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Proceedings

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THE EIGHTH INTERNATIONAL TRIENNIAL CONFERENCE

HEAVY MACHINERY
HM 2014

PROCEEDINGS

Zlatibor, June 25 – June 28 2014.

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PREFACE

The Faculty of Mechanical Engineering Kraljevo has been traditionally organizing the international scientific conference devoted to heavy machinery every three years. The VIII International Scientific Conference HM 2014 is considering modern methods and new technologies in the fields of transport design in machinery, control energy, production technologies, urban engineering and civili engineering through thematic sessions for the purpose of sustainable competitiveness of economic systems. Modern technologies are exposed to fast changes at the global world level so that their timely application both in large industrial systems and in medium and small enterprises is of considerable importance for the entire development and technological progress of economy as a whole.

The VIII International Scientific Conference Heavy Machinery HM 2014 is a place for exchange of experiences and results accomplished in domestic and foreign science and practice, with the goal to indicate directions of further development of our industry on its way toward integration in european and world economic trends. Exchange of experiences between our and foreign scientific workers should contribute to extension of international scientific-technical collaboration, initiation of new international scientific-research projects and broader international collaboration among universities.

The papers which will be presented at this Conference have been classified into seven thematic fields:

A. EARTH-MOVING AND TRANSPORTATION MACHINERY
B. PRODUCTION TECHNOLOGIES
C. CIVIL ENGINEERING AND MATERIALS
D. AUTOMATIC CONTROL, ROBOTICS AND FLUID TECHNIQUE
E. MACHINE DESIGN AND MECHANICS
F. RAILWAY ENGINEERING
G. URBAN ENGINEERING, THERMAL TECHNIQUE AND ENVIRONMENT PROTECTION

Within this Conference, the First International Students Symposium will be held. The aim is to open a scientific discussion on this actual problem in industry among young students.

The sponsorship by the Ministry of Science of the Republic of Serbia is the proper way to promote science and technology in the area of mechanical engineering in Serbia.

On behalf of the organizer, I would like to express our thanks to all organizations and institutions that have supported this Conference. I would also like to extend our thanks to all authors and participants from abroad and from our country for their contribution to the Conference. And last but not the least, dear guests and participants in the Conference, I wish you a good time in Kraljevo – Vrnjačka Banja and see you again at the Eight Conference, in three years.

Kraljevo – Zlatibor, June 2014

Conference Chairman,

М. Гашић

Prof. Dr Milomir Gašić, mech eng.

PLENARY SESSION

SESSION A: EARTH-MOVING AND TRANSPORTATION MACHINERY

SESSION B: PRODUCTION TECHNOLOGIES

SESSION C: CIVIL ENGINEERING AND MATERIALS

SESSION D: AUTOMATIC CONTROL, ROBOTICS AND FLUID TECHNIQUE

SESSION E: MECHANICAL DESIGN AND MECHANICS

SESSION F: RAILWAY ENGINEERING

SESSION G: URBAN ENGINEERING, THERMAL TECHNIQUE AND ENVIRONMENT PROTECTION

CAD Model of Disc Brake for Eliminating Noise Problems

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Highway safety and stopping power are always at the forefront of discussions within the commercial vehicle industry. Air disc brake (ADB) systems have been available for commercial vehicles since the 1970s. The technology initially suffered teething problems, but brake manufacturers say today’s air disc brakes are highly dependable and reliable with superior stopping characteristics that make them an obvious candidate for fleets wanting to make certain they are in compliance with the new stricter regulations. One of vehicle components that occasionally generate unwanted vibration and unpleasant noise is the brake system. As a result, carmakers, brake and friction material suppliers face challenging tasks to reduce high warranty payouts.

This study takes into consideration three major aspects of modelling of a real disc brake so that the model can be built in a more realistic way. There are the structural model, the friction model and the contact model. A fully numerical method is used where all the disc brake components are modelled and analysed using finite element software packages. Having developed the disc brake components, modal analysis is carried out at the brake components and assembly levels. Friction and contact model are included when all the brake components are brought together.

Keywords: CAD, disc brakes, FEM, modeling, noise

1. INTRODUCTION

Highway safety and stopping power are always at the forefront of discussions within the commercial vehicle industry. Air disc brake (ADB) systems have been available for commercial vehicles since the 1970s. The technology initially suffered teething problems, but brake manufacturers say today’s air disc brakes are highly dependable and reliable with superior stopping characteristics that make them an obvious candidate for fleets wanting to make certain they are in compliance with the new stricter regulations.

ADBs are now accepted as the primary foundation brake in Europe. Drum brakes are still used on off-road vehicles (mining, construction, military, etc.) and on vehicles for export to other continents - 18 percent of total European Union (EU) brake demand.

Among the reasons for introduction of ADBs in EU are:

− With ADBs, brake fade is virtually eliminated, proven from Alpine testing.

− Inherent high-efficiency (greater than 95 percent) and low hysteresis ensure a negligible pull (different brake performance left and right) to deliver controlled vehicle steering and braking stability

− This same high-efficiency and stability enable the highest-quality of control functions for electronic control systems like ABS, electronic braking systems (EBS) and electronic stability systems. (ADBs were introduced in parallel with EBS in the EU during 1996)

− ADBs support intelligent functions, such as continuous wear sensors, brake pad wear monitor and, in the future, electronic clearance control

− New ADBs designs reduced stopping distance up to 30 percent at the time of introduction in the EU, compared with then-current drum brakes

− ADBs enable simpler, quicker pad change vs. drum shoes and have an integrated automatic wear adjuster function.

