

Automated Tool for Brake Squeal Analysis

Sanjib NATH*

Scott PAUL*

ABSTRACT

Brake squeal analysis is a difficult process to set-up because selecting proper solver parameters and boundary conditions is complex. For that reason, an automation tool has been developed to mesh, set contacts, set constraints, define the non-linear and complex eigenvalue analysis parameters, and apply a gradual pressure load. The key benefit of this tool is that it helps a non-specialist quickly find unstable modes in the brake system design without prior knowledge of complicated CAE tools.

1. Introduction

Brake squeal is a high-frequency vibration that can be caused by vibrations between the brake pads and rotor, pads and the caliper, or caliper and its mounts. Brake squeal occurs in a high frequency range greater than a 1000 Hz vibration of the brake system during a braking event. Brake noise is annoying to passengers and can lead to customer dissatisfaction.

2. Contributing Factors to Brake Squeal

Figure 1 shows the brake components that can cause brake squeal. Applying the brake increases brake line pressure, which causes a caliper piston to push brake pads against a spinning rotor. An unstable mode occurs when a combination of piston pressure, friction, and damping effects cause two stable modes to merge.

Different brake noise phenomena exist in different frequencies. Figure 2 shows the different noise phenomena that exist in the human hearing range⁽³⁾. This study focused on low and high squeal frequencies in the 1-20 kHz range.

Figure 3 shows a tree of how different simulations

are used to predict brake squeal.

Complex eigenvalue analysis (CEA) is one of the critical simulations for predicting unstable modes for brake squeal. The simulation is very difficult to properly setup. There was no easy tool capable of

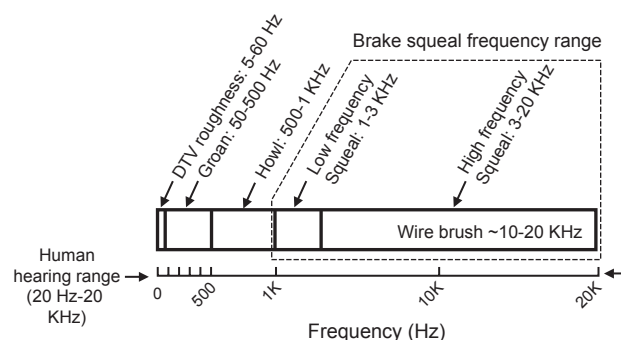


Fig. 2 Frequency range for brake noise phenomena

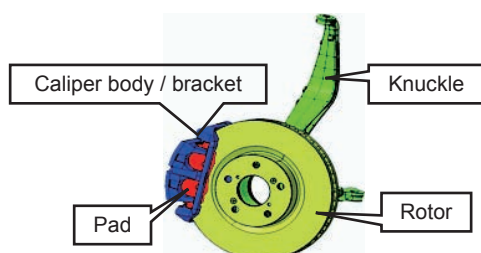


Fig. 1 Components that can cause brake squeal

* HONDA R&D AMERICAS, INC.

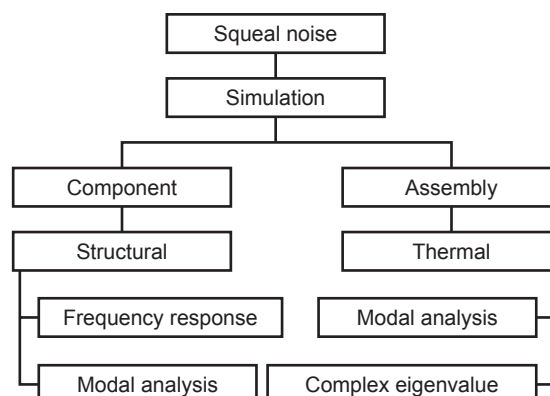


Fig. 3 Simulation dependencies for squeal noise

performing CEA confirmation prior to engine running testing. The main objective is to introduce new NVH techniques in order to improve marketability. CEA runs could be completed to find the best way to optimize the brake corner.

