INTERNATIONAL JOURNAL OF
ENERGY AND ENVIRONMENT

Volume 5, Issue 1, 2014 pp.97-110
Journal homepage: www.IJEE.IEEFoundation.org

A parametric FE modeling of brake for non-linear analysis

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Abstract
A parametric modeling of a drum brake based on 3-D Finite Element Methods (FEM) for non-contact analysis is presented. Many parameters are examined during this study such as the effect of drum-lining interface stiffness, coefficient of friction, and line pressure on the interface contact. Firstly, the modal analysis of the drum brake is also studied to get the natural frequency and instability of the drum to facilitate transforming the modal elements to non-contact elements. It is shown that the Unsymmetric solver of the modal analysis is efficient enough to solve this linear problem after transforming the non-linear behavior of the contact between the drum and the lining to a linear behavior. A SOLID45 which is a linear element is used in the modal analysis and then transferred to non-linear elements which are Targe170 and Conta173 that represent the drum and lining for contact analysis study. The contact analysis problems are highly non-linear and require significant computer resources to solve it, however, the contact problem give two significant difficulties. Firstly, the region of contact is not known based on the boundary conditions such as line pressure, and drum and friction material specs. Secondly, these contact problems need to take the friction into consideration. Finally, it showed a good distribution of the nodal reaction forces on the slotted lining contact surface and existing of the slot in the middle of the lining can help in wear removal due to the friction between the lining and the drum. Accurate contact stiffness can give a good representation for the pressure distribution between the lining and the drum. However, a full contact of the front part of the slotted lining could occur in case of 20, 40, 60 and 80 bar of piston pressure and a partially contact between the drum and lining can occur in the rear part of the slotted lining.

Keywords: Brake; Drum; Lining; Non-Contact element; Non-linear; Modal analysis; Linear elements.

1. Introduction
Drum brakes are commonly used in passenger cars and trucks. The main function of the brake is to reduce the speed and to assure the safety of the vehicle passengers on the road. It is known that, the drum brake generally is installed on the rear wheels of the passenger cars however, the disc brakes on the front wheels. The self-energization of the drum brake is one of the most advantage of the drum brake rather than disc is which help in reducing the required actuating force and its relatively lesser cost. Many researchers used the finite element methods (FEM) to analyze either two dimensional or three dimensional models of the drum brakes. The breakage of the friction material of the brake might occur before the friction material ends its wear life by studying the thermo-elastic on brake friction material

using (FEA), [1]. A three-dimensional finite element model of the disc brake was built by Tirovic et al [2]. He showed the interface pressure distribution was relatively unaffected by disc flexure and the rigid disc was preferable. A method for predicting disc brake pad's contact pressure distribution under different operating conditions was developed in three phase [3], the first was to obtain non-linear elasticity characteristics of pad friction material. The second phase to develop a finite element model of the brake system to simulate actuating test. The third phase was to develop a finite element model of the brake system to simulate the bench test. In a separate piece of work the contact problem by using the Package ADINA sparse solver was studied [4]. It was concluded that the turning moment on the axle rises constantly till the disc change from sticking to sliding conditions. It was also concluded that the sparse solver is efficient for solving the problem of non-linear contact that will be used in this work.
Up to date, there is no experimental method available for the direct measurement of dynamic interface pressure distribution in brakes. But static pressure distributions have been measured in disc brakes by using pressure-sensitive paper in these measurements and the ball pressure method was also used to get an indication to the pressure variation on the contact surface and will be taken as a guide to compare with them in this work [5, 6]. A 2-D finite element method of drum brake has been used to analyze the drum brake model and then, the thermal expansion, imperfect contact, the decay of braking capability and all kinds of transient phenomena were analyzed by using the finite element method [7-12]. It was argued that it is more expensive to use the finite element program in the main frame and that the pressure distribution on the lining plate does not vary significantly along the axis of the brake [13]. The measurement of the torque or the wear can be used to calculate the pressure distribution [14]. Nevertheless, these methods are indirect methods and it difficult to obtain directly the pressure distribution from an experiment. Recently, the applications of the boundary element method (BEM) have been increased and showed that the BEM can reduce the time required to prepare numerical data and can provide good results in the linear problems [15-20]. From these literatures, there is a need to build up a FEM to simulate the real drum brake and study the contact analysis at different operation conditions of pressures and stiffness and also the friction coefficient.

2. Modeling and validation of the drum brake assembly

It is well-known that the drum brake system consists of 5 main parts; the drum and the two shoes and two linings that have been created directly using the ANSYS package (FEM). Figure 1 indicates the meshed coupled drum, shoe and lining after appropriate simplification to the original parts. A solid 45 elements has been chosen from the package library to model the 3-D solid structure that has 8-nodes with three degree of freedom per each node. The drum consists of 11774 brick elements with 18123 nodes, however, the shoe consists of 1260 brick elements with 2666 nodes, and lining contains 2700 brick elements with 2886 nodes. Each lining covers an angle equal to 120º of the drum ring that has an internal diameter of 340 mm as shown in Figure 2. However, the shoe and shoe rib cover an angle equal to 140º of drum ring that has specifications as indicated clearly in Tables 1 and 2.
A commercial vehicle drum brake is chosen as an example for the experimental work and then verification of the theoretical work with the tested results. Eight accelerometers are mounted along the drum that is hit by an impulse hummer. The tested signal is then fed from the accelerometers to a FFT for further analysis as clearly shown in Figure 3. The natural frequencies and modes of the drum and lining were collected in two cases. The first case was a free-free drum and free-free lining however the second case was the coupled drum-lining with a hydraulic pressure. The collected data has been analyzed in both cases and then the finite element model is being adjusted to control the difference between the experimental results and theoretical results. Each component's FE model is refined and adjusted to make the analytical results close to experimental modal analysis results [21-25]. An accurate representation of the component models as well as the statically coupled model is important for good correspondence between experimental squeal characteristics and those in simulations because the brake system's propensity to squeal is very sensitive to the geometry of the system and the material properties. Then the natural frequencies and mass-normalized mode shapes for each component are extracted from the modal analysis of the FE models. These modal characteristics of the components are used to replace the FE models to form the coupled system, and the total degrees of freedom are greatly reduced. To ensure the accuracy of the modal representations of the components and the convergence of the stability analysis results, the upper cut-off frequency for individual component modes was selected at least as high as twice of the squeal frequency of interest. It was found that for a system model constructed in this manner, the frequencies of the statically coupled drum-shoes-lining system exhibited convergence in the

range of interest. The boundary conditions have been applied to the drum and shoe, and apply the appropriate solving to the model. Many trials were made to adjust the meshing elements for the drum and shoe with the appropriate number of elements. A lowest difference between the experimental work, and predicted FE model has been achieved in models shown in Figures 1 and 2. It was found a maximum difference of ± 3 % between the experimental and FEM results for the drum and ± 2.5 % for the shoe with lining. So, this difference in both cases seems to be acceptable to carry on with these models as in Table 3.

