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On the Study of Disc Brake Vane Configuration Effected to Brake Squeal Noise

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Abstract

From many years ago to present, there are many published papers from car manufacturers and academics, are investigated on squeal noise mechanism on the brake system. Brake squeal noise is occurred on the frequency range of 1 kHz to 5 kHz as low frequency squeal and 5 kHz to 20 kHz as high frequency squeal. For self-excited mechanism, the brake disc and pad system is contributed to its mode coupling vibration between out-of-plane and in-plane disc modes acted as friction force feed energy to the brake system and results in the mode locking between the disc and pad mode shapes. This paper is focused on the high frequency brake squeal and countermeasure against the addressed mechanism by vane configuration design. The three brake and pad systems (A, B and C) with some knowledge on their experiment and real commercial use squeal problem, are studied for their unstable behavior using Complex Eigenvalue Analysis (CEA) where the most important brake condition parameters, friction coefficient and brake pressure, taken into account. The four vane configurations with 32, 38, 48, and 52 pieces uniformly distributed along the disc, are studied for all three brake systems. The out-of-plane disc mode has been changed significantly in mode coupling and locking due to the vane configuration design. The vane configurations are designed such that there is at least one configuration that can reduce the self-excited mechanism.

1. Introduction

Brake squeal noise is a significant problem for automotive manufacturing due to its warranty claim cost per year. Although this issue is not effected to brake performance of vehicle but brake noise is mostly make the customer unsatisfied and concerned. Brake squeal noise which occurs in frequency range of 1 kHz to 20 kHz where there are many modes about a hundred theoretically involved. The modes are in different configuration shapes of vibration where their mode frequency and shape order depended on disc brake geometric characteristics. The interaction between brake disc and pad, is generated by a friction contact force during brake operation which identified to be dynamic instabilities [1]. The Complex Eigenvalue Analysis (CEA) is based on the commercial Vanities Element Analysis software which used as a mathematical tool for predicting the unstable modes. The damping coefficient of the brake disc and pad system under friction contact force and disc rotational velocity, is determined such that the high value of negative damping is leading to brake squeal noise. The method is efficient and more insight into the friction-induced dynamic instability [2]. However, CEA method has a limit assessment to more understand what is occurring internally within the system or what component of brake system should modify to increase stability. There are the other methods to get more insight such as feed-in energy indicated vibration energy added to unstable mode, strain energy indicates which component should be modification or which component most effected to cause unstable response and modal participation indicated which component mode is the most dominant in the unstable system mode [3]. North examined geometry influence and symmetry on mode coupling by using simple model relation [4]. Tan and Chen have studied brake squeal mechanism of brake system which indicate 5 significant mechanisms those are (1) stick-slip mechanism indicated different between static friction coefficient and kinetic friction coefficient, (2) negative damping mechanism indicated contact friction coefficient reduced when velocity increase, (3) sprag-slip mechanism occurred when friction coefficient constant (4) mode coupling mechanism indicated mode shapes and frequencies of brake disc and pad modes which closed together and (5) hammering excitation mechanism indicated vibration behavior of disc brake due to uneven contact surface [5]. Quaglia and Chen (2006) have established the design concept and laboratory testing method of brake disc and pad characteristics to get more stability condition and reduce degree of mode coupling [6]. Brake disc design is mainly considered as contribution to brake noise due to a number of modes possessed by the component. The in-of-plane mode is referred as the start of friction induced mode and then coupling with the out-of-plane modes at the frequency nearby. Hence, the aligned or locked mode shapes between disc and pad, in out-of-plane direction, will tend to be self-excited vibration and result in squeal noise. To avoid the aligned mode shapes, the change of disc geometry is the first attempt such as inbound/outbound thickness change [ref.] and disc hat modification [ref.] etc. However, the change of disc thickness is limited by its design strength. Vane or vane configuration is considered as a design parameter in this study. The existing brake disc and pad designs are selected as the brake system A, B, and C where their original squeal noise problem to be studied. The four vane configurations are applied to all brake system such that the mode coupling and locking will be studied such that remedy guideline of vane configuration will be addressed.

2. Brake Squeal Mechanism

Brake squeal mechanism, mode coupling, is revisited with explanation on squeal phenomena.

Negative Damping

Brake squeal analysis is resulted from unstable response during brake disc and pad interaction. The unstable due to negative damping is contributed from introduced friction force, then the unsymmetrical stiffness of the coupling system [7]. Figure 1 illustrated the 2-DOF system where the friction force included. The equation of motion is derived in equation (1) to (3) with unsymmetrical stiffness.

