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EXPERIMENTAL VERIFICATION OF SQUEAL NOISE SUPPRESSION BY MASS ADDITION IN MECHANICAL STRUCTURE WITH FRICTION

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ABSTRACT

Squeal noise suppression has been a challenging topic investigated by some researchers today. Some methods have been developed to overcome this problem. The performances of these methods are still unsatisfied to eliminate this noise. In recent research, dynamic vibration absorber (DVA) application to prevent the unstable vibration that emits squeal noise has been proposed. The investigation was performed numerically using two simple structures. It was found that by the addition of a stiff spring-mass or a rigid mass might prevent unstable mode coupling that emits squeal noise in specific condition. In this paper, experimental analysis to verify the method using mass addition is conducted using a simple L-shape structure. The effect of additional mass is investigated at two points of the structure; the point that moves more flexibly in parallel to the normal contact force, and the point that moves more flexibly in parallel to the friction force. It is verified that the additional mass at the near-point of friction where the point moves flexibly parallel in a friction force direction will has a possibility to remove the unstable mode coupling and to prevent the squeal noise. Adding mass at the point relatively far from the friction contact and moves more flexible to the normal contact friction will increase the possibility of a squeal noise incident, although in restricted normal contact stiffness the squeal noise is still being cancelled. It is confirmed that mass addition to the structure at the near point of friction will prevent unstable mode coupling if the additional mass has more contribution to the horizontal direction and uncoupled from the vertical direction.

Keywords: squeal noise, mode coupling instability, additional mass, normal contact stiffness.

INTRODUCTION

Squeal noise is frequently observed when a mechanical structure received a friction load in specific condition. Three typical mechanical systems found to generate squeal noise are train-wheels on a curved track, brake systems [1,2], and vehicle wiper blades [3]. Various empirical methods have been developed to prevent this noise. For instance, Remington set up lubrication on the contact surface suppressed squeal noise on some wheel, but did not work for other wheels [4]. Eadie used a water-based liquid material on the top of the rail and reduced squeal noise levels about 10-15 dB [5].

Brunel applied a ring damping in rail wheel comprising machined a semi-cylindrical groove under the wheel rim and inserted a metallic circular ring. This method successfully reduced the noise from 5 - 10 dB in some case, but ineffective in other cases [6]. Active noise control was implemented in suppressing squeal noise in the train cabin. Even though the idea did not prevent the squeal noise generation; it is applied to control the noise on the train car [7].

Furthermore, some ideas have been developed to suppress brake squeal. Adding mass to be attached to the backing plate was proposed to reduce the rate of squeal occurrence [8], development of state feedback control to overcome chaotic behaviours and squealing from during braking [9], and modification of lining material and lining head of the pad [10]. Abu Bakar proposed suppression of drum brake squeal through structural modifications using finite element method [11].

Recently, application of dynamics vibration absorber (DVA) in suppressing squeal noise has investigated by the authors [12]. In previous research, it is found that squeal noise is generated in the mechanism of mode coupling instability [13]. The authors investigated numerically the implementation of DVAs to prevent mode coupling instability in a squeal noise generation. It is revealed that additional of a stiff spring-mass or a rigid mass might change the characteristic of unstable mode coupling in specific condition. This method is applicable when the DVA or additional mass is set at the near-point of friction, working at the parallel direction to the friction force, and least coupling with the normal contact direction of the system.

In this paper, an experimental analysis of structural modification by adding mass in squeal noise suppression will be investigated. The analysis is conducted using experimental modal analysis to analyse the dynamics characteristic of the modified structure, and using vibration and sound analysis when the structure is exposed to friction load.

THE NUMERICAL MODEL

First, the analysis of DVA application to prevent squeal noise is modelled using a simple two-degree of freedom (DOF) of a spring-mass system [12]. The DVAs is applied to the system in two directions separately; parallel to the friction force or horizontal axis and perpendicular to the friction force or vertical axis. It is found that added mass with stiff spring or rigid mass in the horizontal direction will prevent unstable mode coupling, as long as it is isolated from influencing the dynamic characteristic in vertical axis. If the additional mass is involved in the dynamic responses of both vertical and horizontal axis, or only in the vertical axis, the instability of the system will increase and the squeal noise still occurs.

