PAPER • OPEN ACCESS

Identification dynamic behaviour of disc brake based on finite element model updating approach

IOP ebooks™

Bringing you innovative digital publishing with leading voices to create your essential collection of books in STEM research.

Start exploring the collection - download the first chapter of every title for free.

Identification dynamic behaviour of disc brake based on finite element model updating approach

M. S. M. Fouzi[1,3], M. S. M. Sani[1,2*] and M. F. R. Ishak[1]

[1]Advanced Structural Integrity and Vibration (ASiVR), Faculty of Mechanical & Manufacturing Engineering, Universiti Malaysia Pahang, 26600 Pekan, Pahang, Malaysia.
[2]Automotive Engineering Centre, Universiti Malaysia Pahang, 26600 Pekan, Pahang, Malaysia.
[3]Jabatan Pendidikan Politeknik & Kolej Komuniti.

*Corresponding author: mshahrir@ump.edu.my

Abstract. Braking system is an essential mechanism that designed to provide a good handling in order to slow down or to stop a vehicle depends on situation and driver's capability. In order to slow down or to stop the vehicle, brake pad will be pushed and make contact with the disc/rotor to form the friction. Resultant from braking event, it will produce vibration on disc/rotor structure. Thus, this study presents numerical prediction approach to identify the dynamic behaviour of disc brake structure using normal mode analysis in FEA software. Since the result of FEA is a prediction, the fidelity of FE model might be questioned. Therefore, measured data from EMA used to validate the predicted result of FEA through correlation process. As FE model used in this study made from assumption and simplification apart of real structure, it drive to discrepancy between FEA result and EMA data during correlation process. Hence, FE model updating approach implemented in order to reduce the discrepancy and increase the trustworthiness of FE model and it is become the main goal in the present study. The updating process has successfully reduced the discrepancy between FEA and EMA from 9.00 % of error to 4.05 % of error from its original value. As conclusion, the use of model updating process is important in preparing a reliable FE model of test structure no matters how simple or complex it is before being used for further analysis.

1. Introduction

Predominantly, each of vehicles has been designed with braking system to control the speed and give better handlings to the driver. Brake system is built from several components such as disc/rotor, piston, pad, calliper and knuckle as illustrated in Figure 1. In order to slow down the vehicle speed, the brake pad will be pushed by the piston to make a sliding friction with the disc/rotor. Due to this sliding friction, the structure tends to experience the vibration formed from dissipated kinetic energy of moving part. This friction-induced vibration has resulted the phenomenon called brake squeal [1]. In addition, [2] have mentioned most of the scientist and engineers have agreed that squeal noise in the disc brake is initiated by the instability due to friction forces, contributing to self-excited vibrations. Hence, the identification of dynamic behaviour of the disc brake structure subjected to the induced-vibration becomes a major interest in this study using modal analysis.
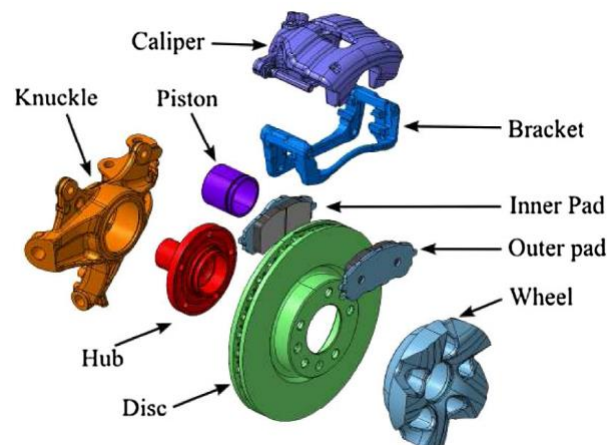
Figure 1. Exploded view of brake system [3].

Regarding to [4], modal analysis was conducted numerically and experimentally to determine the structural dynamic characteristics, in terms of modal frequencies, damping ratio, and mode shape. It is vital to identify the dynamic behaviour of the structure in the early development stage to avoid the resonance. Resonance effects are activated when the frequency content of the loading gets closer to the natural frequencies of the structure [5]. This was agreed by [6], it is paramount importance to periodically monitor the technical condition of components in order to avoid resonance or overload problems that may cause undesirable behaviour or even lead to critical failure. Thus, a reliable finite element (FE) model is required to monitor the structure periodically rather than make a repeated measurement of structure response that required highly expenses on expensive equipments and more efforts.