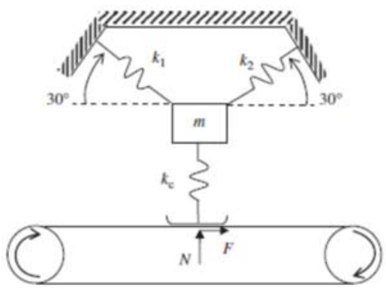

Figure 1 Two degree-of-freedom system with sliding friction by Hoffman [7].

The equation of motion can be expressed as

$$M\ddot{x} + (K + \sum_{i=1}^{n} \mu_i k_{ei} E_i)x = f \quad (1)$$

$$[Ms^2 + (K + \sum_{i=1}^{n} \mu_i k_{ci} E_i)]X(s) = F(s) \quad (2)$$

$$\frac{\partial}{\partial_{s^2}}[det(Ms^2 + K + \sum_{i=1}^{n} \mu_i k_{ci} E_i)] = 0 \quad (3)$$

Where

$$k_{ei} = k_1, k_{2,}, k_3, \dots \dots$$

Mode Coupling and Locking

There is an attempt to distinguish between mode coupling and mode locking [2] where the first is the meaning of natural frequency of two or more brake components are close together and the latter is the vibration mode shape aligned between two contact components. However, because the friction force is able to excite the in-plane disc modes the coupling between in-of-plane and out-of-plane of the disc modes also referred as the mode coupling mechanism. The out-of-plane disc mode will result in excite the air around the disc to move such that generated as the squeal noise. The brake pad is usually designed to damp out the disc vibration but if the pad vibration mode is coupling in frequency and aligning with the disc mode shape, then the vibration energy is still existing during braking event.

Due to small size of the brake pad, the first ten modes are around 2 kHz to 18 kHz. Compared to the brake disc, the modes are considered from 500 Hz to 18 kHz with nearly a hundred modes. The disc/pad mode shape locking or alignment is normally occurred at high frequency so called high frequency brake squeal.

3. The Brake System Element Models

The brake disc and pad, as shown in Figure 2, are modeled such that the thickness layers for all component are divided into two layers and the element number is optimized for describing the vibration modes from 500 Hz to 18,000 Hz with solution convergence and computing time.

There are three existing designs of the brakes disc and pad systems considered as the Design A, B and C. The geometries of the three brake systems are different and the identified material properties are provided in Table 1 where the FE models have been updated with the test results.

To analyze mode coupling (in-of-plane and out-of-plane disc modes) due to contact friction between the brake disc and pad, the CEA must provide with the braking conditions, the friction coefficient, μ, the inherit brake disc material damping ratio of 0.005, the piston brake pressure and the initial disc angular velocity as shown these data in Table 2.

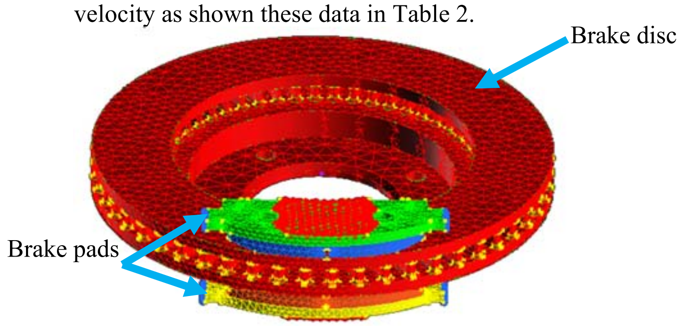

Figure 2 3D Brake system model for CEA

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The CEA by NASTRAN/SimXpert software are utilized for studying mode coupling and locking. The boundary conditions and parameters which used for identified in the FE models come from laboratory testing.

Table 1 Material properties of brake disc and pads

Design A	E (GPa)	Density (kg/m^3)	Poisson's ratio
Disc	114	7300	0.3
Pad lining mat.	9.5	3000	0.3
Pad Back plate	200	7300	0.3
Design B	E (GPa)	Density (kg/m^3)	Poisson's ratio
Disc	110	7100	0.3
Pad lining mat.	7.5	2300	0.3
Pad Back plate	200	7300	0.3
Design C	E (GPa)	Density (kg/m^3)	Poisson's ratio
Disc	128	7100	0.3
Pad lining mat.	11	2600	0.3
Pad Back plate	200	7300	0.3

Table 2 Braking conditions for CEA

Angular Velocity (Rad/s)	μ, friction coefficient	Pressure (MPa)
6.28	0.5	5

4. Simulation results

The CEA of the original vane configuration, or the number of vane distribution around the brake disc, is included for each brake system. There are of four vane configurations used, 32, 38, 48, and 52 vanes. These vane numbers are chosen such the out-of-plane disc modes to be most affected. The mode shape alignment/locking between disc and pads, is then examined.

