ANALYSIS OF DESIGN PARAMETER EFFECTS ON VIBRATION MODES OF A MOTORCYCLE DRUM BRAKE AND BRAKE SHOE USING THE FINITE ELEMENT METHOD

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ABSTRACT

The natural frequencies and vibration modes of the particular motorcycle drum brake and brake shoe have been determined using the finite element method. The results show that the drum brake has 42 mode shapes in the frequency range of 100 Hz to 12 kHz. Most of the mode shapes occurred in pair (repeated root). The brake shoe is found to have 10 mode shapes in the frequency range analysed. Parametric analysis has also been conducted to study the effect of various design parameters to the occurrence of natural frequency of the model and the results have been presented in the frequency response function. The results show high modal density in the frequency range of 1500 – 2000 Hz and 5600 – 6400 Hz. Modal density of the drum can be reduced by increasing its stiffness for the low frequency range and increasing flange stiffness for the high frequency range. These results indicate important parameters for the design and prevention of drum brake noise.

Keywords: *Drum Brake, Noise, Vibration, Finite Element Method, Design*

INTRODUCTION

Vibration of the components of the drum brakes when braking causes noise to be emitted. The motorcycle drum brake consists of two major components, the drum and the brake shoe. The vibration of these two components are related to the squeal noise generated by the motorcycle drum brake which is related to the perceived quality of the motorcycle and noise pollution. Noise emanating from brakes has been studied as early as 1935 when Lamarque [1] collected the experience of manufacturers and operators on the cause and prevention of squeal in drum brakes. There have been a considerable number of works on drum brake squeal and this has been collated in a relatively comprehensive overview by Crolla and Lang in 1991 [2]. Since then there have been a remarkable improvement in the understanding on the cause of drum brake squeal particularly when the problem is treated as a stability problem associated with the complex eigenvalue of the brake system [3,4,5]. One of the latest works by Hamabe [6] clearly showed the effect of modal intensity and the influence of various geometric parameters in drum brake squeal analysis.

The effect of various parametric designs has also been highlighted in the previous studies of drum brake squeal. The effect of suspension system on the drum brake noise has been studied by Fieldhouse [7]. The importance of drum brake back plate on the drum brake squeal has also been reported by many researchers [6,7,8].

The present work attempts to identify the design parameter of a motorcycle drum brake which influence its natural frequency in particular those parameters which can reduce the modal intensities and the diametral mode of the drum. A typical drum brake of a light motorcycle (110 cc) of a Malaysian made is used in this analysis. The importance of this work lies on the fact that unlike disc, a drum brake assembly is relatively more complex and asymmetry exists due to its design. This work will highlight the effect of such geometry on its overall natural frequencies and mode shapes.

DESCRIPTION OF THE MODEL

A motorcycle drum brake chosen for the analysis is shown in Figure 1. The rear wheel motorcycle drum brake consists of a drum brake, a pair of shoes, actuator, and brake linings. The drum itself was attached with the spokes and two bearings.

Because the brake system is complex, it is decided to consider only the drum for the analysis. The specification of

