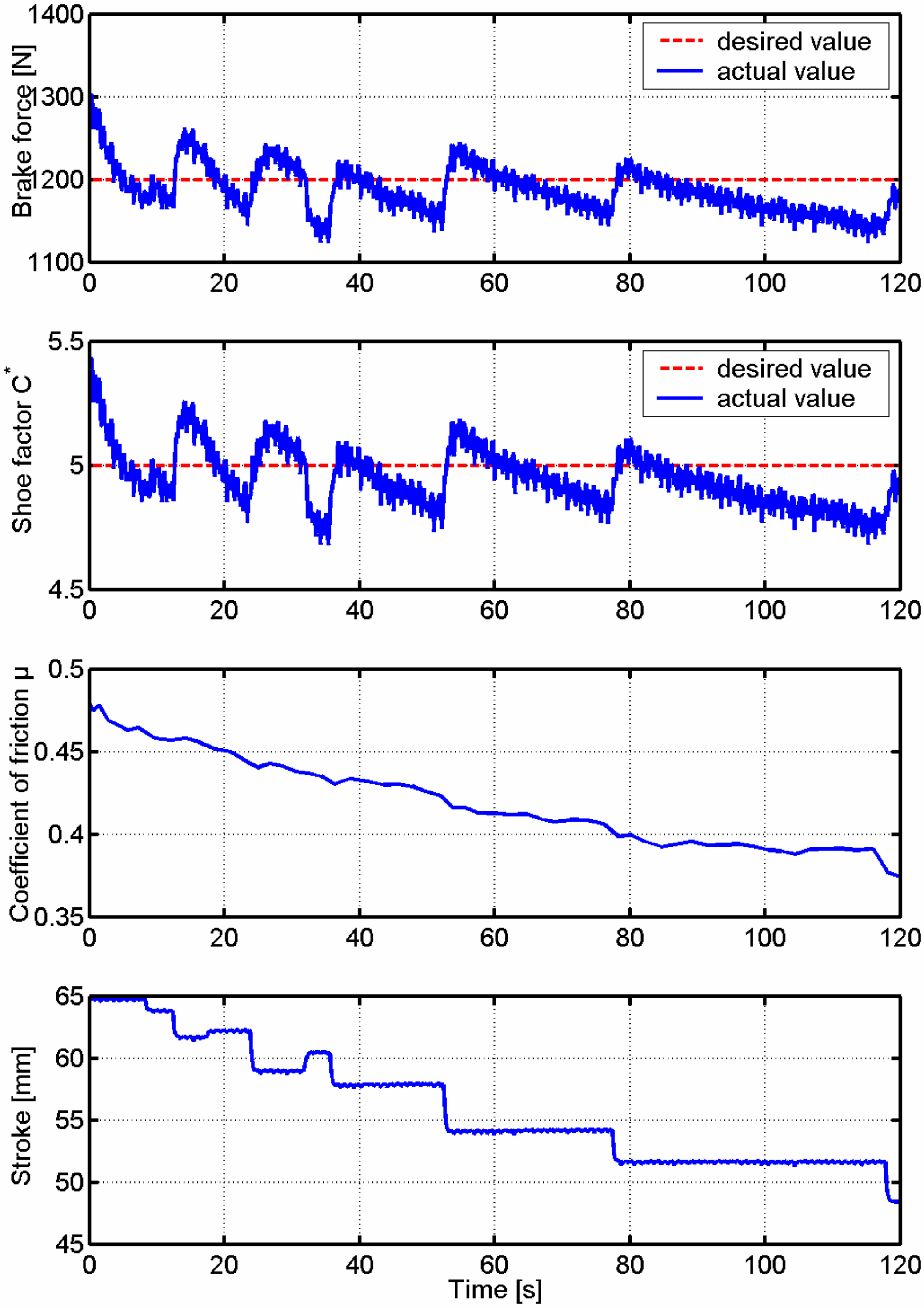


Fig 6.21 Brake force, shoe factor, coefficient of friction and floating link stroke variations when the desired brake shoe factor=5 (applied force =240 N and simulated vehicle speed= 8.5 km/h)

