MODELING AND SIMULATION OF AUTOMOTIVE WIPER NOISE AND VIBRATION USING FINITE ELEMENT METHOD

I M Awang, **C Y Leong**, **A R Abu-Bakar**, **B Abd-Ghani**, **R Abd-Rahman**, **and M Z Md-Zain**

Faculty of Mechanical Engineering
Universiti Teknologi Malaysia
81310 UTM Skudai, Johor
Email: arahim@fkm.utm.my

Abstract

As modern passenger cars become increasingly quieter, wiper operation vibration and noise become more noticeable. As a result of the market information analysis, most complaints about the wiper concern operation noise. Wiper vibration and noise is classified into three main categories namely, squeal noise, chattering, and reversal noise. Squeal noise is a high-frequency vibration of about 1000 Hz. Chattering noise is a low-frequency vibration of 100Hz or less and reversal noise is an impact sound with a frequency of 500 Hz or less produced when the wiper reverses. This paper presents numerical studies on noise and vibration of an automotive wiper blade. A 3-dimensional (3D) finite element (FE) model of a wiper blade assembly is developed and then validated at the component level using modal analysis. Complex eigenvalue analysis available in ABAQUS is employed to determine stability of the wiper blade assembly. It is found that predicted results from complex eigenvalue analysis are fairly close to those generated in the experiment.

Keywords: wiper; noise & vibration; finite element; complex eigenvalue; modal analysis

1. Introduction

A windscreen wiper is indispensable device used to wipe rain and dirt from a windscreen. Today, almost all automobile are equipped with windscreen wiper, often by legal requirement. Clear vision for the car driver is an important prerequisite for safety in road traffic. The wiper faithfully keeps the windscreen clear, moving back and forth across the windscreen countless times as they sweep the water away. Traditional windshield wipers are actuated by a single constant speed motor related to the wipers by a system of connecting rods, often called the wiper arm (Fig. 1).

A wiper generally consists of an arm, pivoting at one end and with a long rubber blade attached to the other. The blade is swung back and forth over the windscreen, pushing water from its surface. The speed is normally adjustable, with several continuous speeds and often one or more intermittent settings. There are generally three speeds: fast, slow and intermittent, whose selection made by the driver.

It is often that the wiper system generates unwanted noise and vibration. Noise and vibration in the wiper system can be classified into three groups, namely, squeal noise, chattering and reversal noise. Squeal noise, sometimes called squeaky noise, is a high-frequency vibration of about 1000 Hz. Chattering or beep noise, is a low-frequency vibration of 100Hz or less. Reversal noise is an impact sound with a frequency of 500 Hz or less produced when the wiper reverses. These types of noise and vibration phenomenon lead to visual and audible annoyance for the driver and passengers [1].

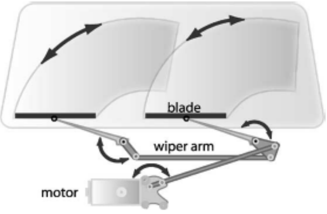


Fig. 1. Wiper system on a windscreen[2]

Numerous studies, using numerical and/or experimental approach, have been carried out to investigate noise and vibration of an automotive wiper system. They studied dynamic analysis of blade reversal behaviour using a 2-dimensional (2D) mechanical model of a wiper system and a spring-mass model of an arm and blade [3]. Extended studies considering a complete 3D model were also performed. Comparison between 2D and 3D model for the arm and blade was made and the result suggested that the 3D model could simulate the reversal behaviour of the wiper system more accurately than 2D model [4].

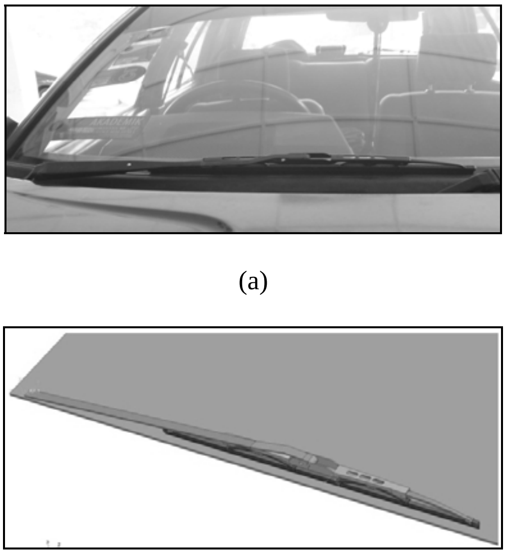
The investigation of squeal noise reduction using a mathematical model has been proposed [1, 5]. From the proposed model, material physical properties and design of the blade were varied. Experiments on squeal noise were also carried out to verify the effectiveness of the proposed material and design changes. A combined approach to study chatter vibrations for a wiper system has been performed [6]. Wiper motion tests were carried out on a developed test rig. Different attack angles and pressure were used and their effect on the wiper motion was observed. The study also developed a 2D mathematical model to demonstrate the influence of the geometrical configuration of the wiper system on the generation of unstable motion.