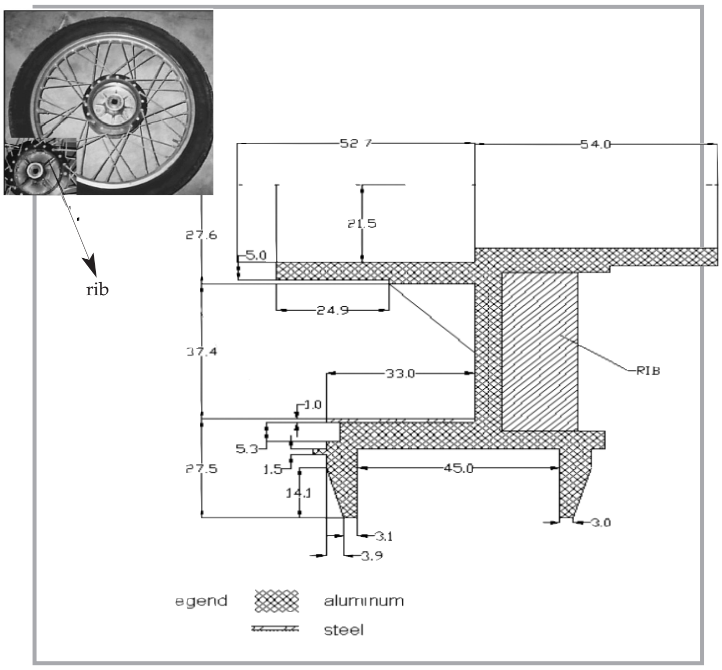

Figure 1: Motorcycle drum brake and its cross-sectional area (dimension in mm)

Table 1: Specifications of the drum brake and brake shoe

Drum Brake			Brake Shoe		
	Drum	Friction Steel		Aluminum back plate	Lining
Modulus Elasticity, E_d, E_s	69 GPa	207GPa	Modulus of elasticity, E	69 GPa	8 GPa
Diameter, d_d, d_s	136 mm	130 mm			
Drum Width, l_d, l_s	30 mm	33 mm	Density, ρ	2720 kg/m^3	2500 kg/m^3
Density, ρ_d, ρ_s	2720 kg/m^3	7520 kg/m^3			

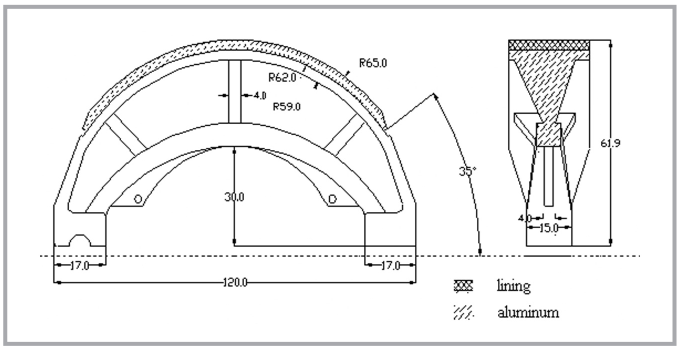

Figure 2: Motorcycle brake shoe and its cross-sectional area (dimensions in mm)

the drum is shown in Table 1. The drum brake is made of cast aluminum alloy and friction part of the drum in contact with the lining is made of cylindrical steel. The motorcycle drum brake is shown in Figure 2. The shoe back plate is made of the same material as that of the drum brake. The specifications of the drum brake and brake shoe are shown in Table 1.

