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experimental work, the main benefits of the FE methods are that they are much faster and create cost efficient solutions.
The advantages of studying the break squeal phenomenon is provide the solutions for the designers and manufacturing to reducing noises also to customer comfort, therefore spending money about the research as result saving money.

Future Work

It is recommended that for future work more disc brake kinds are studies and analyzed to better expect the probability that a brake noise may happen when the brake is applied. Different other methods could be used to better know the happening of noise in the brakes. In this plan, only high frequency brake squeal was studies. This work could be protracted to study the low frequency noises such as groan, moan etc.

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Table (9) Modal pair's frequency

Figure (b) above shown the influence of the variations can be clearly seen from measured and simulated frequencies on system and component levels

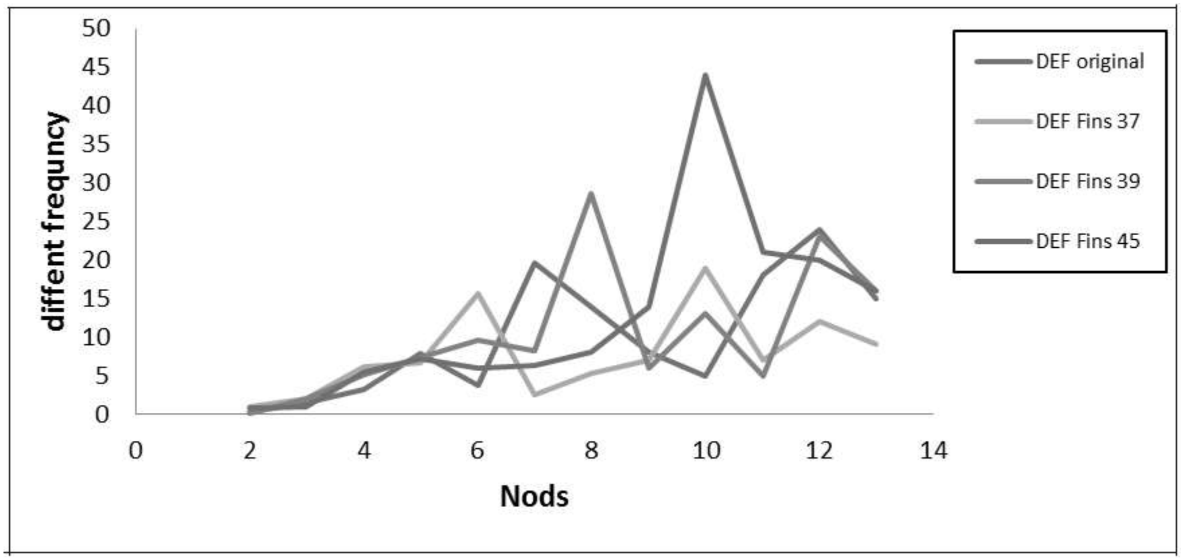


Figure (19) different frequency against nodes model

Figure (19) shows the variation of the number of fins. The result shows that the change of the fins observed that the squeal occurred at frequencies of 5119.9 Hz at the number fins 37. Also the result was showed significant impact of change at the number fins 45.

Conclusion

Brake system is one of the most significant parts of the vehicle. Improper brake system could lead to loss of life and property. Noise from the brake system makes irritation and is annoying for the customers and passengers. It is therefore very important that noise free brake systems are designed and used in the vehicles.
Prediction of disc brake squeal using finite element (FE) systems mainly through complex eigenvalue analysis has been a common practice in the brake research community. As opposed to the

result the squeal noise will be decreased because the number of fine is dropped.

Number of fins	Young's modulus (Pa)	Poisson's ratio	Density (kgm-3)
FE original 42	96725 E^6	0.25	7250
37	96725 E^6	0.25	7250
39	96725 E^6	0.25	7250

Table (8) disc rotor properties

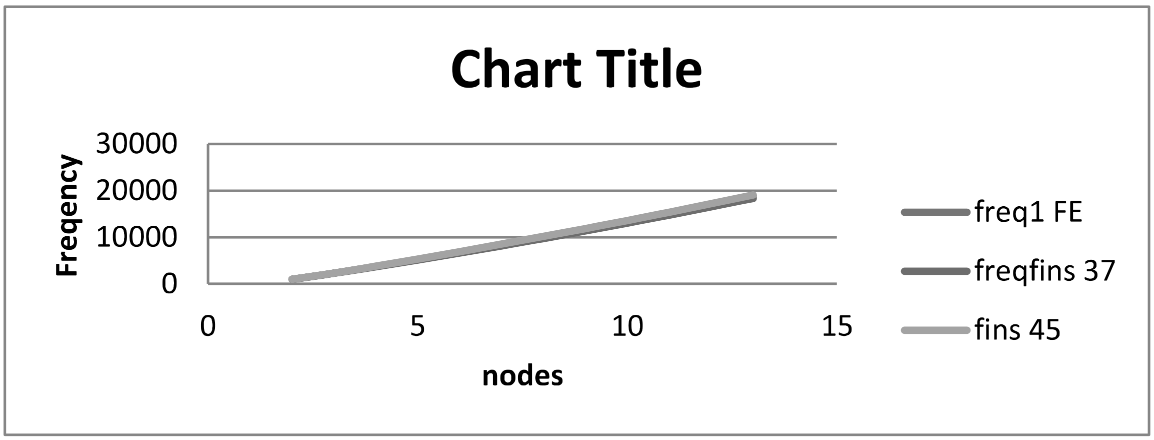


Figure (18) fins number impact

Node model	**Deferent frequency (original) 42**	**Deferent frequency 37**	**Deferent frequency 45**
2	**0.27**	**0.92**	**0.86**
3	**1.4**	**2**	**0.9**
4	**3.2**	**6.2**	**5.5**
5	**7.8**	**6.7**	**7.1**
6	**7.7**	**15.6**	**6**
7	**19.6**	**2.5**	**6.4**
8	**14**	**5.2**	**8**
9	**8**	**7**	**14**
10	**5**	**19**	**44**
11	**18**	**7**	**21**
12	**24**	**12**	**20**
13	**15**	**9**	**16**

Node model	Deferent frequency (original)	Deferent frequency - 10%	Deferent frequency +10
2	0.27	0.6	0.55
3	1.4	2.2	2
4	3.2	5.7	5.2
5	7.8	7.8	7
6	3.7	5.7	7.9
7	19.6	27.5	24.8
8	14	18	16.2
9	8	7	7
10	5	39	36
11	18	20	18
12	24	42	38
13	15	41	37

