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A study of tyre, cavity and rim coupling resonance induced noise

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Abstract: The tyre cavity resonance induced noise and vibration currently have no effective mass production solutions. All proposed solutions are hard to service and difficult to maintain. Different analysis approaches are presented in this paper to verify computer simulation models. Modal analysis results of the tyre cavity, tyre and rim structures were compared and a complex nature of tyre cavity acoustics is understood. Modified air cavities have been simulated. The research in this paper aims to find effective solutions for elimination of the coupled resonance mode by shifting the tyre cavity modal frequency.

Keywords: tyre cavity resonance; modal analysis; resonance induced noise; the fluid-solid coupling resonance.

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1 Introduction

As a vehicle is travelling on a road surface, contact forces between the tyre and the road deform the tyre tread and result in an excited acoustic field within the tyre cavity. The pressure fluctuation of the air cavity is transferred via the wheel carrier and suspension system into the vehicle body which then results in the noise within the cabin. At resonant frequencies the noise is loud and noticeable.

Tyre cavity noise is audible inside the vehicle cabin. Its root cause is still unclear. Scavuzzo et al. (1994) proposed two viable solutions for eliminating or minimising the tyre cavity noise, either to detune the transmission path from the tyre to the passenger compartment or to minimise the input which excites the cavity resonance. Scavuzzo et al.

(1994) developed a model for understanding the coupling mechanism between the fluid cavity and structural resonances. It was believed that passenger car wheels contain a rim structural resonance that can couple with the tyre acoustic cavity resonance to cause an amplification of interior vehicle noise; the frequency of the cavity resonance was defined solely by the size of the tyre and wheel; changes in tyre design or wheel design would have little or no effect on the presence of the acoustic cavity mode or on the frequency of the resonance, but may affect the transmission path between the tyre and the passenger compartment.

On the other hand, Molisani et al. (2003) investigated the coupled resonance of the tyre cavity and tyre structure, rather than the coupled resonance of the tyre cavity and rim structure. The root cause of the noise was believed to be the coupled resonance of the tyre cavity and tyre structure. Molisani et al. (2003) developed a closed form of analytical model for tyre cavity and tyre structure. In this case the tyre was modelled as an annular cylindrical shell where only the outer surface was flexible, the side walls of the tyre are assumed to be rigid. They presented a process or a tool which can be used to investigate the physical coupling between the acoustic cavity of the tyre and the tyre structure without using finite element models. The main contributions from Molisani et al. (2003) are the captures of some of effects of the tyre cavity resonance on the tyre dynamics and spindle forces and establishing the coupling coefficient between the tyre air cavity and the tyre structure. However, with such different results and conclusions from Molisani et al. (2003) and Scavuzzo et al. (1994), the root causes of the noise are still not clear.

Fernandez (2006) studied a rotational wheel in a cylindrical coordinate system where the natural frequencies and mode shapes of a tyre cavity torus were sought from its Eigen equations and expressed in form of the Bessel and Neumann functions. However, Fernandez did not study the coupling effect between the tyre air cavity and tyre or rim structure and did not identify the root cause of the noise. Fernandez presented various methods for materials to be mated to either the wheel rim circumferent surface, or the inner tyre surface in order to achieve sound absorption or damping effects, which provided interesting information in regard to the design concepts or changes within the tyre cavity. These methods are generally called ‘insert tyre cavity absorbers’ which are not durable and hard to be assembled in mass production.

Hayashi (2007) believed that the fluctuation pressure caused by cavity resonance acting on the rim and tyre as compelling force plus the low damping and high gain characteristics of the cavity made the tyre cavity noise reduction hard. Hayashi (2007) identified the out-of-plane torsional stiffness of the rim hub disk as one of the significant design parameters for reduction of the tyre cavity noise, as the fourth order camber resonance mode was identified as one of the important suspension amplification resonance modes which was related to the hub bearing stiffness and wheel inertia in the lateral axis of the y direction, and contributed to the tyre cavity resonance noise heard in the cabin. It can be deducted from Hayashi’s work that the wheel scrub radius may play a role on the suspension amplification or the transmission sensitivity of the noise which should be verified by future investigations.

Waisanen and Blough (2009) placed six microphones in different locations of a vehicle cabin, and measured the sound pressures when vehicle was driven at speeds of 48 km/h and 80 km/h on a concrete road surface. Two noise spectrum peaks were found to have close frequencies and amplitudes which generated booming or drumming like sound inside the cabin. Waisanen and Blough (2009) used transfer path analysis (TPA) to predict the path contributions for the road noise transmission in a vehicle. This was initially performed on the suspension system in order to identify the vehicle sensitivity for a transfer of the coupled wheel-tyre-cavity resonance to the vehicle cabin. Waisanen and Blough (2009) did not explore potential design solutions for the tyre cavity resonance induced noise, their study was not focused on identification of the root cause of the noise.

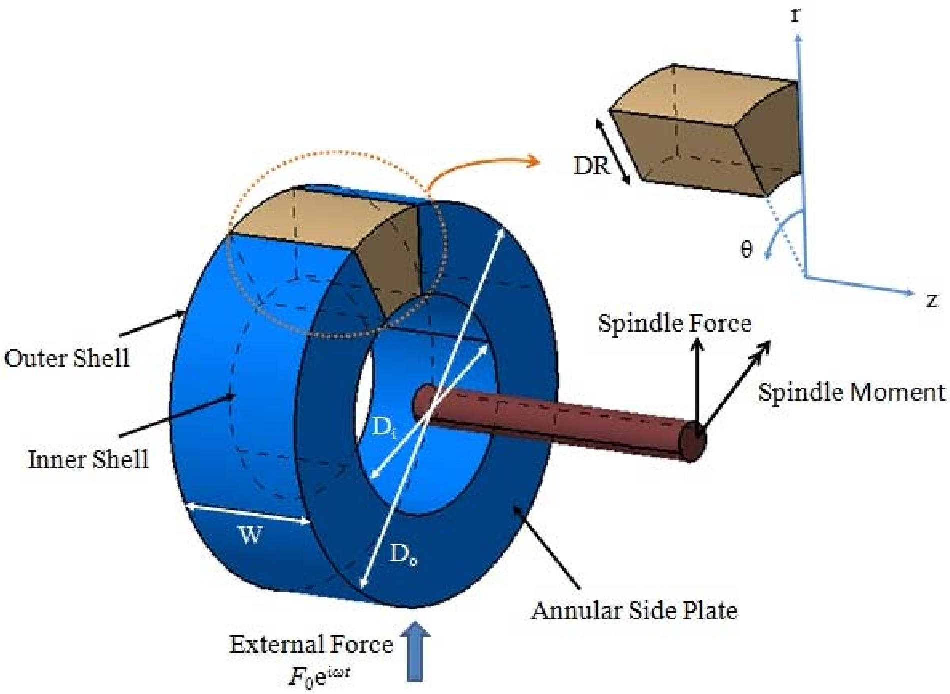
Elimination of the tyre cavity resonance noise as one type of unwanted noise generated and heard in vehicles has become an increasingly important vehicle design consideration as new sustainable power train technologies such as electric power train or hybrid power train technologies develop. There are many different models developed for various studies of the tyre noise. These models were assessed to determine which parameter was important and which model/s best predicted and illustrated the tyre cavity noise. These models and studies, along with experiments, are a very important starting point for analysis of the tyre cavity noise and for justifying CAE models and their modifications later on. The research gaps identified are that none of studies so far clearly indicates whether the coupling resonant mode is caused by the tyre cavity and the rim structure or by the tyre cavity and the tyre structure; none of the solutions is focused on tyre and wheel design changes, none of the solutions is low cost and feasible for mass production.