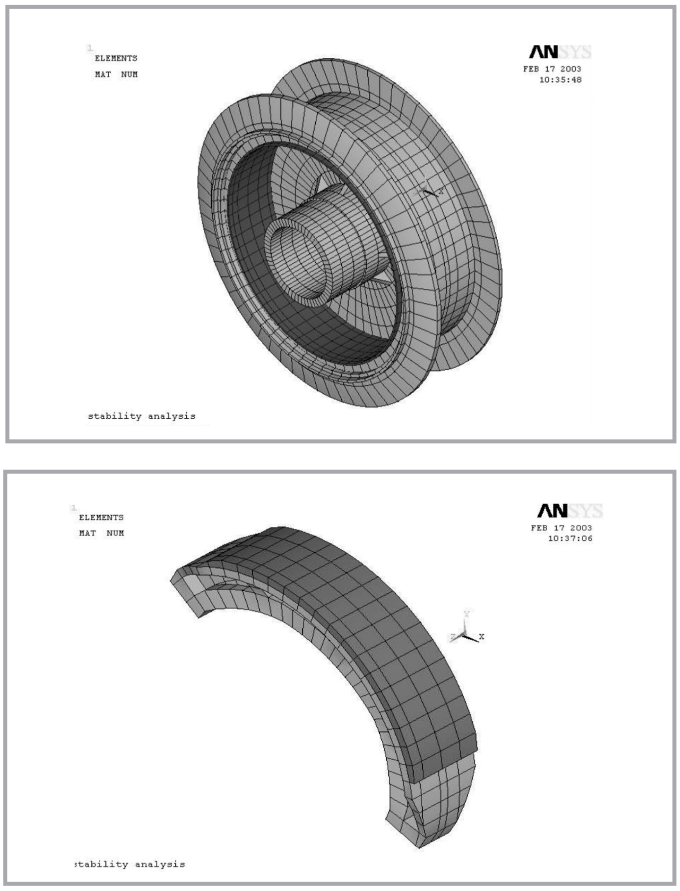

Figure 3: 3-D finite element model; (a) drum, and (b) brake shoe

FINITE ELEMENT MODEL

A three-dimensional finite element model of the drum brake and brake shoe are shown in Figure 3. The models consist of hexahedral elements with eight nodes. The definition of nodes and elements are created using ANSYS 6.0 finite element solver package.

The following assumptions are considered for the analysis:

1. The spokes and the spokes holes are neglected.
2. The bearings are excluded.
3. Details produced by casting process are neglected.

Modal Analysis of the Drum

Natural frequencies of the drum brake are calculated by modal analysis option available in ANSYS 6.0 software. The element type used is 8-node isoperimetric brick known as SOLID45 in ANSYS nomenclature. The modal analysis was conducted in the range of frequency from 100 Hz to 12 kHz. There are 42 mode-shapes occurred in the specific frequency range which indicates that the modal density of the drum brake is high as it has many modes in a small frequency band. It can also be observed as well that drum brake has repeated roots where two modes have the same mode shape but different modal frequency. The repeated roots occur at the structure that is axially symmetrical with identical frequency for both roots. But in the case of this particular drum brake, the repeated roots have different frequencies. This is caused by the presence of back plate ribs and will be shown later in this paper.

Mode shape of the drum can be classified into several groups according to geometrical mode shapes, such as diametral modes and flange modes. The first diametral mode shapes occurred at frequencies of 1170 Hz. The second diametral mode of the drum occurred at frequencies of 1955 Hz and 2233 Hz. The third diametral mode shape of the drum occurred at a frequency of 5213 Hz. The fourth drum diametral mode shapes occurred at frequencies of 8766 Hz and 8799 Hz. Another set of third diametral mode shapes of the drum occurred at frequencies of 11230 Hz and 11055 Hz.

Flange mode shapes dominate the mode shape of the drum brake with 25 mode shapes out of 42 mode shapes. Flange mode shapes occurred at frequencies of, among others, 2873 Hz, 3066 Hz, 3492 Hz, 4341 Hz, 4367 Hz, 5699 Hz, 5967 Hz, 6187 Hz, 7049 Hz, 7054 Hz, 7420 Hz, 7619 Hz, 8350 Hz, 8766 Hz, 9087