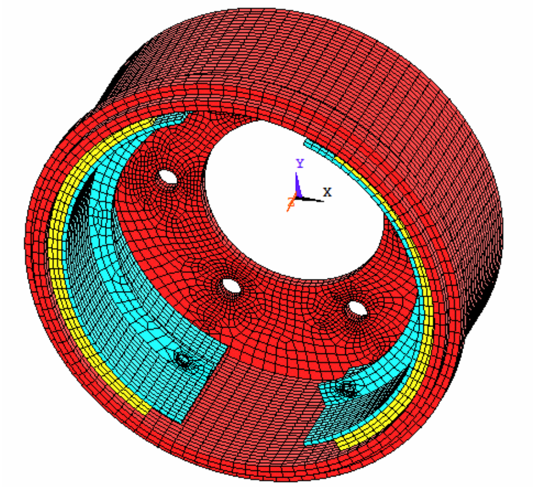


Figure 1. FE of the coupled drum brake system

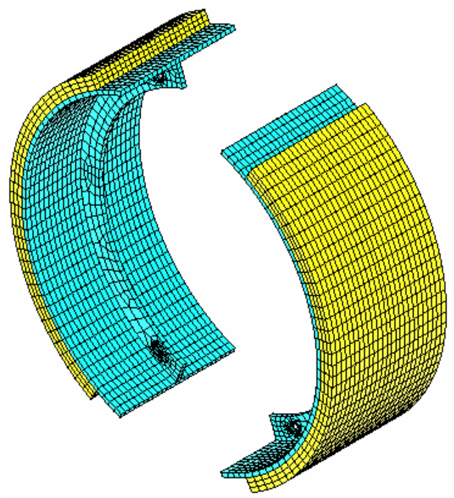


Figure 2. Schematic of the leading and trailing linings

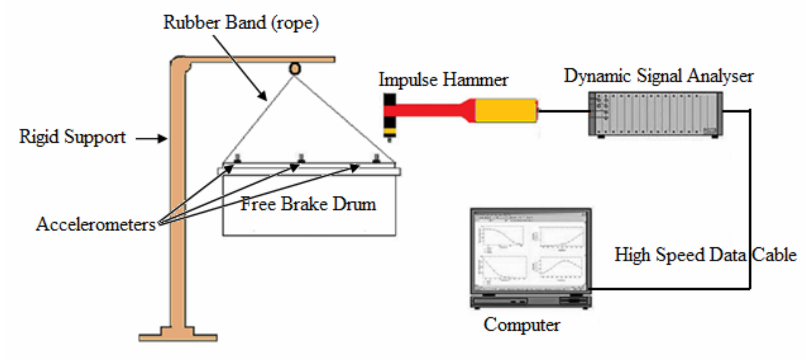


Figure 3. Modal testing for the free-free brake drum

Table 1. Original brake drum properties and dimensions

Drum dimensions and properties	Value
Outer drum diameter,	360 mm
Inner drum diameter,	340 mm
Inner drum cap diameter,	160 mm
Outer drum cap diameter,	340 mm
Height of drum cap,	10 mm
Diameter of holes centreline,	220 mm
Height of drum,	150 mm
Holes diameter (6),	10 mm
Inner diameter of upper ring,	360 mm
Outer diameter of upper ring,	360 mm
Height of upper ring,	15 mm
Density of drum,	7350 kg/m^3
Young's modulus of drum,	1200 GN/m^2
Poisson ratio of drum,	0.27

Table 2. Original brake lining properties and dimensions

Lining and shoe dimensions and properties	Value
Thickness of lining,	12 mm
Width of lining,	120 mm
Thickness of shoe,	4 mm
Width of shoe,	120 mm
Thickness of shoe rib,	4 mm
Height of shoe rib,	20 mm
Lining arc,	120º
Shoe and rib arc,	140º
Hole diameter,	10 mm
Slot depth,	12 mm
Slot length,	10 mm
Density of lining,	1350 kg / m^3
Young's modulus of lining,	200 MN / m^2
Poisson ratio of lining,	0.23
Density of shoe,	7800 kg / m^3
Young's modulus of shoe,	2000 GN / m^2
Poisson ratio of shoe,	0.27

Table 3. Natural frequencies for the modal testing and FE model for free-free brake drum and lining.

Component name	No. of nodes and elements	Mode number	Modal testing (kHz)	FE model (kHz)
Drum	11774 elements and 18123 nodes	1	1483	1503
		2	1488	1508
		3	2087	2068
		4	2140	2200
		5	2302	2332
		6	2340	2345
		7	2917	2906
		8	3087	3097
		9	3129	3109
		10	3203	3253
Lining	Lining 2700 elements and 2886 nodes + Shoe 1260 elements and 2666 nodes	1	1179	1200
		2	1213	1240
		3	1667	1684
		4	1843	1820
		5	2226	2210
		6	2233	2253
		7	2605	2600
		8	2679	2677
		9	2861	2886
		10	3109	3100