The problem will be analysed as three separate parts – one just for the rim structure, one for the tyre cavity and the other for the tyre structure. The resonant frequencies of the three parts will be found and compared with the measurement results to verify the tyre cavity resonant frequencies calculated by the analyses and simulations. By conducting simulations on the three separate parts, it should be demonstrated if there are any rim structure resonant frequencies are close to the tyre cavity resonant ones, whereupon coupling of the corresponding modes occurs. After validating the ANSYS finite element analysis (FEA) models, possible solutions will be explored for elimination of the coupling resonance noise. Alternative gases will be inserted into the tyre cavity to investigate the effects of changing the medium mass density. The advantage of the CAE simulation is that there are no needs for the rare alternative gases to be sourced for experimental purposes. Other potential solutions such as the tyre cavity volume changes will also be investigated by the CAE simulation. It is hoped that through the CAE simulation of the three parts and the changes to the tyre cavity, the tyre and rim structures, more lights should be shed on the potential solutions that are too difficult to test experimentally.

2 Computational model

The tyre acoustic cavity can be modelled as a toroid for an unloaded tyre as shown in Figure 1.

Figure 1 A simplified tyre toroid model (see online version for colours)



As the first order approximation, the frequency of the resonance is calculated using the Rayleigh method by Scavuzzo et al. (1994).

$$f = \frac{c}{L} \tag{1}$$

where L represents the circumference of the toroid at the centroid of the cross-section.

If the tyre cavity is considered a whole toroid with a sound wavelength equal to the circumference of the toroid at the centroid of its cross-section, the sound wavelength corresponding to the first modal frequency of the tyre cavity (i = 1) can be calculated from the tyre cavity dimension as

$$\lambda = \frac{L}{i} = \frac{\pi(D_o + D_i)}{2} \tag{2}$$

where D_o is the outer diameter of the tyre cavity toroid, D_i is the inner diameter of the tyre cavity toroid.

From equations (1) and (2), the first tyre cavity resonance frequency f can be written as

$$f = \frac{c}{\lambda} = \frac{2c}{\pi(D_o + D_i)} \tag{3}$$

If D_o is 0.6014 m, D_i is 0.3710 m, from equation (3), the tyre resonance frequency f is calculated to be 224.558 Hz.

Due to the spatial symmetry of the tyre cavity toroid, the tyre cavity toroid mode shape is assumed to be symmetric about at least one nodal plane passing through the centre. The nodal plane can be vertical or horizontal and assumed to be equivalent to a solid wall. The natural frequencies of the two symmetric cavities divided by the nodal plane are identical to the natural frequencies of the whole tyre cavity toroid and can be calculated from those of a rectangular cavity. The rectangular cavity has its length equal to the half circle arc length of the half symmetric cavity at the centroid of its cross-section, its width equal to the width of the tyre cavity toroid, its height equal to the height of the tyre cavity toroid.

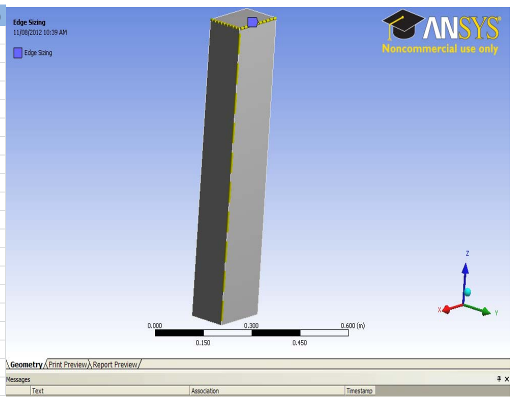
According to Nakanishi and Gerges (1995), the resonant frequencies of a rectangular air cavity prism with rigid walls are given by

$$f_{n_x,n_y,n_z} = c\sqrt{\left(\frac{n_x}{L_x}\right)^2 + \left(\frac{n_y}{L_y}\right)^2 + \left(\frac{n_z}{L_z}\right)^2} \tag{4}$$

where c is the speed of sound in medium (c = 343 m/s in dry air of 20°C); n_x, n_y, n_z are integer indices 0,1,2,3, … etc; L_x = 0.7635 m is the length of the rectangular air cavity prism and equal to a half of the circumference of the tyre air cavity toroid at the centroid of its cross-section; L_y = 0.185 m and L_z = 0.1152 m are the width and height of the rectangular air cavity prism and equal to the width and height of the tyre air cavity toroid. Substitution of L_x, L_y, L_z values into equation (4) gives the natural frequencies of the air cavity prism or the whole tyre air cavity toroid in Figure 2.

