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Development of a test system to measure squeak propensity of materials used in vehicle underbody components

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ABSTRACT

Squeak is a noise generated by a large-scale, unstable vibratory motion between two surfaces in sliding contact; therefore, squeak properties of materials should be measured while they are in such a condition. A test apparatus with a built-in instability mechanism was developed in our previous work, which was designed to induce a large-scale unstable vibratory motion between two specimens, then measure the noise generated under that condition. In this work, a new test system was developed based on the original design to measure squeak propensities of materials used in underbody components of a vehicle, such as a suspension or chassis part. The test system can adjust the contact force and the sliding velocity at desired levels to simulate typical loading conditions of underbody components. The system also has a computer algorithm that rates the measured noise by reflecting human hearing characteristics and psychoacoustics. A finite element model was developed to study characteristics of motion induced by the test apparatus that leads to squeak noises focusing on the effects of the contact force and the sliding velocity to decide the proper operating parameters to be used for the test of underbody components.

Keywords: Squeak, Instability, Finite Element Analysis, Test System, Sprag-slip
I-INCE Classification of Subject Number: 76

1. INTRODUCTION

Due to the effort of reducing persistent noises such as engine and wind noise, the noise, vibration and harshness (NVH) performance of passenger cars has been significantly improved. Because of this, intermittent noises, such as a squeak noise, stand

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out and form detrimental perception of the quality of a vehicle. Squeak is an annoying, undesirable noise generated by large-scale, unstable vibratory motion between two surfaces in sliding contact.[1] In order to quantitatively measure the squeak propensity of materials pairs, the authors previously designed and built a unique squeak test system[2-7] for interior components of a vehicle by employing a sprag-slip mechanism[8]. In this study, a new squeak test system to measure squeak propensities of materials used in underbody components of a vehicle, such as a suspension and chassis part was developed by modifying the previous version of the squeak test system. The new test system can adjust the contact force and the sliding velocity to simulate general loading conditions of underbody components.

A finite element (FE) model of the squeak test apparatus, which is a bent cantilever beam in contact with a flat moving surface, was developed to investigate characteristics of the motion of the test apparatus under typical loading conditions of underbody components of a vehicle and study underlying mechanism of the squeak generation. Using the FE model, the stability of the system was studied by the complex eigenvalue analysis (CEA) in the frequency domain focusing on the effect of the design parameters on the stability of the system. In addition, dynamic transient analysis (DTA) was used to calculate the response of the system in the time domain. The test system developed in this work can be used in conjunction with an automatic squeak detection/rating algorithm previously developed by the authors to build the material database of squeak propensity that can be used by automotive NVH engineers.

2. FINITE ELEMENT MODEL OF THE SQUEAK TEST SYSTEM

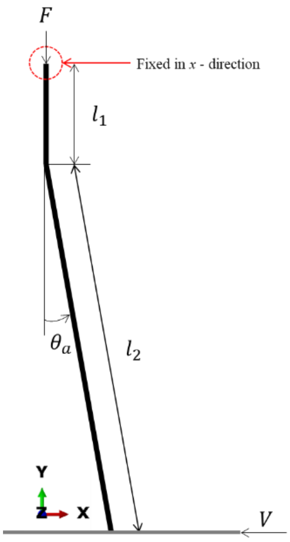


Fig. 1 Finite element model of a squeak test system

The FE model of the system is shown in Fig. 1. 1-D beam elements are used for the cantilever beam comprised of two segments of length l_1 and l_2 with attack angle θ_a. The material property and dimensions of the model are summarized in Table 1. The top end of the bent beam can move only in *y*-direction by constraining the motion in *x*-direction. A normal force *F* is applied at the top end of the beam in *y*-direction. The tip of the beam contacts with the rigid surface that moves in *x*-direction with the velocity *V*.