In order to prepare the reliable FE model, the predicted result computed in finite element analysis (FEA) needs to validate with measured data of frequency response functions (FRFs) that executed from experimental modal analysis (EMA). This kind of validation of FE model was reported by [7] using correlation process which compared the modal parameters (natural frequency and mode shape) of a body-in-white (BIW) obtained from EMA as benchmark and analyzed in FEA as prediction. However, the discrepancy possibly occurred between EMA data and FEA result in correlation process. This happen due to inaccuracies of FE model; poorly known boundary conditions of structure, the unknown material properties of the structure or because of the simplification in the modelling of very complex structural system [8]. In encounter this problem, an approach called FE model updating will be used to reduce the discrepancy and reconcile the predicted result with its measured counterpart by adjusting certain parameters of the FE model.

As mentioned by [9], FE model updating techniques are used to update the finite element model of a structure in order to improve its correlation with the experimental dynamic test data. Regarding [10], model updating techniques can be broadly classified as the direct methods and iterative methods. Direct methods are one-step procedures that seek to make a minimum change in the structural matrices so that the measured natural frequencies and mode shapes are reproduced by the updated model [10]. Despite of direct method, iterative methods are based on minimizing an objective function that is generally a non-linear function of selected updating parameters [11]. Practically, the iterative method seems more useful for larger and more complex structure in model updating technique. In addition, the sensitivity method probably the most successful of the many approaches to the problem of updating FE models of engineering structures based on vibration test data [12]. In the sensitivity method, the eigensolutions are often used to construct the objective function and the physical parameters of analytical FE model adjusted iteratively to minimize the discrepancy of dynamic properties between the analytical FE model and the measured counterpart [13]. This kind of iteratively updating based on sensitivity method successfully reported in their works [7, 14, 15].

Due to outstanding capability of iterative model updating technique reported from the previous researchers, this technique has been adopted in preparing the reliable FE model to replicate disc brake structure and it has become the main goal for the present study. Last but not least, this updating technique managed to shrink the discrepancy between predicted dynamic properties with the measured counterpart. This supported by the previous published works [7, 16, 17] that reported their updating process successfully reduce the discrepancies between FEA and EMA.

2. Finite element analysis (FEA) of disc brake

Disc/rotor is major structure in brake system and directly exposed to friction induced-vibration. Hence, numerical approach using FEA is adopted to identify the dynamic behaviour of the structure using normal mode analysis SOL103 in MSC Nastran/Patran. This normal mode analysis used to extract natural frequency and mode shape of undamped system [18]. The entire process involved during this analysis has been summarized in Figure 2.

Initially, the disc/rotor structure was modelled using computational aided design (CAD) software, SolidWork. Then, it was imported into FEA software, MSC Patran in *"parasolid"* format for pre-processing stage. The similar approach has been reported in the previous work, where the test structure was modelled using CATIA V-5 software and been imported into Hypermesh for pre-processing in FEA and iterated by MSC Nastran solver [19].

During pre-processing stage, the FE model was meshed using solid element CTETRA10 topology that produced 3712 number of elements with 7243 number of nodes. Then, the FE model was assigned with the following nominal value of material properties; Modulus of Young, $E = 130$ GPa, density, $\rho = 7100$ kg/m^3, and Poisson's Ratio, $v = 0.26$. This nominal value was considered in the midst of the study for material properties effect on disc brake squeal by [2].

Once pre-processing stage done, the mathematical model generated for normal mode analysis in bulk data file (*bdf.*) format that will be used in iteration process by MSC Nastran solver to extract the modal parameter (natural frequency and mode shape) of disc brake. Regarding [7], neither boundary condition nor external forces were applied to the model as the model was let to be free-free boundary condition for calculation of modal properties using normal mode analysis SOL103 in MSC Nastran/Patran.