Table (7) different frequency, change properties

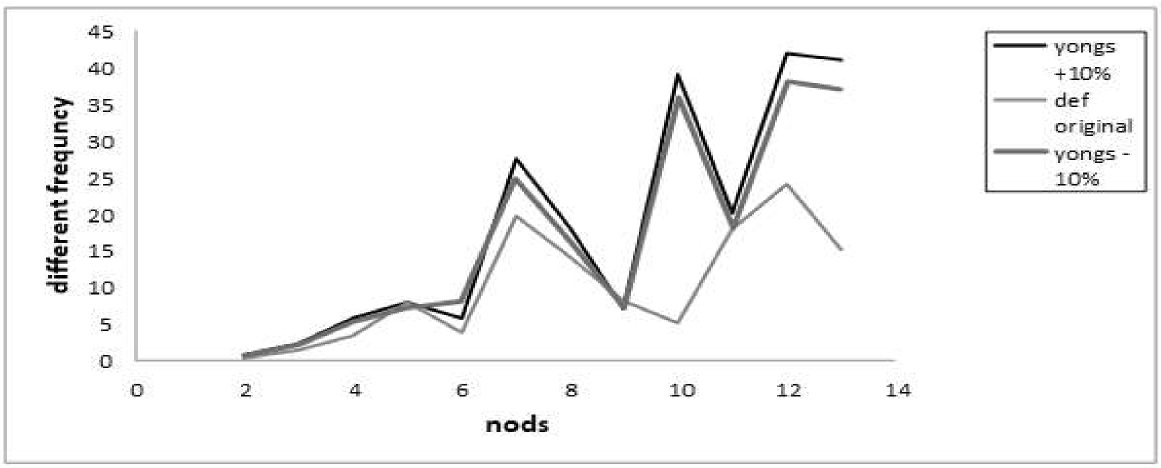


Figure (17) different frequency against nodes model

Effected of Change fins number

Many of the experimental has been done to study the effect of change the geometry shape within change number of fins .however it found there the slight impact in the 37-45 as the figure (18) presented. From the beginning at node 2 the frequency 975 HZ this grows gradually until frequency at node 8 to9734.8 HZ. However the fins 45 is slightly increased and fins 37 slightly decreased. As a increase as well. Also form node 2 to node4 there no change, after that there is slight gap between the original and 10% of young's modulus until node 8, then there significant gap until node13.

Young's modulus (Pa)	-10%	+10%
	8.70525E^{10}	1063975E^{11}
Density (kgm-3)	7250	7250
Poisson's ratio	0.25	0.25

Table (6) proprieties of geometry

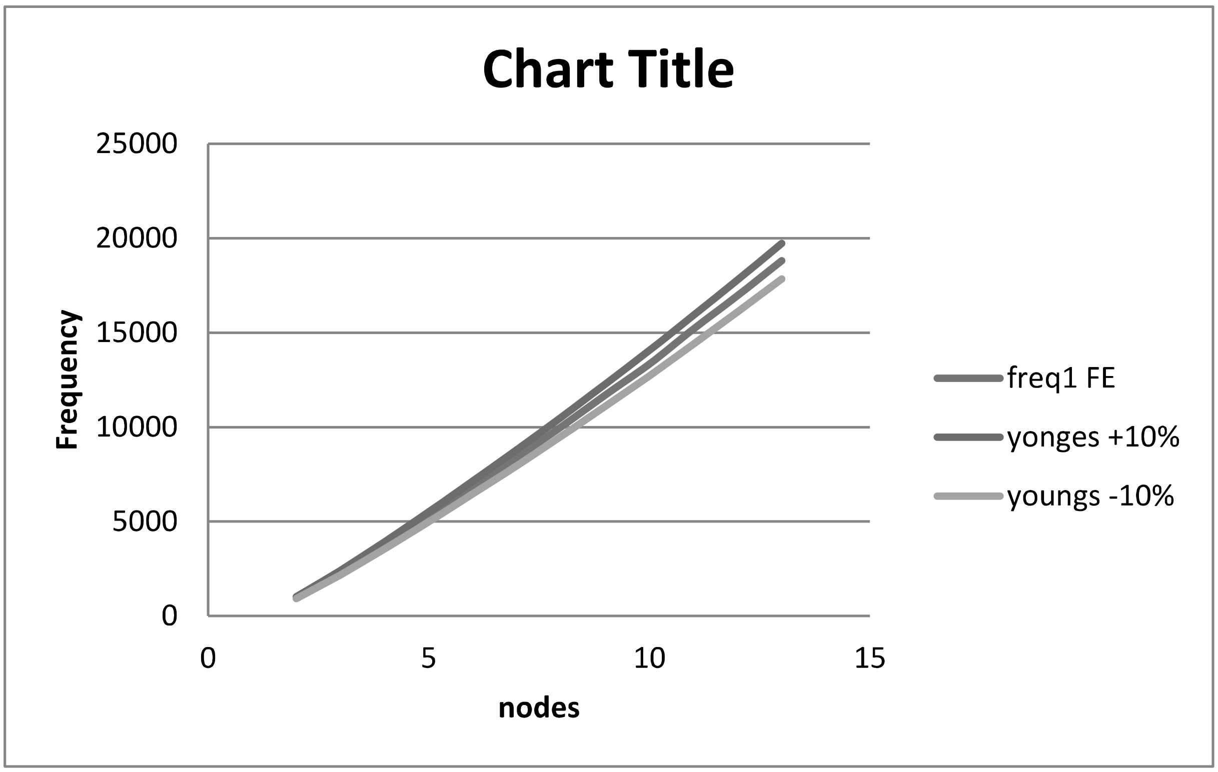


Figure (16) impact of changing young's model

Figure (16) shows the change of the young's mode. The change was increased +10% of young's model and then decreased – 10%. The result was showed that the young's model gives close result to def original which the experimental result.

Figure (65) different frequency against nodes

Table (5) is provides the different frequencies which are obtained from model simulation.

Modal pairs: Different frequency is identifying as the difference between two frequencies in same diameter mode shape due to motion in same node.

Node model	Deferent frequency (original)	Deferent frequency -10%	Deferent frequency +10
2	0.27	0.82	1.1
3	1.4	1.1	1.2
4	3.2	6.2	6.9
5	7.8	1	1.1
6	3.7	6.2	6.8
7	19.6	24.2	26.8
8	14	15	17
9	8	13	15
10	5	31	35
11	18	23	25
12	24	32	36
13	15	32	-

Table (5) different frequency at node model, change density

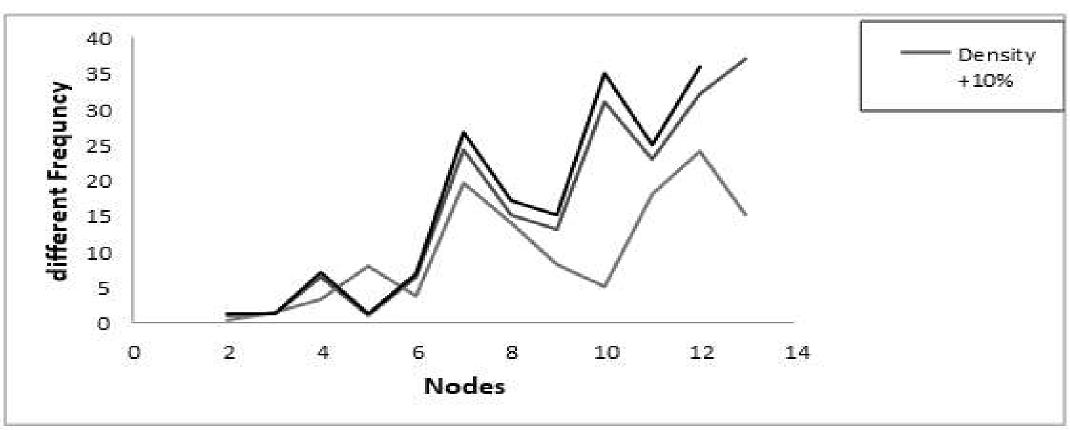
Effected of Change young's modulus

The effect of the stiffness of the disc on the disc brake squeal is studied by changing Young's modulus The figure (16) is providing the comparison of effecting of change the young's modulus between the increase young's modulus + 10% (red line) , decrease young's modulus -10% (green line) and FE results which original shape (blue line), .from graph information we can see that by change young's modulus – 10% the green line is a slight drop as result the frequencies dropped as well and by increase young's modulus +10% the gap between red and blue lines is grow this means the frequency

nodes	Frequency 1	Frequency 2	Deferent
2	1017	1017.6	0.6
3	2407.9	2410.1	2.2
4	3923.5	3923.5	5.7
5	5508.1	5515.9	7.8
6	7139.8	7145.5	5.7
7	8797.7	8825.2	27.5
8	10512	10530	18
9	12270	12277	7
10	14053	14092	39
11	15908	15928	20
12	17789	17831	42
13	19734	19775	41