Table. 1 Material properties and dimension of the FE model

Mass Density	2700 kg/m^3
Young's Modulus, E	70 GPa
Poisson's Ratio, ν	0.33
Total Length of the Beam	$l = l_1 + l_2 = 80$ mm
θ_a	8 deg
Width	20 mm
Thickness	5 – 8 mm
Friction Coefficient, μ	0.3
Velocity of the Moving Plate, V	5 mm/s
Applied Normal Force, F	500 N

3. STABILITY ANALYSIS

3.1. Complex Eigenvalue Analysis

In order to investigate the stability of the given system, a two-step approach is used. At first, the complex eigenvalue analysis (CEA) is used for evaluating the stability condition.

$$\lambda_n = \sigma_n + j\omega_n \tag{1}$$

where, λ_n is the complex eigenvalue of the system, σ_n is the damping factor and ω_n is the natural frequency of the system. Then,

$$\zeta_n = -\mathrm{Re}(\lambda_n)/\left|\lambda_n\right| \tag{2}$$

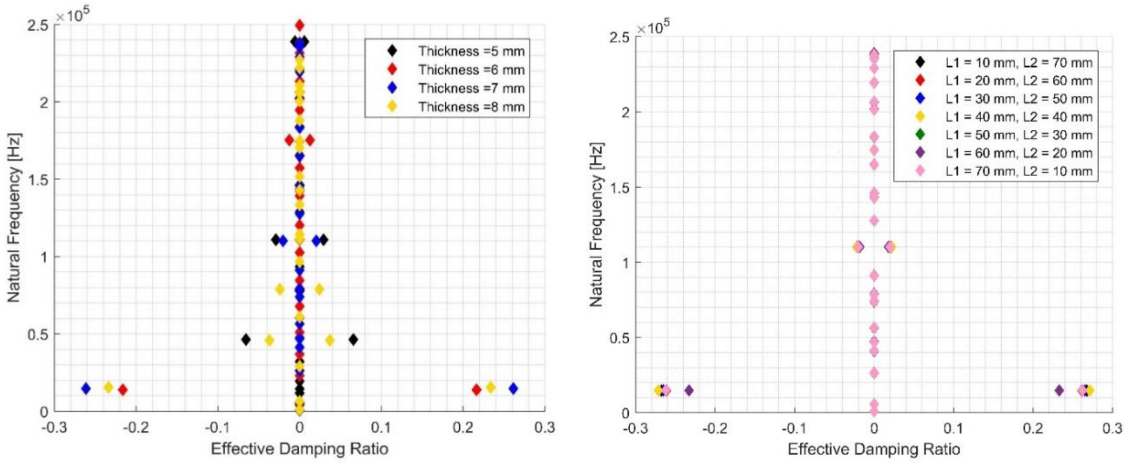
where, ζ_n is the effective damping ratio of the system. The system becomes unstable when the ζ_n becomes negative, that is when the real part of the eigenvalue is positive.

Because of the requirement to test squeak propensity at higher normal load, the thickness of the beam had to be increased. The change of the instability condition due to the thickness of the beam was investigated when l_1 and l_2 were kept constant at 20 mm and 80 mm respectively. The eigenvalues obtained from the CEA with respect to the thickness of the beam are shown in Fig. 2(a). Table 2 shows the mode that becomes the first and the corresponding natural frequency. When the thickness of the beam is 5 mm, the first unstable mode is the 7th mode whose natural is 46,398 Hz. When the thickness was 6, 7 or 8 mm, the 3rd mode was the mode that becomes unstable first.

The instability condition affected by the proportion of the bent length of the beam is investigated by changing l_1 while keeping the total length ($l = l_1 + l_2$) the same. The eigenvalues obtained from the CEA with respect to the length of the beam is shown in Fig. 2(b). The unstable mode of all cases was identified as the same mode and the corresponding frequency changed very little, indicating that the stability of the system is not affected by the portion of l_1 if the total length of the beam is kept the same.

Table. 2 First unstable mode and corresponding natural frequency for each thickness