It is fully-known that there is a small gap between the drum and the two shoes during the rotation of the wheel. However, this gap becomes zero at the full contact between the drum and the shoe. The coupling between the brake shoe and lining assembly and the brake drum is made by the contact between them when the brake is actuated and friction force between the lining and drum is generated. The coupling can be regarded as a "contact stiffness" modeled by springs connecting the brake shoe assembly and the brake drum [26]. However, the model proposed here expects that the degree of coupling, and thus the contact spring stiffness, will be determined by the contact force between the brake lining and brake drum. This means that the contact stiffness over the whole contact area is dependent not only on the brake force applied, but also on the friction interface pressure distribution. The contact stiffness will therefore vary

ISSN 2076-2895 (Print), ISSN 2076-2909 (Online)

around the contact surface, being higher as the local contact pressure increases. The coupling between the lining and drum can be represented by springs whose stiffnesses represent the local interface contact pressure and the brake shoe can thus be modeled as being coupled to the drum via two springs, one representing the contact stiffness $K_{contact}$ and one representing the brake lining dynamic stiffness K_{lining} [27]. These two springs can then be combined to give a single "coupling" spring whose stiffness is

$$\frac{1}{K_{coupling}} = \frac{1}{K_{contact}} + \frac{1}{K_{lining}} \tag{1}$$

Figure 4 shows the APDL (Ansys Parametric Design Language) flowchart of a contact analysis of drum brake assembly system. This APDL includes 3 main stages such as preprocessor, solutions and post-processors. The pre-processors step include construction of FE models of drum brake assembly, parametric meshing generation, parametric material definition, parametric boundary conditions, parametric analysis options, parametric solving and parametric post-processing [28]. The post-processing step includes the summary, reading and plotting of required results. Figures 1 and 2 show the used drum brake that contain the main 5 components which are the drum, the two brake shoes (leading and trailing) and the two lining (leading and trailing), participate in the vibrational response of a drum brake system. The attached linings to the shoes will be in contact with the drum during braking to produce the friction forces. These leading and trailing shoes can be moved in different direction opposed to each other through hydraulic cylinders which contain two pistons to assist the shoes in the braking action. It is well-known that the friction-induced vibration is generated by the stiffness and friction coupling between the drum and the shoes through the shoe lining. A simplified coupled model that includes the drum, the shoes, and the shoe lining has been modeled to incorporate the effect of different boundary conditions on the occurrence of squeal. Finite Element models are built-up by ANSYS package for the five components of drum brake using 3-dimensional brick elements called SOLID45 [29]. The drum is clamped in the bolt hole positions while the shoe model uses free boundary conditions. To include the inertial and stiffness influences of the shoe lining on modal characteristics (eigenvalues and eigenvectors) of the shoe, the shoe lining is modeled as an integral part of the FE model of the shoe.
The equations of motion of the uncoupled system including one drum and two identical shoes can be written as [23, 24, 27, 30]

$$\{\ddot{q}\} + \lfloor \omega^2 \rfloor \{q\} = \{0\} \tag{2}$$

where $[\omega^2]$ is a diagonal matrix of the extracted *N* natural frequencies of the components, and {q} is an N-vector of generalized coordinates. However, the number of degrees of freedom of the system, *N, is* equal to the total number of extracted component modes. In considering the coupling between the drum and the shoe through the contact lining, the contact interface between the drum and shoe is discretized into a mesh of 2-dimensional contact elements. The lining material is then modeled as a spring located at the contact elements. So, the equation of motion of the coupled system is as follows;

$$\{\ddot{q}\} + \left[\lfloor \omega^2 \rfloor + [A] + \mu[[B] + [C]]\right]\{q\} = \{0\} \tag{3}$$

where, [A] and [C] are stiffness contribution due to the lining and shoe supports respectively and [B] arises from friction coupling between the shoes and drum which is assemmetric. In the absence of lining coupling i.e., [A] and μ equal to zero, the eigenvalues are purely imaginary that is being the natural frequencies of the drum components and the shoes coupled through the hydraulic cylinder stiffness and backing plate stiffness. The solution of the Equation 3 gives the eigenvalues of:

$$S = \sigma_i \pm j\omega_i \tag{4}$$

However; in the presence of the lining stiffness coupling but without friction coupling, the eigenvalues are again purely imaginary and correspond to the natural frequencies of an engaged brake system which is not rotating. This is referred to what is called statically coupled system. In the presence of the lining stiffness coupling but with friction coupling and the [B] is non-symmetric. When all of the eigenvalue are purely imaginary, these correspond to the natural frequencies of an engaged and rotating system. If any of the eigenvalues is complex, it will appear in the form of complex conjugate pairs, one with positive real part and the other with negative real part. The existence of complex roots with positive real parts indicates the presence of mode merging or what is called coupled mode, instability, which causes the brake to squeal. The value of friction coefficient that demarcates stable and unstable oscillations will be referred as a critical value of friction coefficient μ_{cr}. The imaginary part of the eigenvalues with a doublet root at this μ_{cr} is the squeal frequency and the corresponding mode of the complex structure is the mode shape at this squeal frequency.