3D-Modelling using CAD software   Pre-processing in FEA package

Post-processing in FEA   Solver
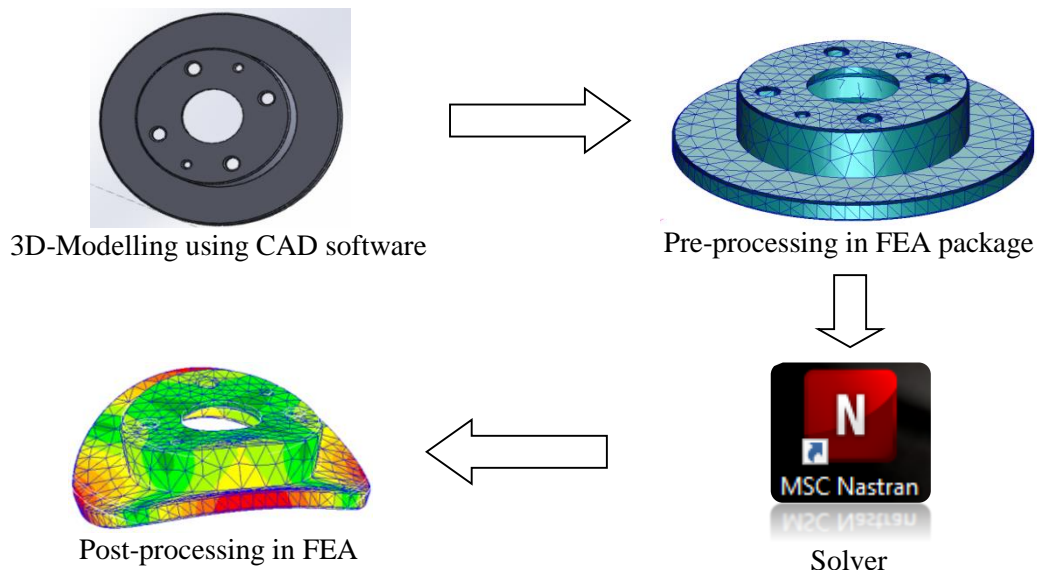
**Figure 2**. Flow for normal mode analysis of exhaust structure.

3. Modal testing procedure of disc brake structure

In order to validate the predicted result computed in FEA, the measured dynamic properties of disc brake was carried out using modal testing. Modal testing or familiar as experimental modal analysis (EMA) was conducted to measure the structural response. As reported by [20], one of preferable technique to measure the structural response is impact hammer excitation owing to its fast, convenient, and fast diagnostic. This technique was implemented to measure the structural response of disc brake in the present study.

In the beginning of experimental procedure, the configuration of equipment and test structure were setup as depicted in Figure 3. The disc brake was suspended by elastic cord such as Figure 4 to represent free-free boundary condition and this approach was reported by [21] in their work. The measurements were made using EMA software with several equipment such as 4 channels National Instruments (NI) data acquisition system (DAQ), impact hammer with transducer sensitivity 2.25 mV/g, tri-axial accelerometer with transducer sensitivity 100 mV/g for x-axis, 104 mV/g for y-axis, and 102 mV/g for z-axis. Roving accelerometer technique was employed in this setting same as practiced by[7]. In this technique, the disturbance will be created by impact hammer at fixed point while the accelerometer will be roved around 48 measurements point that assigned on the structure. The response will specifically relate to each point sketched in wire-frame structure such as Figure 5 in EMA software to simulate the mode shape once the measurement completed.