Figure 2 Comparison of calculated natural frequencies using the ANSYS FEA and analytical approach (see online version for colours)

Mode	n_x	n_y	n_z	Analytical (Hz)	ANSYS (Hz)	Errors (%)
1	0	0	0	0	0	0.00
2	1	0	0	224.6	225.3	-0.28
3	2	0	0	449.2	454.5	-1.15
4	0	1	0	927.0	940.8	-1.47
5	1	1	0	953.9	957.7	-0.41
6	0	0	1	1488.7	1494.8	-0.41
7	1	0	1	1505.6	1508.5	-0.19
8	0	1	1	1753.8	1761.0	-0.41
9	1	1	1	1768.1	1768.1	0.00
10	2	1	1	1810.4	1816.3	-0.33
11	0	2	0	1854.1	1884.7	-1.63
12	2	2	0	1907.7	1898.1	0.51
13	1	2	1	2388.4	2391.8	-0.14
14	2	2	1	2419.8	2416.4	0.14
15	0	0	2	2977.4	2973.9	0.12
16	2	0	2	3011.1	3020.5	-0.31
17	1	1	2	3126.5	3132.5	-0.19
18	2	1	2	3150.6	3138.2	0.40
19	0	2	2	3507.5	3508.2	-0.02
20	1	2	2	3514.7	3529.8	-0.43



It is seen that the tyre resonant frequency f =224.56 Hz calculated by the Rayleigh method of equation (3) is very close to the natural frequency of the tyre cavity toroid, $f_{1,0,0}$ = 224.62 Hz calculated by equation (4) which is one of the natural frequencies of the air cavity prism.

In order to validate the FEA method using the software of ANSYS, the natural frequencies of the above rectangular air cavity prism are calculated using the ANSYS FEA method and are compared with those using the analytical approach.

The ANSYS FEA model for the air cavity prism is shown Figure 2 where the dimensions of L_x, L_y and L_z are given as above, each dimension of the air cavity prism is divided ten increments for a mesh size. Air mass density is assumed to be 1.12 kg/m^3. The results of both the analytical calculation and ANSYS FEA are shown in Figure 2.

It is seen from Figure 2 that the natural frequencies calculated by the ANSYS FEA method are very close to those calculated by the analytical approach. For example, for the mode of 1, 0, 0, the natural frequency calculated by the ANSYS FEA method is 225.3 Hz, while the natural frequency calculated by the analytical approach is, $f_{1,0,0}$ = 224.6 Hz as shown in Figure 2, which has verified the ANSYS FEA method. In the analytical approach, it is assumed that the whole tyre air cavity mode shape is symmetric about at least one nodal plane passing through the centre. The natural frequency of the whole tyre air cavity and the divided symmetric cavities is identical. From the ANSYS FEA method, the natural frequency of 225.3 Hz has been obtained which has verified the assumption in the analytical approach. The difference of the natural frequencies of 224.6 Hz and 225.3 Hz obtained from the analytical approach and ANSYS FEA method could be caused by meshing size.

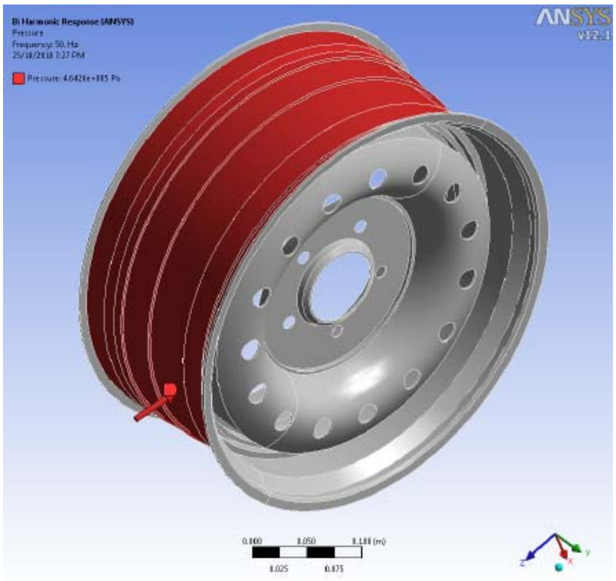
The initial CAD models of the wheel rim, tyre structure and tyre cavity were built using CATIA V5R18. The three separate CAD models of the rim structure, tyre structure and tyre cavity were imported to the ANSYS software for the finite element modal analysis. The rim structure, tyre structure and tyre cavity are analysed separately, as they have different material properties. In order to model the wheel, the wheel profile dimension was measured and used to generate ANASYS FEA model. It is important that the measurement errors and the difficulties encountered during the measurement of the complex tyre side wall profile shapes are taken into account when considering the modelling accuracy.

Shown in Figure 3 is a rim structure model imported from CATIA to ANSYS where meshing was done automatically by the ANSYS software. As mentioned before, for the rectangular air cavity prism, meshing size was specified as ten divisions of each of its orthogonal leading edges. For the complex geometry model such as tyre air cavity, 20 divisions of each of its orthogonal leading edges were applied in the ANSYS software which led to finer meshes.

After meshing, material properties and boundary conditions and constraints were assigned to the FEA models of the rim, tyre and the cavity. For example, the rim was fixed and supported at the mounting holes and the central hub bearing hole, 34 psi pressure (234.3 kPa) was applied onto the rim surface as shown in the red arrow in Figure 3. Young’s modulus of the steel rim is 210 GPa, Poisson’s ratio of the steel rim is 0.3, and the mass density of the steel rim is 7850 kg/m^3. The tyre was fixed and supported at the bead cord edges which contact with the rim, the 34 psi pressure (234.3 kPa) was applied onto the tyre inner surface in the normal directions. Young’s modulus of the tyre is 481 MPa, Poisson’s ratio of the tyre is 0.49, and the mass density of the rubber tyre is 1200 kg/m^3.

Modal analysis was performed using the ANSYS software where modal frequencies and mode shapes of the tyre air cavity, tyre structure and rim structure were calculated.

Figure 3 Rim structure geometry model imported from CATIA to ANSYS and applied tyre pressure distribution and constrained boundary conditions for the rim FEA model (see online version for colours)



3 Experimental model

A wheel was suspended by two bungee cords as shown in Figure 4 where the wheel was a standard steel wheel, 15” diameter by 6” wide. The tyre is a tubeless Bridgestone RE 92 in 205/65/15 guise with standard steel belt and a tyre pressure of 34 psi for all tests.