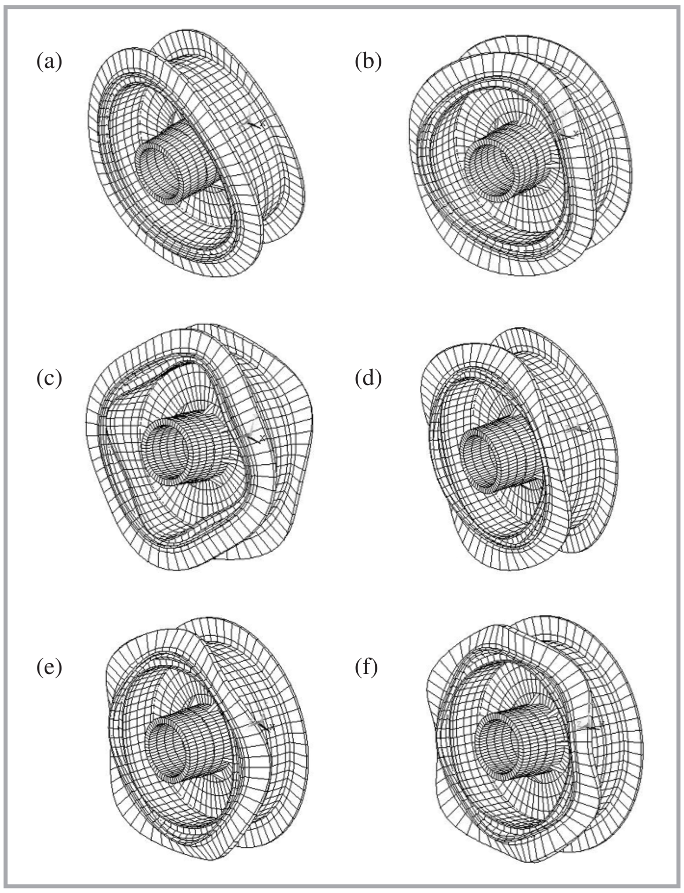

Figure 4: (a) Second diametral mode shape at 1955 Hz, (b) Third diametral mode shape at 5213 Hz, (c) Fourth diametral mode shape at 11203 Hz, (d) Fourth flange mode shape at 5699 Hz, (e) Fifth flange mode shape at 7054 Hz, and (f) Sixth flange mode shape at 8766 Hz

Hz, 9210 Hz, 10603 Hz, 10604 Hz, and 10974 Hz. Some of the mode shapes are presented in Figure 4.

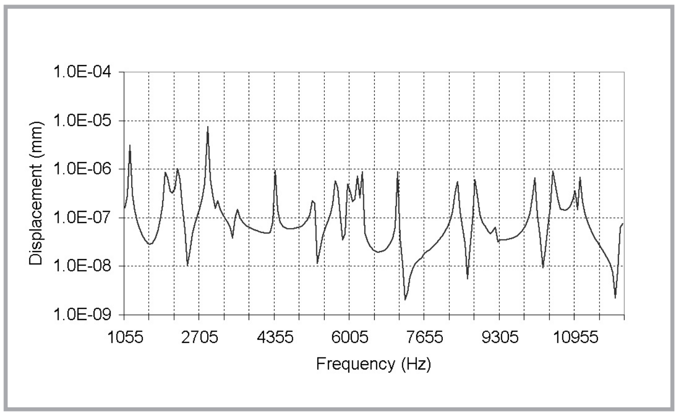

Figure 5: Frequency response function of the drum brake (in radial direction)

Other mode shape include back plate mode shapes which occurred at frequencies of 1369 Hz and 2873 Hz and cylindrical mode shapes occurred at frequencies of 2873 Hz, 10061 Hz, 10476 Hz, and 11970 Hz. The frequency response function of the drum brake is shown in Figure 5.

Classification of the mode shape according to the frequency range is given in Table 2. The most numbers of the mode shapes occurred in the frequency range between 10 kHz and 11 kHz with 7 mode shapes.

Frequency range (Hz)	Number of mode shapes
1000 – 2000	5
2000 – 3000	3
3000 – 4000	3
4000 – 5000	2
5000 – 6000	5
6000 – 7000	1
7000 – 8000	5
8000 – 9000	6
9000-10000	2
10000-11000	7
11000-12000	3
Total	42

Table 2: Classifications of mode shapes according to frequency range

Modal Analysis of the Shoe

Modal analysis of the brake shoe is conducted in the frequency range of 100 Hz and 12000 Hz. The analysis resulted in 10 mode shapes of the brake shoe. The mode shape can be grouped into bending mode shapes and twisting mode shapes. The bending modes are those that involve major deformation in the radial direction while the torsion modes are when the displacement involves major displacement in the axial direction.