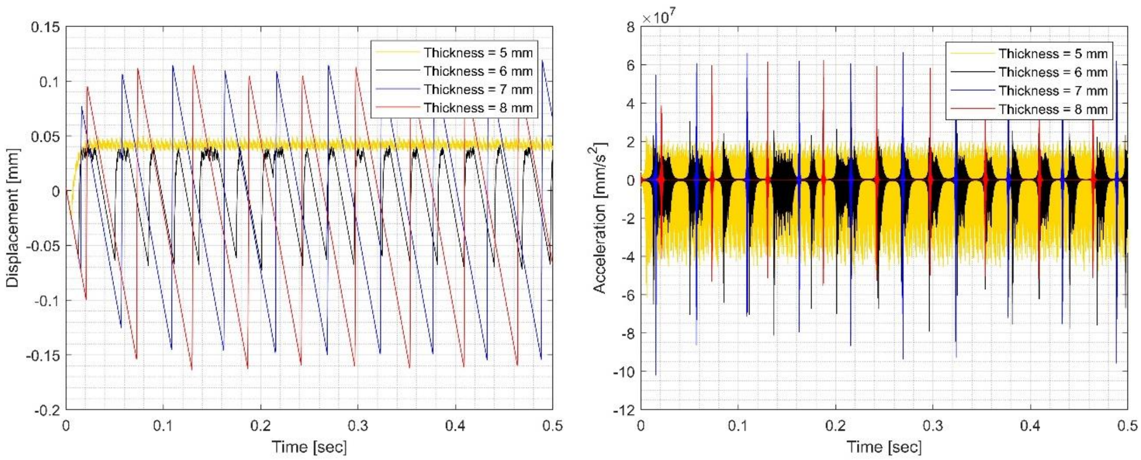
Thickness (mm)	First Unstable mode	Natural Frequency (Hz)
5	7	46398
6	3	13867
7	3	14679
8	3	15477



(a) *(b)*

Fig. 2 Eigenvalues of the FE model of the squeak test apparatus with respect to (a) thickness of the beam and (b) length of the beam

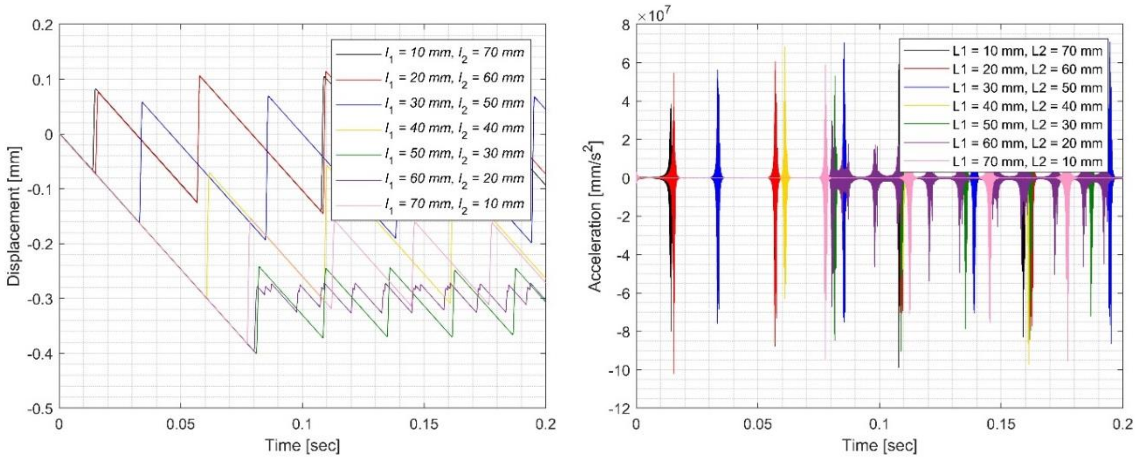
3.2. Dynamics Transient Analysis



(a) *(b)*

Fig. 3 Time domain response at the tip of the beam with respect to the thickness. (a) Displacement and (b) acceleration

The response of the beam was obtained by dynamics transient analysis (DTA) of the FE model numerically in time domain. The displacement and acceleration time histories at the tip that is in contact with the rigid moving surface are shown in Fig. 3. When the thickness of the beam is 5 mm, the response shows fast repetitions of sprag and slip stages. When the thickness is 6 mm or higher, the sprag and slip stage can be observed very clearly. During the sprag stage, the tip of the beam sticks to the bottom surface moving together. When the elastic recovery force of the beam due to the deformation in the sprag stage increases to exceed the friction force, the tip starts to slip. During the slip stage, the motion shows a high frequency oscillatory motion with very large amplitude as shown in Fig. 3(b). Therefore, the response of the system forms a large-amplitude limit cycle which leads to a squeak noise. In addition, the response of the system with respect to the length of the beam was also calculated as shown in Fig 4. For all cases, the sprag and slip motion can be observed clearly, therefore a large-amplitude limit cycle and squeak noise is expected.