Figure 4 Testing wheel tyre assembly suspended by two bungee cords (see online version for colours)

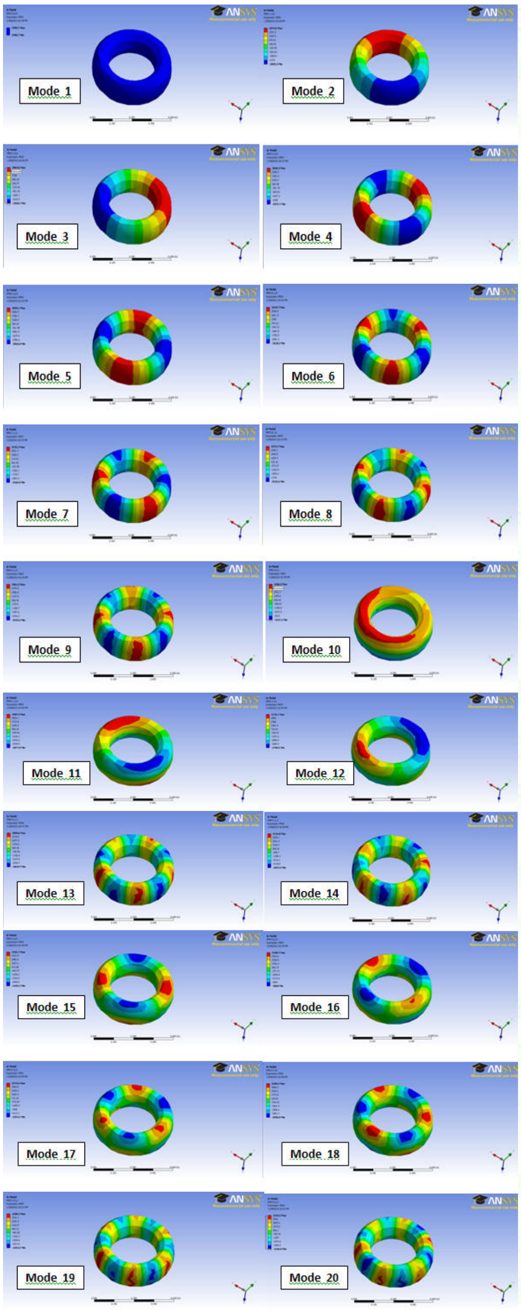
A tri-axial accelerometer Bruel & Kjaer 4506B was mounted on the rim disk surface near the central wheel hub as shown in Figure 5. An impact hammer mounted with force transducer (PCB 086C03) and steel hammer cap was used to impact one spot of the tyre side wall in its normal direction. The output of the force transducer was connected to PCB Piezotronics Model 480E09 signal conditioner, the output of the signal conditioner and the three outputs of the tri-axial were connected to Bruel & Kjaer Pulse data acquisition front end. Frequency response and coherence functions were measured and analysed by use of the Brüel & Kjær Pulse system and its software LabShop v12.5 from which the resonant frequencies of the wheel-tyre assembly and components were identified. Each of the measured results was averaged on three times. The measured FRF amplitude and coherence curves in the x, y and z directions are shown in Figure 15.

Figure 5 The mounting position of the tri-axial accelerometer on the rim disk (see online version for colours)

4 Results and discussions

From results of both the simulation and experiment, it is observed that there are slight differences in the calculated and measured resonant frequencies. It is seen that there are pairs of close natural frequencies appear as shown in Figure 6 because of the tyre cavity spatial symmetry. The mode shapes of 229.23 Hz and 231.32 Hz are shown to have the two half cavity toroids in the whole tyre cavity toroid with the anti-symmetric pressure distribution and are symmetric about one horizontal or vertical nodal plane which have verified the assumption in the above analytical approach. Scavuzzo et al. (1994) also observed a pair of the two modal frequencies close to each other around 220 ~ 230 Hz. They believed one modal frequency corresponded to the vertical cavity resonance mode shape with a horizontal nodal symmetric plane, the other modal frequency corresponded to the fore/aft cavity resonance mode shape with a vertical nodal symmetric plane. The pair of the two close cavity resonant frequencies were concluded to be caused by the tyre deformation under a load. However, the above presented pair of cavity resonance modes in Figure 6 are shown to have their natural frequencies close to each other for unformed tyres, which have directly denied the conclusions from Scavuzzo et al. (1994).

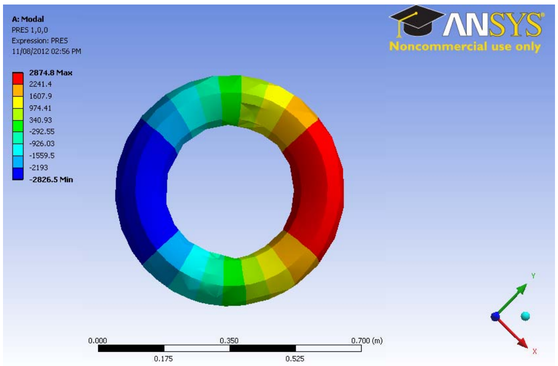
Figure 6 CAE simulation model – the tyre air cavity resonant frequency and mode shape pair, (a) the cavity modes (b) mode 2: 229.23 Hz (c) mode 3: 231.32 Hz (see online version for colours)



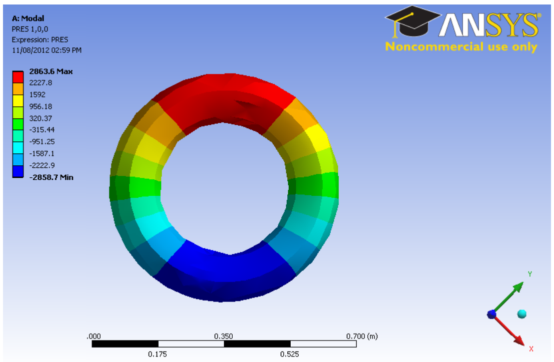
Mode	Frequency [Hz]
1.	0.
2.	229.23
3.	231.32
4.	460.42
5.	462.27
6.	692.76
7.	696.19
8.	926.2
9.	937.6
10.	1079.
11.	1107.
12.	1114.8
13.	1171.5
14.	1180.8
15.	1195.4
16.	1201.5
17.	1330.9
18.	1334.1
19.	1423.3
20.	1436.3

(a)

Figure 6 CAE simulation model – the tyre air cavity resonant frequency and mode shape pair, (a) the cavity modes (b) mode 2: 229.23 Hz (c) mode 3: 231.32 Hz (continued) (see online version for colours)



(b)



(c)

The FEA simulation with the ANSYS software presents a pair of the 2nd and 3rd modes of the tyre cavity having modal frequencies of 229.23 Hz and 231.32 Hz, whereas the experiment yielded only a resonant mode at 232 Hz as shown in Figure 15. The other mode might not be excited out in the experiment. The difference of about 1 Hz between the simulation model and experimental model is expected due to the approximation in construction of the tyre cavity geometry model. The CAD geometry model approximations would lead to a small variation in the volume of the cavity. This immediately would lead to different calculation results for the resonant frequencies of the cavity. The difference of about 1 Hz would also be because that the simulated modal frequency is only related to the tyre air cavity while the measured modal frequency is related to the whole wheel tyre assembly. Both the CAE and experimental models show that their results are close to each other, which has validated the simulation and testing methods.