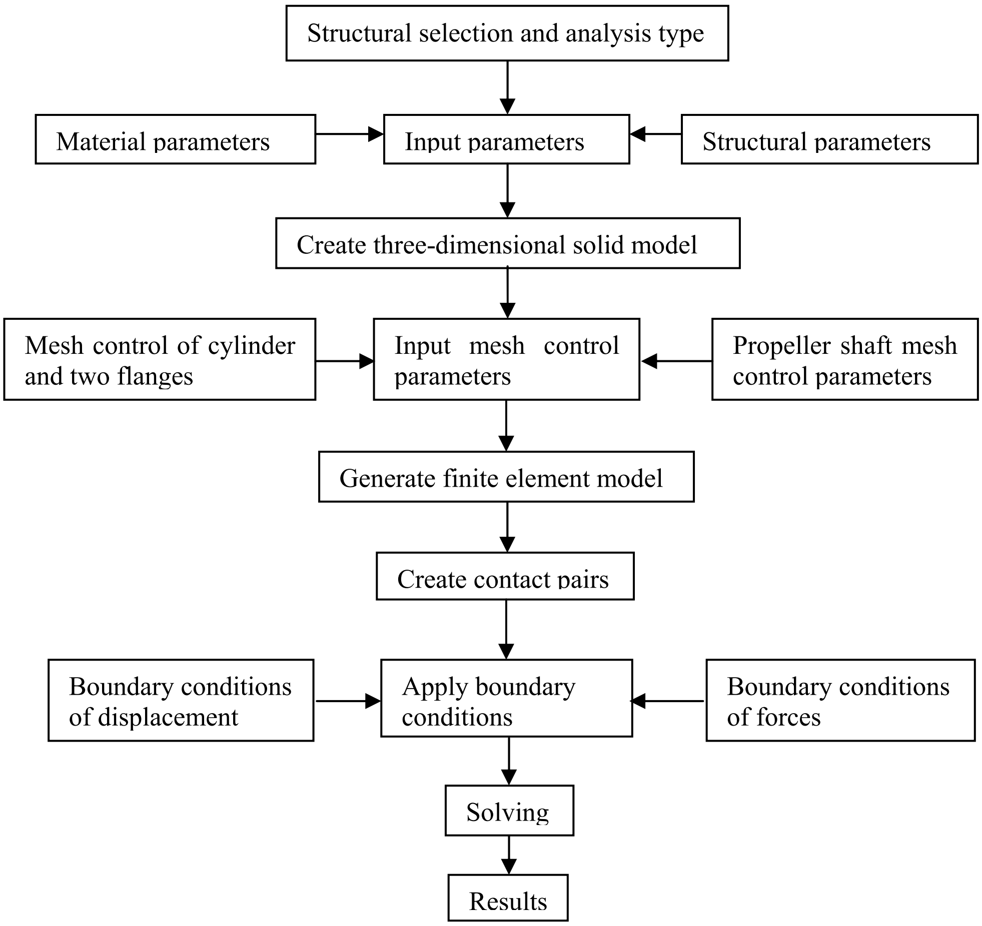


Figure 4. Parametric modeling and modal analysis flowchart, [28]

3. Contact analysis and contact pairs identifying

The contact problems are non-linear and require significant computer resources to solve. Therefore, it is important to understand the physics of the problem and take the time to set up the model of your problem to run as efficiently as possible. Contact problems present two significant difficulties. First, the regions of contact are not known to run the problem. Depending on the loads, material, boundary conditions, and other factors, surfaces can come into and go out of contact with each other in a largely unpredictable and abrupt manner. Second, most contact problems need to account for friction. Frictional response can be chaotic making solution convergence difficult. Contact problems fall into two general classes; rigid-to-flexible and flexible-to-flexible. In this work which is rigid-to-flexible contact problems; one or more of the contacting surfaces are treated as rigid (drum) that has much higher stiffness relative to the other part. The other surface (friction material), which has softer stiffness, is treated as the contact surface. Finally, the purpose of the non-linear contact analysis is to calculate the interfacial contact boundary, normal contact forces and circumferential friction forces under frictional braking conditions.

In this work the finite element supports rigid-to-flexible surface-to-surface contact elements offered by Ansys has been used for the contact analysis. The rigid surface referred to the (target) surface and is

ISSN 2076-2895 (Print), ISSN 2076-2909 (Online)

modeled with TARGE170 for 3-D works and the target surface in this work is the drum surface. The surface of the deformable body is referred to the (contact) surface and is modeled with element CONTA 173 for elements without midside nodes and the contact surface is leading and trailing friction material. The target and contact elements known as a contact pair (drum and lining) are identified through a shared real constant number ID. One condition should be taken into account in identifying the contact pair where contact might occur during the deformation of the model. Once the contact surfaces have been identified via target and contact elements, which will then track the kinematics of the deformation process, target and contact elements that make up a contact pair are associated with each other via a shared real constant set. The contact zone can be arbitrary; however, for the most efficient solution should be defined by smaller, localized contacting zones, but these contact zones should be adequate to capture all necessary contact. Different contact pairs must be defined by a different real constant set. There is no limit on the number of surfaces allowed.

Depending on the geometry of the model, two target surfaces could interact with the two friction materials, which are the contact surfaces. The rigid target surface in this work is a 3-D application (drum brake application). So, the shape of the target surfaces is described by a pilot node, which is really an element with one node, whose motion governs the motion of the entire target surface. Forces or rotations for the entire target surface can be prescribed on just the pilot node. When the pilot node is defined, FE program will check for boundary conditions only on the pilot node and ignores any constraints on other nodes. To create the deformable contact surface, that surface should be defined using contact elements CONTA 173. The contact surface is defined by the set of contact elements that comprise the surface of the deformable body. These contact elements have the same geometric characteristics as the underlying elements of the deformable body. The nodes on the meshed deformable body should be selected first to generate contact elements. For each surface, the node list should be viewed first. If there are some particular nodes will never come into contact, it could be omitted. However, it could be included to sure there are unexpected areas of contact.

Any contact problems require stiffness between the two contact surfaces. The amount of penetration between the two surfaces depends on this stiffness. Higher stiffness values can lead to ill conditioning of the global stiffness matrix and to convergence difficulties. Ideally, a high enough stiffness wanted that contact penetration is acceptably small, but a low enough stiffness that the problem will be well behaved in terms of convergence or matrix ill conditioning. The value of this contact stiffness controls the accuracy of the contact behavior as well as the convergence characteristics of the problem. Low contact stiffness may cause excessive penetration and large relative displacement between the contacting nodes and may therefore inaccurately simulate the contact behavior. On the other hand, high contact stiffness may result in convergence problem, [31]. Therefore, appropriate contact stiffness must be determined before commencement of the main simulation of works. In the basic Coulomb friction model, two contacting surfaces can carry shear stresses up to a certain magnitude across their interface before they start sliding relative to each other. This state is known as sticking. The Coulomb friction model defines an equivalent shear stress τ at which sliding on the surface begins as a fraction of the contact pressure P ($\tau = \mu$. P, where μ is the friction coefficient which is defined as a material properties. Once the shear stress is exceeded, the two surfaces will slide relative to each other; this state is known as sliding. It is found that the optimum contact stiffness in this work is 1.1 GN/m.