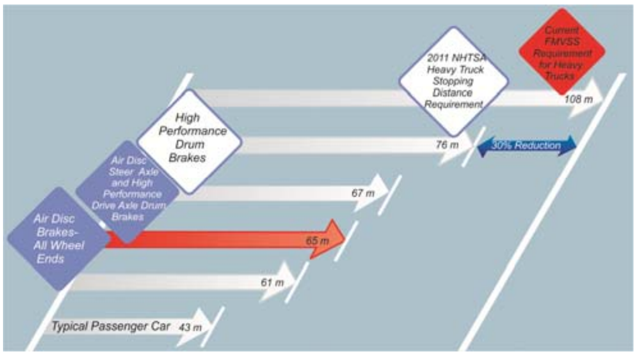

Figure 1: Comparsion of FMVSS Stopping Distance Requirements from 60 MPH

More than 90 percent of trucks in the United States still spec s-cam drum brakes. However, ADBs are widely used on refuse trucks and transit vehicles. There are several reasons contributing to the slow adoption of ADBs in North America (NA). Firstly, in Europe, the truck OEMs decide the vehicle specs, whereas NA is predominately a customer spec market. Next, there is a different service infrastructure. Trucks are serviced at OEM dealers in Europe by factory-trained and equipped technicians, so technology changes can be more easily managed and facilitated. In NA, vehicles are serviced at a wide variety of service locations, so conversion to new parts and training is more complex. Another factor is that trucks in North America are dynamically different so the impact of ADBs is less. Plus, the upfront costs for ADBs are more than drum brakes. Further, in general, technology application lags in North America compared to Europe.

Some of the brakes trends in the North American market are:

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− CSA (Compliance, Safety, Accountability) is putting more emphasis on brakes for service and reliability.

− Weight is becoming a factor as larger drum brakes, emissions equipment and pending GHG (Greenhouse Gas) regulations all shift the focus.

− Air disc brakes are widely available - a factor that influences choice.

− An array of safety innovations, such as collision mitigation and lane departure warning, compete for ADB dollars

− Data sophistication is growing with real-time telematics-based systems.

Among the major differentiators of ADB vs. drum brakes are feel and safety. As for feel, the ADB's linear output and stability drive the preference. With regard to safety, the difference is multi-faceted. With better brake feel there is less driver fatigue. Because there is less brake fade, less skill is required, making for safer drivers. In addition, stopping distance is slightly better. ADBs cost more than drum brakes but this has to be factored against future truck residual value, maintenance and service savings (pad changes are up to 75 percent faster than drum shoe changes and no periodic lubrication is required) and uptime improvement (adjuster mechanism and pistons are environmentally sealed for life and there are no current "out-of-adjustment" conditions). Greater adoption of ADBs in North America will continue to progress as vehicle owners and operators become more educated on their benefits and advantages compared to drum brakes [1].

In the testing sequence, two tractor-trailers were driven side-by-side on a closed test track with simultaneously applied full brake pressure to stop the vehicles at 75 mph. The stopping distance for air disc brake-equipped truck was within the range of 305 to 325 feet. The drum brake-equipped truck stopped in the range of 450 to 518 feet initially when cold, but as the drums heated up, the stopping distances became progressively longer. Stopping distance for the hot drum brake-equipped vehicle exceeded 750 feet, while the air disc brakes consistently stopped at around 320 feet.

The performance advantages for air disc brakes at higher speeds are particularly noteworthy when considering that during night-time driving, low-beam headlights only provide 350 feet of visibility. This is within the range of the air disc brakes' ability to stop a vehicle, but is not the case for drum brakes.

At 60 mph, the air disc brake-equipped truck stopped in the range of 185 to 210 feet in both the hot and cold brake temperature conditions. The drum brake-equipped truck stopped in the range of 255 to 292 feet with cold brakes and more than 425 feet with hot brakes [2].

The disadvantage of disc brakes is the high sensitivity (susceptibility) to self-excited vibrations. Most of the kinetic energy of a moving vehicle is converted into heat through friction. However, a small part of the kinetic energy is converted into acoustic energy and creates noise. The squealing brake is difficult and expensive to fix. It is better to solve the noise problem in the design phase [3]. Modern disc brakes with floating caliper are highly developed mechanical engineering device. They have to work reliably over a long lifetime, tolerating huge mechanical and thermal loads.

On the other hand, in recent decades, there has been a significant increase of engine power, but also the expectations in terms of comfort. This means that the noise levels, and especially brake noises, which are to be acceptable 20 or 30 years ago, are no longer tolerated by the modern user. Noise and vibration have become an important issue in the design of braking systems for motor vehicles. Efforts to improve today's braking systems must take into account the problems of noise and vibration. Good understanding of the generation mechanism of brake noise in this manner has become an important factor in the competition to design successful braking systems. It should be noted that, according to the manufacturers of brakes, brake noise is generally only a problem of comfort, and that according to them, does not affect the operation of the brakes. Although there are some new solutions in the field of braking systems ("brake by wire"), it did not affect the problem of brake noise until the brakes are working with the energy dissipation due to dry friction.

Vibrations in the braking process are a major problem of today's engineers, as can be seen by the existence of the NVH Department (Noise, Vibration and Harshness) in a number of companies. Frequently, these NVH teams focus on the problem of braking systems in terms of brake noise caused by high frequencies vibration with low amplitude. These oscillations are produced in the process of friction when the brake linings come into contact with the rotating elements. Created sound is much like that produced when writing or scraping on a chalkboard, an energy-dissipating frictional vibration occurs. Because frictional or self-excited vibrations are so different from resonant and forced vibrations, different methods of study need to be implemented when trying to understand, measure, and remedy these situations [3].

2. COMPUTER AIDED DESIGN (CAD) OF DISC BRAKE FOR ELIMINATING NOISE PROBLEMS

Computer aided design has evolved from the simple replacement of traditional drafting equipment to a very sophisticated, highly visual design tool. The earlier CAD programs used the computer to generate lines for 2D drawings. As the software and hardware advanced, these 2D drawings could be converted into 3D objects. Modern software used for solid modelling often functions in the reverse order; the three-dimensional object is drawn and then two-dimensional, orthographic drawings are generated from that model.