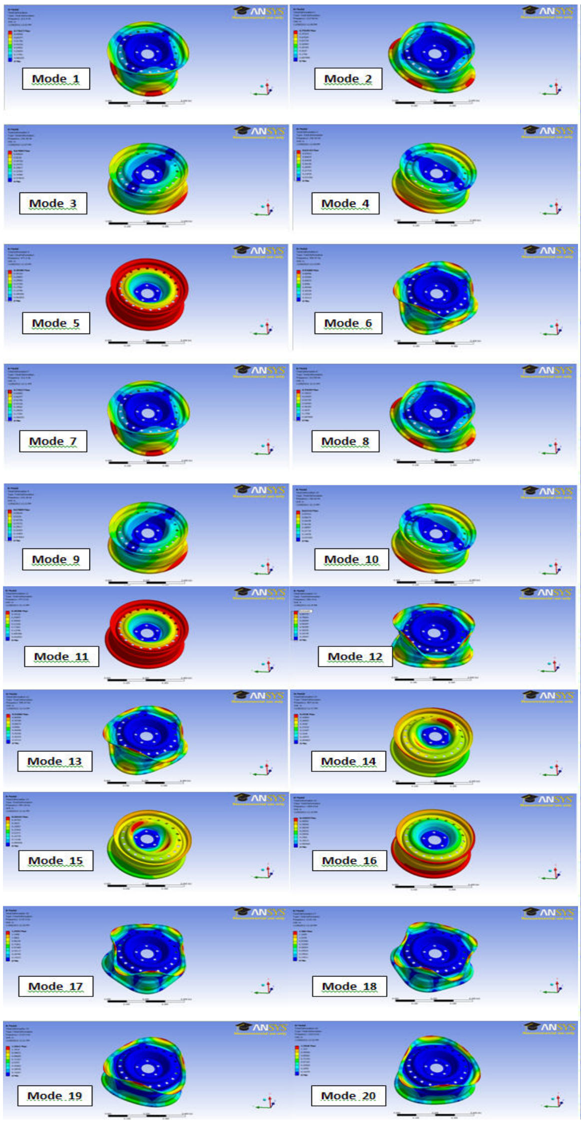
The calculated results according to equations (3) and (4) are 224.56 Hz and 224.62 Hz, which are close to 225.3 Hz, 229.23 Hz and 231.32 Hz calculated from the ANSYS FEA models. Equation (3) is based on the Raleigh's energy method, which ignores the geometry details of the wheel and tyre. Equation (4) is based on the assumption that the whole tyre cavity toroid consists of two half symmetric cavity toroids which are symmetric about one nodal plane. The two half symmetric cavity toroids have the same natural frequency as the whole tyre cavity toroid. The half cavity toroid can be treated as a rectangular cavity prism for the natural frequency calculation where the cylindrical wave is approximated by a planar wave in the radial direction. This has explained the causes of the differences of 224.56 Hz calculated by equation (3) and 224.62 Hz calculated by equation (4), 225.3 Hz calculated from the rectangular air cavity prism model using the ANSYS FEA method, 229.23 Hz and 231.32 Hz calculated from the measured tyre air cavity CAD geometry model using the ANSYS FEA method. The latter is most accurate since there are the errors in the dimensions and volume in comparison of the simplified tyre air cavity toroid model as shown in Figure 1 with the measured tyre air cavity CAD geometry model as shown in Figure 6. There are also the calculation errors of the natural frequency due to the planar wave approximation in the rectangular air cavity prism model for the cylindrical wave in the radical direction in the tyre air cavity model.

The simulation results of the rim structure modal frequencies using the ANSYS FEA method are shown in Figure 7. Pairs of close natural frequencies appear for the reason of the rim structure symmetry which is same as that for the tyre cavity. A pair of the rim structure modes are shown to have natural frequencies of 234.4 Hz and 236.6 Hz which are close to the tyre cavity modal frequency pair of 229.23 Hz and 231.32 Hz. This would lead to a coupling resonant mode of the rim structure and the tyre air cavity around those frequencies, would generate and transmit resonant vibration or noise energy from the wheel to vehicle chassis around this frequency.

As seen from Figure 7 the mode shapes of the modal frequency pair 234.4 Hz and 236.6 Hz have a contour pattern phase difference of 90 degree from each other. The mode shapes of 234.4 Hz and 236.6 Hz have shown the out-of-plane torsion motions of the rim hub disk which have verified the conclusions from Hayashi (2007). In addition, the rim mode shapes of 954.4 Hz and 957.6 Hz have also shown the similar motions to those of 234.4 Hz and 236.6 Hz. The rim modal frequencies of 954.4 Hz and 957.6 Hz are close to the tyre cavity modal frequencies of 926.2 Hz and 937.6 Hz which could also lead to another coupling mode around those frequencies and contribute to the coupling resonance induced noise transmitted into the cabin.

Elimination of the coupled resonance induced noise is then reduced to separation of the modal frequencies of the tyre cavity and the rim structure, or to increasing the gap between the two modal frequencies. The modal frequencies of the rim structure measured from the impact frequency response functions are 212 Hz and 248 Hz as shown in Figure 13, which have verified those pairs of 212.4 Hz and 212.6 Hz, and 234.4 Hz and 236.6 Hz calculated from the ANSYS FEA simulation as shown in Figure 7. One mode corresponding to the modal shape pair of 212.4 Hz and 212.6 Hz was not excited out in the experiment.