3. Modeling Techniques

The rear brake assembly model was considered to illustrate the modeling technique and analysis. Altair HyperMesh software was used to setup the model and complex eigenvalue analysis carried out using MD Nastran solver.

3.1. Brake Components

The rear brake assembly includes rotor, caliper body, caliper bracket, piston, inner pad lining, outer pad, retainer, back plates, and shim. The brake assembly with all the components is shown in the Fig. 4.

3.2. Material Property

Anisotropic material property was used for the lining material and isotropic material property was used for the rest of the components.

3.3. Contact Definition

Touch and glue types of contacts (in MD Nastran solution) were used between the mating parts. Touch contact was used where friction is involved and movement is allowed between components. The glue contact was used where no friction is involved and

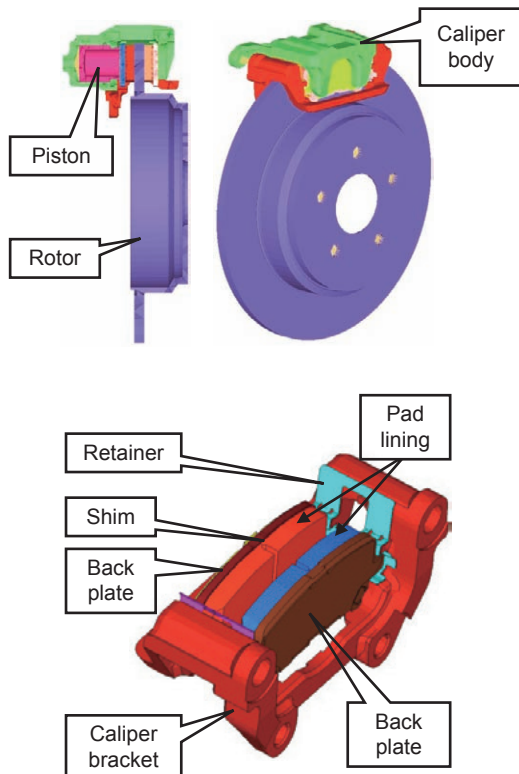


Fig. 4 Brake assembly

Table 1 MD Nastran parameters

Parameters	Definition
SOL 400	Nonlinear static and transient analysis
NLPARAM	Nonlinear static analysis parameter selection
NLIC	Defines nonlinear initial condition
METHOD	Selects real eigenvalue extraction parameters
CMETHOD	Selects complex eigenvalue extraction parameters
EIGRL	Defines data needed to real eigenvalue analysis
EIGC	Defines data needed to perform complex eigenvalue analysis
MAT1	Defines material properties for linear isotropic materials
MAT9	Defines material properties for linear anisotropic materials
BSQUEAL	Specifies data for brake squeal analysis
BSURF	Defines a contact body or surface defined by elements IDs
BCONTACT	Initiates and controls 3D contact
BCTABLE	Defines contact table
BCBODY	Defines flexible or rigid contact body

a rigid connector between components is allowed. Three different friction values of 0.1, 0.3 and 0.5 were applied between the rotor and pad and default values (blank) were used for the rest of the contact pairs. Contact pair for different mating parts is shown in Fig. 14 in section 4.4.

3.4. Constraining the Model

The rotor and caliper bracket bolt holes were fix-constrained. Rigid body elements were used instead of modeling pin and caliper contacts (virtual pin). The pin between the caliper body and bracket was only allowed to move in the Z axis (axial). The constraints details are shown in Fig. 10 and Fig. 11.

3.5. Loading Condition

The same and opposite pressure was applied to piston (bottom) and caliper body (bore bottom) which is shown in Fig. 12 in section 4.3.

3.6. Analysis Solver

MD Nastran Solution 400 was used to solve the complex eigenvalue analysis. Some of the important brake squeals and contact parameters used in the analysis are listed in Table 1⁽¹⁾.