4. Methodology of non linear contact analysis

A complex eigenvalue analysis technique that is available in ANSYS package is firstly used to determine the stability of drum brake assembly. The real and imaginary parts of the complex eigenvalues are responsible for the degree of instability (unstable frequencies and unstable modes) of the drum brake assembly and are thought to imply the likelihood of squeal occurrence. The importance of this method lies in the asymmetric stiffness matrix that is derived from the contact stiffness and the friction coefficient at the drum-lining interface [32]. In order to perform the complex eigenvalue analysis using ANSYS, four main steps are required [33]. They are given as follows: 1-Nonlinear static analysis for applying drum brake-line pressure. 2-Nonlinear static analysis to impose rotational velocity on the drum. 3-Normal mode analysis to extract natural frequency of undamped system. 4-Complex eigenvalue analysis that incorporates the effect of friction coupling. Finally the non-linear contact analysis provides information on the size of the areas of contact between the linings and rotor for a certain coefficient of friction, actuation load and initial gap [34]. The number of finite element nodes contained within these predicted lining contact areas are consequently coupled with the radially opposite rotor nodes in order to perform a dynamic modal analysis [35]. To investigate the mode shapes, natural frequencies and instability measurements by this type of analysis, the existing non linear structural system must be linearised. This means replacing the contact elements with equivalent linear elements as stated previously. Therefore matrix elements were used which have their elastic kinematic response specified by a form of stiffness matrix which relates the displacements of the two nodes across the lining/rotor interface. The amount of penetration or incompatibility between the two coupled nodes depends on the contact stiffness included in the stiffness matrix.

A non-linear contact analysis for the drum brake assembly is used to determine the pressure distribution for 20, 40, 60 and 80 bar respectively. The contact analysis between two brake components were carried out using either spring elements non-linear gap elements that allows for separation and small relative sliding of the contact surfaces. Any researcher usually faces two major problems: a) a lack of prior knowledge of the exact location of the adjacent contact nodes in order to generate this type of element, and b) the inability of introducing friction forces at the interface without previously knowing the magnitude of the normal forces acting between the components. Many studies have introduced the "friction stabilization" repetitive process [36] in which the friction forces are derived from the normal contact forces calculated in the previous load step and continues until normal and tangential forces are related by Coulomb's law. The curved geometry of the drum brake contact areas adds a degree of complexity when trying to accurately capture the behavior of the friction interface. Therefore, contact surface elements are used with elastic friction capabilities. A contact (conta173) and a target (Targe 170) surface need to be defined and in this case, the linings' outer surfaces were meshed with contact elements while the adjacent surfaces on the drum were meshed with target elements. Contact occurs when the contact surface penetrates the target segment elements. The determination of the pressure distribution occurs as follows; displacement loads are imposed firstly on the shoes in order to close the installation gap and therefore bring them in contact with the drum. At the end of this loadstep, the displacements are replaced by forces which are gradually applied to both leading and trailing shoes in case of plain and slotted linings. These two steps together enable the calculation of the static pressure distribution. The solution procedure ultimately reaches a limit at which the torque reacted is at its maximum and any further rotational increment will have no effect on the pressure distribution. Hence, the simulation predicts the "pseudo-dynamic" contact boundaries between the two linings and the drum.

5. Results and discussions of contact analysis

The next group of figures covers the pressure distribution on lining contact surface at 20, 40, 60 and 80 bar loading for plain and slotted linings respectively. Figure 5 shows the vector direction at piston pressure of 40 bar as an example of this vector directions for leading and trailing lining respectively. The effect of the piston pressure on the leading and trailing linings is mainly concentrated in the longitudinal centerline of the lining. Generally, literature showed the importance of studying the contact analysis of the drum brake system. It was found the nodal reaction force is the most important in this study. The effect of the applied load on the leading and trailing linings is presented in Figures 6 to 15. So, Figures 6 and 7 represent the reaction force for leading and trailing linings at 20, 40, 60 and 80 bar of piston pressures for the plain and slotted linings. A mapping on the contact surface of the ling has been captured to show the 3-D representation of the reaction force on both kinds of linings. The number of nodes to be studied on the contact surface of the plain lining is 962 nodes distributed through 900 brick solid element as shown in Figures 6 to 15 either leading or trailing. However; the number of nodes to be studied on the contact surface of the slotted lining is 936 nodes distributed through 850 brick solid elements.

Figures 6-a and 6-b show the reaction force on the contact surface of the leading lining for both plain and slotted lining at piston pressures of 20, 40, 60 and 80 bar respectively at a contact stiffness of 1100 MN/m and coefficient of friction of 0.42. It shows that at a low piston pressure of 20 bar, the contact area of the plain leading lining is smaller than the contact areas at higher piston pressure. The maximum reaction force reached at 20 bar is 37 N and 66 N at piston pressure of 40 bar. It increases till 110 N in case of 80 bar. It showed also that a 30 % of full contact area is reached at 20 bar, however, it reaches around 90 % of full contact between the lining and the drum at 80 bar of piston pressure. Figure 6-b shows the reaction force at the same values of piston pressures but for the slotted leading lining. The normal contact area between the lining and the drum is less than the plain lining by around 6 %. This less of contact area leads to a higher value of reaction force which also leads to a higher friction between the lining and the drum. The main advantage of using a slot in the middle of the lining is to decrease the lining wear due to the wear particle removal through this slot but it is not covered through this investigation. The nodal reaction force for each node on the contact surface is displayed on a 3-dimensional graph as shown in Figures 7, 8, 9 and 10 (a and b) for the plain and slotted lining. This 3-D graph shows a mapping of the contact surface to illustrate the nodal reaction force. These figures also indicate that the reaction force increase and the contact force increase as the piston pressure increase due to the increase in the elastic deformation of the shoe which lead to a contact change pattern for the leading shoe from original crown contact to an almost full contact between the lining and the drum. The self-energization phenomena of the leading lining also help in increasing the contact area between the lining and the drum.