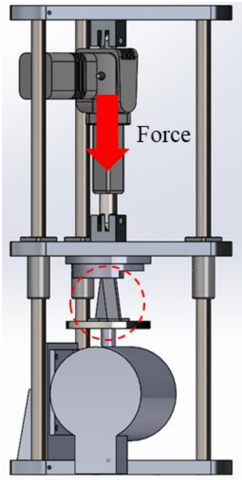
(a) *(b)*

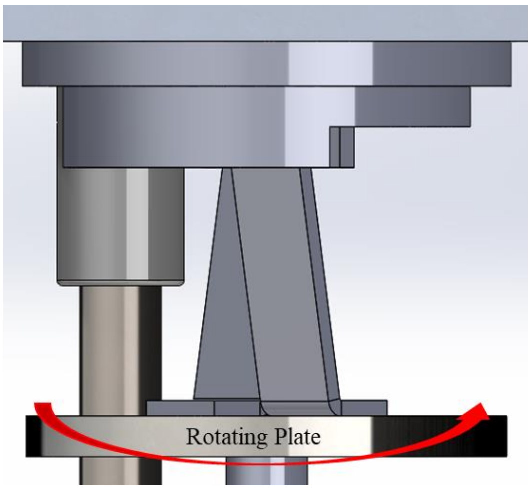
Fig. 4 Time domain response at the tip of the beam with respect to the length of the beam (l_1 and l_2). (a) Displacement and (b) acceleration

4. DESIGN OF THE SQUEAK TEST SYSTEM

The squeak test system was designed for the purpose to measure squeak propensity of materials used in underbody components of a vehicle. Fig. 6 (a) shows the 3D model of the squeak test system and Fig. 6 (b) shows the bent beam fixed to the frame of the test device. One specimen is attached to the tip of the bent beam and the other specimen is mounted on the rotating surface at the bottom. An electro-magnetic actuator applies the normal contact force between two specimens up to 3000 N, the maximum normal force of the suspension parts subjected in a typical passenger vehicle. The rotating surface at the bottom is driven by a DC motor that can set the sliding velocity at a desired speed.

An accelerometer attached to the tip of the beam is to measure the spag-slip motion the beam. The normal force is measured by a loadcell located between the actuator and fixed-end of the beam. A microphone is installed at a point close to the contact region to record sound pressure of generated squeak sound.





(a) (b)

Fig. 6 (a) 3D model of a squeak test system, (b) Zoom-in of the bent beam fixed to the frame

5. CONCLUSION

A new test system for measuring squeak propensities of materials used in underbody components of vehicle was designed. The system has a built-in instability mechanism, a sprag-slip mechanism, to induce friction-induced, self-excited vibration with large-amplitude between two surfaces in sliding contact so that squeak propensity of a par of materials can be measured at actual "squeak condition". The squeak test system is composed of the bent beam with a specimen fixed to the stationary frame and the other specimen mounted on a rotating frame. In order to design the system for the intended purpose, measurement of under-body parts, a FE model was developed. The effect of the major design parameters, the thickness and the length of the beam, on the stability was studied to ensure the system can always produce squeak conditions. Dynamic responses of the system were also studied using the FE model and the complex eigenvalue analysis and dynamic transient analysis under typical loading conditions of underbody components of a vehicle. The results clearly show that the test system can induce the large-amplitude limit cycle composed of sprag and slip stages, thus will generate squeak noises if the thickness of the bent beam is 5 mm or thicker. Therefore, the beam can be designed so that it can induce squeak noises for most material pairs under the given loading condition as well is strong enough to sustain the large load.

Utilizing what was learned from the analytical study, a squeak test machine was designed to measure the squeak propensity of material pairs under various loading condition. The machine can apply a variable normal contact force between two specimens, control the sliding velocity between them, and measure the time history of the sound pressure of the squeak noises. The measured noise is processed by the automatic squeak detection/rating algorithm previously developed by the authors[9-10] to obtain the squeak propensity of any given material pairs, which can be used to build a material database for automotive NVH engineers. The database can guide selection of material pairs for places with potentially serious squeak problems.

6. ACKNOWLEDGEMENT

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