The numerical investigation then is conducted using a three-dimensional finite element model (FEM) of a steel L-shape frame with a friction contact illustrated in Figure-1 [12,14]. The frame is modelled asnine-finite-element, and the friction force is applied to nodal number 9 in the x-axis.

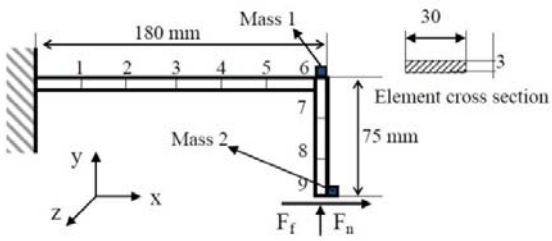


Figure-1. A simple L-shape frame for numerical and experimental mode [12].

The frequency response function of FEM model confirmed by experimental impact testingis depicted in Figure-2. The experimental FRF is obtained from the impact force at point 9 to the x-axis and vibration response at point 1 in the y-axis. There are 7 natural frequencies related to mode shape in xy-plane up to 5000 Hz. By this confirmed model, the squeal noise generation then is predicted by mode coupling instability diagram like presented in Figure-4. This figure shows the natural frequencies at friction coefficient 0.3.It is obvious that the normal contact stiffness affects to the squeal noise generation [15]. The value of the normal contact stiffness is modeled up to$1\times10^{8}\,N/m$, and the tangential contact force is accumulated into friction forces.

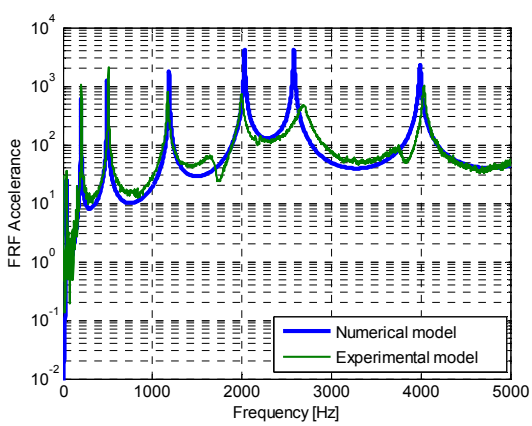


Figure-2. Frequency response function of the structure.

Figure-3 shows eigenvalues interaction presented by natural frequencies as the function of normal contact stiffness. It is found that two eigenvalues (mode 6th and mode 7th in xy-plane system) are merged and produce a unstable mode coupling at frequencies range around 1900 and 2200 Hz within the normal contact stiffness $1\times10^{7}-4\times10^{7}$ N/m that depends on the friction coefficient. Some natural frequencies are shown constantly to the normal contact stiffness because the natural modes work in the z-axis of three-dimensional frame model.

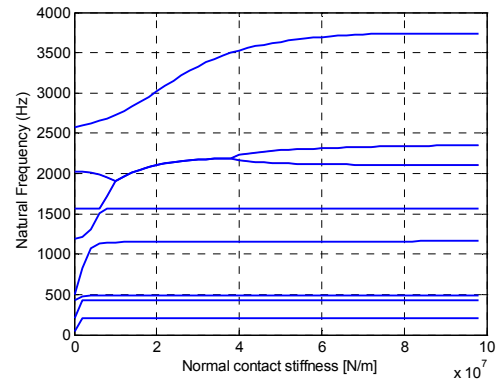


Figure-3. The natural frequencies of L-shape frame as the function of normal contact stiffness at friction coefficient of 0.3.

In experimental observation, when a stiff slider bar moves into x-direction and scratches point 9 of the L-shape structure (Figure-1), the squeal noise is emitted during the sliding. The vibration response at point 1 and the sound pressure level of the squeal noise are depicted in time and frequency domain in Figure-4. The squeal noise is found comprising only a few spectra at 2009 Hz (fundamental harmonics), 4017 Hz (second harmonics) and 6026 Hz (third harmonics).