There are 5 mode shapes that can be categorised as bending mode shapes and 4 mode shapes as the twisting mode shapes. The bending mode shapes occurred at frequencies of 358 Hz, 2141 Hz, 6485 Hz, 10253 Hz, and 10986 Hz. The twisting mode shapes occurred at frequencies of 927 Hz, 2690 Hz, 5937 Hz, and 9323 Hz. Several brake shoe mode shapes are given in Figure 6. The frequency response function of the brake shoe is given in Figure 7.

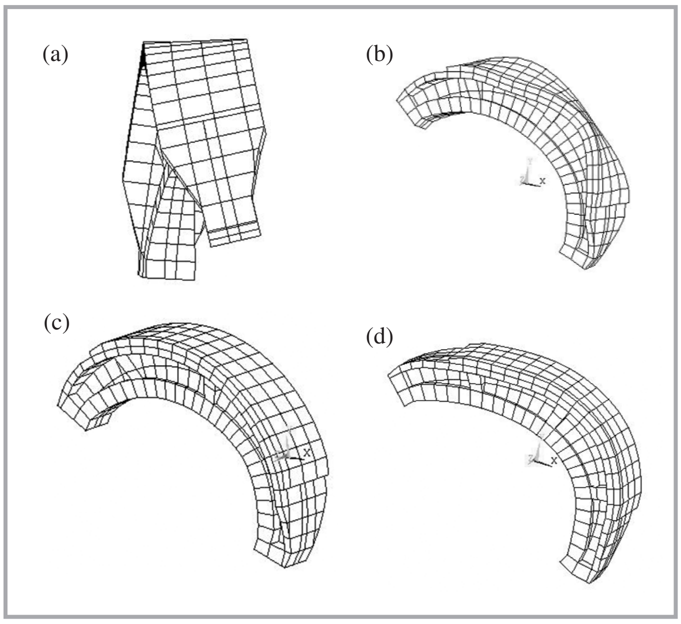

Figure 6: (a) Twisting mode at 2690Hz, (b) twisting mode at 9323Hz, (c) bending mode at 2141Hz, and (d) bending mode at 10984Hz

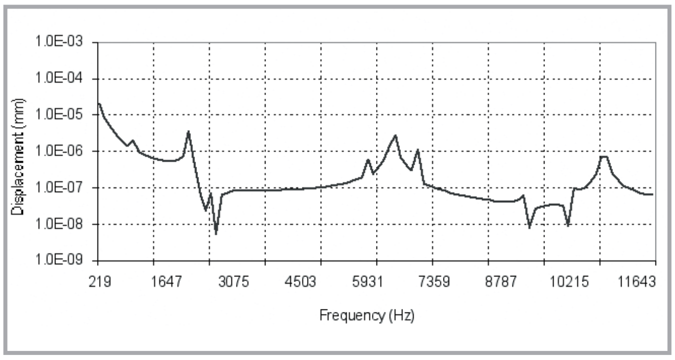

Figure 7: Frequency response function of the brake shoe (in radial direction)

PARAMETRIC ANALYSIS

Parametric analysis is conducted to find out the design parameters effect for natural frequencies, mode shape and modal density of the model. The interested parameters for the analysis are stiffness of the drum back plate ribs, flanges, shoe, as well as flange thickness.

Effect of Drum Back Plate Ribs

The particular motorcycle drum brake used in this analysis has four ribs at its back plate ribs. The back plate ribs of the drum brake are the part where torque from the sprocket is applied. The influence of the ribs to the overall mode shape of the drum brake is analysed by conducting the modal analysis after removing the ribs. The results showed that the frequencies of each of the repeating roots (mode shape pair) become identical to each other. For example, one of the mode shape pair have the frequencies of 1955 Hz and 2233 Hz. As the ribs are removed, this mode shape frequency and its pair occurred at a frequency of 1927 Hz. The comparison of the frequency response function for this mode pair with and without the presence of back plate ribs is given in Figure 8. With the presence of rib (indicated by continuous line) there are two peaks for the same mode shape, at frequencies of 1955 Hz and 2233 Hz. As the ribs are removed (indicated by dotted line), there