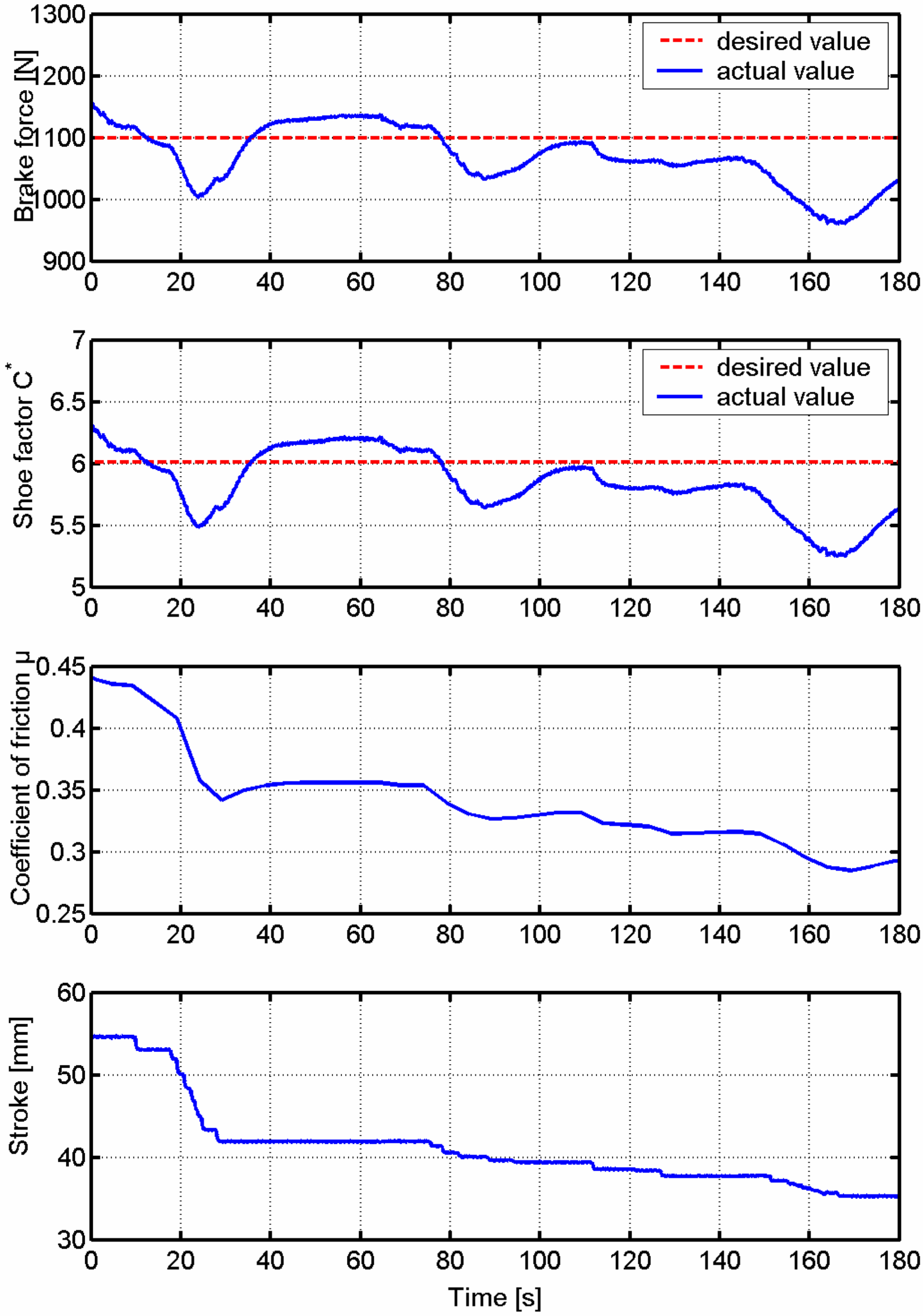


Fig 6.22 Brake force, shoe factor, coefficient of friction and floating link stroke variations when the desired brake shoe factor = 6 (applied force =191 N and simulated vehicle speed= 8.5 km/h)

7 Conclusions

The aim of this work was to investigate self-amplifying drum brakes and to study how the brake shoe factor can be kept at a high and constant value independently from the changes in the coefficient of friction. In this final chapter the newly developed concept will be evaluated and it will be studied to which extend the aims have been achieved.

The evaluation of the adaptive duo servo drum brake will be based on a comparison between the brake force performances with the adaptive duo servo drum brake and the classical duo servo drum brake. The experimental measurements illustrated in the last chapter will be taken into account.