Table (4) different frequency at node model

Figure (15) shows the change of the density from -10 % to +10%. It was showed the change of the proprieties of disc rotor. In this figure the red colour is the density +10% and black colour is the density when decreased to -10%. The results show that the change of the density was gives the same shape as the def original from nodes 2 to nodes 8 but after that the shape was increased with increase the nods, different frequency is identifying the gap between shape motions for any nodes at simulation Procedure. The squeal is increasing within decreased density, for example at node 5 frequency 13420 HZ is number grow to 14825HZ.hunc the squeal will be dropped.



mass will lead to decrease natural frequency, Huns the vibration will be decrease.
The properties of disc rotor have been changed, table (3) is describe a new properties.

Density (kgm-3)	10%	+10%
	6525	7975
Young's modulus (Pa)	96725E6	96725E6
Poisson's ratio	0.25	0.25

Table (3) specification of disc brake

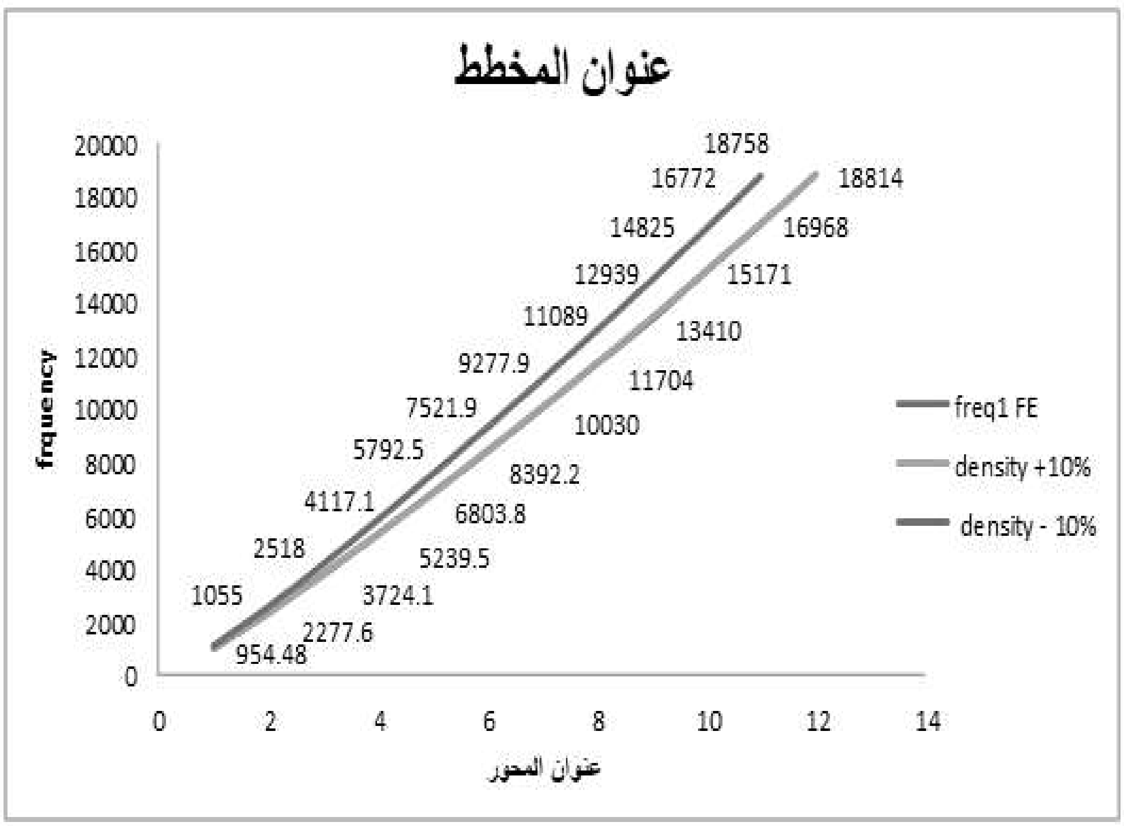


Figure (54) the effect of changing density

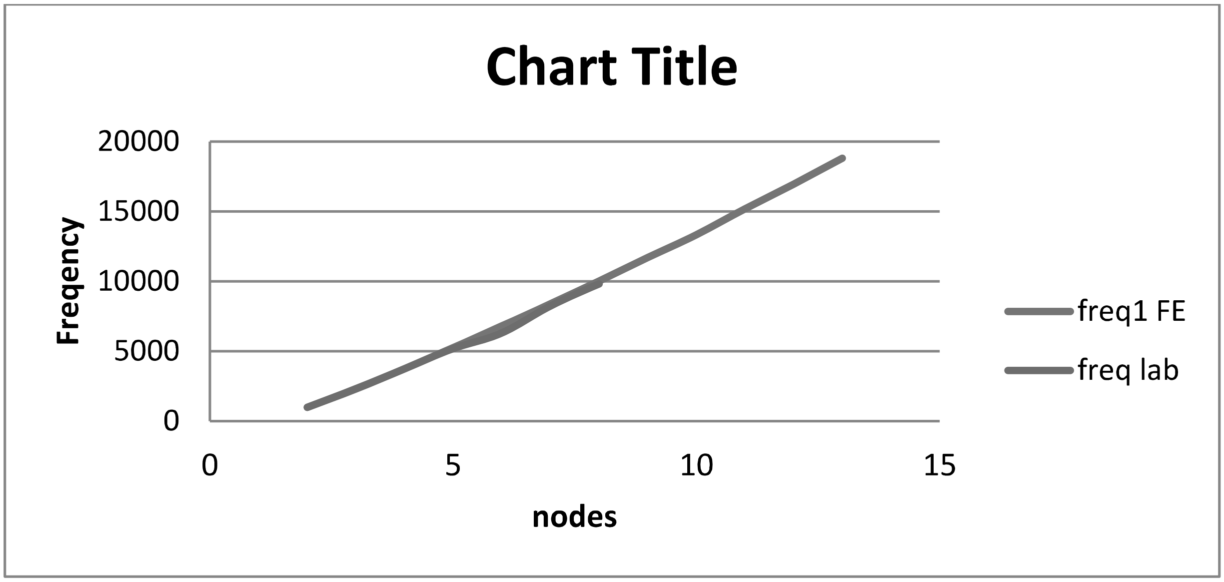


Figure (43) describe the lab results and FE results

Characteristic modifications

Researchers have reported links between the occurrence of squeal and the material properties of the brake components, The modification which has done to study the influence of properties change such as adding ± 10% of density value, number of fins and young's model.