4. Developing Brake Squeal Analysis Tool

CEA is one of the established methods to predict squeal noise; however, it is a difficult analysis to set-up because selecting the proper solver parameters and boundary conditions are complex. For that reason, a wizard based GUI was developed for a non-expert to set-up a CEA step-by-step. The wizard is accessed in the Altair HyperMesh workbench, which is used for setting-up and viewing the results of the analysis. It uses the MD Nastran advanced non-linear solution sequence to solve the CEA. The overall process is shown in Fig. 5.

4.1. Brake Squeal Simulation GUI

The tool (as shown in Fig. 6) enables the designer to go through the following steps, with meshed parts as the starting point:

1. Update parts to represent correct material properties and orientation
2. Define the right boundary conditions
3. User defined contact pairs and contact properties
4. Export the input data to solve in MD Nastran

4.2. Part Update Function

After the user imports a mesh model, the Part Update function (as shown in Fig. 7) is used to adjust the materials and properties and create coordinate

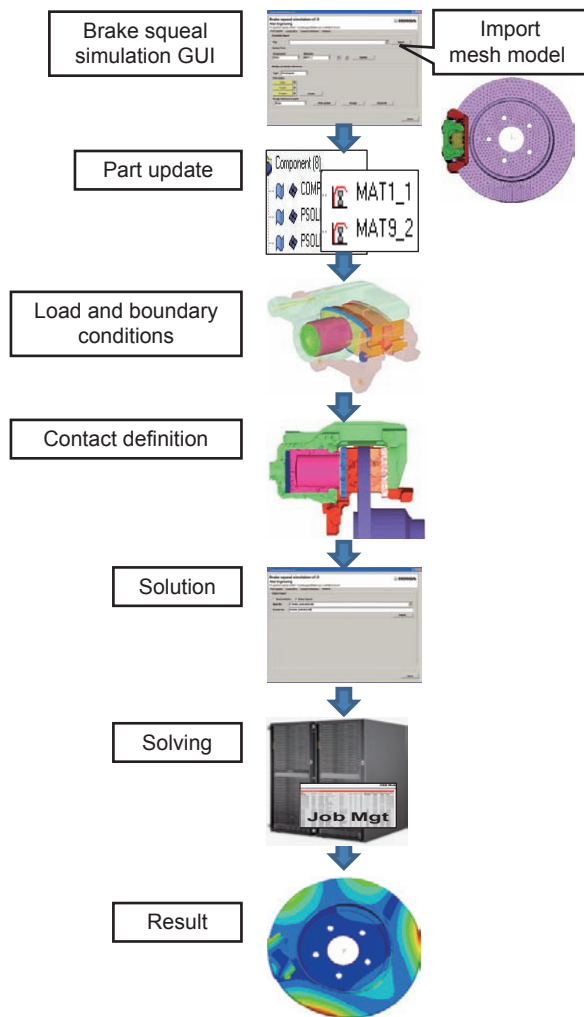


Fig. 5 Overall analysis process

systems of the model. The process involves the following steps:

- (a) Select a part from the Components list box.
- (b) Select a material from the Materials list box.
- (c) Display the part in the graphics window.
- (d) Update the component properties.
- (e) Specify a rectangular coordinate system (for pads) or cylindrical system (for rotors).
- (f) Assign new references to parts.

The imported mesh model is shown in Fig. 8.