As shown in the experimental results presentation, the brake force of the duo servo drum brake could be maintained at constant values with fluctuations or peaks which are not more than 10 % around the desired values. Also, the brake shoe factor could be kept constant at high values 4, 5 and 6 respectively.

The overall brake system size is greater than the normal duo servo drum brake because of the presence of the actuator. The overall brake system volume could be reduced if a smaller actuator is chosen and the integration is performed more consequent. When the adaptive duo servo drum brake is compared with the similar adaptive duplex drum brake like that is presented by [Leber et al, 1998], it is smaller and hence lighter in weight.

The performance of the adaptive duo servo drum brake can be evaluated by making a comparison between the brake force, or brake shoe factor, once with the new concept and with normal duo servo drum brake. Measurements of the brake force at constant desired brake shoe factors have been made. They will be re-evaluated here by comparing them to the brake forces that would have been obtained under the same conditions if a classical brake were used. The calculated non-controlled brake force will be compared with the brake force at constant brake shoe factors of 4, 5, and 6. Fig 7.1 illustrates the comparison between the measured controlled and calculated non-controlled brake forces with desired brake shoe factor of 4. It can be seen that the brake force with the concept is higher than that with a normal duo servo drum brake. The brake force with the new concept ranges between 920 N und 1220 N, while, at the same coefficient of friction variations, the brake force with the normal duo servo drum brake ranged approximately between 580 N and 1150 N.

The comparison with the desired value can be based on the mean value of the measured controlled force as well as the non-controlled and calculating the deviation:

$$deviation = \frac{mean\ \ value - desired\ \ value}{desire\ \ dvalue}$$

The deviation for example of the measured controlled force at shoe factor of 4 is about 7.27% and the deviation of the calculated non-controlled force at the same shoe factor is about 21%.

Fig 7.2 illustrates the comparison between the measured controlled and calculated non-controlled brake forces with desired brake shoe factor of 5. From this figure, it can be seen that the brake force with the concept is higher than that with a normal duo servo drum brake especially with increasing the brake time. The difference was firstly about 220 N and then increased up to about 550 N. Referring to Fig 6.21 the coefficient of friction μ is decreased from about 0.47 to 0.38. Where, the brake force with the concept ranged approximately between 1150 N and 1300 N, while, at the same coefficient of friction variations, the brake force with the normal duo servo drum brake ranged approximately between 620 N and 1150 N.

The deviation for example of the measured controlled force at shoe factor of 5 is about 2,23 % and the deviation of the calculated non-controlled force at the same shoe factor would have been about 33.05 %.

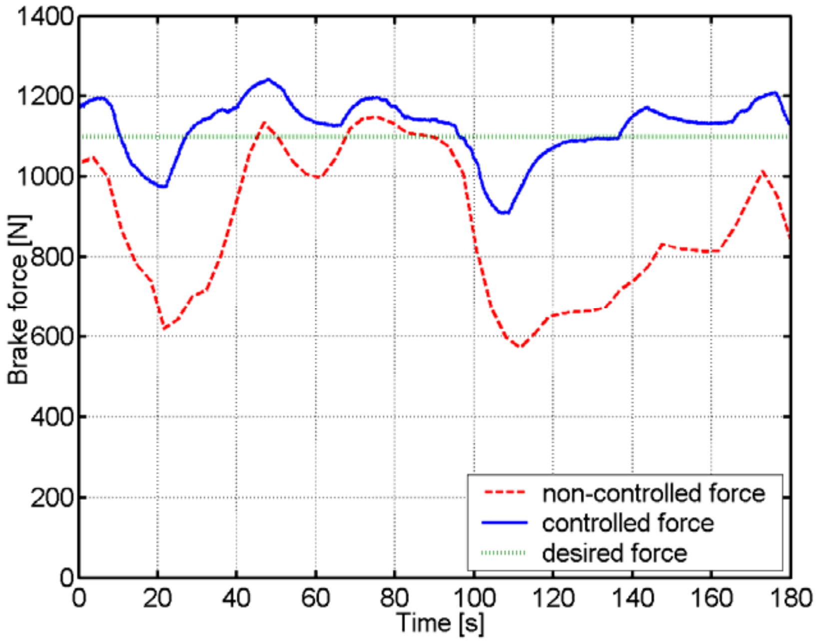


Fig 7.1 Comparison between the measured controlled brake force and the calculated non-controlled brake force at the same coefficient of friction when the brake shoe factor=4