Complex eigenvalue analysis of 3 design brake systems

For the brake system Design A, the original vane configuration is of 48 vanes and the brake squeal is possible due to high negative damping ratio as shown in Figure 3. The original vane configurations of the Design B and C are of 42 and 48 vanes, respectively, where the unstable modes are existed with less significant of damping ratio values compared to the Design A, see Figures 4 and 5. Apart from explanation about each Design characteristics, the 52 vane configuration for all Design is tended to be unstable or brake squeal noise mechanism exist. Then the mode coupling and locking analyses are need to be examined.

All the effect of vane number to all Design is provided in Table 3 where summarizes a number of the unstable modes with significant of negative damping ratio more than 0.01 and the frequency more than 5000 Hz where the high squeal frequencies are interested. Noted that all the brake system design is unstable in the 52-vane configuration.

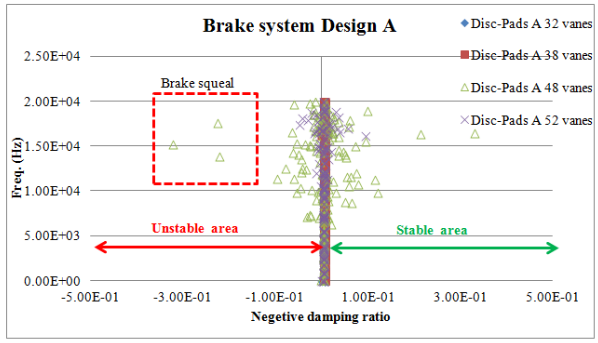

Figure 3. Negative damping ratio of Complex eigenvalue Analysis of brake system Design A

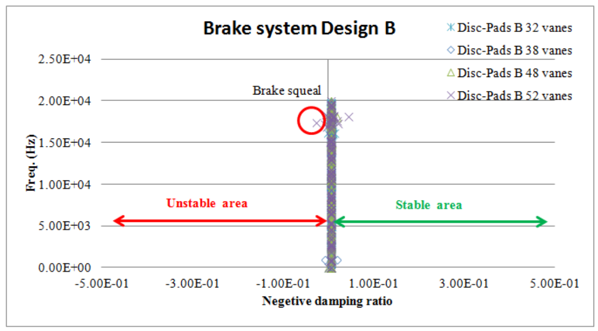

Figure 4. Negative damping ratio of complex eigenvalue analysis of brake system Design B

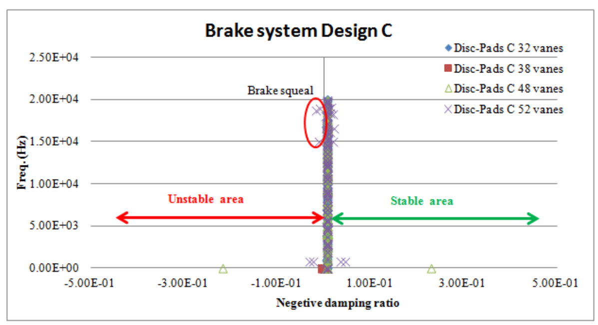

Figure 5. Negative damping ratio of complex eigenvalue analysis of brake system Design C

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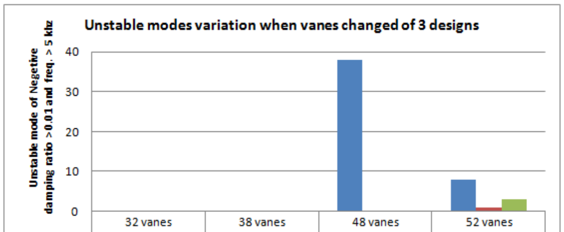
Table 3. Number of unstable modes for all vane configuration and brake system designs (threshold for negative damping > 0.01 and frequency > 5000 Hz)



	32 vanes	38 vanes	48 vanes	52 vanes
Design A	0	0	38	8
Design B	0	0	0	1
Design C	0	0	0	3

Mode coupling and locking analyses

For the Design A with original vane number of 48, the most dominant unstable mode is the mode of 15,200 Hz as shown in Figure 6. The response to the excitation friction force is the in-of-plane modes, around that frequency, are comprised of pure circumferential mode at 15,517 Hz and in-of-plane radial-circumferential mode at 14,316 Hz. Due to the unstable mode shape recognized as 6 nodal lines, the 6ND out-of-plane modes is about the fundamental frequency of 7,471 Hz and the unstable mode, at 15,200 Hz, is nearly its second harmonic frequency. Then the air surrounded the disc, is radiated as high frequency noise. This mechanism is known as the 'mode coupling'.