4.3. Load and Boundary Conditions Function

The Load and Boundary Conditions function (as shown in Fig. 9) constrains and applies load to the model. The process involves the following steps:

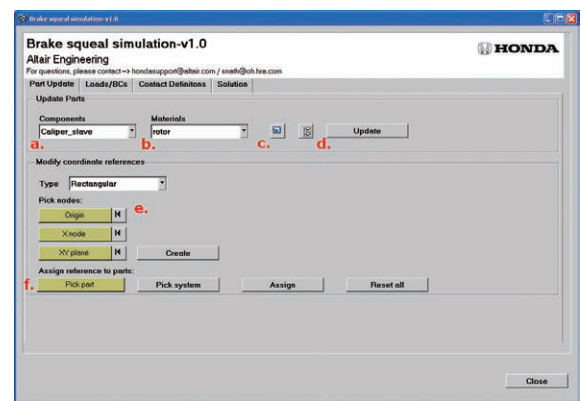


Fig. 7 Part update function

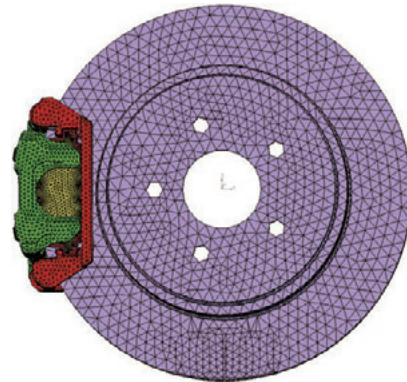


Fig. 8 Imported mesh model

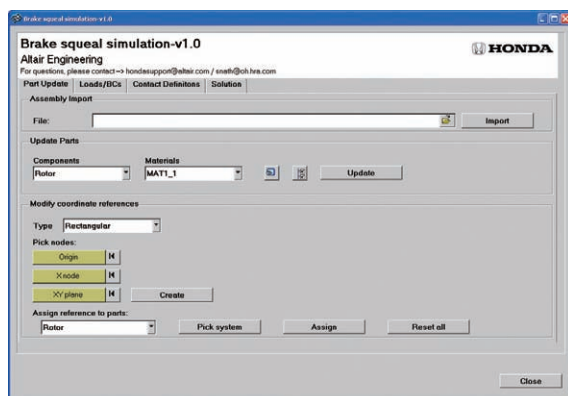


Fig. 6 Brake squeal simulation GUI

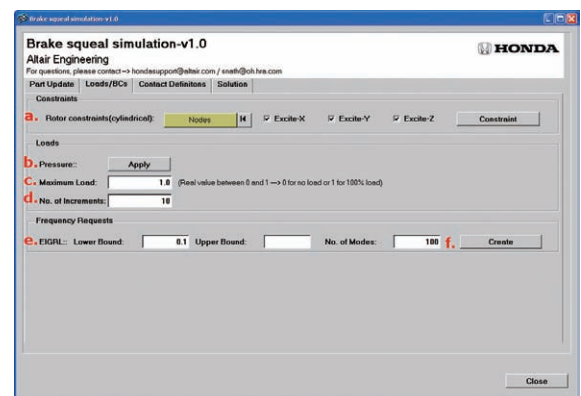


Fig. 9 Load and boundary conditions

- Specify directions for nodal constraints. Since solid tetra elements are being used, only 3 degrees of freedom is necessary.
- Specify pressure loads to be applied to the piston or the caliper, usually in equal and opposite pressure values (5 to 10 bars).
- Indicate the maximum fraction of total load to be applied.
- Enter the number of load increments for the non-linear static case.
- Specify the lower and upper frequency bounds. The squeal mode usually occurs between 3-20 kHz. User also can define number of modes to be extracted.
- Create the request for real and complex Eigenvalues.

Rotor and caliper were constrained shown in Fig. 10.

Virtual pin is allowed to move along the pin axis which is shown in Fig. 11.

Same and opposite pressure load were applied to piston and caliper body which is shown in Fig. 12.

4.4. Contact Definitions Function

Glue and touch contacts were defined between the mating parts using the GUI (as shown in Fig. 13). There is no user input required to select specific elements of the rotor and pad, which will potentially participate in contact surfaces. MD Nastran parameters automatically calculate the contact between deformable bodies by selecting rotor and pad components. The process involves the following:

- Select a contact pair name.
- Graphically pick master and slave parts.
- Select Touch or Glue type of contacts.
- Create a contact pair.
- Select a coordinate system for the rotor.
- Apply a friction coefficient between rotor and pad at different steps.
- Review/modify/delete contact pairs.
- Assign contact stiffness.