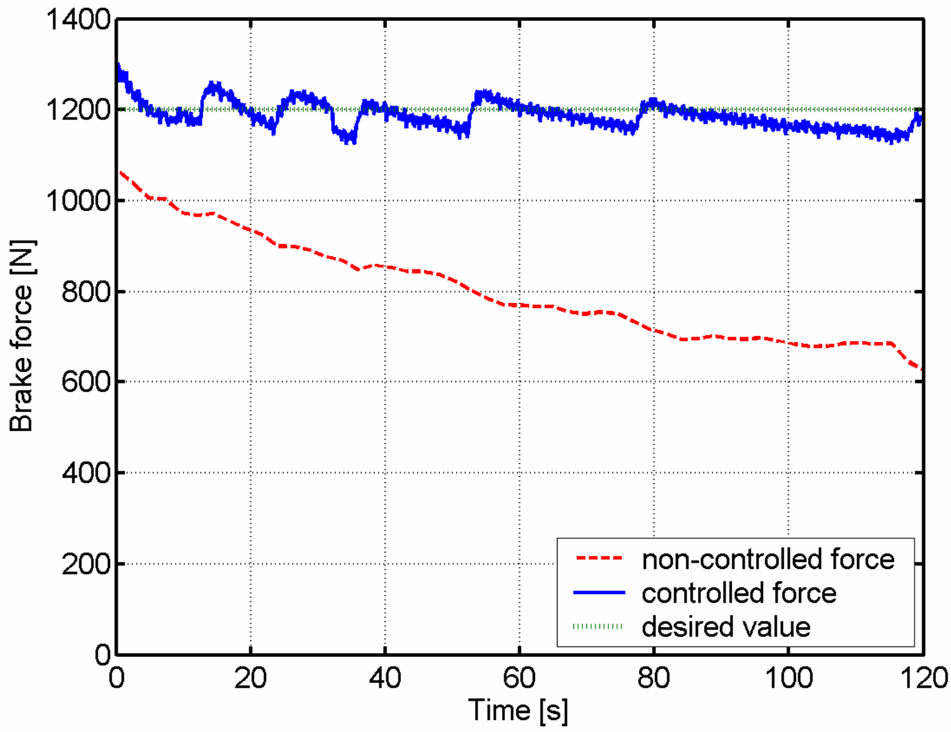


Fig 7.2 Comparison between the measured controlled brake force and the calculated non-controlled brake force at the same coefficient of friction when the brake shoe factor=5