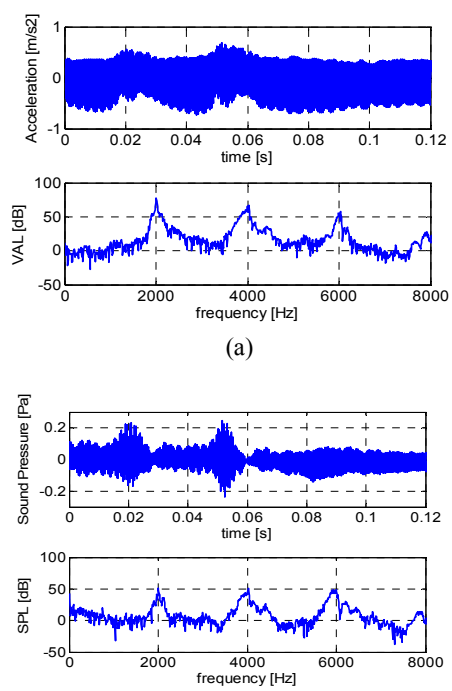


Figure-4. Time response and frequency spectrum during squeal; (a) vibration response at point 1, (b) Sound pressure level near point 9.

The numerical analysis in two-degree of freedom model suggests that added mass with stiff spring or added rigid mass in the x-axis or parallel to friction force is possible to remove unstable mode coupling [12]. The next discussion will show the analysis of mass addition application. The particle rigid mass is added to the point number 6 and 9 of the L-shape frame in (Figure-10). In finite element, model the mass is working in three translation nodes. Added mass is varied from 5 grams and 10 grams. Figure-5and Figure-6 shows the effect of the additional mass to the unstable mode coupling between mode 6th and 7th. It is shown that additional mass of 5 grams and 10 grams at point 6 change the natural frequency and the mode coupling possibility of the 6th and 7th modes insignificantly (Figure-6). The squeal noise will still occur by this way and the possibility will increase at higher normal contact.

Moreover, Figure-6 shows that additional mass of 5 grams and 10 grams at point 9 have a significant effect to prevent the mode coupling possibility of the 6th and 7th modes. Adding mass 7 grams and 10 grams totally suppress the possibility of squeal noise generation. It means that the mode coupling of the 6th and 7th modes are more sensitive to the mass addition at point 9 rather than the additional mass at point 6.

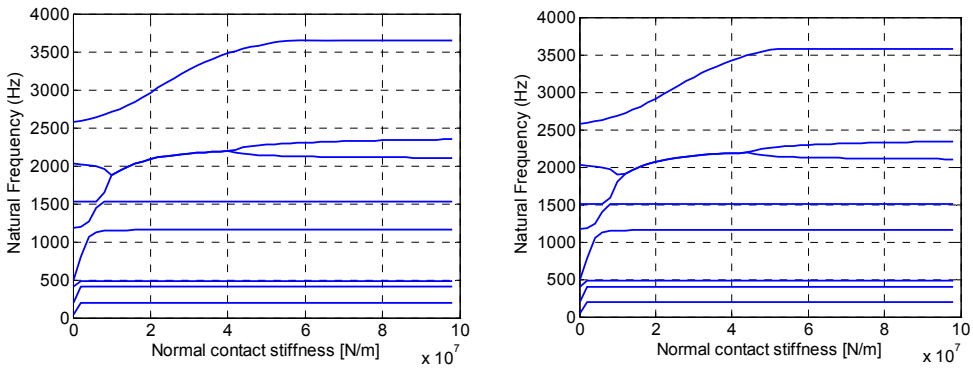


Figure-5. Natural frequencies with added mass on point 6; 5 grams (left), 10 grams (right).

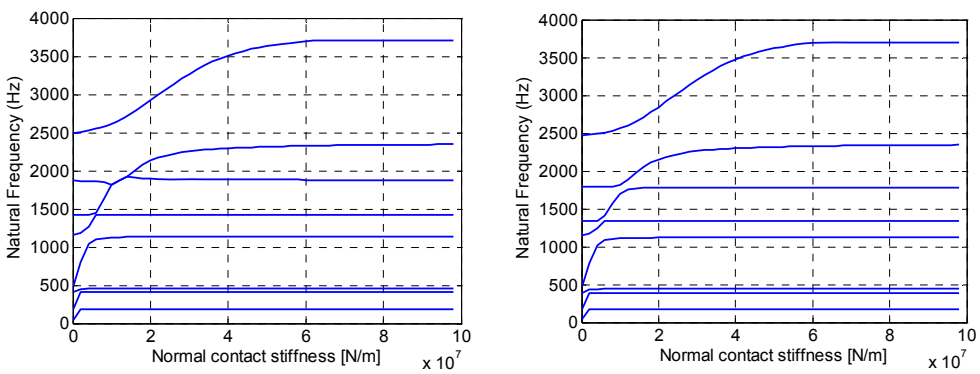


Figure-6. Natural frequencies with varied added mass on point 9; 5 grams (left), 10 grams (right).