During the testing, the output response measured by the accelerometer is in the form of Fast Fourier Transform (FFT). In order to extract the modal parameters, DAQ is used as FFT analyzer which converter the response in the form of FFT into frequency response functions (FRFs). Due to measured FRFs, modal parameters can be extracted using curve-fitting method in EMA software as illustrated in Figure 6. As supported by [4], EMA techniques have been dramatically improved within the past two decade with accurate modal post processing curve-fitting technique (modal extraction).
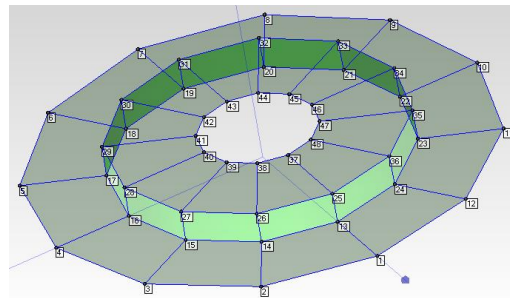
The extracted modal parameters (natural frequency and mode shape) then were used to validate the predicted result from FEA in correlation process.
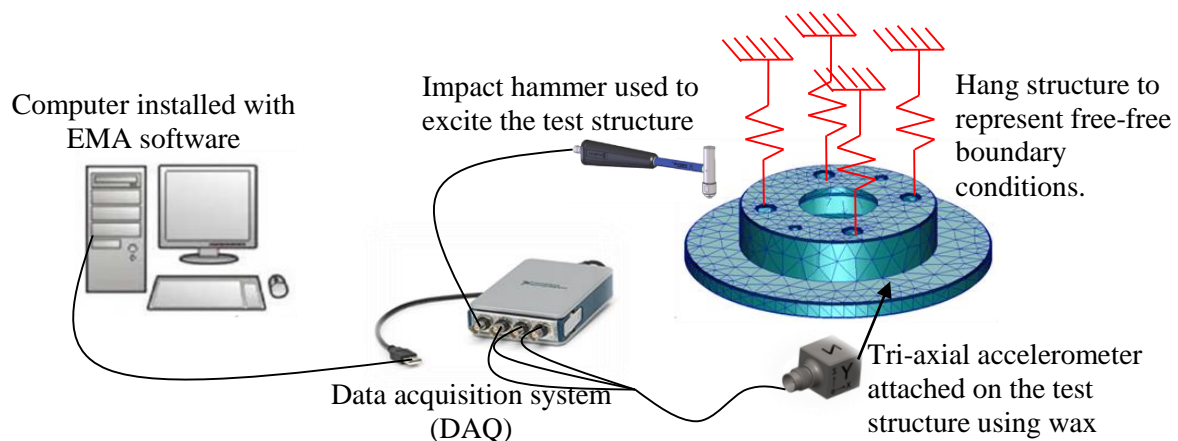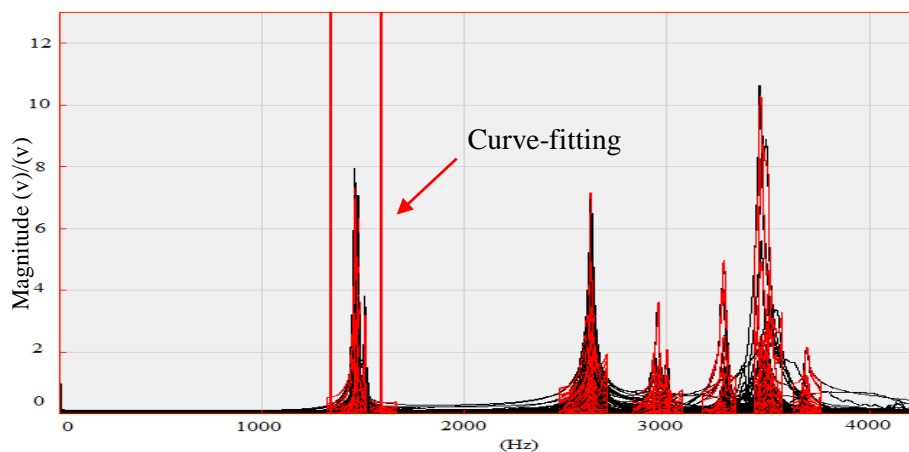

Figure 3. Configuration of equipments and disc brake in experimental modal analysis (EMA).
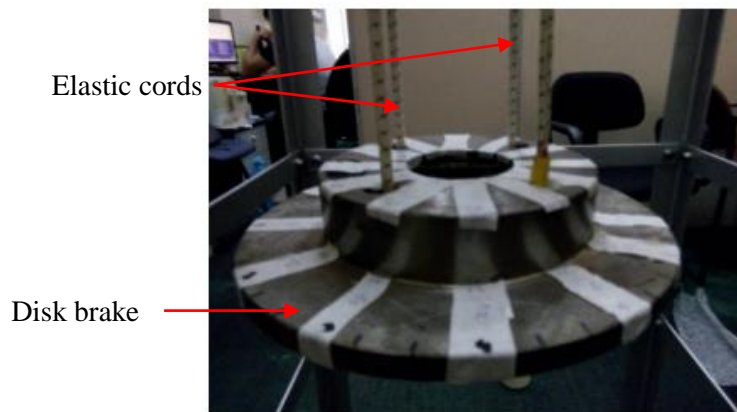
Figure 4. Test structure hanged to replicate free-free boundary condition.