Effected of Change Density

Figure (14) shows the impact of change the density, it was increase +10% to be (7975) and as we see clearly there is no change, but by decrease the density -10%, there were large different between the FE and -10%. As figure shown when the frequency increases the nodes will be increase as will. For example nodes was2 the frequency is 1055 HZ this number is grow to reach 18758 at nodes 13.

From the equation Wn = $\sqrt{\frac{k}{m}}$ Where Wn: is the natural frequency

and K: is stiffness confection of the material, m: is the mass. We know that the density is equal to the mass divided by volume, so the relationship between the mass and natural frequency is increasing the

Table (2) Mode Order

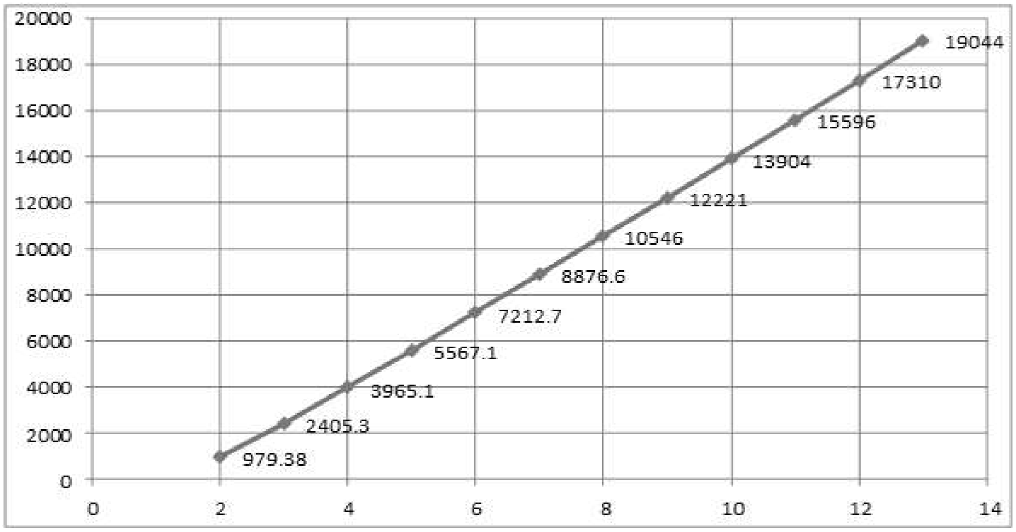


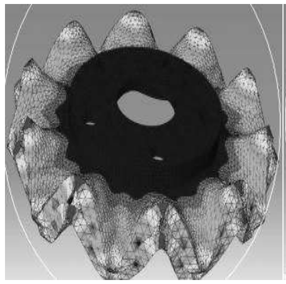
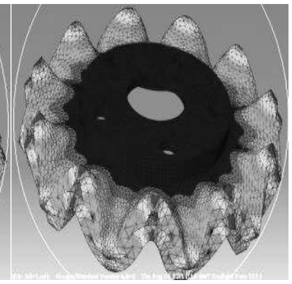
Figure (32) Mode Orders against the Frequency of the Squeal Noise

Figure(12) presented the diameter Mode shapes of the disc compare with frequency, as the number of modes included increases , the frequency is increase from 979.38HZ at mode 2 to 19044 HZ at model 13 , also the number of nodes are grow the squeal propensity increases with an increased value of the frequency.

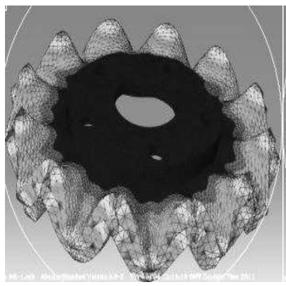
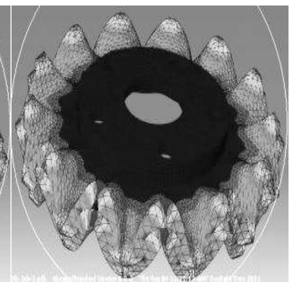
Validation

The detailed designing, development and verification process of a disc brake model throughout this study will be described in this section. Using of the modal analysis will constitute the first stage of the verification process under which comparison of the natural frequencies and its mode shapes will be done with experimental results from the FE model analysis that employed the ABAQUS software.

Figure (13) show the comparison experimental and numerical results.

i) 10 nodal diameter at 13904 Hz j) 11 nodal diameter at 15596 Hz

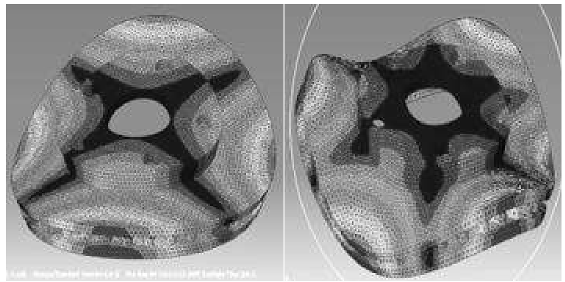
k) 12 nodal diameter at 17310 H l) 13 nodal diameter at 19044 Hz

Figure (21) Mode shapes of the disc at free-free boundary condition

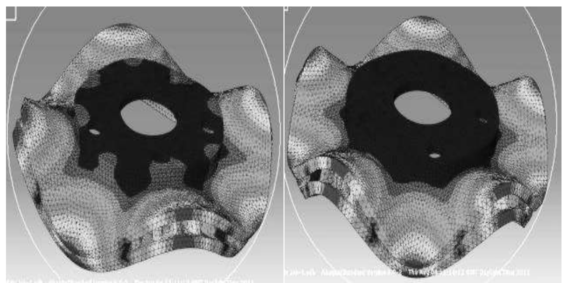
The images above are giving the change of shape thought the change the mode diameter, the effect of increasing the number of modes diameter Unstable (squeal) mode shapes for the baseline model.

Table (2) provide the mode order, frequency, and node pitch in degrees for the car disk rotor for example if the frequency at 5567.1Hz is assumed to be a fifth diameter mode:

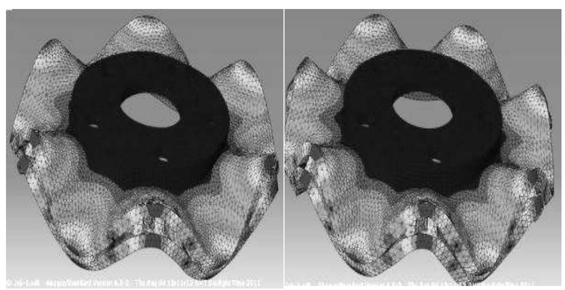
Mode Order	**Node pitch(degrees)**	**Frequency (HZ)**
2	**360**	**979.38**
3	**120**	**2405.3**
4	**90**	**3965.1**
5	**72**	**5567.1**
6	**60**	**7212.7**
7	**51.4**	**8876.6**
8	**45**	**10546**
9	**40**	**12221**
10	**36**	**13904**
11	**32.7**	**15596**
12	**30**	**17310**
13	**27.69**	**19044**