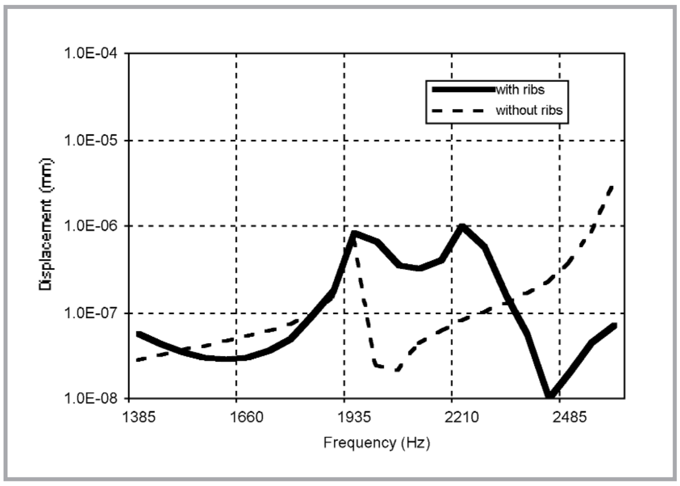

Figure 8: Effect of back plate ribs for the particular pairing mode shapes

is only one peak left, indicating that this mode pair occurred at the same frequency of 1927 Hz. Another example is the mode pair at frequencies of 4341 Hz and 4367 Hz. After the ribs are removed, this mode pair occurred at frequency of 4334 Hz. Both of these mode pair are shown in Figure 9.

Of all the 42 mode shapes of the drum brake at the frequency range between 1 kHz and 12 kHz, 17 of them are mode pair with

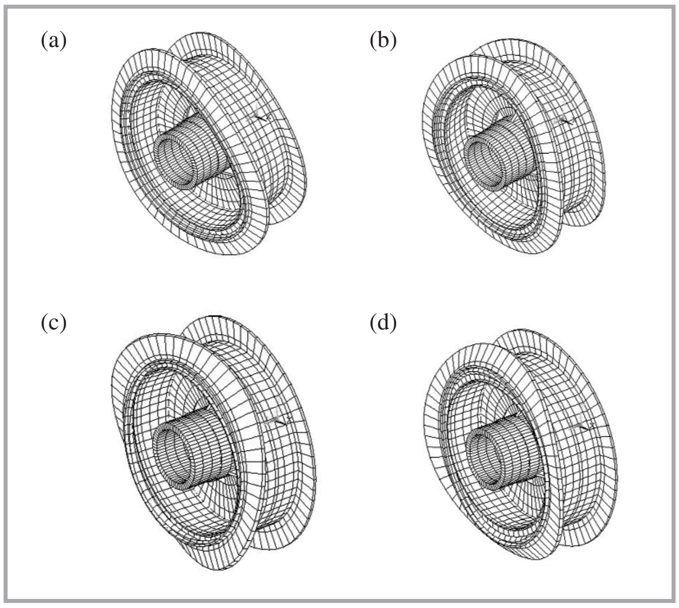

Figure 9: (a) Mode shape at 1955 Hz, (b) pairing mode of (a) at 2233 Hz, (c) mode shape at 4341 Hz, and (d) pairing mode of (c) at 4367 Hz

different frequencies. As the ribs are removed, all of these mode pair shapes occurred at the same frequency.

The effects of various components' stiffness can be observed in Figure 10. The drum modulus of elasticity and the back plate rib modulus of elasticity were changed to 207 GPa with density of 7820 kg/m3 for each case. The baseline for the comparison is the frequency response function of the drum brake having modulus elasticity of 69 GPa (indicated by continuous line).

Stiffening the drum (indicated by a dash-dot lines) resulted in some decrease in the natural frequencies of the drum brake while stiffening the drum back plate ribs has resulted in a more significant increase in the natural frequencies of the drum brake.