EXPERIMENTAL INVESTIGATION

An experiment is carried out using a simple L-shape frame corresponding to the model in Figure-7. The free-side tip of the frame is scratched by a rigid sliding device with a constant speed in the horizontal axis (y-axis in Figure-7). When accelerometer is used to measure FRFs on a test structure, it affects the dynamics of the structure as additional mass and damping. Since the experimental specimen in this study has relatively small mass, this effects appear in some degrees of difference by lowering the natural frequency at measurement points, by changing the magnitude of FRFs and by raising the damping ratio at high resonance frequencies.

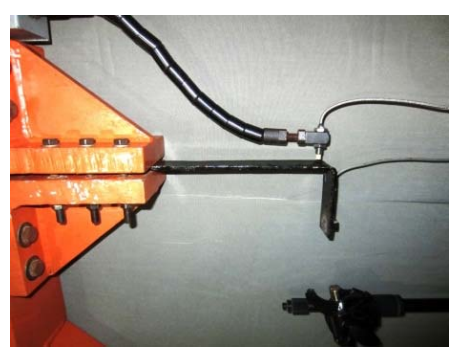


Figure-7. A simple L-frame model for experiment.

First, a particle mass is added to point 6. Based on the initial prediction numerically, it is found that the sensitivity of the natural frequency of the structure to the mass addition is low; so the additional mass of 10 grams gives a small contribution to the change of natural frequencies and squeal noise generation possibility (Table-1). By this case, the additional mass is varied from 5 grams to 40 grams. The FRFs of the structure is shown in Figure-8. Some that the additional mass of 5 to 30 grams gives insignificant change to the natural frequency. It is shown in Figure-8 and Table-1 that the natural frequency changes insignificantly by mass addition, especially at the natural frequency about 1995 Hz, where the squeal noise is generated firstly.

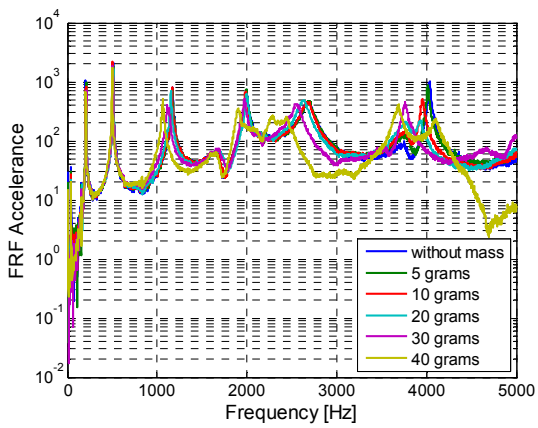


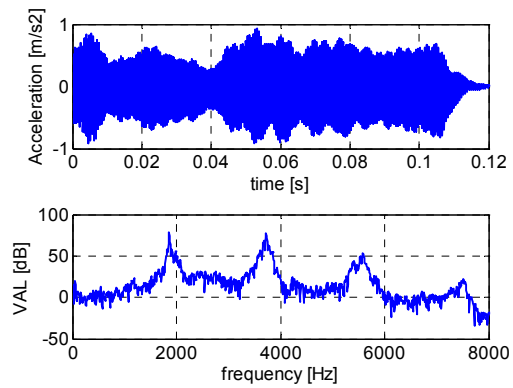
Figure-8. Frequency response functions with additional mass at point 6.