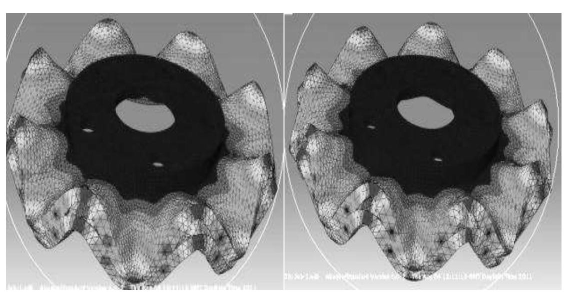
a) 2 nodal diameter at 979.38 Hz b) 3 nodal diameter at 2405.3 Hz



c) 4nodal diameter at 3965.1 Hz d) 5 nodal diameter at 5567.1 Hz



e) 6 nodal diameter at 7212.7Hz f) 7 nodal diameter at 8876.6 Hz



g) 8 nodal diameter at 10546 Hz h) 9 nodal diameter at 12221 Hz

ABAQUAS part model

A number of modes for up to frequencies of 20 kHz have been extracted and captured for the free-free boundary condition of the brake disc rotor. The numerical results describe several mode shapes, but being the dominant ones in the squeal events, only the nodal diameter type mode shapes have been taken into account. Figure (3) shows the mode shapes and the natural frequencies which were calculated, including 2ND up to 13ND (nodal diameters). Clearly visible on the rubbing surfaces of the disc are the number of nodes and anti-nodes upon which the number of nodal diameters are based. The anticipated frequencies are not well correlated with the experimental results after using the standard material characteristics for cast iron. Therefore, change of the density and Young's modulus is required to study the effects change of material properties of the modes shape.

Table (1) these results are based on the material properties given in Table

Density (kgm-3)	7250
Young's modulus (Pa)	96725E6
Poisson's ratio	0.25

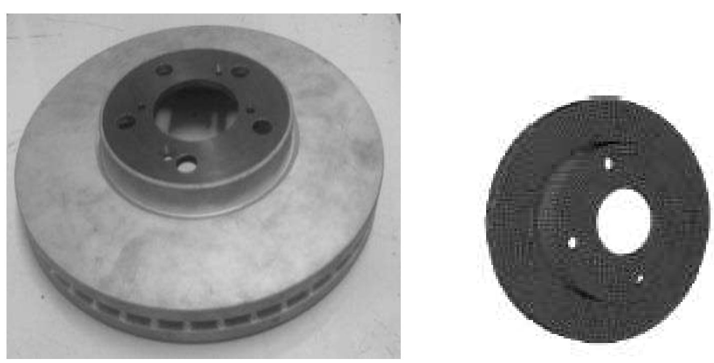


Figure (10) ABAQUS part model &Meshed part

Images below have been provide the change of squeal throughout the period of test.

The figure(9) has explained the different section of the function frequencies responds at the important point of the lab brake assembly system ,grows up to a certain magnitude for the corresponding nonlinear model, as demonstrated by Thompson and Stewart (2002) for a different nonlinear system.

Simulation

In this study three different contact schemes are simulated. This is performed in order to find out which contact scheme provided in ABAQUS can predict similar results to complex eigenvalue analysis and dynamic transient analysis.

CAD model of brake disc

For achieving the best accurate representation of a real disc brake, the Solid works software has been used to model the disc brake rotor, all of shape model details are taken from real disc rotor in brake noise research laboratory.

Finite Element Model of Disc Brake (ABAQUAS)

The next step is to import the model into ABAQUS. To perform a (free-free) modal analysis. This means, to obtain the natural frequencies and mode shapes, assuming that the brake disc is not supported by anything i.e. floating in air. Therefore not be applying any constraints or boundary conditions to the model. Each natural frequency will have a mode shape and this investigation is interested in.

The model shape was creating In Solid works, saved the model as an ACIS .sat file. Then In ABAQUS import the file. Make sure that Topology is set to "solid" and then click OK to import the part. After this steps need to follow this procedure to perform a modal analysis within the model tree. The materials were Create a new material with Young's modulus of 96725x10^6 Pa and Poisson's ratio of 0.25.Next step is create Sections, Create a solid Homogeneous section with the material Section assignments (within the parts branch), thane Assign the section to the brake disc part

Reside table

The reside table is to put all the mechanical system in one table and also to make them arranged for the mechanical system.

Experimental procedures

In the laboratory disc brake determine of the car disc brake system that can be the outer diameter of the 30cm and also the inner of 18.7cm.

1. The brake disc rotor is checked to be ready for tasting.
2. The hammer was connected to the computer trough the Scada III.
3. Make sure that the amplifier is connected with Scada iii and computer together.
4. Ensuring that the sponge is located under the disc rotor that for (Free-Free) condition.

The brake rotor has been knocked repeatedly by an impact hammer, consequently the signal of vibration will be create , so this signal will be grow and thought the amplifier then from scada will be analysis this signal by using software (ABAQUSE)

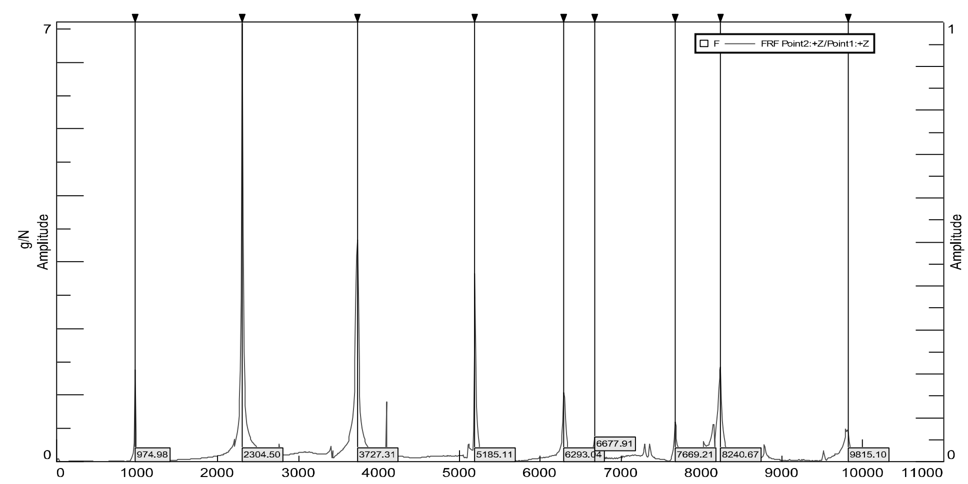


Figure (9) below shows the results that opting from lap test

and also the most efficient way to move the high speed of the different channels, that can count application in a different wide range for the electrical system. The different mechanical and structure testing scenarios of the Scadas III can be the result of the LMS instruments' extensive experience in supplying complete to customized different solutions giving to optimized and performance in the functions. The Scadas III is actually the primary modular and expandable system of the different machines. The system expansion can virtually be unlimited numbers of the different channels possible to be used of one of the Scadas III systems.