Normally, the brake pad with sufficient damping will suppress this noise. However, if the pad mode shape at nearly the unstable frequency is aligned in shape with the disc mode, then the self-excited vibration is occurred. The disc response is growing in magnitude due to pad motion excitation. This is so called 'mode shape locking'.

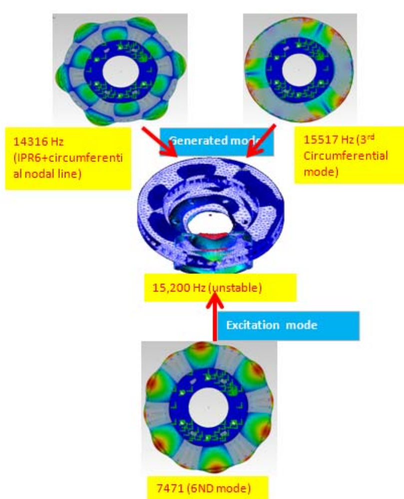


Figure 6. Mode coupling description of Design A with original vane number of 48 [9].

When vane configuration has changed from original Design A with 48 vanes to 52 vanes, the unstable mode at 17.44 kHz is contributed by disc in-plane radial mode with 2 circumfertial and 4 radial nodal lines at 14.37 kHz where the mode shape is aligned with the pad (see Figure 7). Figure 7 shows the deformation for out-of-plane motion along the disc circumferential and also along the outside edge of the pad. There is a central location that the disc and pad motion is in phase and resulted in self-excitation. Figure 8 shows the mode shapes around the unstable modes where some of them are in-plane disc mode and some of them are out-of-plane disc mode.

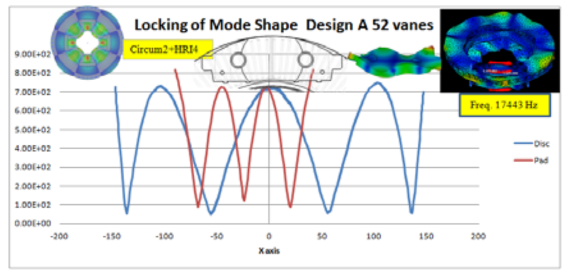


Figure 7. Mode shape locking of unstable mode (17.44 kHz): brake system **Design A** with 52 vanes (disc (blue), pad (red))

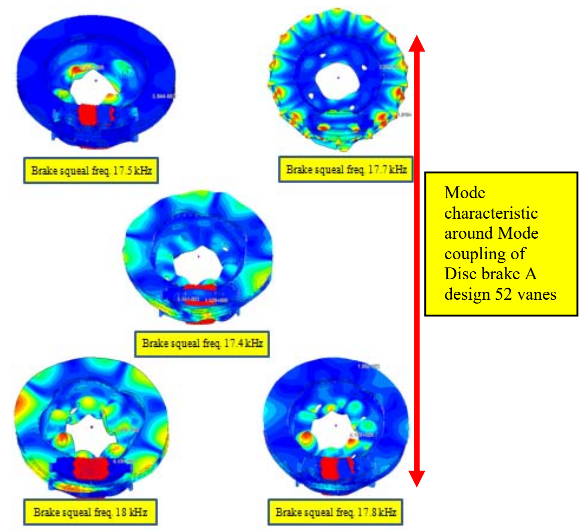


Figure 8. Mode coupling of brake system Design A with 52 vanes

If considered the stable mode near the unstable mode frequency as shown in Figure 9, the disc deformation shape is the out-of-phase compared to the pad deformation shape at the contact area location. The unstable mode shapes for Design B and C are shown for the locking of shapes in Figures 10 and 11, respectively.

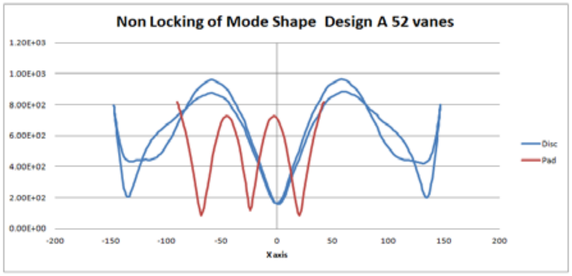


Figure 9. Non-locking mode shape of stable mode (15.56 kHz): brake system **Design A** with 52 vanes (disc (blue), pad (red))