Figure 5. Wire frame structure sketched in EMA software.

Figure 6. Overlay traces of FRFs.

4. Correlation

In verifying and validate the predicted result from FEA, correlation process is implemented in this study. Regarding [22], correlation is a process where data from the experiment are compared with theoretical results. In addition, this correlation process also implemented by [7] where numerical analysis for dynamic behaviour of a body-in-white (BIW) structure are compared with modal testing results.

Comparison of eigenvalue or natural frequency is tabulated in Table 1 while in Table 2 is comparison of eigenvector or mode shape between predicted result and measured counterpart. In equation (1), the percentage of error is calculated to demonstrate how far the predicted of FEA agreed with measured data in EMA. The bigger value of percentage of error indicates huge disparity while the small error
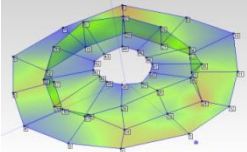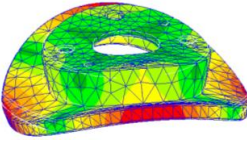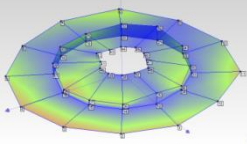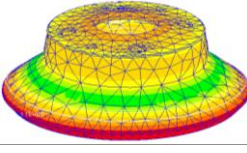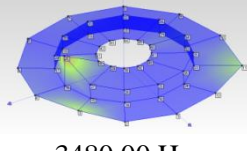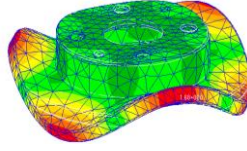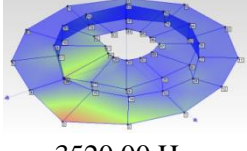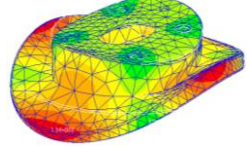
shows the agreement between EMA and FEA. Hence, the smaller error means the FE model is reliable to represent actual structure. The calculated percentage of error showed there is 9.00 % of error between EMA and FEA in Table 1. The discrepancies are able to be minimized using FE model updating approach that be discussed in section 5.

$$Percentage\ of\ error = \left| \frac{f^{predicted} - f^{measured}}{f^{measured}} \right| x\ 100 \tag{1}$$

Table 1. Comparison between predicted results from FEA with measured data in EMA.

| Mode	Natural frequency (Hz)		
	EMA	FEA	Error (%)
1	1480.00	1659.50	12.13
2	2640.00	3061.40	15.96
3	3480.00	3455.60	0.70
4	3520.00	3873.00	10.03
5	3700.00	3929.30	6.20
Total average error			9.00

Table 2. Comparison of mode shape between measured FRF in EMA and numerical prediction eigenvector by FEA for the first 5 mode of interests.

Mode	EMA	FEA
1	1480.00 Hz	1659.50 Hz
2	2640.00 Hz	3061.40 Hz
3	3480.00 Hz	3455.60 Hz
4	3520.00 Hz	3873.00 Hz

5
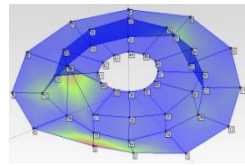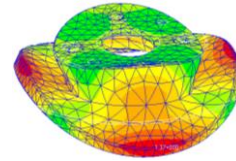
| 3700.00 Hz | 3929.30 Hz |

5. FE model updating

Since FEA is numerical prediction method, it is possible that disparity will occurred when correlate with the measured counterpart from EMA. The difference occurred due to inaccurate FE model since simplification of structure details for complex structure and lack of knowledge of exact material properties [10]. FE model updating adopted in this study to overcome this obstacle. Model updating is essentially a process of adjusting certain parameters of FE model [12]. The following equation (2) is sensitivity analysis used to identify the most sensitive parameter in the analysis.

$$\mathbf{S} = \boldsymbol{\phi}_i^{\mathrm{T}} \left[\frac{\partial \mathbf{K}}{\partial \theta_j} - \lambda_i \frac{\partial \mathbf{M}}{\partial \theta_j} \right] \boldsymbol{\phi}_i \tag{2}$$

where **S** indicates the sensitivity matrix, **K** and **M** are the stiffness and mass matrices respectively, while ϕ, λ and θ represent eigenvector, eigenvalue and parameter respectively. Furthermore, i indicate the i-th eigenvalue and j for the j-th parameter [23-25].