Brake rotor

The mean part of this research is disc brake rotor, properties and specification of this rotor flowing retiles below

a. Outer diameter = 30 cm
b. Inner diameter = 18.7 cm
c. Young's model = 96725e6 Pa
d. Density = 7250 kgm-3
e. Number of fins = 42

This details have been taken from real disc rotor in lab of brake noise research, figure (5) shows the image of this rotor

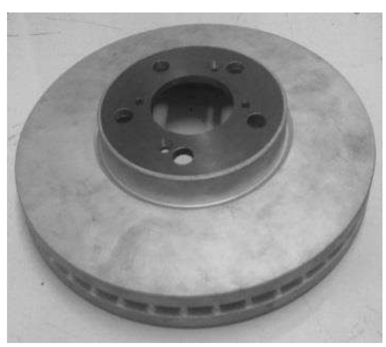


Figure (8) brake disc rotor

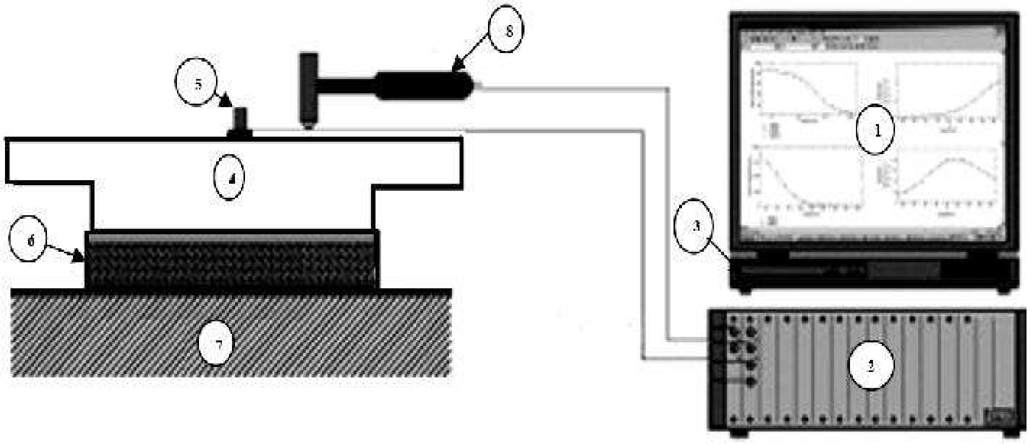


Figure (7) rigs test

SCADA III

For instant, Krutz. (2005). " the Scalds III offers complete different high quality and cost effective solutions for the high speed data acquisition and the different signal conditioning of the machines". In a wide range of the different applications, the Scadas III front and the end is tightly integrated with the LMS testing methods in the Lab with some software and the optimally tuned to reach the specific needs of the different noises and the vibration for the engineering methods. Therefore, it can actually give different flexible methods to choice with some hardware frames and the modules and outstanding performance into the modular system as well. According, to Clarke and Reynders (2004). "the Scadas III is the different state of the art signal conditioning and the data acquisition that can actually offers powerful and dynamic signal processing of the machine. It can also analyses on the large numbers of the machine channels for the convenient and the cost effective of the different operations of the machines systems". The Scadas III function need to be built from versatile signal conditioning with an exceptional high performance digital signal progressing and user with programmable signal generation; which can extremely give high channel count possibilities and the data through of the put capacities. Furthermore, the Scadas III totally have integrated system concept is the fastest

Application of the finite element method in brake noise

For several purposes, researchers have used the finite element method in brake squeal studies. The investigation of modes and natural frequencies of the brake rotor was one of its earlier uses. The computation of the M and K matrices in disc brake models constitutes the most common use. Following on, to ascertain the system's modes, stability and frequencies a linear eigenvalue analysis is carried out. As quoted by the other mentioned analyses, squeal propensity is linked with the lack of linear stability, verified by one or more than one non-dissipating Eigen modes.

Only a handful of the studies focus on the evaluation perspective of some control methods and operational parameters, though in relation to finite element modelling of the squeal phenomenon in disc brake systems the technical literature can be said to be rich with respect to the number of executed investigations.

For predicting the dynamics/ noise, parameters like additional damping, brake temperature, wear and braking pressure should be considered in a numerical model as a must as they can strongly impact the mechanism of noise generation in the brake system.

Description of equipment used in the experiment

Following shows the specification of the test rig system in the laboratory used to test the type of brake car disc. In terms of experimentation the figure (8) have explained the highest level to view the how the disc brake system can set up during the experimentation system

The devices have been used in the lap research that including

- Monitor
- SCADA III
- Amplifier
- Disc brake
- Sensor
- Sponge
- Impact hammer

Methods to Eliminate Brake Squeal

For the purpose of reducing brake squeal in disc brakes, several empirical methods have been developed. In this regard, examination of various design alterations like modification of the backing plate, changing the calliper stiffness and geometry and increasing the damping between the brake pad and backing plate and brake piston has been done.

Some of the generally employed solutions in the automotive maintenance sector entail the increasing of the damping in the brake system for suppressing brake squeal.

a) Between the calipers and backing plates, the use of an anti-squeal product like the disc brake quiet (this product contains ethylene, water and glycol and is manufactured by a number of companies).

b) Applying a grease which can also be an anti-seize compound to the piston-backing plate contact locations (at high temperatures found in a braking system, an anti seize compound may help in preventing the fusing or welding together of the backing pads and pistons in addition to acting as a lubricant).

c)Between the backing pads and calipers, the use of vibration shims. Constrained layer dampers are usually found in these shims (a constrained layer damper is laid up with a viscoelastic polymer, limited between dual layers of a stiff material like a metal). Quite a lot of off-the-shelf brake pads contain these shims.

d) Slotting and/or chamfering of the friction material's pads.

e) Subjecting the surfaces of brake rotors to sanding.

f) Lubricating the pins connecting the calliper to its mounting bracket.

In the 1950s, the first concentrated attempt to study and dissect car disk brake squeal was made at the MIRA (Motor Industrial Research Association, UK) as observed by North [10].
By performing stability analysis, the real parts of the eigenvalues can be extracted to construct the characteristic equation of the system. If the real parts are positive then the system is unstable, and if they are negative then the system is stable. In a disc brake system, friction force is the main cause for the excitation of vibration and is unevenly distributed force between the brake disc and the brake pad on either sides of the rotor.
Research on the vibrational instabilities which lead to break squeal has been conducted for more than fifty years. Today's better understanding of the vibrational instabilities that produce brake squeal has required a great deal of detailed and sophisticated analysis and experimental research due to the complexity of a brake system. The ability to design a brake system and identify its causes of squealing in advance would offer a very helpful design tool. The literature also suggests that different brake systems radiate sound in very different ways. Murakami (1984) studied a brake system and found that the brake calliper and pad played major role in brake squeal. Five years later, Nish Iwaki conducted a research on a different brake system and concluded that rotor was responsible for the most noise. McDaniel, Moore, Chen and Clarke [11] presented a study of acoustic radiation from the stationary brake system. The objective of this experiment was to better understand the acoustic radiation from squealing brake systems. The researchers found that the great majority of squeal mechanisms occur due to the resonant behavior of the operating brake system. They also presented an analysis that equated the natural frequencies and modes of mechanically-excited stationary brake systems to those of an operating brake system. Acoustic radiation efficiencies and intensities of the modes were computed by importing experimentally measured velocities into a BEM software package, which revealed that for a particular brake system, the highest radiation efficiency occurred at frequencies above 2-3 kHz [11].