Effect of Flanges

The flanges are the part where spokes are mounted to connect

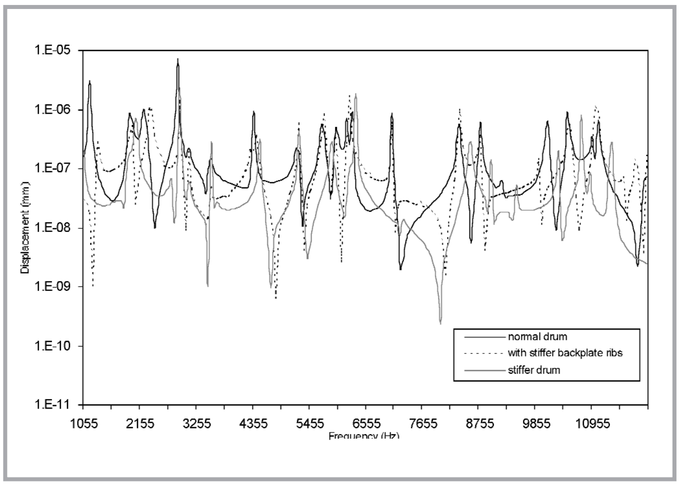

Figure 10: The effects of drum and ribs stiffness to the drum brake frequency response function

the drum to the rim. Modal analysis conducted after removing the flanges decreased the modal density of the drum brake to 33 mode shapes. It indicates that the flanges contributed 25 mode shapes to the overall drum brake mode shapes. The effects of other parameters of the flanges were analysed by altering the thickness of the flanges and changing the modulus elasticity of the flanges.

The effects of various parameters related to flanges to the drum brake frequency response function can be observed in Figure 11. The initial drum brake has E of 69 GPa and flange thickness of 4 mm. Increasing the thickness of the flange has resulted in the reduction of the natural frequencies of the drum brake. Increasing the stiffness of the flange has significantly reduced the natural

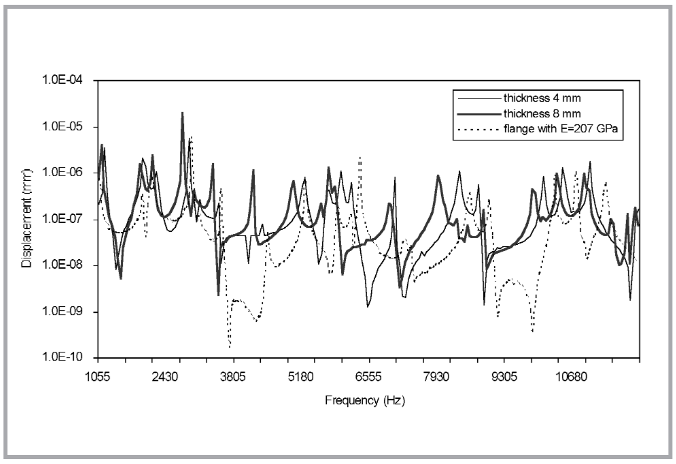

Figure 11: Effects of parameters related to flanges to the drum brake frequency response function

frequencies of the drum brake. The effect of various parameters for drum diametral mode shape frequencies is given in Table 3.

Effect of Shoe Back Plate Modulus of Elasticity

The flanges are the part where spokes are The influence of shoe back plate stiffness is analysed by changing the modulus elasticity of the shoe back plate from 69 GPa and density of 2720 kg/m^3 to 210 GPa and density of 7800 kg/m^3. Figure 12 gives the frequency response function in radial direction of the drum brake in reference to the lining node as the effect of this change.