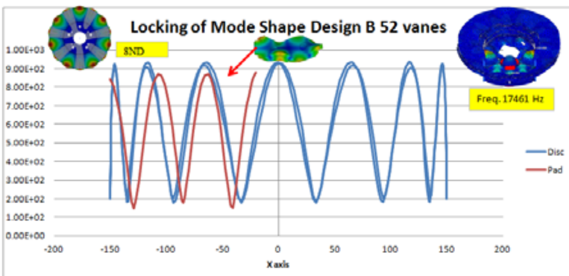


Figure 10. Mode shape locking of unstable mode (17.44 kHz): brake system **Design B** with 52 vanes (disc (blue), pad (red))

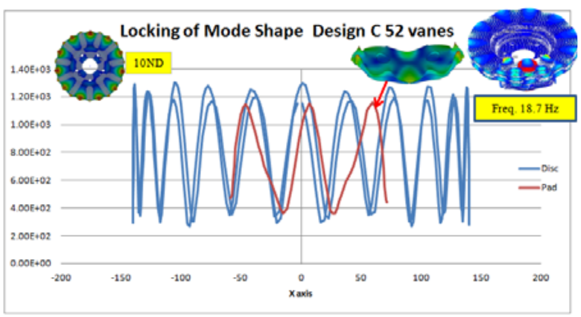


Figure 11. Mode shape locking of unstable mode (18.7 kHz): brake system **Design C** with 52 vanes (disc (blue), pad (red))

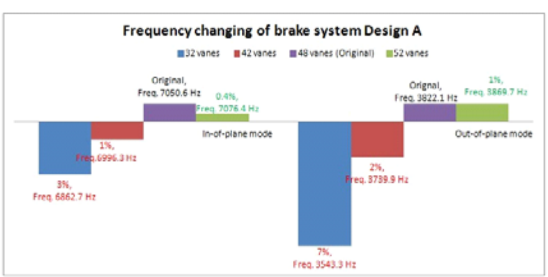


Figure 12. **Design A**: Percentage of changing in mode frequencies both in-of-plane and out-of-plane of the disc related to the unstable mode for 4 vane configurations

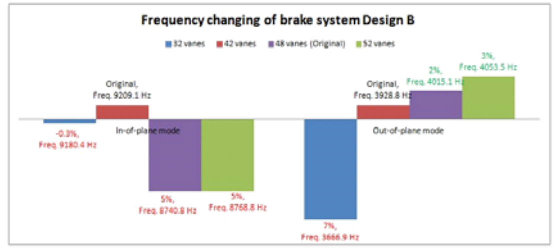


Figure 13. **Design B**: Percentage of changing in mode frequencies both in-of-plane and out-of-plane of the disc related to the unstable mode for 4 vane configurations

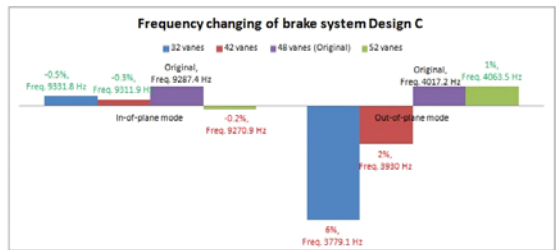


Figure 14. **Design C**: Percentage of changing in mode frequencies both in-of-plane and out-of-plane of the disc related to the unstable mode for 4 vane configurations

Figures 12 to 14 show the percentage of frequency changes in both in-plane and out-of-plane of the disc mode for Design A, B, and C, respectively. More the vane number, higher the out-of-plane natural frequency. This is due to contribution of vane bending stiffness acted as cantilever beam-like to the modes. The vane number is less an effect to the in-plane mode. Usually, the mode coupling or frequency coupling can be separated between the in-plane and out-of-plane modes by vane configuration design. In case of 52 vane configuration, all of the brake design comes with unstable modes. However, the Design B is more stable than others. From Figure 13, the in-plane modes have significantly changed in frequencies and opposite effect to the other designs. The vane shape of Design B is different from the Design A and Design C. This is to be studied in future about the vane shape effected to the in-plane disc mode.

5. Conclusion

This research studied on three commercial brake disc-pad designs which maintain original concept. All of design are studied on stablity during braking condition by using Complex Eigenvalue Method, where the negative damping is considered on the friction force and angular velocity are constant. When brake disc and brake pads interfere then friction force is generated, this mechanism can excited mode coupling and brake squeal due to negative dampling variable. The mechanism of unstable mode is showed about the alignment of the deformed shapes between disc and pad. The vane configuration changes are performed where the out-of-plane modes are most effected in frequency changed. The design with vane spacing can then reduce the effect of mode coupling and locking.

6. Acknowledgement

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