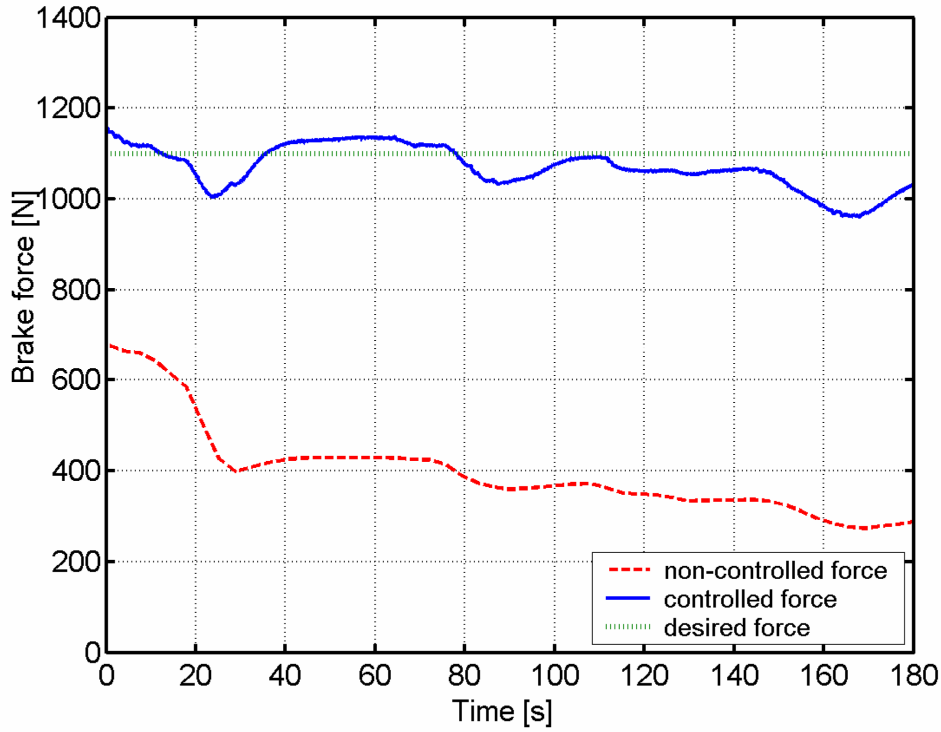


Fig 7.3 Comparison between the measured controlled brake force and the calculated non-controlled brake force at the same coefficient of friction when the brake shoe factor=6.

Fig 7.3 illustrates the comparison between the measured controlled and calculated non-controlled brake forces with desired brake shoe factor of 6. From this figure, it can be seen that the brake force with the concept is higher than that with a normal duo servo drum brake especially with increasing the brake time. The difference was firstly about 470 N and then increased up to about 830 N. The brake force with the concept ranged approximately between 970 N and 1320 N, while, at the same coefficient of friction variations, the brake force with the normal duo servo drum brake ranged approximately between 290 N and 670 N. The deviation of the measured controlled force at shoe factor of 6 is about 4.47 % and the deviation of the calculated non-controlled force at the same shoe factor is about 63.66 %.

It is noticed that, with the increase of brake shoe factor, the difference between the controlled measured and calculated non-controlled forces increases. Also, with the increase brake time, the difference also increases. This increase is duo to generating of high temperature. This probably caused a decrease in the coefficient of friction μ and hence the calculated non-controlled force decreases while with the use of concept application, the decrease in the coefficient of friction μ was compensated by shifting the floating link position and hence to regulate the brake force at the desired value.

The newly developed concept is capable to maintain the brake shoe factor C^* at high and constant values but there are some requirements for the duo servo drum brake:

- Wear automatic adjuster construction
- Friction blocking avoidance
- Fast actuator dynamics
- Different brake performance in forward and reverse directions

In the following these problems will be discussed in detail.

There are many possible designs for automatic wear adjusters. All of these designs depend on the change of the hydraulic piston lever stroke and the remedy is to change the floating link with a suitable value. Classical wear automatic adjuster mechanisms for the duo servo drum brake are shown in Fig 7.4 and Fig 7.5. The automatic adjuster mechanism consists of an actuator link, an actuator lever, and an adjusting screw. The automatic wear adjuster of duo servo drum brake depends on the clearance (shoe movement) between the primary shoe and the anchor pin. If the clearance increases, the actuator lever will rotate about the lever pivot. The adjuster nut then converts the rotational movement of the actuator lever to an extension of the floating link.

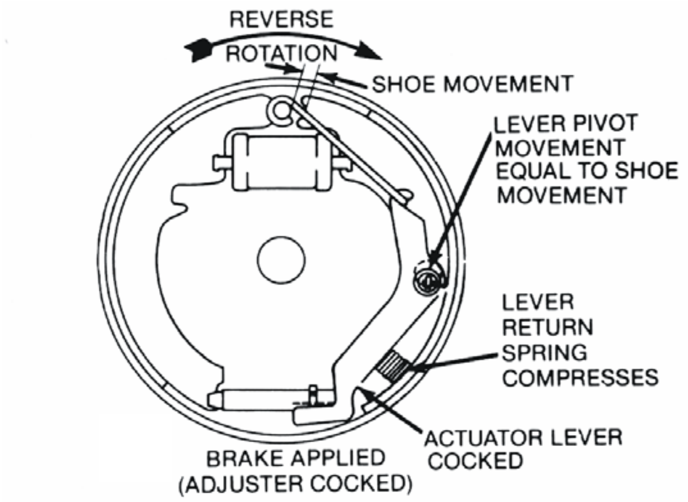


Fig 7.4 Automatic adjuster mechanism [Brake system P. M. council, 1991].