DISCUSSION

Comparison of results from Figure 5 and Figure 7 when

Table 3: Effect of various parameters for drum diametral mode shape frequencies

	Initial Drum E=69 GPa, 4 mm flange	Drum with E=207 GPa 4mm fla	Drum with rib E=207 GPa	Drum with 8mm flange	Drum with 4mm flange
First diametral mode frequency	1170Hz	1039Hz	1309Hz	1102Hz	899Hz
Second diametral mode frequency	1955Hz and 2233Hz	1952Hz and 2119Hz	1974Hz and 2346Hz	1899Hz and 2146Hz	1734Hz and 1906Hz
Third diametral mode frequency	5213Hz	5243Hz	5228Hz	4993Hz	4488Hz

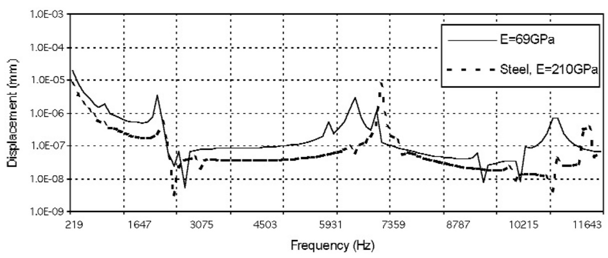

Figure 12: Frequency response function of the brake shoe for different stiffness and material properties, with damping ratio 0.04%, translation in radial direction

compared indicated high modal density of the drum brake and brake shoe with one mode at frequencies of 1500-2000 Hz and 3 modes at frequencies of 5600–6400 Hz, respectively. This mode proximity is potentially vulnerable to squeal since squeal in general occurs when two separate and distinct modes coalesce to form a mode pair, one of which is stable and the other unstable. The unstable mode manifests itself as squeal mode [10]. Since modal proximity is one of the criteria that can be used to prevent drum brake squeal at the design stage, parameters affecting the modes can be used to achieve this. In particular are the size and shape of the drum brake ribs whereby in the current design, the ribs consisted of radial stiffeners at 90 degrees angular space. The ribs cause mode separation of 278 Hz for the second radial mode which is essential for stability purposes and should be maintained. Removing the ribs causes the second radial mode pair to exist at a common frequency of 1955 Hz and this is to be avoided.

Changing the modulus of elasticity of the material does not have major influence on the number of mode shapes, which remains 42. The frequencies of the mode shifted to higher values, commensurate with the overall increase in stiffness.

The drum back plate is a dynamically important part in squeal generation where earlier works have shown that modifications carried out on drum back plate on automotive brake can stabilise the system [7,8]. With stiffer back plate ribs the overall displacement level goes down by one order of magnitude for mode shape which are related to the back plate deformation. Thus the back plate stiffening is beneficial for squeal reduction.

The effect of flanges thickness is to increase the mass of the drum but with no appreciable increase in stiffness of the back plate. This resulted in the overall decrease of the natural frequencies of the drum, particularly good frequency separation was obtained for modes in the frequency range of 5180 – 7930 Hz. If squeal occurs within this range, the modification of the flanges is one of the suitable remedies.

The shoe back plate that is currently made of cast aluminum alloy can be modified by the use of alternative materials of higher modulus of elasticity. The use of steel (E=207 GPa) will shift the natural frequency of the back plate. The overall results indicated that for the current motorcycle drum brake analysed, modal density is relatively high in the region of 1500 – 2000 Hz and 5600 – 6400 Hz. In the first range, the drum, stiffness can be increased to reduce the modal density as well as proximity. Adjusting the flange thickness to 8 mm reduces the modal density for the drum at the frequency range of 5600 – 6400 Hz, which will be one of the steps for reducing drum brake squeal propensity within the above range.

CONCLUSIONS

The analysis can be concluded as follows:

1. There are 42 mode shapes for the drum and 10 mode shapes for the shoe within the frequency range of 100 Hz and 12000 Hz.
2. Modal density is high in the frequency region of 1500 – 200 Hz and 5600 – 6400 Hz.
3. Increasing the stiffness of the drum is useful to reduce modal density for the lower frequency range (1500-2400 Hz) while increasing the flange stiffness reduces modal density in the higher frequency range (5600-6400 Hz).
4. The results have shown the important design parameters effect on the natural frequencies of the drum brake components. ■

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