Modern software provides all the necessary tools for advanced designers and specialists involved in structural analysis. The processes covered include stress, frequency, thermo-mechanical, buckling and contact analysis with multiple load, restraint and mass complex configurations. Analysis can be performed on single parts as well as on hybrid models mixing solid, shell and beam elements. This allows for a wider number of mechanical behaviour and sizing assessments of parts and assemblies earlier in the product development process.

Analytical methods have proven to be inadequate to achieve complete understanding of squeal phenomena, as well as providing tools for the prediction and suppression of squeal. Besides, the analytical approaches are often limited to the study of a certain influential parameter. However, analytical methods are, despite its limitations, very useful for a concise explanation of the instability of the system. These disadvantages can be overcome by numerical methods, using the finite element method, which allows the development of models with a large number of degrees of freedom. Numerical methods take into account the deformability of elements during modelling, while the analytical approach is often treated them as rigid. Experimental methods are essential not only to quantify the nature of squeal noise and impact of different working conditions on this phenomenon, but also to ensure the validation of the results of the numerical approach and quality of brakes in terms of brake noise before going to market.

In recent years, the finite element method has become most commonly used tool for studying disc brakes squeal among researchers of this problem. The reason for this lies in the fact that this method offers a much faster and more economically cost effective solutions with regard to the experimental methods and can predict the performance of squealing noise in the early stage of the structural development of the product [4]. It also can achieve more realistic representation of disc brakes, including non-linearity and elasticity of disc brake's components. The previously listed great advantages suggest to the high promising future of finite element method with respect to the other methods. However, much research remains to be done to make the method reliable in predicting the occurrence of squeal. During development of the disc brake's model using the finite element method, it is important to validate it, in order to get the model that correctly represents the actual structure in terms of geometry and material characteristics. Validated model should be able to sufficiently accurately predict the occurrence of squeal [5, 6].

There are generally two major categories among simulation and analysis methods in the prediction of squealing brakes: the complex eigenvalues analysis in the frequency domain and the dynamic transient analysis in the time domain. Both analyzes have their advantages and disadvantages. The complex eigenvalue analysis can reveal which system modes of vibration are unstable but a shortcoming of this technique is that they do not allow time-dependent material properties and could not take into account full effect of nonlinearity away from steady sliding [7]. Meanwhile, divergence of a transient solution indicates that instability is present in the system and this technique could overcome the shortcomings in complex eigenvalue analysis. But the drawback of such technique is its long computing time and slow turnaround time for design iterations. A comparison between the two analyses is also made.

Dynamic analysis of transient processes (sometimes called the time-history analysis) is a technique used to determine the dynamic response of the structure under the effect of any time-dependent load. This type of analysis can be used to determine the time-varying displacements, deformations, stresses, and forces in the structure, because it is suitable for any combination of static, transient and harmonic loads. The load range during time is such that the effects of inertia and damping are considered important. If the effects of inertia and damping are not important, you may be able to use static analysis.

In recent years, the complex eigenvalue becomes the most preferred method in the brake research community to study brake squeal than the transient analysis. The positive real parts of the complex eigenvalue indicate the degree of instability of the linear model of a disc brake and are thought to show the likelihood of squeal occurrence or the noise intensity [8]. On the other hand, instability in the disc brake can be associated with an initially divergent vibration response using transient analysis. Liles [8] was the early researcher who incorporated complex eigenvalue analysis with the finite element method whilst Nagy et al. [9] pioneered dynamic transient analysis with the finite element method. Complex eigenvalue analysis allows all unstable frequencies to be found in one run for one set of operating conditions and hence is very efficient. However, not all unstable frequencies thus obtained can be observed in experiments. Transient analysis is able to predict true unstable frequencies (those found in experiments) in principle if the system model is correct. However it is very time-consuming. Moreover it does not provide any information on unstable modes.

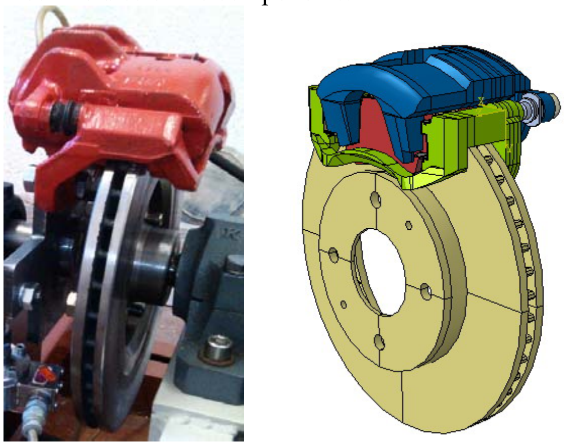
It can be seen from the previous works [8,10] that the complex eigenvalue analysis required using a number of linear spring elements at the friction interface disc/pad in order to create the friction connected members (asymmetric stiffness matrix), which leads to complex eigenvalues, or unstable behavior where positive real parts indicate the likely occurrence of squeal. Fortunately, with the contribution of some researchers [11,12] and the initiative of a finite element software companies [13,9], linear spring elements are no longer required as friction coupling terms can now directly implemented into the stiffness matrix. As a result, the effect of non-uniform contact pressure and the influence of residual stresses can be included in the complex eigenvalue analysis [13]. Another advantage of this approach is that the surfaces in contact do not need to have the matching meshes, and in fact it can reduce data preparation time. Some former used approaches required nodes on two contacting surfaces to coincide and similar meshes. In some previous studies the authors have assumed full contact at the pads and disc interface [8, 7]. However, previous works related to the break contact pressure analysis [15, 16] has shown that the contact pressure distributions at the disc/pads interface are not uniform and that there exists partial contact over the disc surface.