Glue and touch contacts were defined in the following contact pairs shown in Fig. 14.

4.5. Solution Function

Two simulation cases can be created: Normal Modes and Brake Squeal simulation. The analysis was solved in MD Nastran SOL 400 and the input file was exported from the GUI (as shown in Fig. 15).

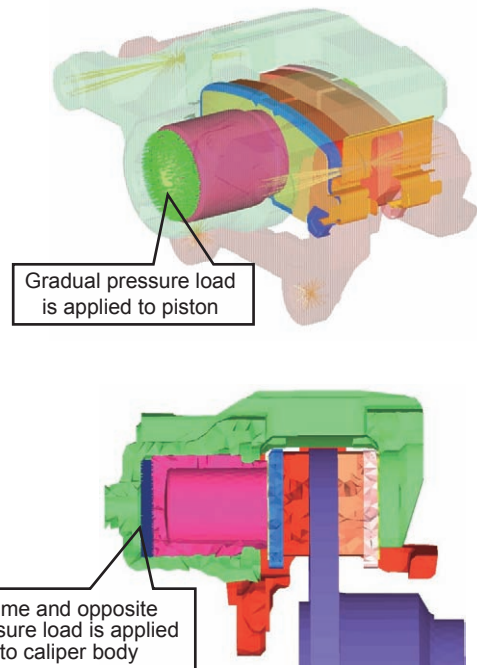


Fig. 12 Pressure load applied to piston and caliper

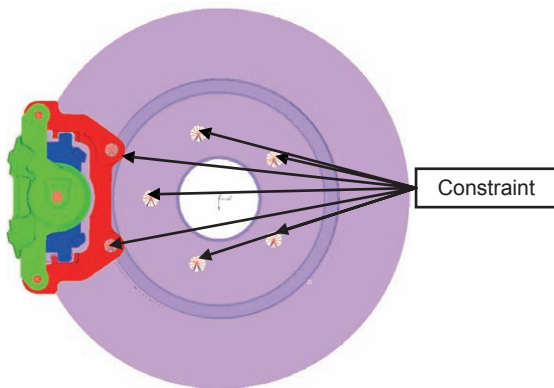


Fig. 10 Constraining model

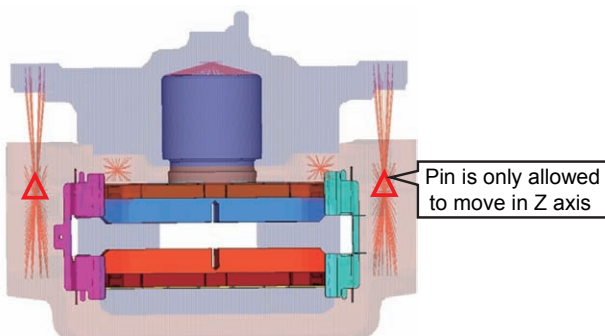


Fig. 11 Constraining virtual pin

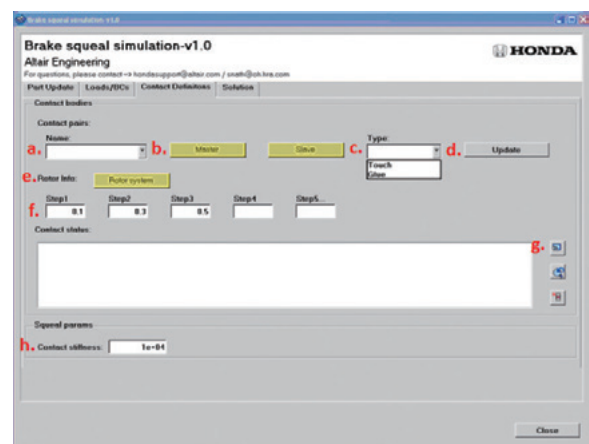


Fig. 13 Contact settings

4.6. Solving the Analysis

The MD Nastran model was submitted to a remote Cluster through a web-based job submission process.