The employing of dither control to stabilize squeal noise in the wiper system also has been studied [7]. A FE model was developed in order to support the optimization of the control configuration. The study showed that with a proposed dither control, wiper squeal noise was effectively suppressed. The developing of a FE model to study dynamic instability of a flexible wiper system also has been developed [8]. The FE model was validated by experimental tests with different value of arm forces and attack angles of a rubber blade. The results of the study show that the predicted instabilities were close to those obtained in the experiments.

From the previous study, none of existing works studied noise and vibration of an automotive wiper using FE method particularly through complex eigenvalue analysis. This paper presents numerical studies on noise and vibration of an automotive wiper blade. A 3-dimensional finite element model of a wiper blade assembly is developed and then validated at the component level using modal analysis. Complex eigenvalue analysis available in ABAQUS is employed to determine the stability of the wiper blade assembly.

2. Development of Wiper Model

A detailed 3-dimensional FE model of a Proton windscreen wiper assembly is developed using Solidworks modeling software. Fig. 2(a) and 2(b) show a real wiper design and its FE model respectively. The FE model consists of a windscreen, a rubber blade, primary yoke and a wiper arm as shown in Fig.3. The windscreen is simplified as a flat surface in the FE model in order to avoid convergence issue. The assumption of flat surface is made based on the previous studies [6, 9].



(a)

(b)

Fig. 2. Windscreen wiper; (a) an actual wiper (b) FE model

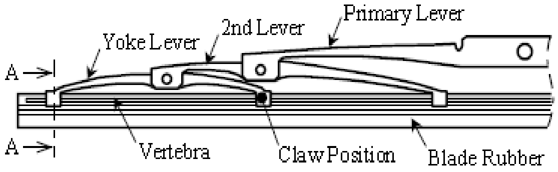


Fig. 3. Structure of wiper blade[3]

2.1 Validation of FE model

An experimental modal analysis (EMA) was performed on the individual wiper components in order to determine their normal mode response [10]. The EMA was performed at free-free boundary condition for the rubber blade, primary lever and yoke and secondary lever, whilst at fixed boundary condition for the windscreen.

The impact hammer test method is used to obtain natural frequencies of those components (Table 1). In doing so, a Kistler type 9722A500 impact hammer is used to produce the excitation force while a Kistler Type 8636C50 uni-axial accelerometer is fix-mounted onto the tested components [11]. The results showed very good correlation between predicted and measured natural frequencies for all wiper components. This is obtained by tuning Young’s modulus and the density value of the wiper components. The updated material data of the wiper components are given in Table 2.

Table 1. Correlation of experimental modal analysis with FE analysis

Component	Mode No	Experimental Frequencies (Hz)	FE Analysis (Hz)
Windscreen	1	219	207
	2	242	248
	3	308	319
Rubber blade	1	61	42
	2	153	152
	3	286	281
Primary lever	1	247	247
Levers (second and yoke lever)	1	723	705
	2	835	857

Table 2. Material properties of wiper components

Component	Density (kg/m^3)	Young's Modulus (GPa)	Poisson's Ratio
Windscreen	2500	8.7	0.25
Rubber blade	1000	7.1e-3	0.49
Primary lever	7981	242.8	0.29
Levers	7981	464	0.29

2.2 Assembly Model of Wiper Components

The next stage is to bring all components together to form an assembly model. A combination of tie element and self-contact/surface to surface contact element are used to represent contact interaction between wiper components and windscreen/rubber blade interface respectively. Table 3 shows details of windscreen wiper couplings that are employed in the FE model assembly.