In the past, simulation of disc brake squeal using the complex eigenvalue analysis, together with the finite element method was time consuming compared to the normal mode analysis. It is already known that the contact geometry between disc and friction material interface has a significant contribution to squeal generation [10, 17]. These researchers believed that squealing can generate at particular conditions of pads topography. This is true, because the material properties of friction materials are much lower compared to the disc as a result the friction materials are more prone to wear. Furthermore, the friction material has a much irregular/corrugated surface compare to disc. From the literature review, it was determined that none of the finite element models considered the friction material surface's topography. All the models assumed that the friction material had the smooth and flat interface, while, in reality, it is a rough surface. As previously mentioned, most of the FEM models are validated only at the component level or a combination of the components and assembly levels. In the literature, it was shown that the complex eigenvalue analysis was the most common method and most adopted by the industry to study their problems with squealing noise. This method depends largely on the results of contact analysis, which can determine the instability in the disc brake assembly. Determination of the dynamic contact pressure through the experimental methods still remains impossible. However, there are methods to obtain the static contact pressure, when the disc is stationary. The reference [14] shows that the static contact pressure distribution and its magnitude can be used as a validation tool where the correlation between the calculated and the measured results can be established. Therefore, this level of validation can enhance one's confidence in the developed model, as well as to provide better prediction of squeal.

It must be understood that the disc/pads contact is not complete. There are gaps in the contact interface, and the contact area varies during brake's vibration. There are several methods for modeling the contact in the literature, such as the gap element, the spring element, etc. The surface element for disc/pad interface was used for the contact model, while the spring element is used to represent the contact interaction between the other components of the disc brakes that are in contact. The real contact surface of the friction material can be used instead of the assumed ideal contact surface. This can lead to new insights how to get better predictable results. Due to wear, the contact between the disc and the pads can be changed over time. Perhaps this may explain the elusive nature of the squeal phenomena [14]. Another important aspect is the friction model. It is believed that the friction is the primary cause of squeal. The basic Coulomb friction model is used in this paper. It can be assumed that the friction coefficient is constant or depends on the speed. Previous studies of squeal occurrence were based on the hypothesis that the negative slope of μ-v function greatly contributes to the occurrence of squeal. However, this hypothesis was subsequently replaced by other mechanisms called sprag-slip and modal joining that did not require this friction characteristic. Instead, it is also shown that the constant coefficient of friction generates a squealing. The effects of both friction characteristics on the squeal occurrence are simulated in this study. In addition to these most important characteristics, the impact of heat on the contact pressure distribution and the occurrence of squeal can also be an important influential parameter [14, 18]. Research of the complex combined effects of thermal expansion and the contact loads between pads and disc at a moment when they are exposed to temperature changes during the braking process is presented in [18].

3. MODAL ANALYSIS OF FEM MODEL OF DISC BRAKES ASSEMBLY

A detailed three-dimensional finite element model (FEM) of disc brake assembly is developed. Figure 2a) and 2b) show a real disc brake assembly with floating caliper, and its FE model. The FE model consists of a disc, a piston, a caliper, a mounting bracket, interior and exterior pads, two bolts and two guide pins. The rubber seal (attached to the piston), and the two rubber washers (attached to the guide pins) are not included in the FE model. Damping shims are also not present in the model since they have been removed in the squeal experiments. The FE model uses 35169 solid finite elements and approximately 37,100 degrees of freedom (DOFs). This figure excludes the spring elements that have been used to connect the disc brake components.

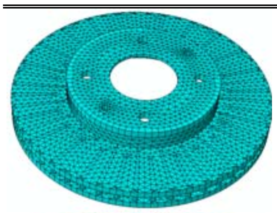
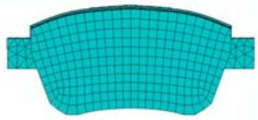


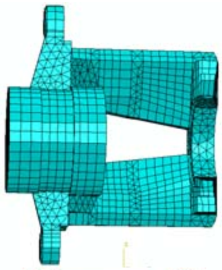
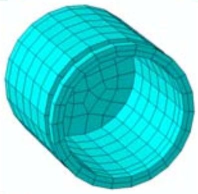
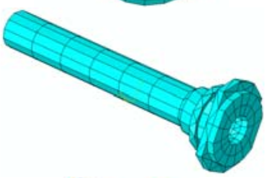
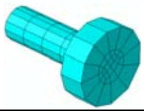
a) b)

Figure 2: Disc brake assembly a) the real disc brake b) FE model [19]

The disc, brake pads, piston, guide pins and bolts are developed using a combination of 8-node (C3D8) and 6-node (C3D6) linear solid elements, while the other components developed using a combination of 8-node (C3D8), 6-node (C3D6) and 4-node (C3D4) linear solid elements. Details for each of the components are given in Table 1. Since the contact between the disc and friction material surface is crucial, realistic representation of these interfaces should be made. Friction material has a rougher surface and is softer in terms of properties than the disc, which has quite smoother and flat surface, and is less prone to wear.

Table 1: Finite element models of disc brake's components

Disc brake's components		***Type of finite element***	***Number of elements***	***Number of nodes***
	Disc	C3D8 C3D6	8177	7866
	Friction Material The back plate	C3D8 C3D4	2735	1804

	Caliper	C3D8 C3D6 C3D4	8138	5135
	Mounting Bracket	C3D8 C3D6 C3D4	7204	3803
	Piston	C3D8 C3D6	383	678
	Guide Pin	C3D8 C3D4	802	662
	Bolt	C3D8	112	186

3.1. Components Interfaces

Many different methods can be used in the FEA modeling of a contact between the components. These methods are (in order of simplest to most complex):

- Merged nodes
- Multi-point constraints
- Linear spring elements
- Contact elements.