Then, an experiment is conducted to generate squeal noise. The specimen of L-shape structure is loaded by friction in the horizontal direction at point 9. A slider bar has relatively much more rigid than the specimen and scratches the structure at the measurement point 9. Sliding speed is adjusted manually approximately about 5 cm/s in the x-direction. The acceleration response at point 1 and the sound pressure level without additional massin time and frequency domain is depicted in Figure-9. The graphs in Figure-9 show that the squeal noise comprises only a few spectra at 2006 Hz (fundamental harmonics), 4017 Hz (second harmonics) and 6026 Hz (third harmonics).

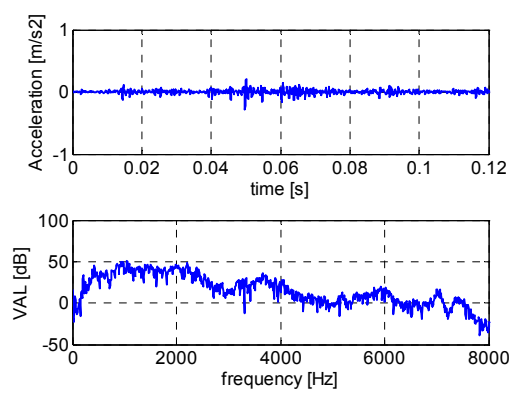
Table-1. Natural frequency and squeal frequency with various masses at point 6.

Additional mass (grams)	**Natural frequency 6** (Hz)	**Squeal frequency** (Hz)
0	1995.3	2009
5	1995.3	1915
10	1990	1927
20	1995.5	1861.5
30	1975	1830
40	1903.7	No squeal

The experiments then are conducted by when additional mass is at point 6. Table-1 shows the frequency of squeal by various additions mass. Although the nearest natural frequency of the structure to the squeal noise (1995.3 Hz) has no significant change by additional mass, the frequency of squeal decreases significantly. The squeal noise is generated at the almost equal sound pressure level at a slightly different frequency according to the additional mass. However, when the mass 40 grams is added at point 6 the vibration response become stable and the squeal noise disappears. The vibration response in 20 grams and 40 grams additional mass are depicted in Figure-9. It is observed the sound pressure level decrease significantly and the squeal frequency spectrum disappears. It means that squeal noise has been suppressed.



(a)



(b)

Figure-9. Vibration response by additional mass (a) 20 grams, and (b) 40 grams.

Similarly, the particle mass then is added to point 9. Based on initial prediction numerically, it is found that the sensitivity of natural frequency of the structure to the mass addition is relatively higher that point 6 one; so the additional mass of 10 grams shifts natural frequencies and reduces the squeal noise generation possibility. It is shown in Figure-6, by addition 10 grams at point 9 the squeal noise is predicted suppressed because the additional mass can prevent unstable couple modes. By this case, the

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additional mass is varied from 2 grams to 10 grams. The FRFs of the structure is shown in Figure-10 and Table-2 show that the additional mass of 2 to 8 grams changes the natural frequency significantly especially at the nearest natural frequency to the squeal noise spectrum.

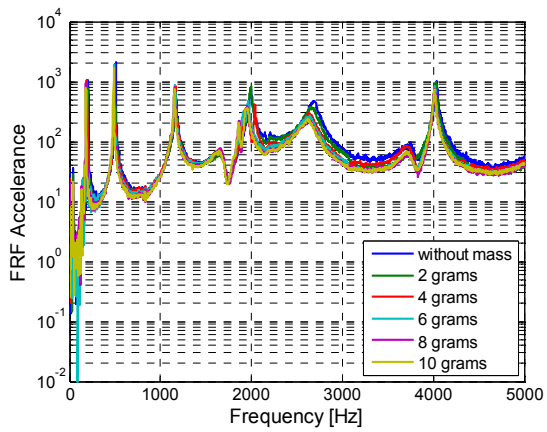

Figure-10. Frequency response function with additional mass on point 9.

Now, experiments are conducted to obtain squeal noise. Table-2 shows the frequency of squeal by various additions mass. The natural frequency of the structure and the squeal noise frequency change easily by the additional mass. The frequency of squeal decreases significantly. The squeal noise is generated at an almost equal sound pressure level at a different frequency according to the additional mass for 2 grams to 8 grams mass. However, when the mass 10 grams is added at point 9 the vibration response become stable and the squeal noise disappears. The vibration response by 4 grams additional mass for instance and vibration response by 10 grams additional mass are depicted in Figure-11

Table-2. Natural frequency and squeal frequency with various mass at point 9.