4.7. The Result

After the analysis was solved, MD Nastran extracted the number of modes (based on frequency range) in different frequencies. If the mode contains a positive damping coefficient in a particular frequency called a stable mode and with a negative damping coefficient called an unstable mode. In addition, wherever an unstable damping mode exists there will also be an unstable positive real part (eigenvalue). These unstable real parts have a higher propensity for brake squeal.

All the unstable modes at different friction values are listed which is shown in Table 2. The highest Real Part values at that particular frequency and friction the more potential for brake squeal. The highest values were considered in each friction value.

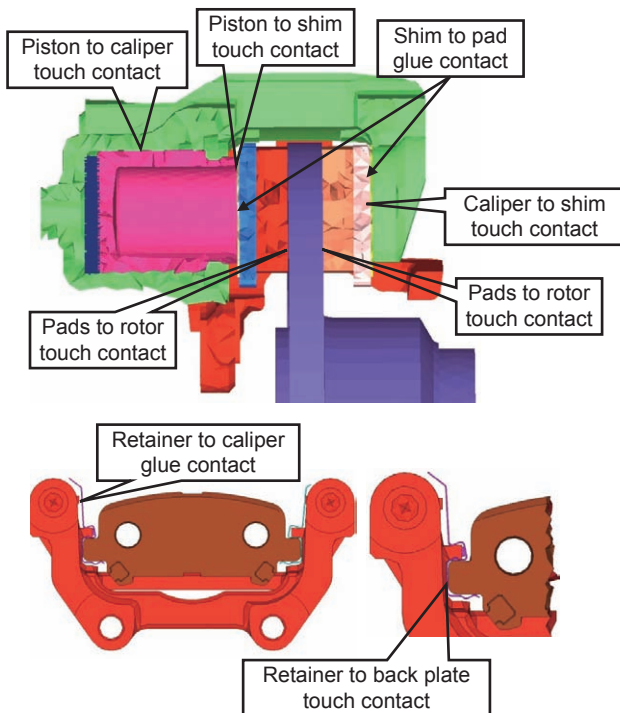


Fig. 14 Contact pair between mating parts

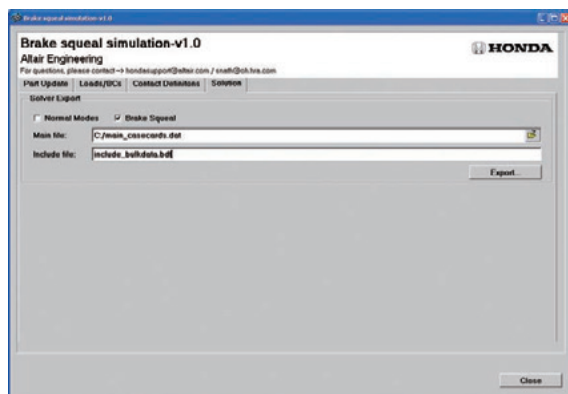


Fig. 15 Exporting Nastran input file

Table 2 List of unstable modes

Friction Coefficient					
0.1		0.3		0.5	
Frequency	Damping coefficient	Frequency	Damping coefficient	Frequency	Damping coefficient
1436.1	-1.0879E-02	765.9	-1.6644E-01	334.2	-3.9757E+00
1581.8	-7.3663E-02	1089.2	-2.1251E-01	1205.1	-9.1807E-02
2329.9	-1.2960E-02	1389.2	-2.6517E-02	2327.9	-5.0113E-02
2218.8	-1.3124E-01	2528.0	-1.6859E-01	2552.6	-1.1247E-01
1661.3	-9.5757E-01	2268.2	-5.0868E-01	3090.9	-2.6059E-01
2563.3	-9.3130E-03	3852.6	-4.4087E-02	3463.1	-2.7524E-02
4468.5	-2.5986E-03	4505.9	-1.8205E-02	3740.3	-4.6142E-02
5722.9	-3.4972E-02	4926.9	-9.4900E-02	4478.7	-8.0819E-02
5877.6	-2.2904E-02	5712.1	-5.0749E-02	3933.4	-5.2205E-02
7175.0	-4.5128E-03	5827.2	-1.2575E-02	5100.6	-5.8908E-02
7056.3	-4.9961E-02	7089.4	-6.6410E-01	6018.7	-5.2367E-02
		12719.1	-2.5516E-01	5644.7	-1.0792E+00
		11472.6	-1.0793E+00	11392.2	-8.9748E-01
		12897.3	-1.8317E+00	15360.0	-2.2695E-01
				10462.0	-2.6509E+00