Table 3. Windscreen wiper model couplings

No.	Connections	Type of connection
1	Windscreen-Rubber blade	Surface to surface
2	Neck-shoulder	Self contact
3	Rubber blade-Yoke lever	Tie
4	Yoke lever-Second lever	Tie
5	Second lever-Primary yoke lever	Tie
6	Primary yoke lever-Wiper arm	Rigid body

3. Experiment of Noise and Vibration on Wiper

The measurement set-up is shown in Fig. 4, where a Kistler Type 8794A500 tri-axial accelerometer is attached to the primary yoke. The wiper used in the experiments is of the uni-blade type that typically found in the PROTON cars. Noise and vibration measurements of the wiper were carried out at wet environmental conditions and were measured at two different average speeds of 1.8 and 2.5 rad/s [12].

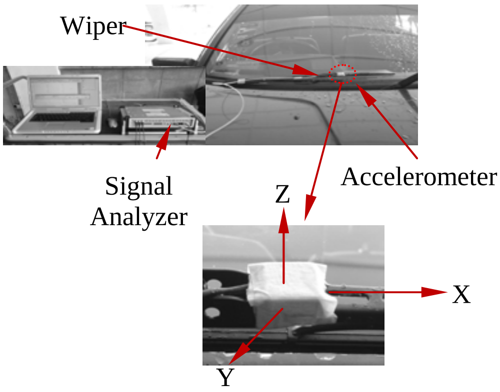
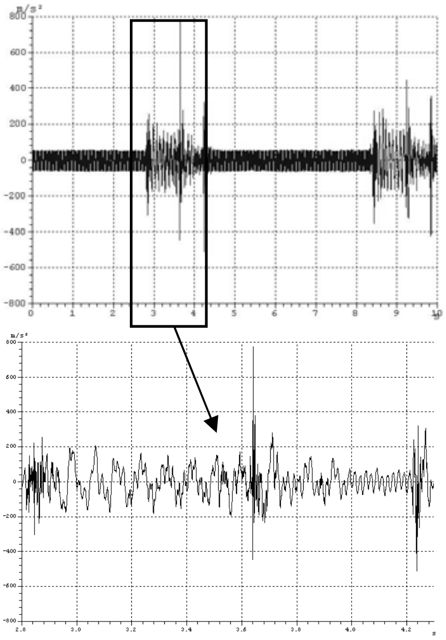
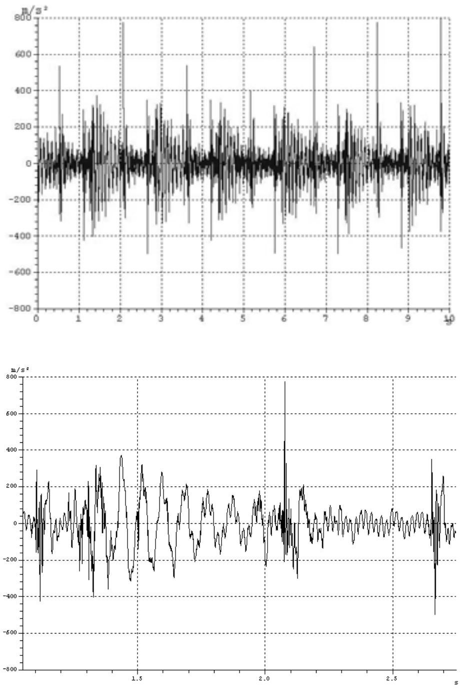


Fig. 4. Setup for noise and vibration measurement

For a rotational speed of 1.8 rad/s, acceleration response is shown in Fig. 5(a). It is seen that high vibration amplitude occurred rightly at the beginning and end of the wiper stroke. This may due to two reasons: stick-slip and/or negative velocity-friction characteristic mechanisms [6]. There is less idle time after one complete stroke for rotational speeds of 2.5 rad/s (Fig. 5(b)), compared to 4s idle time for rotational speed of 1.8 rad/s.



(a) Average speed of 1.8 rad/s at Z direction



(b) Average speed of 2.5 rad/s at Z direction

Fig. 5. Acceleration responses during wet condition

The above study suggests that the results are concurred with the findings of Goto et.al [1], where the study stated that noise could easily be generated before and after wiper stroke as shown in Fig. 6.