Merged nodes are shared between neighboring elements so that the components are effectively connected together. Although this is simple, it does not allow the application of any type of interfacial property such as contact stiffness or damping. Contact surfaces are by far the most advanced methods of coupling components together. There are sophisticated contact surface models that allow some level of motion between the components, which include the specifying normal and tangential contact stiffness. Here a more realistic representation of what is a highly non-linear feature can be applied. Unfortunately, it requires a considerable computational process compared to the other three methods.

Upon completion of the modeling, all the disc brake's components must be integrated into the assembly model. Contact interaction between the disc brake components is represented by the linear spring elements (SPRING 2 in *ABAQUS* nomenclature), with the exception of the disc/pad interface where surface-to-surface contact are introduced (see Table 2). This selection was made due to the fact that the contact pressure distributions at the disc/pads interface are more significant than other component contact interface. This type of spring element has three degrees of freedom in the translational direction and the relative displacement across the spring element is the difference of the *i*-th component at the spring's first node and *j*-th component of the spring's second node:

$$\Delta u = u_i^1 - u_j^2 , \qquad (1)$$

where i and j are the degrees of freedom in the translational direction. This spring element allows the users to specify different spring stiffness for different directions.

Figure 3 shows a schematic diagram of the contact interaction that has been used in a model of the disc brake assembly. A rigid boundary condition is imposed at the bolt holes of the disc and of the mounting bracket, where all six degrees of freedom are rigidly constrained.

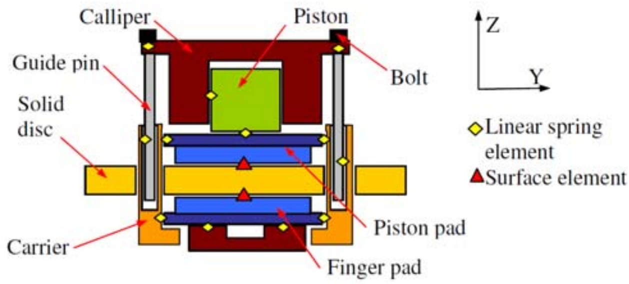


Figure 3: Schematic diagram of contact interaction in a disc brake assembly [14]

Some of the components need to be fixed-should be tied, for example, bolt and caliper.

Table 2: The contact interaction between components

Components	Interaction
Disc-Brake Pads	*Surface-to-surface*
Caliper- Brake pads	*Node-to-surface*
Piston-Brake pads	*Node-to-surface*
Clip-Brake pads	*Node-to-surface*
Guide Pin-Mounting Bracket	*Node-to-surface*
Piston- Caliper	*Node-to-surface*

The model used for analysis of this braking system has the three-dimensional elements that are used from the library of elements:

- C3D6 - first order 3D continuous wedge element with 6 nodes,
- C3D8 - first order 3D continuous hexahedral element with 8 nodes,
- C3D4 - first order 3D continuous tetrahedral elements with 4 nodes.

Contact surface elements are used in areas where the contact is occurred.

It is not necessary to apply the final sliding of contact pairs in any contact location within the model of the brake system. Most of these locations show negligible relative slip when the brake is loaded. The only exception is the disc/pads interface, where obviously the rotation of the disc leads to the high sliding speed in a physical brake system. However, in the contact analysis used in the *ABAQUS*, it is simply a case of defining the boundary conditions of the disc's speed. Table 3 shows an overview of the parameters used for each contact interface within the brake assembly.

Table 3: Contact interface in the ABAQUS model of disc brake assembly

Interface	Type	The initial clearance, mm	μ
Disc / Friction lining	Small sliding	.005	0.336
Inner plate/Piston	Small sliding	.005	0.12
The outer plate/Housing	Small sliding	.005	0.12
Plate/Mounting Bracket	Small sliding	.005	0.12
Piston/Caliper Housing	Small sliding	.001	0.05
Guide pin/Mounting Bracket	Small sliding	.001	0.05
Guide pin/Caliper Housing	Tied	.01	-

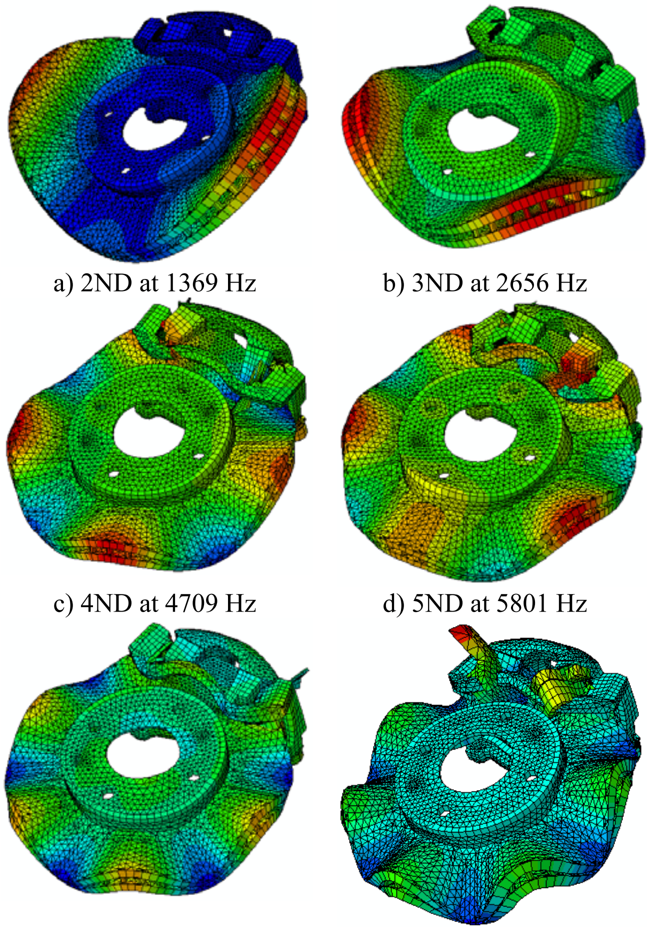
The chosen values of initial contact clearance are 0.005 mm for all the pad's surfaces, because they all should have surfaces that lie on each other at the beginning of the analysis, and 0.005 mm represents the geometric resolution of the geometry. Clearance between pad and caliper's surfaces is 0.001 mm because these surfaces are not designed to be in the initial contact and are modeled with a finite clearance. The value of 0.001 mm ensures there is no adjustment of nodes on these contact surfaces prior to analysis. The value of clearance between caliper housing and guide pin is 0.01 mm, and this ensures that all contact surfaces are completely adjusted and connected even if some nodes are separated even 0.005 mm for tolerance modeling.