Meanwhile, the minimisation of the discrepancies between the measured and predicted values was carried out via a residual based objective function in the form of equation (3)

$$min \sum_{j=1}^{m} W_j \left(\frac{\omega_i^n}{\omega_i^e} - 1 \right)^2 \tag{3}$$

Where ω_i^n is the i-th predicted frequency and ω_i^e represent the i-th measured frequency; W_j indicates the weighting coefficient through which certain modes that required particular attention are assigned [14]. Both equation (2) and (3) are embedded in the optimization algorithm SOL200 that been running into MSC Nastran. The outputs of optimization accessible via F06 file such as updated eigenvalue and the sensitivity of parameters.

As depicted in Figure 7 and tabulated in Table 3, three parameters which are Modulus Young (E), Density (RHO) and Poisson's Ratio (Nu) are considered in the sensitivity analysis for five mode of interests. It showed Modulus Young and density are sensitive parameters while Poisson's Ratio is light sensitive. There are only 4 iterations are taken to converge in this analysis such as showed in Figure 8. Regarding to [12], the updating procedure are significantly be made only to erroneous assumptions such material parameters not on model-structure errors. Hence, only three parameters are available to update in this study as mentioned earlier. This was supported in the published work by [2], there are only three parameters (Modulus Young, density and Poisson's Ratio) were considered in identifying material properties effect on disc brake squeal.
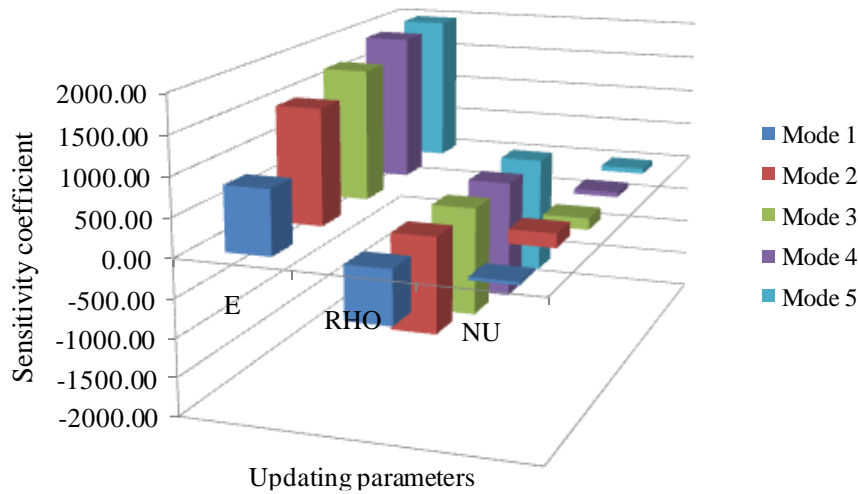
Figure 7. Graph of sensitivity matrix analysis for the present study.

The updated natural frequency of FE model accessed from F06 file then compared with the initial value of natural frequency as tabulated in table 3. The discrepancy successfully reduced from initial value 9.00 % of total average error to 4.05 % of total average error. Updated value of parameters that be suggested from F06 is tabulated in Table 4 with the changes calculated from its original value.
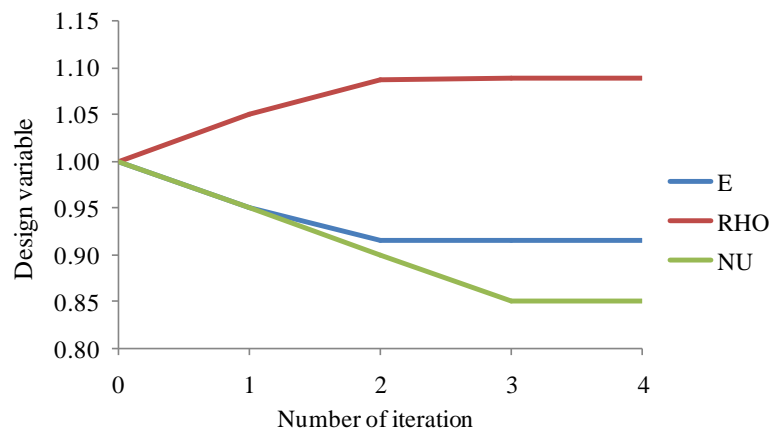
Figure 8. Graph of iteration.