Additional mass (grams)	**Natural frequency 6** (Hz)	**Squeal frequency** (Hz)
0	1995.3	2009
2	1993.75	1905
4	1951.5	1930
6	1962.5	1968
8	1954.7	1850
10	1948.5	No squeal

Numerical analysis of squeal noise prediction by mass addition is shown in Figure-12 and Figure-13. Figure-12 shows that the two-gram added mass on point 9 changes the mode coupling between mode 6th and 7th, restricting it to occur within the normal contact stiffness of $1\times10^7 - 3\times10^7$ N/m at frequency range 1900 to 2200 Hz corresponding to normal contact stiffness. Hence, the possibility of unstable mode coupling occurrence has been reduced. As indicated by Figure-12, the sixth-gram added mass almost eliminates the possibility of squeal noise by reducing the region of instability to that within the normal contact stiffness of $1\times10^7 - 1.3\times10^7$ N/m. Finally, attaching ten-gram added mass can entirely suppress the possibility of squeal noise generation at those modes.

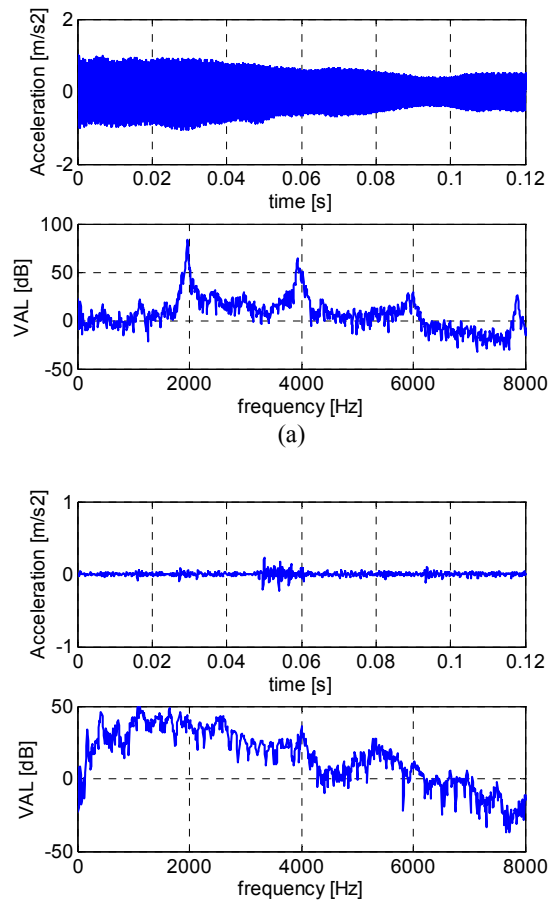

Figure-11. Vibration response by additional mass; (a) 6 grams and, (b) 10 grams.

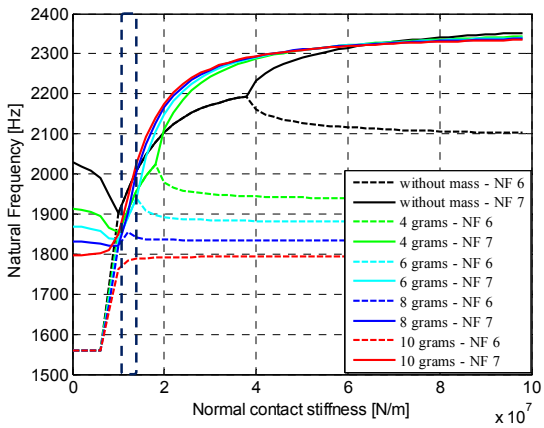

Figure-12. Numerical analysis of squeal noise prediction by mass addition at point 9.

In the next, a particle rigid mass is attached to point 6 of the L-shape frame. The added mass may vary from 10, 20, to 40 grams. In Figure-13, it was shown that by adding a ten-gram mass on point 6, mode coupling between mode 6th and 7th still occur at within frequency of 1850 to 2150 Hz and a wider range of normal contact stiffness of $1\times10^7 - 12.5\times10^7$ N/m. The possibility of occurrence of unstable mode coupling to generate squeal noise is increased. Also, adding bigger masses (20, 30 and 40 grams), raises the possibility to generate the unstable mode coupling. However, the normal contact stiffness where the instability occurs is shifted to higher normal contact stiffness. It can explain why the instability is preventing when additional mass is 40 grams.