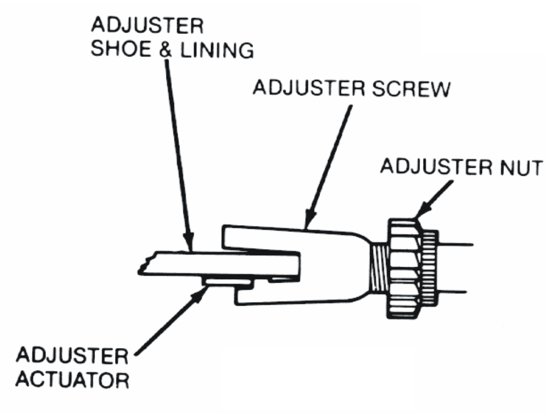


Fig 7.5 Floating link with automatic adjuster mechanism [Brake system P. M. council, 1991].

For the newly designed concept, all these mechanisms can not be used because the floating link in not kept at a fixed position. It could be moved up and down in the radial direction. Though, it is necessary to design a new wear automatic adjuster.

According to the relation between the brake shoe factor C^*, the coefficient of friction μ, and the floating link position stroke in the radial direction a the brake shoe factor was as follow:

$$C^* = C_1^*(1+k_1)+k_2 \tag{7.1}$$

where $k_1 = \left(\frac{a+r}{r}\frac{\sqrt{1+\mu^2}}{(1-\mu)}\right)$, $k_2 = -\frac{(a+b)}{r}\frac{\mu}{(1-\mu)}$

and $C_1^* = \frac{R_1}{F} = \frac{\mu(a+b)}{a\sqrt{1+\mu^2} - \mu \cdot r}$

From this equation, it can be seen that the brake shoe factor C^* goes to infinity at:

$a\sqrt{1+\mu^2} - \mu \cdot r = 0$, or

$$1-\mu=0$$

In this case the brake system will be blocked. This must of course be avoided. The coefficient of friction μ between the drum and the brake shoes is ranged between 0.2 and 0.6. Hence, the second root which is $\mu = 1$ is not critical. To avoid blocking as consequence of the first condition, it must be avoided that:

$$a = \frac{\mu\ r}{\sqrt{1+\mu^2}} \tag{7.2}$$

I.e. with the drum radius of this concept and with the above range of the coefficient of friction, the floating link position must not be lied between 18 mm and 47 mm.

8 Summary and outlook

An adaptive duo servo drum brake concept was designed and constructed to maintain the brake shoe factor C^* at high and constant values independently from the coefficient of friction variations. The brake shoe factor C^* can be kept at constant and high values with the variations of the coefficient of friction μ by actuating the distance between the floating link and the resultant brake force acting point. In this concept the applied force will be maintained at constant level during the brake process.

The classical duo servo drum brake is a self-amplifying brake. The generated brake force of the self-amplifying brakes increases with the increase of the coefficient of friction. The shoe factor of the duo servo drum brake has the greatest value of all brake types. It can reach very high value for example up to 10. At the same time it is very sensitive to the coefficient of friction. The shoe factor depends not only on the coefficient of friction between the brake shoe and drum but also on the distance between the floating link and the resultant brake force acting point.

The main problem of the self-amplifying brakes, especially the duo servo one, is the high sensitivity to the coefficient of friction variations. A little reduction in the coefficient of friction may cause an enormous reduction in the brake force especially with the duo servo drum brake. The coefficient of friction varies with many parameters such as normal force, speed, contact temperature, number of stops …etc. These parameters tend to vary the coefficient of friction with the time. Consequently the shoe factor will not be constant i.e. at the same pedal force, different brake forces will be obtained. The worst case, however, is when the brake linings get too hot and the coefficient of friction will be decreased so strongly that the brake loses its effectiveness. This phenomenon is known as brake fade.

The main idea of the prototype developed in this thesis was to design an adaptive duo servo drum brake such that its shoe factor is maintained at high and constant values independently from the coefficient of friction variations. The brake force could be changed by shifting the floating link axially up or down.

The test rig which was used to carry out the experimental part of this study has an AC electric motor as a brake power source. A gearbox is attached to the motor to change the moment and the rotational speed to suitable values. The motor with the gearbox was coupled to the brake system through a torquemeter which is used to measure the brake torque.