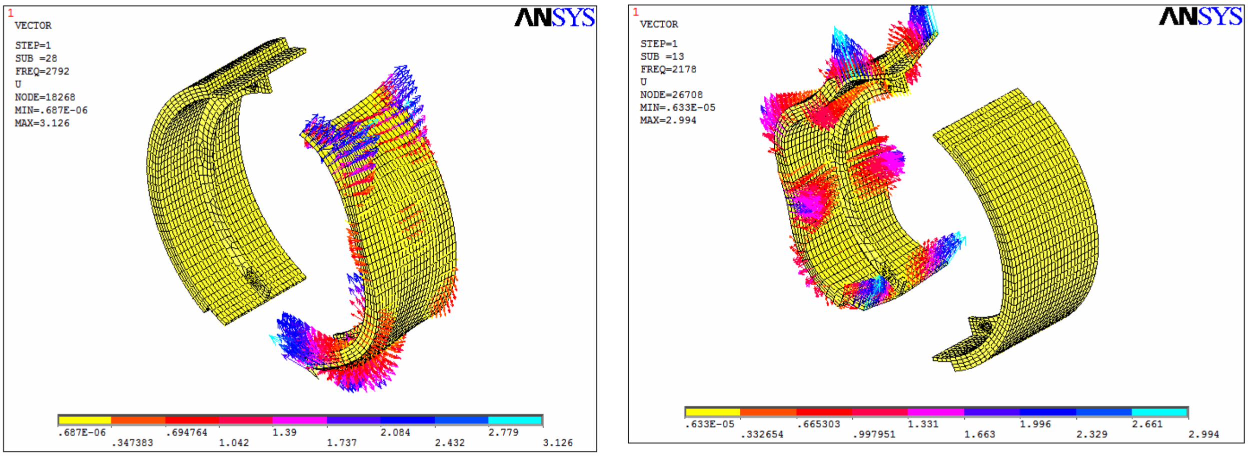
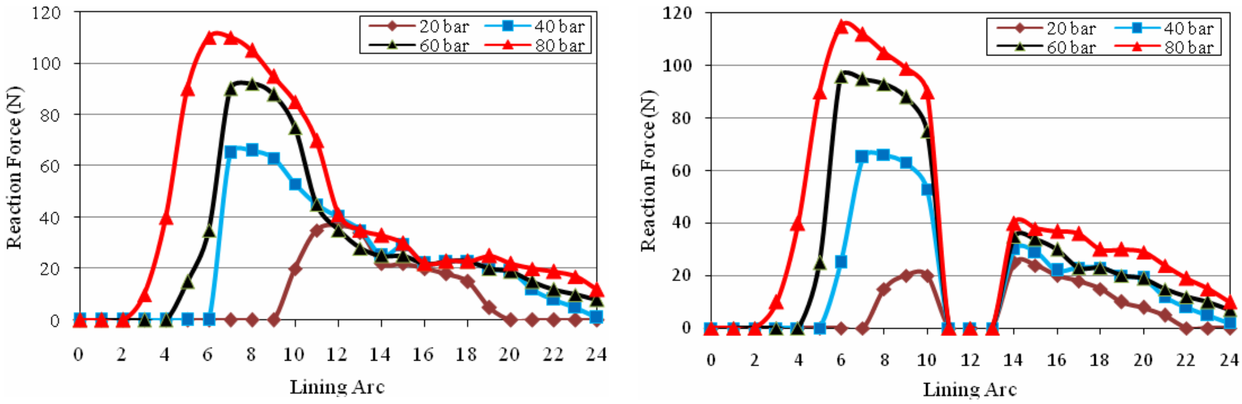
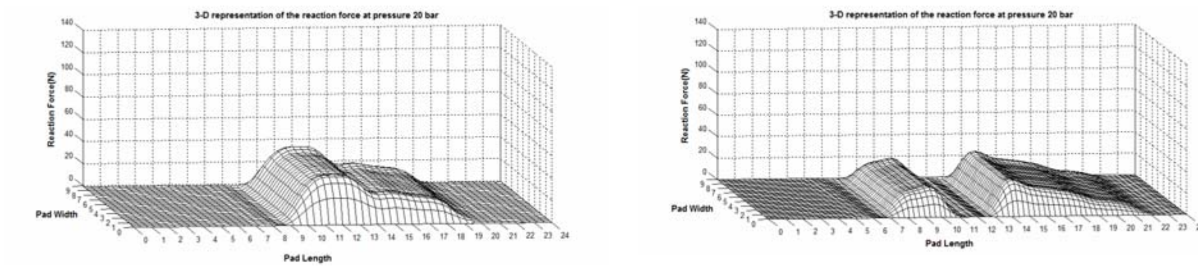


Figure 5. Vector direction at piston pressure of 40 bar for the leading and trailing linings



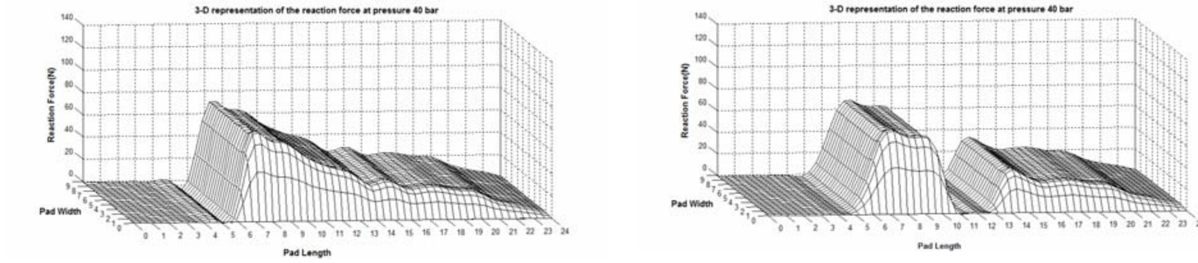
(a) Plain lining (b) Slotted lining

Figure 6. Reaction force against the lining arc under different pressure loads



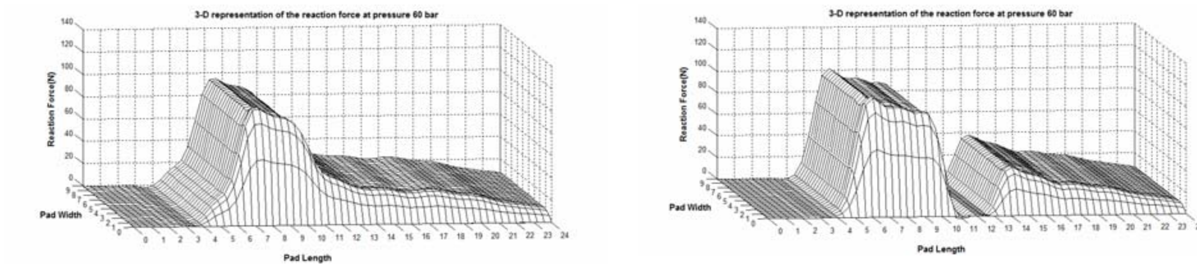
(a) Plain lining (b) Slotted lining

Figure 7. 3-D representation of reaction force on the leading lining due to 20 bar piston loading