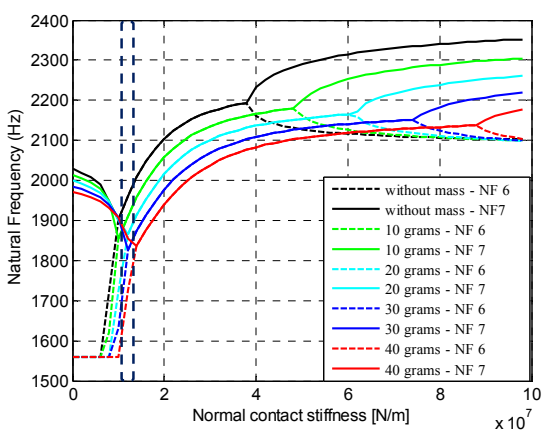

Figure-13. Numerical Analysis of squeal noise prediction by mass addition at point 6.

Based on the investigation by Hoffman [16,17], it was found that energy can be transferred from the frictional to the vibratory system due to a simple mechanism: the normal force oscillations induce the oscillations of the tangential friction force. If there are simultaneously relative tangential displacements at the friction interface not exactly in phase with the force oscillations, a cyclic growth of the vibrational energy may occur. Moreover, the friction force acts like a structural coupling force linking the normal motion to tangential motion and that instability possibly exists if these friction-induced cross-coupling forces balance the corresponding coupling forces of the structure.

The added mass on point 6 moves flexibly in the vertical direction but very difficult in the horizontal direction, because of the structural rigidity. Therefore, the added mass leads to increase of the normal force oscillation. The added mass in the horizontal direction at the near-point-of-friction absorbs more energy to move the structure and added mass. The added mass on point 9 reduces the possibility of instability occurrence. The added mass on point 9 moves flexibly in the horizontal direction. Figure-14 shows the 6th and 7thmode shapes of L-shape frame. The 6th and 7th mode shapes that make a couple-mode when to generate squeal noise have high eigenvectors to the horizontal direction and low eigenvector to the vertical direction in point 9, despite point 6 with much lower eigenvector in vertical and horizontal direction. It means that additional mass in point 9 will give more effect to reduce the velocity or vibration response because by increasing the mass the systems will need more kinetic energy to have a high magnitude. Therefore, higher friction forces or higher friction coefficients are needed to link the out-of-plane motion to in-plane motion.

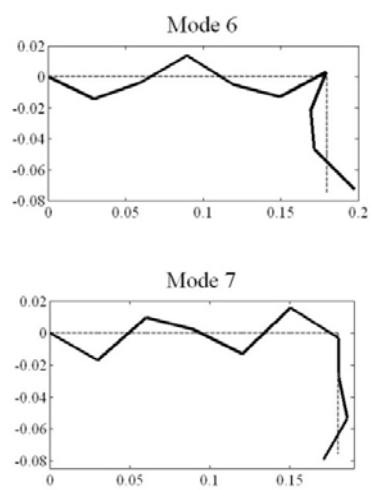

Figure-14. 6th and 7th mode shapes.

CONCLUSIONS

Applying structural modification by additional mass in suppressing squeal noise has been investigated. It was shown that by adding a mass to the point will shift the instability region of squeal noise generation. It is found by adding a mass at a point where moves flexibly in vertical direction or perpendicular to friction force will increase the possibility of squeal noise incident, but in restricted normal contact stiffness the squeal noise will be cancelled. The additional mass at the near-point-of-friction that can move flexibly in the horizontal direction or parallel to the

friction force has the possibility to avoid unstable mode coupling by moving the unstable region to one with higher normal contact stiffness. Therefore, an added mass will prevent unstable mode coupling although long coupled from the vertical direction. The additional mass will give more effect to reduce the velocity or vibration response because by increasing the mass the systems will need more kinetic energy to have a high magnitude. Therefore, higher friction forces or higher friction coefficients are needed to link the out-of-plane motion to in-plane motion.

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