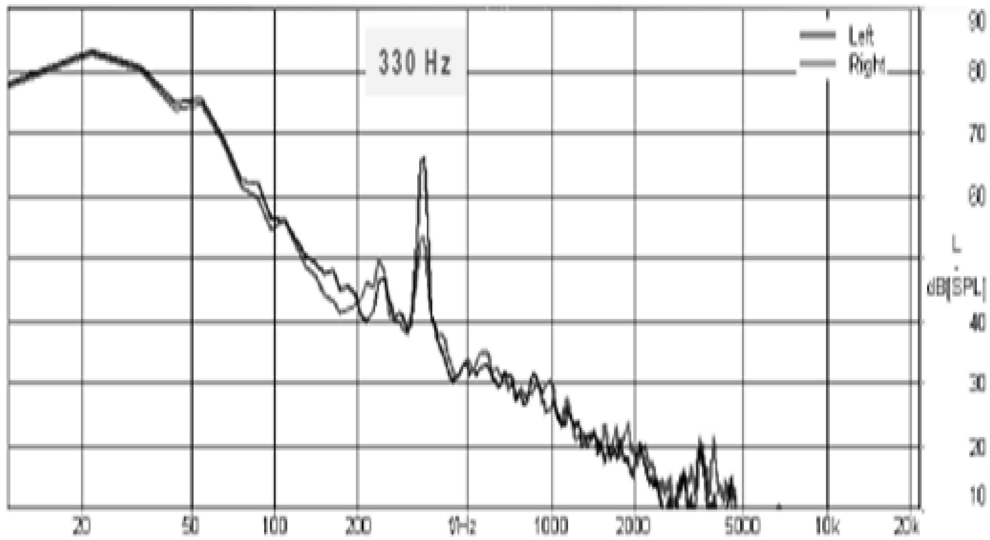


Figure (5) Moan noise ranges

Brake Squeal theories

When several brake parts like the disc and the pad vibrate together, a coupled system is created resulting in the occurrence of squeal noise. Geometrical matching and phase vibration will be triggered in consideration of bending modes coupling encompassing similar frequency and wavelength of the components (Fieldhouse, 1999). [11] Figure (4) shown the brake squeal noise.

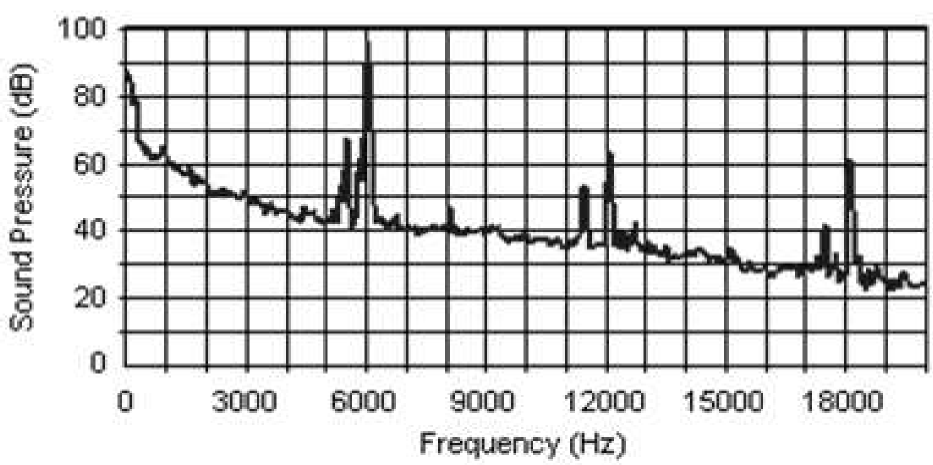


Figure (6) squeal noise ranges

vibrations at right angles to the disc plane in such a scenario. The in-plane vibrations are excited with the decreasing braking power amongst the brake disc and the pads, effectively impacting the disc plane [9]. As the figure (4) below presented the range of frequencies

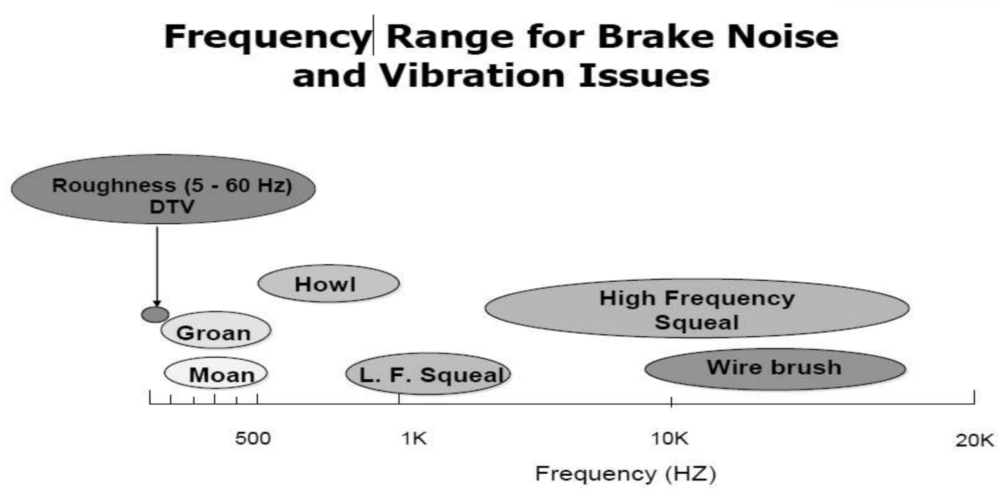


Figure (4) range and type of frequencies

Brake moan

Dunlap et al (1999) investigated various categories of brake noise namely low frequency noise, low frequency squeal and finally high-frequency squeal. Low frequency noise typically occurred at frequency between 100 and 1000 Hz where grunt, groan, grind and moan generally fall into this category. This class of noise was due to friction material excitation between the disc and pad interface. Low frequency squeal defined as a noise having a narrow frequency bandwidth in the frequency above 1000 Hz and yet below the first in-plane mode of the disc [18]. The Mong noise has been ranged between 30-600 HZ, figure (4) shows the Moan noise.

frictional excitation is caused by this kind of noise. With regards to brake squeal, optimum conditions are produced by this sort of coupling. At frequencies above 5 kHz, high frequency squeal occurs which is quite a trouble maker in brake development. The frequency of squeal matches with the circumferential resonance frequencies of the rotor disc according to experimental proofs [7]. Commonly referred to as grunt, grind, moan and groan is the low frequency brake noise, occurring between 100 Hz and 1 KHz. The excitation of the friction material at the rotor and lining interface serves as its point of origin. As the vehicle moves at a low speed, a vibration in the form of an annoying high-pitched squeal is produced owing to the contact of the brake pads with the rotor, thus resulting in the occurrence of squeal in disk brakes.

A piston, guide pins, sealing ring, dust boot, carrier bracket, rotating disc, non-rotating friction pads, yoke and Caliper are the constituents of a car brake system. The pads are loosely located in the caliper and are situated alongside the carrier bracket. In a floating calliper design, the caliper is permitted to slide quite freely along the two mounting guide pins. The disc rotates at a similar speed to the wheel and is bolted to the car wheel. The two pads come in contact with the rotating disc surfaces with the application of the disk brake. Sound is generated when a small portion of the vehicle’s kinetic energy is converted to sound energy, though most of it transforms to heat through friction [8].