Table 3. Comparison between initial FEA and updated FEA with measured data from EMA.

Mode	Natural frequency (Hz)				
	EMA	Initial FEA	Error (%)	Updated FEA	Error (%)
1	1480.00	1659.50	12.13	1527.45	3.21
2	2640.00	3061.40	15.96	2777.72	5.22
3	3480.00	3455.60	0.70	3190.52	8.32
4	3520.00	3873.00	10.03	3560.00	1.14
5	3700.00	3929.30	6.20	3611.93	2.38
	Total average error		9.00		4.05

$$Changes\ value\ of\ parameters = \left|\frac{Updated\ value - Initial\ value}{Initial\ value}\right| \tag{4}$$

Table 4. Value of parameters based on design variable of model updating approach.

Parameter	Initial value	Updated value	Changes
Modulus of Young, E (GPa)	130	119	0.08
Density, ρ (kg/m^3)	7100	7728.35	0.09
Poisson's Ratio, v	0.26	0.22	0.15

6. Conclusion

The present work is identified the dynamic behaviour of disc brake rotor using modal analysis approach. The analysis has been carried out through numerical prediction method via finite element analysis (FEA) software. Since the accuracy issue of material properties used to assign in the modelling process made from assumption, the discrepancy occurred when compared with measured dynamic parameters. Modal testing was carried out to measure the structural response in order to verify the predicted result computed in FEA. In attempt to reduce the discrepancy between predicted and its measured counterpart, FE model updating technique applied in this study with respective updating parameters. After been updated iteratively, the discrepancy between EMA and FEA successfully minimized from 9.00 % of total average error to be 4.05 % of total average error. As conclusion, FE model updating approach is feasible to produce a reliable FE model in numerical analysis before been used for further analysis.

Acknowledgement

References

[1] Ouyang H, Nack W, Yuan Y, and Chen F 2005 "Numerical analysis of automotive disc brake squeal: a review" *International Journal of Vehicle Noise Vibration* vol. 1 pp. 207-231

[2] Kharate N K and Chaudhari S S 2018 "Effect of Material Properties On Disc Brake Squeal And Performance Using FEM and EMA Approach" *Materials Today: Proceedings* vol. 5 pp. 4986-4994

[3] Nacivet S and Sinou J-J 2017 "Modal Amplitude Stability Analysis and its application to brake squeal" *Applied Acoustics* vol. 116 pp. 127-138

[4] Eslaminejad A, Ziejewski M, and Karami G 2019 "An experimental–numerical modal analysis for the study of shell-fluid interactions in a clamped hemispherical shell" *Applied Acoustics* vol. 152 pp. 110-117 doi:https://doi.org/10.1016/j.apacoust.2019.03.029

[5] Kihm F, Langelier A, and Munson K 2018 "Influence of the modal damping on the estimated fatigue life" *Procedia engineering* vol. 213 pp. 270-281 doi:10.1016/j.proeng.2018.02.028

[6] Huňady R, Pavelka P, and Lengvarský P 2019 "Vibration and modal analysis of a rotating disc using high-speed 3D digital image correlation" *Mechanical Systems and Signal Processing* vol. 121 pp. 201-214 doi:https://doi.org/10.1016/j.ymssp.2018.11.024

[7] Abdullah N A Z, Sani M S M, Rahman M M, and Zaman I 2017 "Correlation of numerical and experimental analysis for dynamic behaviour of a body-in-white (BIW) structure" *MATEC Web of Conferences* vol. 90 p. 01020 doi:10.1051/matecconf/20179001020

[8]   Abdullah N A Z, Sani M S M, Rahman M M, and Zaman I 2015 "A review on model updating in structural dynamics" *IOP Conference Series: Materials Science and Engineering* vol. 100 pp. 1-6 doi:10.1088/1757-899X/100/1/012015