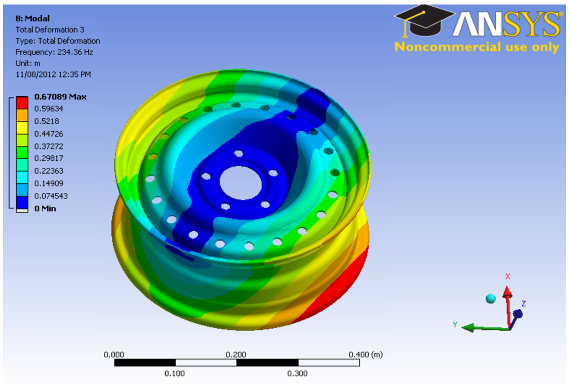
Figure 7 CAE simulation model – the rim structure resonant frequency and mode shape pairs, (a) the rim modes (b) mode 3: 234.4 Hz (c) mode 4: 236.6 Hz (see online version for colours)



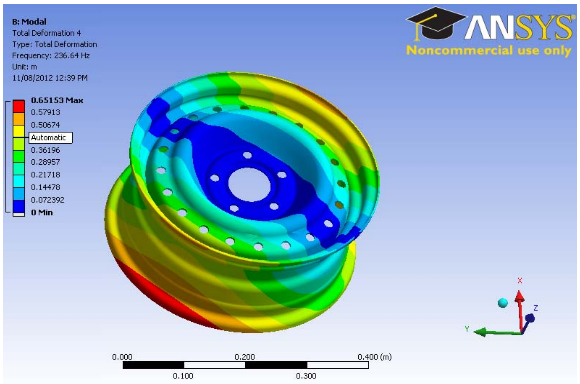
Mode	Frequency [Hz]
1.	212.4
2.	212.58
3.	234.36
4.	236.64
5.	477.2
6.	595.87
7.	596.4
8.	954.38
9.	957.62
10.	1004.5
11.	1118.
12.	1119.3
13.	1219.4
14.	1222.6
15.	1333.2
16.	1333.4
17.	1526.4
18.	1531.
19.	1719.7
20.	1720.

(a)

Figure 7 CAE simulation model – the rim structure resonant frequency and mode shape pairs, (a) the rim modes (b) mode 3: 234.4 Hz (c) mode 4: 236.6 Hz (continued) (see online version for colours)



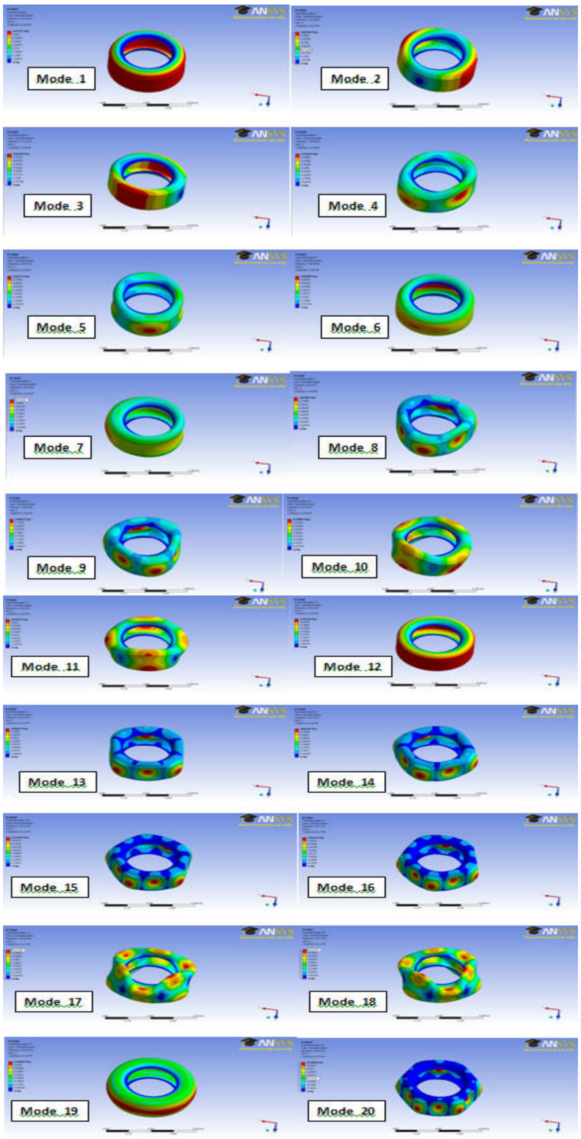
(b)



(c)

The tyre structure geometry model was also imported from the CATIA software to the ANSYS software, to form a tyre structure FEA model. Meshing, material properties and boundary conditions were applied to the FEA model. The focused tyre structure modal frequency pair was found to be 222.66 Hz and 224.03 Hz. The focused rim structure modal frequency pair of 234.4 Hz and 236.6 Hz are closer to the tyre air cavity modal frequency pair of 229.23 Hz and 231.32 Hz than those of the tyre structure of 222.66 Hz and 224.03 Hz. It is clear that the rim structural resonance will more likely couple with the tyre air cavity resonance. Therefore, the root cause of the coupled resonance induced noise includes both the coupling possibilities between the tyre air cavity and the rim structure resonance modes and between the tyre air cavity and tyre structure resonance modes. Decoupling of the tyre air cavity and the tyre and rim structure resonance modes is expected to play a key role in elimination of the tyre noise. The focused tyre structure modal frequency measured from the impact frequency response functions is 222 Hz as shown in Figure 14, which has verified the pair of 222.66 Hz and 224.03 Hz calculated from the ANSYS FEA simulation as shown in Figure 8.