Static analysis establishes the basic state of the system with typical load of the brake, then perform complex eigenvalue analysis to determine the stability of the system around this basic state:

1. Static preload, nonlinear static analysis. The pressure is applied to the back of the piston and inside the cylinder in the caliper housing. No rotation is applied to the rotor for this step and the system reflects a stationary brake with pressure applied. This allows non-linear solver to more easily determine the contact conditions at the disc/pad, guide pin and piston interfaces without the complication introduced by rotation. Stabilization of solutions, which involves applying artificial damping to control rigid body motions, is applied to the bodies that are not constrained prior to contact being established. The damping is small enough not to affect the final static solution when all the contact conditions have been properly established.
2. Adding rotation, non-linear static analysis. Velocity boundary condition was added to the disc from the static loaded state from step 1. The pads react to frictional forces at the disc/pad interface and begin to translate until they are fully captured by the pad abutment regions on the mounting bracket. The system converged into its basic state during a brake application. This provided the basic state for the analysis steps that followed.
3. Normal modes. The normal modes solution provides a subspace of modes to be used for complex eigenvalue solution in the step 4. Number of modes extracted was 107 and covers a frequency range from 1 to 10 kHz. The number of modes in this step needed to be greater than the number of complex modes requested for step 4 adequately to allow the complex modes to be represented.
4. Complex modes. Complex eigenvalue solution to provide the stability response of the base statically loaded state. Complex modes were extracted taking into account the effect of friction interface [5, 6].

4. RESULTS

The second phase of the methodology is to determine the dynamic characteristics of the disc brake assembly's model. The previous separated components of the disc brake must be now coupled together to form the assembly model. Modal analysis was carried out to obtain the natural frequency of the assembly (Table 4). The resulting out-of-plane modes of disc brake assembly are shown in Figure 4.



a) 2ND at 1369 Hz b) 3ND at 2656 Hz
c) 4ND at 4709 Hz d) 5ND at 5801 Hz
e) 6ND at 7782 Hz f) 7ND at 9055 Hz

Figure 4: Mods of disc brake assembly [19]

Table 4: Modes of disc brake assembly in a free-free boundary conditions

Mode Nº	Frequency, Hz	Mode Nº	Frequency, Hz
2ND	1369	5ND	5801
3ND	2656	6ND	7782
4ND	4709	7ND	9055

4.1. Nonlinear contact analysis

ABAQUS defines the contact pressure between the surfaces at a point, p, as a function of the over-closure, h, of the surfaces. A hard contact model is considered where the disc and pad surfaces will separate (or contact constraint is removed), when the contact pressure between them becomes zero or negative and on the other hand, the disc and pad surface will interact (or contact constraint is applied) when the contact pressure between them is larger than zero. Two regimes for *p=f(h)* are given in the formulations below [13]:

$$\begin{cases} p = 0 \text{ за } h < 0\,(open) \\ h = 0 \text{ за } p > 0\,(closed) \end{cases} \quad (2)$$

When surfaces are in contact, they usually generate shear (friction) and the normal forces across the sliding interface. A relation between these two components of force is described in terms of friction between the bodies in contact. Typically, when deriving friction in a theoretical context, the critical value of the tangential force is defined as:

$$F_{crit} = \mu \cdot F_N\,, \quad (3)$$

where F_{crit} is the critical shear force, μ is the friction coefficient, and F_N is the normal force. Due to the discretization process used by the finite element method, the critical value is not defined in terms of a critical load (F_{crit}), but as a critical shear stress (τ_{crit}) that is a function of the pressure (p), as given below:

$$\tau_{crit} = \mu \cdot p\,. \quad (4)$$

The value of the shear stress that compares with the critical value, defined above, is the magnitude of the resultant shear stress in the x and y-directions:

$$\tau_{eq} = \sqrt{\tau_x^2 + \tau_y^2}\,. \quad (5)$$

If the value of the equivalent shear stress is greater than value of the critical shear stress, sliding contact will be initiated, and the restoring shear stress will be equivalent to τ_{crit}. In the case of sticking condition, the shear stress will balance that applies to the contact interface.

ABAQUS provides various friction models to describe the relative tangential motion of the contact surfaces. A basic Coulomb friction model is used, where, by default, friction coefficient can be defined as a function of sliding speed, contact pressure and average temperature at the contact point. The users can also define different coefficients of friction, ie. static friction and kinetic friction coefficients (Figure 5). In this model, it is assumed that the friction coefficient exponentially decreasing from the static value to kinetic value based on the following equation:

$$\mu = \mu_k + (\mu_s - \mu_k)e^{-d_c v}\,, \quad (6)$$

where μ_k is the kinetic friction coefficient, μ_s is the static coefficient, d_c is a decay coefficient and v is the sliding speed. During the specifying static and kinetic friction coefficient, *ABAQUS* allows the users to change the friction coefficient during the analysis. This is adopted in the entire study where the static friction coefficient is used during the first step, and the kinetic friction coefficient in the following steps. *ABAQUS* also provides an anisotropic friction model that allows the users to specify different friction coefficients in the two orthogonal directions on the contact surface. The users can also develop their own friction model using user-defined subroutine [13].