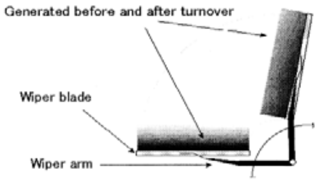
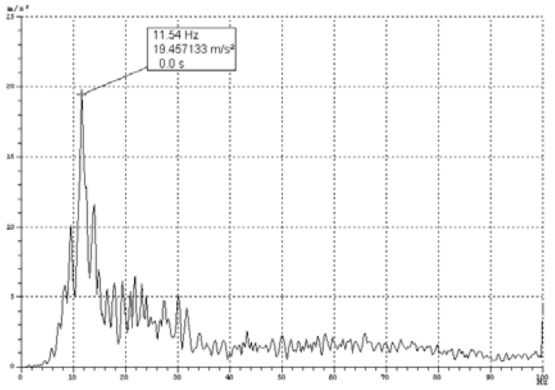
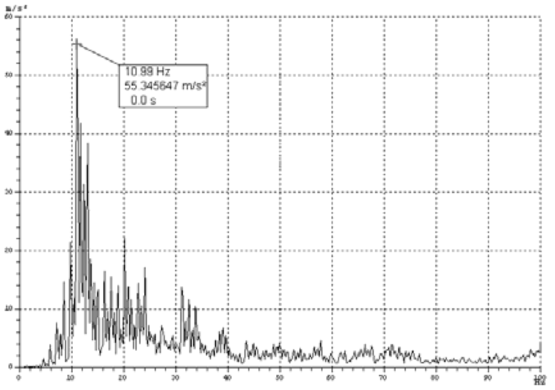


Fig. 6. Location of noise generation [1]

From acceleration responses in Fig. 7(a), it is found that at the speed of 1.8 rad/s, the noise is dominated at frequency around 12 Hz. For speed at 2.5 rad/s as depicted in Fig. 7(b) noise is generated at dominant frequency of 11 Hz. This has similar trend with previous speed. It seems that the noise frequency of the wiper is almost identical. From those measured frequencies, it can be said that current wiper system is experiencing chatter noise.



(a) Average speed of 1.8 rad/s



(b) Average speed of 2.5 rad/s

Fig. 7. Noise frequencies measured from the rubber blade

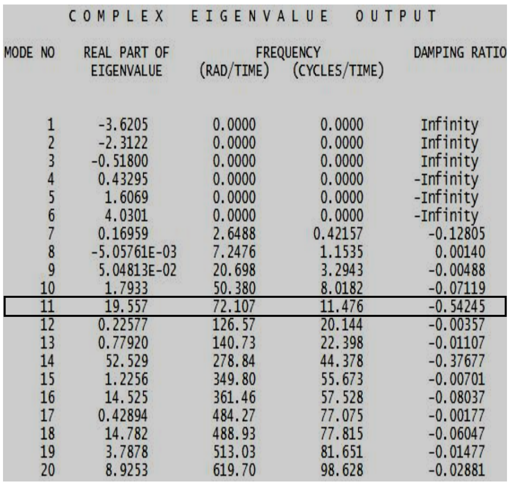
4. Complex Eigenvalue Analysis

This section focuses on prediction of unstable analysis and describes wiper vibration characteristic at a system level. A preferred, complex eigenvalue analysis method is used to predict the unstable frequency. The complex eigenvalue are solved using the subspace projection that is available in ABAQUS.

In order to perform the complex eigenvalue analysis using ABAQUS, four main steps are required [12].

4.1 Simulation Result

Complex eigenvalue analysis defines instability of the system by positive real parts. Fig. 8 presents the results obtained from complex eigenvalue analysis. It is found that the frequencies in two different speeds are almost identical. This suggests that the rotational speed of the wiper may not influence the noise generated in the wiper system.