[9]   Nehete D V, Modak S V, and Gupta K 2015 "Structural FE model updating of cavity systems incorporating vibro–acoustic coupling" *Mechanical Systems and Signal Processing* vol. 50-51 pp. 362-379 doi:http://dx.doi.org/10.1016/j.ymssp.2014.05.028

[10]   Modak S V 2014 "Model updating using uncorrelated modes" *Journal of Sound and Vibration* vol. 333 pp. 2297-2322 doi:http://dx.doi.org/10.1016/j.jsv.2014.01.013

[11]   Pradhan S and Modak S V 2012 "Normal response function method for mass and stiffness matrix updating using complex FRFs" *Mechanical Systems and Signal Processing* vol. 32 pp. 232-250 doi:http://dx.doi.org/10.1016/j.ymssp.2012.04.019

[12]   Mottershead J E, Link M, and Friswell M I 2011 "The sensitivity method in finite element model updating: A tutorial" *Mechanical Systems and Signal Processing* vol. 25 pp. 2275-2296 doi:10.1016/j.ymssp.2010.10.012

[13]   Zhu T, Tian W, Weng S, Ge H, Xia Y, and Wang C 2018 "Sensitivity-based finite element model updating using dynamic condensation approach" *International Journal of Structural Stability and Dynamics* vol. 18 pp. 1-23 doi:10.1142/S0219455418400047

[14]   Mohd Zin M S, Rani M N A, Yunus M A, Wan Iskandar Mirza W I I, Mat Isa A A and Mohamed Z 2017 Modal and FRF Based Updating Methods for the Investigation of the Dynamic Behaviour of a Plate *Journal of Mechanical Engineering* 4 175–89

[15]   Husain N A, Khodaparast H H, and Ouyang H 2010 "FE model updating of welded structures for identification of defects" *International Journal of Vehicle Noise and Vibration* vol. 6 pp. 163-175

[16]   Jelani K M, Razip M H M, Nazri N A, Sani M S M, and Yasar M 2019 "Dynamics investigation on motorcycle chassis based on Finite Element (FE) modelling and updating" *IOP Conference Series: Materials Science and Engineering* vol. 469 pp. 1-10 doi:10.1088/1757-899X/469/1/012103

[17]   Fouzi M S M, Sani M S M, and Muchlis Y 2019 "Finite Element Modelling and updating of welded joint for dynamic study of exhaust structure" *IOP Conference Series: Materials Science and Engineering* vol. 469 pp. 1-10 doi:doi:10.1088/1757-899X/469/1/012099

[18]   AbuBakar A R and Ouyang H 2006 "Complex eigenvalue analysis and dynamic transient analysis in predicting disc brake squeal" *International Journal of Vehicle Noise Vibration* vol. 2 pp. 143-155

[19]   Vinutha B S and Hemanth K 2017 "Study of static and frequency responsible analysis of hangers with exhaust system" *IJASRE* vol. 3

[20]   Carne T G and Stasiunas E C 2006 "Lessons learned in modal testing—part 3: Transient excitation for modal testing, more than just hammer impacts" *Experimental Techniques* vol. 30 pp. 69-79 doi:10.1111/j.1747-1567.2006.00046.x

[21]   Renault A, Massa F, Lallemand B, and Tison T 2016 "Experimental investigations for uncertainty quantification in brake squeal analysis" *Journal of Sound Vibration* vol. 367 pp. 37-55

[22]   Rajadurai M S, Kavin R, Rejinjose, Prabhakaran, and Rajeshraman 2016 "A system approach to dynamic characteristics of hanger rod in exhaust system" *International Journal of Innovative Science, Engineering & Technology* vol. 3

[23]   Fouzi M S M, Jelani K M, Nazri N A, and Sani M S M 2018 "Finite Element Modelling and Updating of Welded Thin-Walled Beam" *International Journal of Automotive and Mechanical Engineering* vol. 15 pp. 5874-5889 doi:10.15282/ijame.15.4.2018.12.0449

[24]   Abdullah N A Z, Sani M S M, Hussain N A, Rahman M M and Zaman I 2017 Dynamics properties of a Go-kart chassis structure and its prediction improvement using model updating approach *International Journal of Automotive and Mechanical Engineering (IJAME)* 14 3887-97

[25]   Sani M S M and Ouyang H 2016 Identification of Nonlinearities in Joints of a Wing Structure *MATEC Web of Conference* 83 (03006)