The following plot (as shown in Fig. 16) was created to see the most unstable modes in different way. Real and imaginary parts of eigenvalues were plotted (shown in the left image in Fig. 16) and positive and negative values of damping coefficient were plotted (shown in the right image) with corresponding frequencies at a particular friction values of 0.3. The real part of eigenvalue and negative values of damping coefficient are the unstable regions. The farthest point from the axis center in unstable regions in both plots is the most unstable mode shown below.

Displacement contour of one of the unstable modes is shown in Fig. 17.

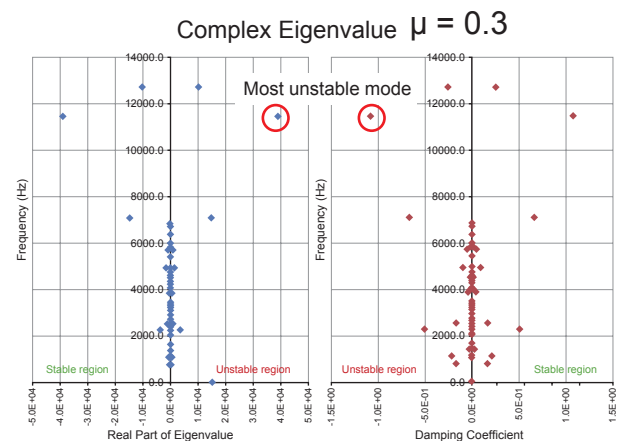


Fig. 16 Most unstable mode

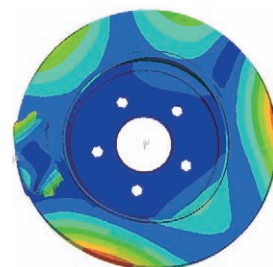


Fig. 17 Displacement contour of unstable mode

The most unstable mode listed above may not necessarily guarantee squeal on vehicle. Depending on the concerned frequencies, friction values, and applied pressure, a number of unstable modes could be considered. In verification with the test result will also be required to judge the simulation result.

5. Conclusion

Developing and implementing a brake squeal analysis tool have contributed to vehicle development in the following ways:

- (1) The tool makes it easy to quickly set-up and run a Complex Eigenvalue Analysis.
- (2) The wizard-based GUI helps the user quickly set-up and execute a simulation.
- (3) A design engineer can use the tool without prior knowledge of complicated CAE tools, such as Hypermesh and Nastran.

References

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- (2) Kinkaid, NM O'Reilly, and P. Papadopoulos. "Automotive disk brake squeal." *Journal of Sound and Vibration* 267 (2003): 105-166
- (3) Liu, W. and J.L. Pfeifer. *Introduction to Brake Noise & Vibration*. Honeywell Friction Materials.

■ 著 者 ■



Sanjib NATH



Scott PAUL