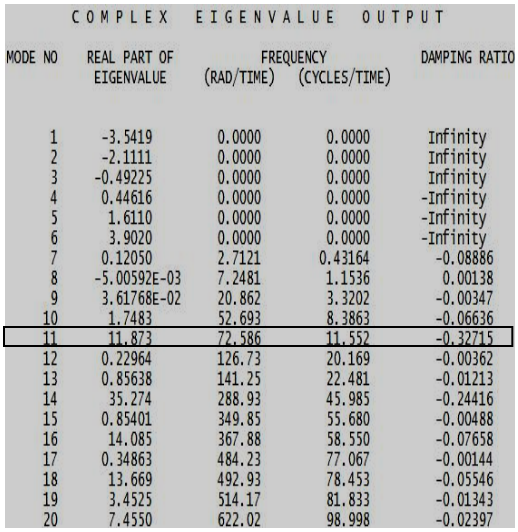


```
        C O M P L E X   E I G E N V A L U E   O U T P U T

MODE NO    REAL PART OF         FREQUENCY           DAMPING RATIO
            EIGENVALUE     (RAD/TIME)  (CYCLES/TIME)


     1       -3.6205        0.0000        0.0000        Infinity
     2       -2.3122        0.0000        0.0000        Infinity
     3       -0.51800       0.0000        0.0000        Infinity
     4        0.43295       0.0000        0.0000       -Infinity
     5        1.6069        0.0000        0.0000       -Infinity
     6        4.0301        0.0000        0.0000       -Infinity
     7        0.16959       2.6488        0.42157       -0.12805
     8       -5.05761E-03   7.2476        1.1535         0.00140
     9        5.04813E-02   20.698        3.2943        -0.00488
    10        1.7933        50.380        8.0182        -0.07119
    11        19.557        72.107        11.476        -0.54245
    12        0.22577       126.57        20.144        -0.00357
    13        0.77920       140.73        22.398        -0.01107
    14        52.529        278.84        44.378        -0.37677
    15        1.2256        349.80        55.673        -0.00701
    16        14.525        361.46        57.528        -0.08037
    17        0.42894       484.27        77.075        -0.00177
    18        14.782        488.93        77.815        -0.06047
    19        3.7878        513.03        81.651        -0.01477
    20        8.9253        619.70        98.628        -0.02881
```

(a) Average speed of 1.8 rad/s



```
        C O M P L E X   E I G E N V A L U E   O U T P U T

MODE NO    REAL PART OF         FREQUENCY           DAMPING RATIO
            EIGENVALUE     (RAD/TIME)  (CYCLES/TIME)


     1       -3.5419        0.0000        0.0000        Infinity
     2       -2.1111        0.0000        0.0000        Infinity
     3       -0.49225       0.0000        0.0000        Infinity
     4        0.44616       0.0000        0.0000       -Infinity
     5        1.6110        0.0000        0.0000       -Infinity
     6        3.9020        0.0000        0.0000       -Infinity
     7        0.12050       2.7121        0.43164       -0.08886
     8       -5.00592E-03   7.2481        1.1536         0.00138
     9        3.61768E-02   20.862        3.3202        -0.00347
    10        1.7483        52.693        8.3863        -0.06636
    11        11.873        72.586        11.552        -0.32715
    12        0.22964       126.73        20.169        -0.00362
    13        0.85638       141.25        22.481        -0.01213
    14        35.274        288.93        45.985        -0.24416
    15        0.85401       349.85        55.680        -0.00488
    16        14.085        367.88        58.550        -0.07658
    17        0.34863       484.23        77.067        -0.00144
    18        13.669        492.93        78.453        -0.05546
    19        3.4525        514.17        81.833        -0.01343
    20        7.4550        622.02        98.998        -0.02397
```

(b) Average speed of 2.5 rad/s

Fig. 8. Results of complex eigenvalue analysis

It is also found that the predicted results are reasonably close to those generated in the experiment as listed in Table 4.

Table 4. Comparison of FE and experimental results

Speeds (rad/s)	Experimental Frequency(Hz)	FE analysis (Hz)
1.8	11.5	11.5
2.5	10.9	11.6

4.2 Attack Angle (Deformation of Rubber Blade)

From the FE analysis, the attack angle for the rubber blade can be measured. The attack angle is the angle between the plane of the blade holder and a vector normal to the glass surface [9] as shown in Fig. 9.

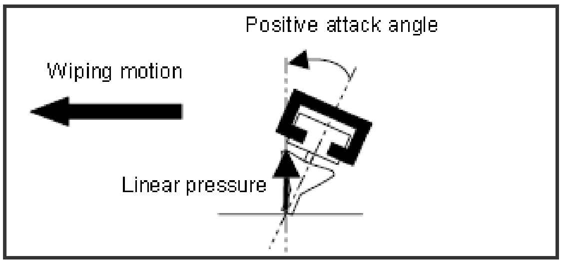


Fig. 9. Definition of attack angle [9]

It is found that, attack angle which is for the two different speeds are almost identical, i.e., about 16°. The deformed rubber blade in ABAQUS is shown in Fig. 10.