During braking, the brake force is measured and the coefficient of friction is calculated by an observer. The observer compares between the actual and the modeled brake forces. The modeled brake force is based on an assumed value for the coefficient of friction while the actual brake force is measured using a torquemeter. The coefficient of friction signals are converted to floating link position signals. The signals are amplified and are received to an electromechanical actuator which shifts the floating link to the desired value. Also, to eliminate the error in the actuator stroke as a result of internal friction especially when the signals are small, a stroke sensor is used to measure the actuator stroke.

The first part of the experimental measurements was to investigate the factors which affect the coefficient of friction such as applied force, wheel speed and position between the floating link and the resultant brake force acting point. Also the measurements repeatability was investigated.

The other part of the experimental measurements was to control the brake shoe factor of the duo servo drum brake. The measurements illustrate that it is possible to achieve a brake shoe factor at high and constant values even if the coefficient of friction varies. For example the brake shoe factor C^* was experimentally maintained at values of 4, 5, and 6 with a deviation up to 2.63 % from the desired value.

Although the concept promising, there are also some problems that have to be overcome before the concept can be used in practice. These problems may be interesting starting points for future investigations.

- The adaptive duo servo drum brake is an application of the self-amplifying friction mechanisms. Since it is capable to generate high and constant brake shoe factor it also can lead to self-blocking, which must be avoided under all circumstances.
- The actuator must meet the variations of the coefficient of friction. In order to obtain good dynamic performance an actuator with high speed is required. Also, the actuator size should be small that is can be fully integrated inside the drum.
- The classic wear adjuster mechanisms can not be used here, new means for an automatic wear adjustment must be designed.

References

1. Anderson, J., R., Ferri, A., A. : ***Behaviour of a single-degree-of-freedom system with a generalized friction law:*** Journal of sound and vibration 140(2), 287-304, 1990.
2. Arovnov, V., D'Sousa, A., F., Kalpakjian, S.: ***Interactions among friction, wear, and system stiffness- Part 1: effect of normal load and system stiffness***: Transactions of the ASME, vol 106, 1984.
3. Avilés, R., Hennequent, G., Hernández, A., Llcrente, J., I.: ***Low frequency vibrations in disc brakes at high car speed. Part I : experimental approach***: Inderscience Enterprises Ltd, 1995.
4. Bannatyne, R. : ***Future developments in electrically controlled braking systems*** : Society of automotive engineers 1999.
5. Bermann, A.: ***Amontons' law at the molecular level***: Tribology letters 4, 95-101, 1998.
6. Birch, T. : ***Automotive braking systems***: Delmar publishers 1999.
7. Blau, P. J.: ***Friction science and technology***: Marcel Dekker, Inc. 1996
8. Bosch : ***Bremsanlagen für Kraftfahrzeuge***: Robert Bosch GmbH, 1994.
9. Börber, M., Straky, H., Weispfenning, T., Isermann, R. : ***Model based fault detection of vehicle suspension and hydraulic brake systems***: Mechatronics 12, 999-1010, 2002.
10. Brake system parts manufacturers council: ***Automotive brake systems***: Prentice Hall, Englewood Cliffs, New Jersy 1991.
11. Curran, P., J., Crombez, D., S., Napier, L. : ***Brake-by-wire system having conditioned brake boost termination at key off***: Patent, US 6325471B1,2001.
12. Dietrich, J, Gombert, B., Grebenstein, M.: ***Elektromechanische Bremse mit Selbstverstärkung***: Patent, DE 198 19 564 C2, 2000.
13. Doericht, M.: ***Elektromechanische Bremsvorrichtung, insbesondere für ein Kraftfahrzeug***: Patent, EP 1138971 A3, 2004.
14. Doericht, M.: ***Method of controlling an electromechanical wheel brake actuator in a motor vehicle***: Patent, EP 1141573B1, 2000.