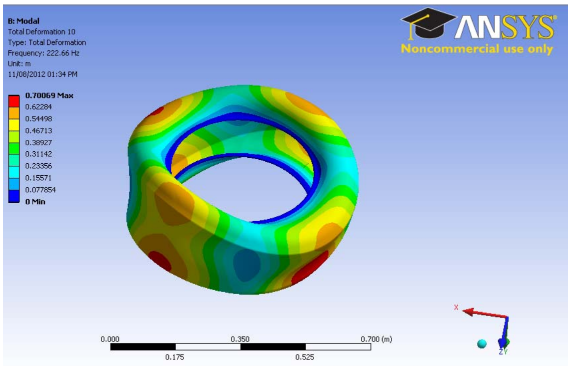
Figure 8 CAE simulation model – the tyre structure resonant frequency and mode shape pairs, (a) the tyre structure modes (b) mode 10: 222.66 Hz (c) mode 11: 224.03 Hz (see online version for colours)



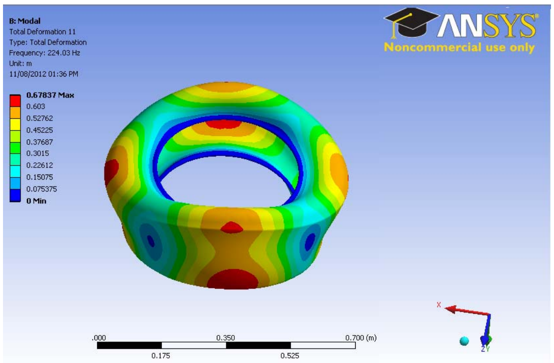
Mode	Frequency [Hz]
1.	69.479
2.	113.28
3.	113.43
4.	194.98
5.	195.61
6.	198.58
7.	198.74
8.	220.5
9.	220.92
10.	222.66
11.	224.03
12.	242.23
13.	254.24
14.	254.53
15.	292.46
16.	292.7
17.	324.64
18.	326.17
19.	334.76
20.	336.97

(a)

Figure 8 CAE simulation model – the tyre structure resonant frequency and mode shape pairs, (a) the tyre structure modes (b) mode 10: 222.66 Hz (c) mode 11: 224.03 Hz (continued) (see online version for colours)



(b)

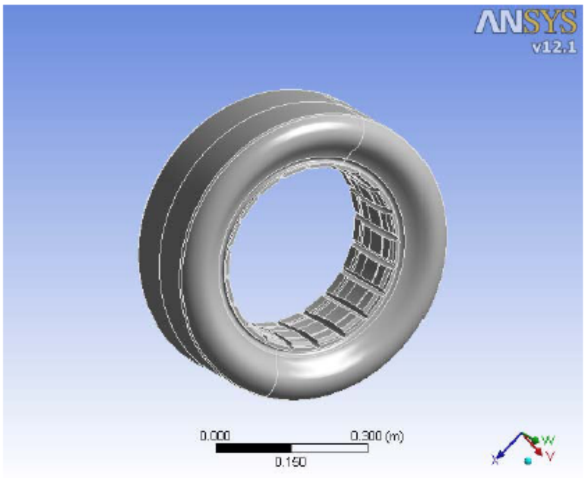


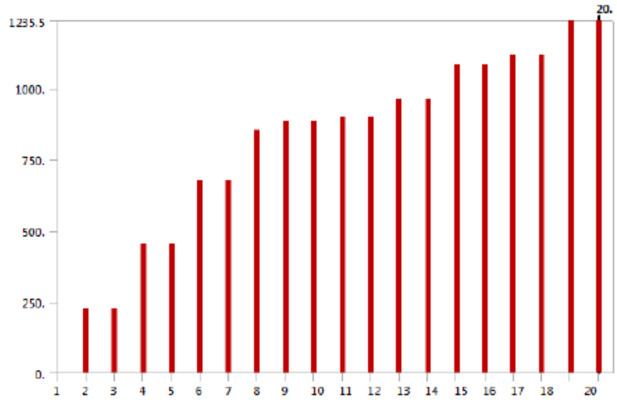
(c)

Shown in Figure 9 are the simulation results of the tyre cavity with ribs and slots added to the rim circumference surface. The pair of the 2nd and 3rd modal frequencies are 225.14 Hz and 225.22 Hz in comparison with the baseline pairs of the tyre air cavity modal frequency pair of 229.23 Hz and 231.32 Hz, which reduces the frequency gap between the tyre cavity and tyre structure resonance modes and slightly increases the frequency gap between the tyre cavity and rim structure resonance modes. It is not able to eliminate the coupled resonance modes of the tyre air cavity, tyre and rim structures. Shown in Figure 10 are the simulation results of the tyre cavity with textures added to the inner tyre surface. The pair of the 2nd and 3rd modal frequencies are 227.02 and 227.03 Hz and not changed much. Both the solutions of adding ribs or slots on the rim circumference surface and adding textures on the inner tyre surface are not effective to shift the tyre air cavity modal frequency pair away from the tyre and rim structure modal frequency pairs, therefore are not effective to remove the coupling resonant mode.

Figure 9 CAE simulation model – the modal frequencies for tyre cavity with the rim circumference surface ribbed (see online version for colours)

Mode	Frequency [Hz]
1.	0.
2.	225.14
3.	225.22
4.	449.86
5.	450.
6.	673.86
7.	674.
8.	853.73
9.	882.4
10.	882.64
11.	896.9
12.	897.17
13.	963.42
14.	963.85
15.	1084.9
16.	1085.5
17.	1119.1
18.	1119.6
19.	1234.8
20.	1235.5
Bounding Box	
Length X	0.5952 m
Length Y	0.2285 m
Length Z	0.5952 m
Properties	
Volume	3.4842e-002 m³
Mass	273.51 kg
Density	1.20 kg/m^3
Speed of Sound	343 m/s