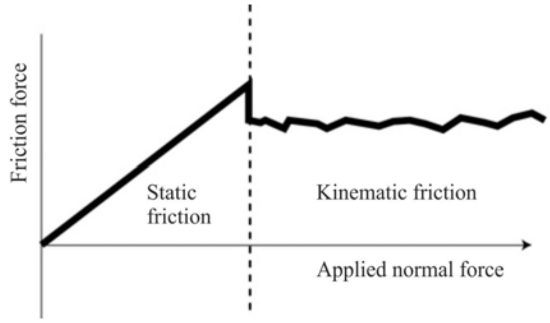
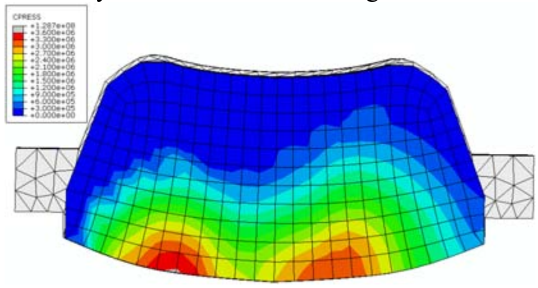


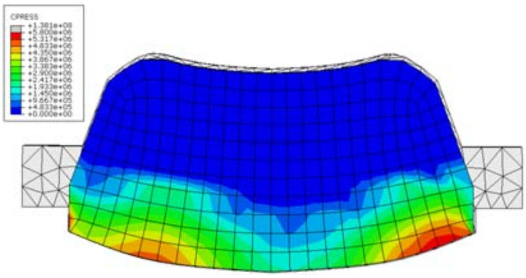
Figure 5: The relationship between static and kinetic friction coefficient

There are three types of contact schemes available in *ABAQUS* namely, small, finite and infinitesimal sliding. By default, *ABAQUS* treats finite sliding by which contact surfaces may allow for arbitrarily separation, sliding and rotation. Using finite sliding, the slave nodes may come in contact anywhere along the master surface and the load transfers are updated throughout the analysis. Whilst for small sliding the contact formulation assumes that the contact surfaces may undergo arbitrarily large rotations, but that a slave node will interact with the same local area of the master surface during analysis. Therefore the slave nodes are not monitored that in contact along the entire master surface. With the final and small sliding consider geometric nonlinearity, infinitesimal sliding ignores this effect and assume both relative motions and the absolute motions of the contacting bodies are small. Accordingly, infinitesimal sliding is unsuitable for the disc brake analysis.

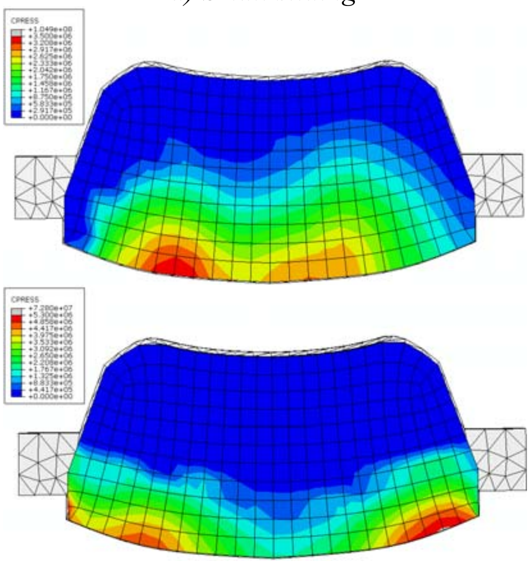
Further, comparison between the two sliding schemes is made in terms of the contact pressure distribution, the contact area and simulation time. The previously developed model will be used in analysis. The experimental data were used for the maximum braking performance regime, and the corresponding maximum friction coefficient. The pressure in braking installation of 1.84 MPa and the rotation speed of 44.52 rad/s were introduced in the model. For the contact interface between the pads and the disc, the kinetic coefficient of friction of μ=0.336 is applied. A penalty friction constraint is chosen for comparison. The obtained results will suggest which sliding scheme should be adopted throughout this research.

From figures 6a) and 6b), it can be seen that contact pressure distributions are almost the same for both the piston and finger brake pads. Maximum contact pressures are also nearly identical for both sliding schemes.





a) Small sliding



b) Final sliding

Figure 6: Contact pressure distribution between small (a)and the finite sliding (2)schemes at the piston (left) and finger (right) pads.Left of the diagrams is the leading edge

Comparisons between small and finite sliding in terms of the contact area, maximum pressure and simulation time are described in Table 5. As previously mentioned, the finite sliding scheme is more demanding in terms of computation time that the small sliding scheme. This is proved to be true as indicated in Table 5, in which the finite sliding takes about 2893 s to complete the simulation, while small sliding only takes about 2000 s, which is a reduction of 30.87%. It appears that the two schemes have little difference in the contact analysis in particular for the disc brake contact analysis. Based on the results, small sliding scheme will be adopted for subsequent analysis due to its computational advantages over the finite sliding, while a similar contact pressure distribution, contact area and maximum pressure can be obtained for both schemes. Furthermore, using finite sliding should be paid more attention in smoothing the master contact surfaces and nothing need to be done for a small sliding.