15. Dorißen, H., Dürkopp, K. : ***Mechatronics and drive-by-wire systems advanced non contacting position sensors***: Control Engineering Practice 11, 191-197, 2002.
16. Dow, T. A.: ***Thermoelastic effects in brakes***: Wear,59, 213-221, 1980.
17. Eriksson, M., Bergman, F., Jacobson, S.: ***On the nature of tribological conatct in automotive brakes***: wear 252, 26-36, 2002.
18. Gissenger, G. L., Menrad, C., Condtans, A.,: ***A mechatronic conception of a new intelligent braking system***: Control engineering practice, vol. 11, issue 2, 2003
19. Harris, A. L., Martin, P. : ***A brake-by-wire electro-hydraulic braking system***: Patent, GB, 2364355A, 2002.
20. Hartmann, H., Schautt, M., Pascucci, A.,Gombert, B.: ***eBrake- The mechatronic wedge brake***: Society of automotive engineers, 2001.
21. Hoffmann, C., Schiffner, K., Oerter, K., Reese, H. : ***Contact analysis for drum brakes and disk brakes using ADINA***: Computers and structures 72, 185-198, 1999.
22. Jarvis, R., P., Mills, B.: Vibrations induced by dry friction: Proc Instn Mech Engrs, vol 178, No 32, 1963.
23. Jonner, W., Hermann, W., Dreilich, L., Schunck, E. : ***Electrohydraulic brake system-The first approach to brake-by-wire technology***: Society of automotive engineers 1999.
24. Kato, K., Iwabuchi, A., Kayaba, T. : ***The effects of friction-induced vibration on friction and wear***: Wear, 80, 307-320, 1982.
25. Kelling, N. A., Heck, W. : ***The brake project-Centralized versus distributed redundancy for brake-by-wire systems***: Society of automotive engineers, 2002.
26. Leber, M. : ***Radbremse mit mechatronischer Kennwertregelung- Untersuchung von Betriebsverhalten und Fahreranbindungsproblematik hinsichtlich Brake-by-Wire Systemen***: VDI Verlag GmbH Düsseldorf, 1998.
27. Leber, M., Karheinz, B., Dusil, V.: ***Selbstversträkende Reibungbremse***: Patent, DE 195 39 012 A1, 1998.
28. Leffler, H. : ***Electronic brake management EMB- prospects of an integration of brake system and drive stability control***: Society of automotive engineers 1999.
29. Littlejohn, D., Riddiford, B. : ***Verfahren zur Regelung eines Bremssystems***: Patent, DE 695 19 181 T2, 2001.
30. Mitschke, M., Sellschopp, J., Braun, H. : ***Regelung der Bremsen an Kraftfahrzeugen im unterkritischen Bereich***: VDI-Verlag GmbH. Düsseldorf 1995.
31. Pavlov, K. J., Oh, P., Anwar, S.: ***Piezoelectric actuator for vehicle brakes***: Patent, GB 2 369 661 A, 2001.
32. Phillips, M. L., Harris, A. L.: ***Improvements in vehicle hydraulic braking systems of the brake-by-wire***: Patent, EP 0889817B1, 2002.

33. Poertzgen, G.: ***Electro-hydraulic Brake system***: Patent, EP 0 895 502 B1, 1997.
34. Rhee, S. K. : ***Wear mechanisms for asbestos-reinforced automotive friction materials***: Wear 29, 1974.
35. Roberts, R., Gombert, B., Hartmann, H., Lange, D., Schautt, M.: ***Testing the mechatronic wedge brake***: Society of automotive engineers, 2004.
36. Roberts, R., Schautt, M., Hartmann, H., Gombert, B.: ***Modelling and validation of the mechatronic wedge brake***: Society of automotive engineers, 2003.
37. Schenk, D., Wells, L., Miller, J. : ***Intelligent braking for current and future vehicles*** : Society of automotive engineers, 1999.
38. Severin, D., Dörsch, S.: ***Friction mechanism in industrial brakes***: Wear 249, 771-779, 2001.
39. Sherriff, P. W. : ***Electro-hydraulic braking systems***: Patent, US 0005661A1, 2002.
40. Shih, S., Somnay, R., Hannon, R., Kay, J.: ***Improved drum brake shoe factor prediction with the consideration of system compliance***: Society of automotive engineers, 2000.
41. Smesch, M., Breuer, B. : ***Die mechatronische Teilbelagscheibenbremse im zukünftigen Kraftfahrzeug***: Wiley-VCH Verlag GmbH Weinheim, 2002.
42. Stölzl, S. : ***Fehlertolerante Pedaleinheit für ein elektromechanisches Bremssystem (Brake-by-Wire)***: VDI Verlag GmbH Düsseldorf, 2000.
43. Wiehan, C., Neuhaus, D. : ***Potential of electronically controlled brake systems***: Society of automotive engineers, 1999.
44. Wang, N.: ***Drum brake actuator***: Patent, EP 0 531 496 B1, 2001.