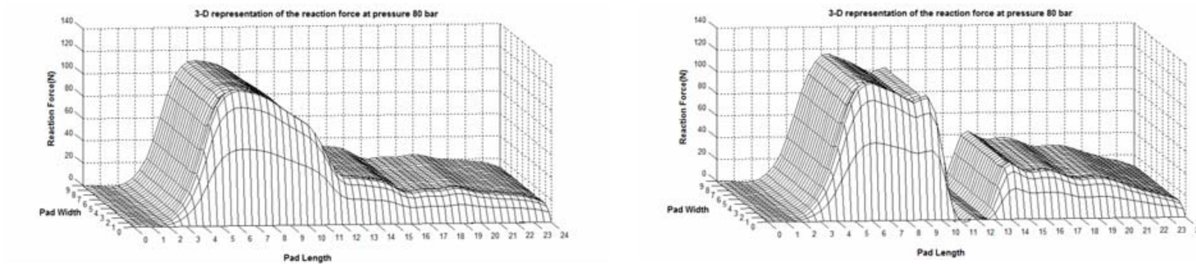
(a) Plain lining (b) Slotted lining

Figure 8. 3-D representation of reaction force on the leading lining due to 40 bar piston loading



(a) Plain lining (b) Slotted lining

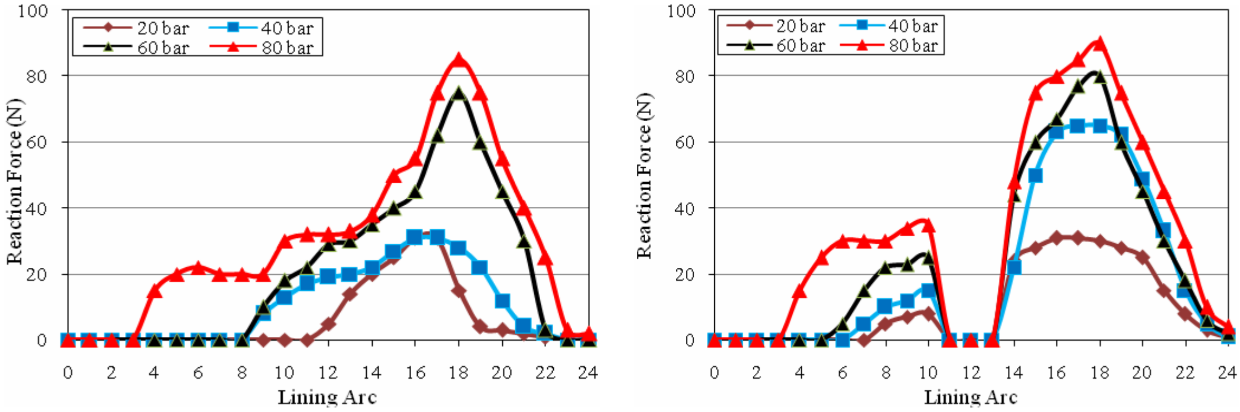
Figure 9. 3-D representation of reaction force on the leading lining due to 60 bar piston loading



(a) Plain lining (b) Slotted lining

Figure 10. 3-D representation of reaction force on the leading lining due to 80 bar piston loading

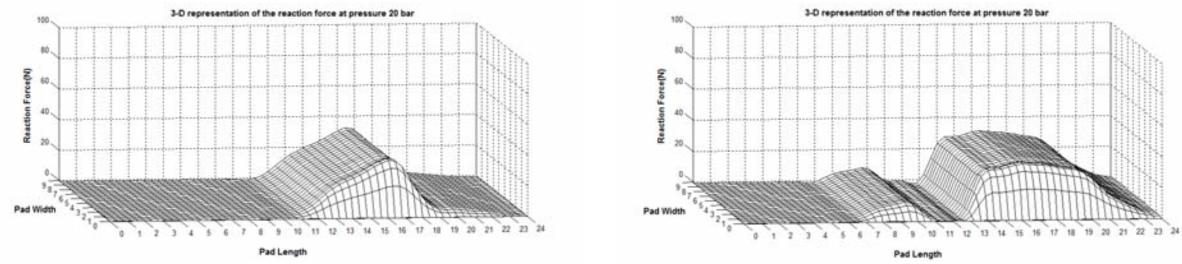
Figures 11-a and 11-b show the reaction force on the contact surface of the trailing lining for both plain and slotted lining at piston pressures of 20, 40, 60 and 80 bar respectively at a contact stiffness of 1100 MN/m and coefficient of friction of 0.42. It shows that at a low piston pressure of 20 bar, the contact area of the plain trailing lining is smaller than the contact areas at higher piston pressure. The maximum reaction force reached at 20 bar is 31 N and 31 N at piston pressure of 40 bar. It increases till 75 N at 60 bar however; it reaches 85 N in case of 80 bar. It showed also that a 36 % of full contact area is reached at 20 bar, however, it reaches around 63 % and 85 % of full contact between the lining and the drum at 60 bar and 80 bar of piston pressure respectively. Figure 11-b shows the reaction force at the same values of piston pressures but for the slotted trailing lining. It shows a maximum of 90 N at 80 bar of piston pressure. It is realized from this figure that a high peaks of the reaction force ate the first half of the lining however, these peaks decrease to nearly 50 % of its value in the second half of the lining. This could be due to the missing of the self-energization phenomena of the leading part in the trailing lining, because it only exists in the leading lining. It is also cleared that a full contact between the first half of the trailing lining with the drum however; a missing contact area in the second half of the lining at four cases of piston pressures. The nodal reaction force for each node on the contact surface is displayed on a 3-dimensional graph as shown in Figures 12, 13, 14 and 15 (a and b) for the plain and slotted trailing lining. This 3-D graph shows a mapping of the contact surface to illustrate the nodal reaction force. These figures also indicate that the reaction force increase and the contact force increase as the piston pressure increase due to the increase in the elastic deformation of the shoe which lead to a contact change pattern for the leading shoe from original crown contact to an almost full contact between the lining and the drum.