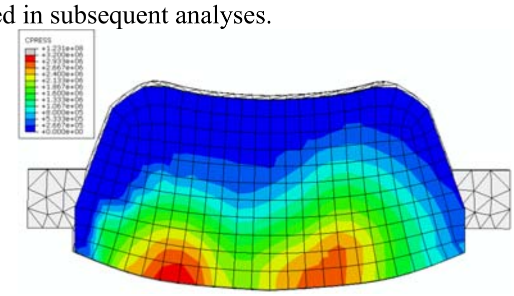
Table 5: Comparison between the small and finite sliding

Parameter	**Small slip**		**Final slip**	
	Piston	**Finger**	**Piston**	**Finger**
Contact area, m²	$2.209 \cdot 10^{-3}$	$1.475 \cdot 10^{-3}$	$2.565 \cdot 10^{-3}$	$1.459 \cdot 10^{-3}$
The highest contact pressure, MPa	3.6	5.8	3.5	5.3
Time simulation, s	2000		2893	

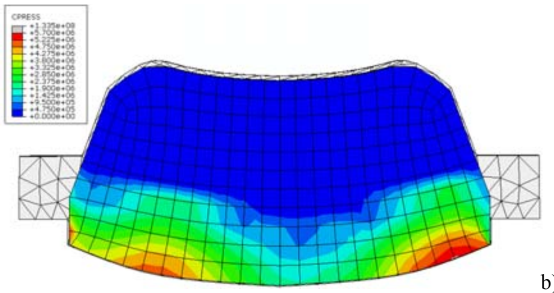
In this study, it is assumed that the pads will come in interaction with the same profile of the rotating disc surface. Therefore, small sliding scheme is chosen. The convergence could also be easily obtained, compared with the finite and infinitesimal sliding formulation. Furthermore, small sliding scheme provides considerably computation time savings in comparison with the finite sliding model.

There are two stiffness methods for friction constraints that are available in *ABAQUS*, namely, a penalty method and the Lagrange multipliers method. The penalty method (default by *ABAQUS*) permits some relative motion of the surfaces when the surfaces should be sticking whilst the Lagrange method should be used when no slip is allowed in sticking condition. Using the Lagrange method can increase the computational cost of the analysis because it adds more degrees of freedom to the model and quite often increases the number of iterations needed to obtain a converged solution. In addition, the Lagrange formulation may prevent convergence of a solution. In the case of the finite sliding, the considered model of disc brake, there was no convergence of solution. Therefore, in this study the penalty method is employed to ease convergence restriction, as well as to obtain minimum computational cost [13].

The results obtained using the method of Lagrange multipliers are presented in Table 6. The results obtained using the small sliding scheme are used. By looking at Table 5 and 6, particularly with respect to the contact area and the maximum pressure can be seen that there are no differences between the two schemes. Similarly, the contact pressure distributions as shown in Figures 7a) and 7b) in both schemes are identical. However, in terms of the computational cost, Lagrange multipliers scheme requires more time for completing the simulation compared to the penalty scheme. Lagrange multipliers scheme requires about 2819 s for a single analysis, whereas the penalty method only takes about 2000 s, which is an increase of 29% in the computational time. Results indicate that for disc brake squeal problem exact sticking condition is not necessary. It has been observed that one of the main characteristic of squeal is that no obvious sticking state is present at the disc/pad contact interface. Even though one can argues that this (no apparent sticking) may be applied at the macroscopic level, but not in the microscopic state. Since this paper only considers squeal occurrence at the macroscopic level, any conditions or behavior that is present at the microscopic level is not considered. Therefore, it is considered that the penalty scheme is most suitable for this study due to its advantages in terms of computational cost over Lagrange multipliers and will be used in subsequent analyses.



a)



b)

Figure 7: Contact pressure distribution using Lagrange multipliers formulation at the piston (up) and finger (down) pads. The left side of the diagram is the leading edge [19]

Table 6: Simulation results of contact analysis

Parameter	Lagrange Multipliers	
	Piston	Finger
Contact area, m^2	$2.3 \cdot 10^{-3}$	$1.457 \cdot 10^{-3}$
The highest contact pressure, MPa	3.2	5.7
Time simulation, s	2819	

5. CONCLUSION

This paper describes the development and validation of the FE model of disc brake. The proposed methodology has two stages as follow:

- Validation of the disc brake mechanism's components using modal analysis,
- Validation of the disc brake assembly using modal analysis.

Using modal analysis has shown that a good agreement is reached at the component's level and brake disc assembly's level. This can only be achieved after the adjusting process or an update in which the material characteristics' values of the components and the spring stiffness is adjusted at each level. It was also established that there are a number of natural frequencies of the brake components are close to each other.

Previous studies using the FE method assume a perfectly flat surface on the disc/pad interface. Improved FE model should include the actual topography of a pad's friction material surface, which can be measured by using the linear micrometers. It is also shown that current mesh of individual FE model, particularly of the brake pads, are sufficiently dense to give a realistic prediction of the contact pressure distribution, and also to capture mode shapes of natural frequencies up to 9 kHz. However, the current predicted results can be improved by using a better mesh quality. Due to the accurate representation of components and brake assembly, the later simulation can achieve much better prediction.

Next part of paper was focused on the non-linear contact analysis of the disc brake model with the main objective of determining of contact pressure distribution, the contact area and maximum contact pressure. These three parameters are useful for subsequent work, especially in comparing predicted results from one model to another. Several potential contact interaction schemes that are available in *ABAQUS* were described. The first comparison is made between small sliding and finite sliding schemes. It was found that, although the small sliding scheme assumes the slave node could slide relatively a small amount at the master surface compared to the finite sliding scheme, predicted results between the two schemes are almost identical. The main advantage of the small sliding scheme over the finite sliding scheme is that it saves about 30.87% of computational time.

The second comparison is made to examine two friction stiffness constraints, namely the penalty method and Lagrange multipliers method. The penalty method allows some relative motion of the surfaces during sticking, while Lagrange multipliers do not allow at all the relative motion during sticking state. In addition, Lagrange multipliers can enforce more precisely the sticking and sliding constraints than the penalty method. However, computational cost is an issue as Lagrange method takes more time for a single analysis. This is proved to be true because Lagrange method requires 2819 s compared to 2000 s using the penalty method, which is an increase of 29% in computational time. On the other side, the predicted results for both methods are identical. By looking at those results, the penalty method is more suitable and will be used together with small sliding scheme for further research.

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