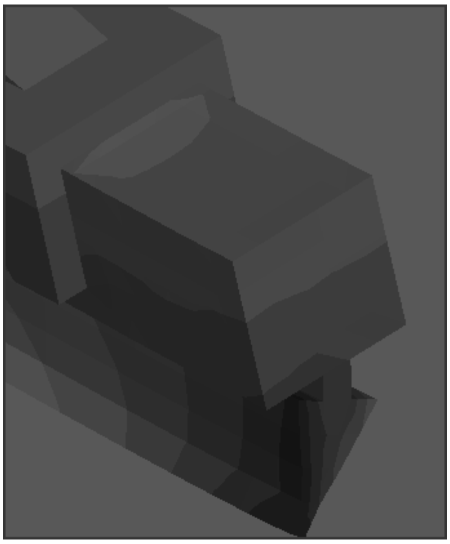


Fig. 10. Deformation of Rubber Blade

5. Conclusion

The wiper produces a low frequency vibration and noise called chatter, at dominant frequency of 11 Hz. It is found that, at different wiper average speeds, the chattering noise is generated before and after the wiper turnover. The complex eigenvalue analysis has been utilized in a finite element analysis to study low frequency squeal vibration problem. The measured chatter noise has been successfully replicated in the analysis. It has been shown in this paper as well as in the literature that the complex eigenvalue analysis is useful tool for low frequency noise analysis. The approach could generate almost identical chatter frequency.

It is the authors' intention to make further investigation to improve noise and vibration of the wiper system by proposing several structural modifications to rubber blade in subsequent work.

Acknowledgements

This project is funded by Ministry of Higher Education Malaysia (MOHE) under Vot No.78190. The first author would also like to thank to Mr. Elfandy Jamaludin for his helpful guidance through experimental work.

References

1. S. Goto, H. Takashi, T. Oya, Clarification of Mechanism of Wiper Blade Rubber Squeal Noise Generation, JSAE Review 22 (2001), 57-62.
2. J. Lévine, On the Synchronization of a Pair of Independent Windshield Wipers, IEEE Transactions on Control Systems Technology, Vol. 12 No. 5 (2004).
3. S. Okura, T. Sekiguchi, T. Oya, Dynamic Analysis of Blade Reversal Behavior in a Windshield Wiper System, SAE Technical Paper, 2000-01-0127.
4. S. Okura, T. Oya, Complete 3D Dynamic Analysis of Blade Reversal Behavior in a Windshield Wiper System, SAE Technical Paper, 2003-01-1373.
5. S. Goto, H. Takahashi, T. Oya, Investigation of Wiper Blade Noise Reduction Measures, SAE Technical Paper, 2001-01-1410.
6. R. Grenouillat, C. Leblanc, Simulation of Mechanical Pressure in a Rubber-Glass Contact for Wiper Systems, SAE Technical Paper, 2002-01-0798.
7. B. Stallaert, F. Doucet, J. Rys, A. Diallo, S. Chaigne, J. Swevers, P. Sas, Application of Dither Control for Automotive Wiper Squeal, Proceedings of ISMA (2006), 263-272.
8. C.R. Chevennement, T. Dreher, P. Alliot, E. Aubry, J.P. Lainé, L. Jézéquel, Flexible Wiper System Dynamic Instabilities: Modelling and Experimental Validation, Experimental Mechanics, 47 (2007), 201–210.
9. R. Grenouillat, C. Leblanc, Simulation of Chatter Vibrations for Wiper Systems, SAE Technical Paper, 2002-01-1239.
10. D.J. Ewins, Modal Testing: Theory and Practice, Research Studies Press, Letchworth, 1984.
11. A.R. Abu Bakar, E. Jamaludin, C.Y. Leong, An Experimental Investigation into Noise and Vibration of an Automotive Wiper, Proceedings of the regional Conferences on Advance in Noise, Vibration and Comfort (2007), 261-269.
12. H. Idris, Study of Windshield Wiper Noise, Bachelor Engineering Project Universiti Teknologi Malaysia (2007).
13. A.R. Abu Bakar, A. Sharif, M.Z.A Rashid, H. Ouyang, Brake Squeal: Complex Eigenvalue versus Dynamic Transient Analysis, SAE Technical Paper, 2007-01-3954.