An important concern for new vehicle designs is the optimization of this automotive noise phenomenon. To assuage efficaciously the kinetic energy of a vehicle is the primary purpose of a brake. This is accomplished by its conversion to heat energy which is produced when friction occurs between the pads and the brake disc. High frequency vibrations of the brake disc can be excited during the process of braking under particular parameters of operation. These substantial brake structural vibrations are considered undesirable, with the squealing of brakes having no impact on the actual braking effect. When squealing occurs, the out-of-plane vibrations are deemed physically accountable for the emission of noise from the brake disc. The surface of the brake disc’s friction ring is subject to

Drilled Brake Rotors

The image bottom shows the general shape for this type

Figure (2) drilled brake rotor

http://auto.howstuffworks.com/auto-parts/brakes/brake-parts/brake-rotors2.htm

Figure (3) disc brake slotted

http://auto.howstuffworks.com/auto-parts/brakes/brake-parts/brake-rotors2.htm

Brake Noise

Three categories are mostly used to classify brake noise. Low frequency squeal, high frequency squeal and low frequency noise. Low frequency squeal occurs in the frequency range above 1 KHz and below the first circumferential mode of the rotor [6]. Coupling of more than two modes of various components of the brake system and

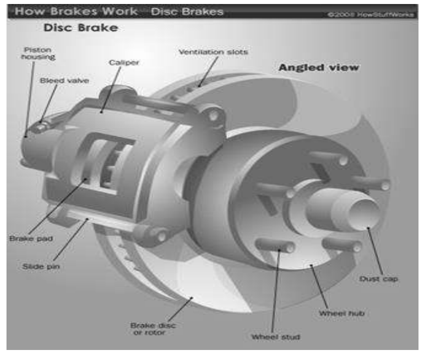


Figure (1) Disc Brake System

[Source:http://auto.howstuffworks.com/auto-parts/brakes/brake-types/disc-brake1.htm]

Brake disc involves of three main components: Brake Pads, Rotor, and Caliper. Each of these components is defined above in detail.

Brake Rotor

According Rahim and Bakar. (2008) [5]. The disk brake is a different device for slowing and stopping several rotations for the machine wheels and whiles in terms the motion of the machine. Therefore, the brake disc is usually made of the metal cast iron, in the different cases the disc brake is made of some composites such as ceramic matrix composites and reinforced carbon to actually connect to the wheels and also the axles from different devices. In terms of stopping the wheels some different materials friction in the form of brake pads that can be connected to the device called a brake calliper. Therefore, the bake disc can actually be forced in the mechanical machines. For example: hydraulically, pneumatically and electromagnetically as well. Significantly, the important of both sides of the disc are the friction causes of the disc brake and also attached into the wheel to slow or stop the some vibrations from the different machines. The brakes can be converting into the motion to heat the vibrations if the brakes get very hot and the different brakes will comes fewer effectives with the phenomenon of the brake fade.

with vehicle vibration and noise via improvement. This can be validated from the literature which describes how awareness regarding brake vibration and noise grew from the early 1930's. The mechanism of generation can be used to explain the various available categories of brake vibration and noise like squeal, hum, squeak, groan and judder. Three primary classifications namely judder, squeal and creep-groan were suggested for brake vibration and noise by a recent review. From these classifications, the most expensive for automobile manufacturers owing to warranty payouts and the most irritable and annoying for the environment and the customers alike is the category of "squeal' [1] and this type is most common.

In this paper, detailed explanations regarding the working of the brake disc and its various parts are provided. Being the major cause of irritation and dissatisfaction for customers, the background to the problem of brake noise is explained in this section as it is one of the major concerns for the vehicle and brake system manufacturers. The occurrence of brake noises, their various types and the methods used to eliminate or reduce the noises is hence explained in this section as well [2].

A pair of brake pads, a calliper –piston assembly where the piston slides within the calliper on top of the vehicle suspension system and a disc capable of rotating about the axis of a wheel are the basic components of a disc brake system. The piston pushes forward and presses the inner pad against the disc and at the same time, the calliper presses the outer pad against the disc with the application of hydraulic pressure [3].

Brake Disc

By clamping brake pads onto a rotor mounted to the hub, a braking force is generated by the disc brake system. A small lever input at the handlebar is converted to a great clamp force at the wheel due to the high mechanical advantage offered by the mechanical and hydraulic disc brakes. Brake power is generated prior to which the rotor is pinched with friction material pads because of this large clamp force.

المستخلص

تعتبر الضوضاء والاهتزازات الناتجة عن نظام الكبح في سيارات الركاب من المشكلات الفنية والاقتصادية المهمة في صناعة السيارات. في السنوات الأخيرة، تم العثور على طريقة العناصر المحدودة (FE) كأداة مفيدة في التنبؤ بحدوث الضوضاء في نظام فرامل معين أثناء مرحلة التصميم. تقدم هذه الورقة نموذج FE أكثر دقة لزاوية فرامل القرص التي تحتوي على محور العجلة ومفصل التوجيه. النموذج هو امتداد لنماذج فرامل قرصية FE السابقة. تم استخدام تحليل تجريبي على نمذجة لنظام الفرامل القرصية في البداية للتحقق من صحة نموذج FE.

ثم تم التنبؤ بالترددات غير المستقرة من خلال تطبيق تحليل القيمة الذاتية المعقد على نموذج FE . وأخيرًا، تم إجراء عدد من التعديلات الهيكلية ومحاكاتها لتقييم صرير الفرامل في مرحلة التصميم.

الهدف العام : الهدف العام من هذا البحث هو دراسة ظاهرة الضوضاء في نظام الفرامل وطريقة مطورة لتقليل هذه الضوضاء من خلال التجارب والمحاكاة.

الأهداف

- توفر دراسة وتحليل ظاهرة صرير الفرامل طريقة لمساعدة المصممين والصناعات على فهم تأثير الاهتزاز على دوار الفرامل القرصية.\
- فهم طرق تحليل البيانات المتقدمة باستخدام برنامج ABAQUSE .

Introduction

The theory underpinning the brake disc and brake noise pertaining to the brake system forms the basic text of this literature review. Reducing or eliminating altogether the vibration and noise of a vehicle structure and system helps in gaining a competitive edge for the automobile manufacturers as vehicle comfort has become a crucial element in indicating the quality of a car. Finite elements in brake vibration and noise are inevitable, with the advancement towards other elements of vehicle design improvement contrasted

Study the noises phenomenon in the brake system

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Abstract

The noise and vibration generated by the braking system in passenger cars are important technical and economic problems in the automotive industry.

In recent years, the finite element (FE) method has been found to be a useful tool in predicting the occurrence of noise in a particular brake system during the design stage.

This paper presents a more refined FE model of the disc brake corner that contains the wheel hub and steering knuckle. The model is an extension of earlier FE disc brake models. Experimental has been modelled analysis of the disc brake system is initially used to validate the FE model. The unstable frequencies were then predicted by applying a complex eigenvalue analysis to the FE model. Finally, a number of structural modifications are made and simulated to evaluate brake squeal at the design stage.

Overall Aim

The over all of aim of this paper is to study the noises phenomenon in the brake system and developed method to minimise this noise though the experiments and simulation to help designers and industries to understand the effect of vibration on disc brake rotor, and also to understand advanced data analysis methods by using ABAQUSE software.