(a) Plain lining

(b) Slotted lining

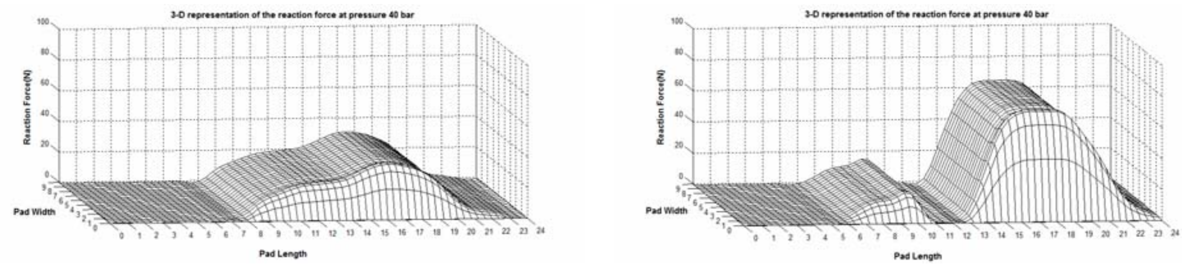
Figure 11. Reaction force against the lining arc under different pressure loads



(a) Plain lining

(b) Slotted lining

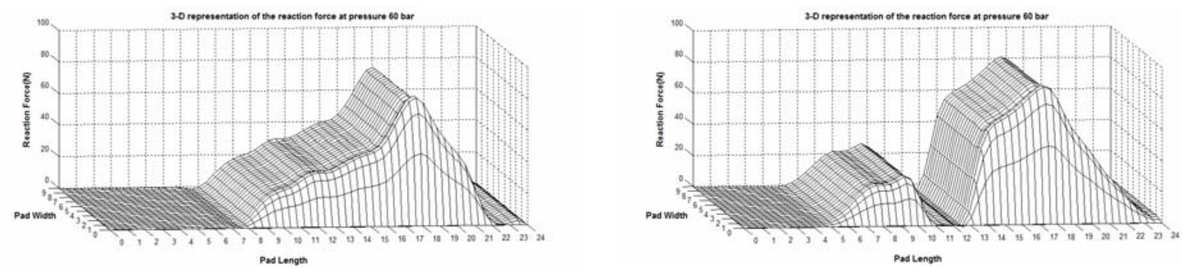
Figure 12. 3-D representation of reaction force on the leading lining due to 20 bar piston loading



(a) Plain lining

(b) Slotted lining

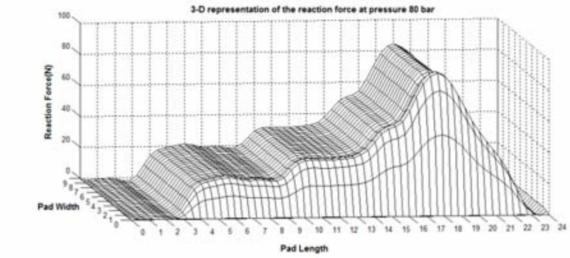
Figure 13. 3-D representation of reaction force on the leading lining due to 40 bar piston loading



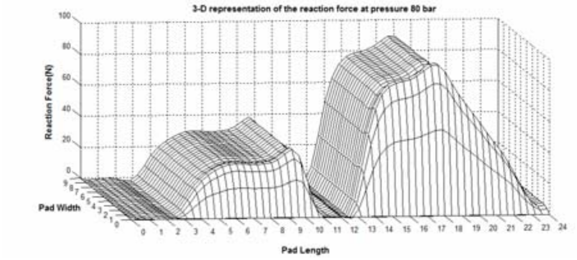
(a) Plain lining

(b) Slotted lining

Figure 14. 3-D representation of reaction force on the leading lining due to 60 bar piston loading



(a) Plain lining



(b) Slotted lining

Figure 15. 3-D representation of reaction force on the leading lining due to 80 bar piston loading

6. Conclusion

A FEM for the drum brake assembly is developed to study the contact analysis in conjunction with the modal analysis to predict the squeal of the brake. It is shown that the Unsymmetric modal analysis is efficient enough to solve this linear problem after transforming the non-linear behaviour of the contact between the drum and the lining to a linear behavior. A linear element which is used in the modal analysis is transferred to non-linear elements which are Targe170 and Conta173 to study the contact analysis. The contact analysis using the technique of surface-to-surface contact is applied which give a compatible representation to the real contact behavior. However; the contact problems are highly non-linear and require significant computer resources to solve it, however, the contact problem give two significant difficulties. Firstly, the region of contact is not known based on the boundary conditions, line pressure, and drum and friction material specifications. Secondly, these contact problems need to take the friction into consideration. It showed that the distribution of the nodal reaction force depends mainly on piston pressure on the lining either leading or trailing. The slotted lining gave a good distribution of the nodal reaction forces and existing of the slot in the middle of the lining can help in wear removal due to the friction between the lining and the drum. An accurate contact stiffness of 1100 MN/m can give a good representation for the pressure distribution between the lining and the drum. A full contact of the front part of the slotted lining could occur in case of 20, 40, 60 and 80 bar of piston pressure however; a partially contact between the drum and lining can occur in the rear part of the slotted lining. This can lean lead to a wear in the front part faster than the rear part of the lining.

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