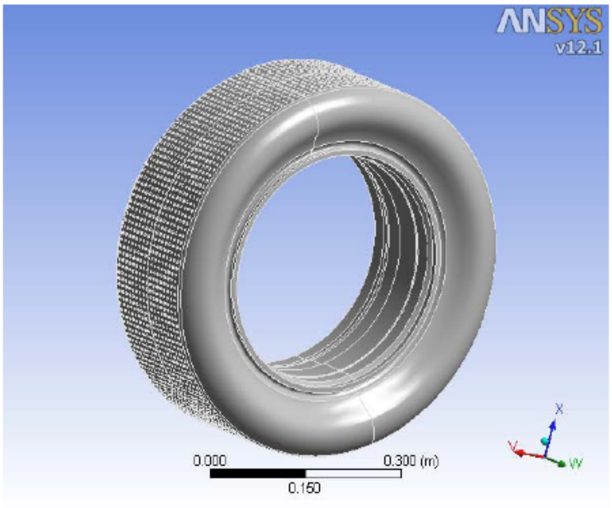




It is not easy to alter the acoustic modes of the tyre cavity without performing substantial modification to the tyre or wheel design, in particular, to the tyre cavity volume. Sound absorption foam or fibre felt fitted on the rim circumference surface within the cavity is a plausible answer but it is not a satisfactory solution because of the durability and manufacturing assembly readiness. Interestingly, the testing results for the tyre filled with nitrogen gas only shift the pair of the second and third modal resonant frequencies toward slightly higher frequencies as shown in Figure 11. Unfortunately the frequency shifts of these two modes are not large enough to make their modal frequencies stay away from the modal resonant frequencies of the tyre and rim structures. There is a high possibility for a coupling resonance mode of the tyre air cavity, the tyre and rim structures which would induce the cabin noise. The tyre filled with nitrogen gas was tested, but both the experimental and simulation results show very little change of the modal resonant frequency of the coupled mode, as shown in Figures 11 and 15. The reason may be that the mass density of the nitrogen gas is very close to that of the air. The most promising simulation is the tyre cavity filled with helium gas which shows the pair of the second and third resonant frequencies shifting upwards by about 400 Hz as shown in Figure 12. As the mass density of the helium gas is much smaller than that of the air, the tyre cavity resonant frequency pair becomes 643.44 Hz and 643.66 Hz which are now far away from the rim resonant frequency pair of 234.4 Hz and 236.6 Hz and the tyre structure resonant frequency pair of 222.66 Hz and 224.03 Hz. Inserting the helium gas into the tyre cavity eliminates the coupling resonant mode.

Figure 10 CAE simulation model – the modal frequencies for tyre cavity with inner tyre surface textured (see online version for colours)

Mode	Frequency [Hz]
1.	2.0164e-005
2.	227.02
3.	227.07
4.	453.24
5.	453.42
6.	678.17
7.	678.33
8.	854.2
9.	882.85
10.	883.1
11.	901.34
12.	901.51
13.	963.71
14.	964.22
15.	1085.
16.	1085.2
17.	1122.7
18.	1122.9
19.	1233.8
20.	1234.5
Bounding Box	
Length X	0.5952 m
Length Y	0.2285 m
Length Z	0.5952 m
Properties	
Volume	3.4842e-002 m³
Mass	273.51 kg
Density	1.20 kg/m^3
Speed of Sound	343 m/s



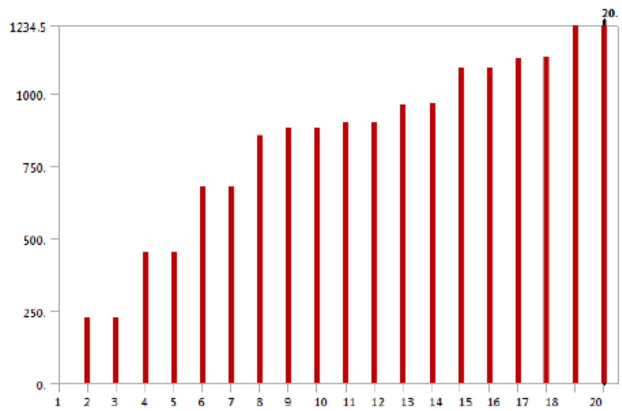
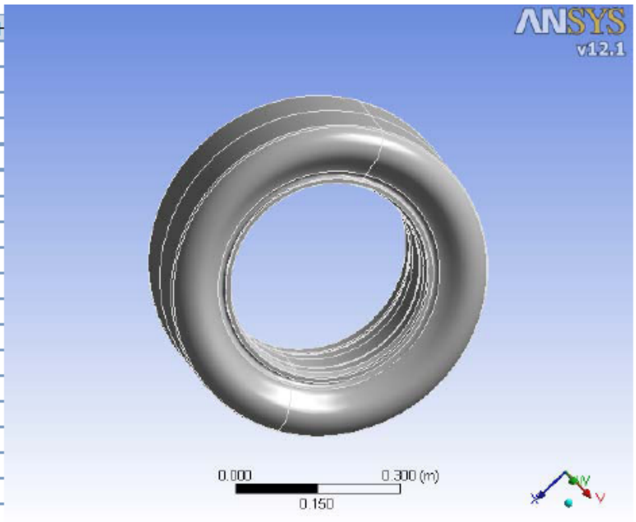


Figure 11 CAE simulation model – the modal frequencies for tyre cavity filled with nitrogen gas (see online version for colours)

Mode	Frequency [H
1.	0.
2.	233.68
3.	233.74
4.	466.67
5.	466.85
6.	698.59
7.	698.7
8.	880.31
9.	910.03
10.	910.41
11.	929.07
12.	929.21
13.	994.28
14.	994.52
15.	1120.1
16.	1120.6
17.	1158.1
18.	1158.6
19.	1274.9
20.	1275.9
Bounding Box	
Length X	0.5952 m
Length Y	0.2285 m
Length Z	0.5952 m
Properties	
Volume	3.4842e-002 m³
Mass	273.51 kg
Density	1.185 kg/m^3
Speed of Sound	353 m/s



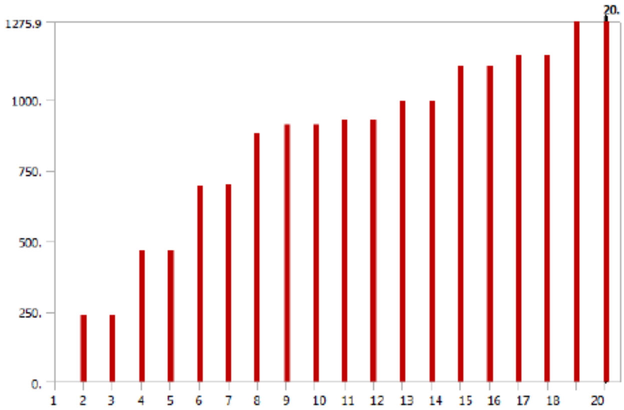


Figure 12 CAE simulation model – the modal frequencies for tyre cavity filled with Helium gas (see online version for colours)

Modal Analysis

Mode	Frequency [Hz]
1.	0.
2.	643.44
3.	643.62
4.	1285.
5.	1285.5
6.	1923.6
7.	1923.9
8.	2424.
9.	2505.8
10.	2506.8
11.	2558.2
12.	2558.6
13.	2737.8
14.	2738.5
15.	3084.2
16.	3085.6
17.	3188.8
18.	3190.4
19.	3510.4
20.	3513.2
Bounding Box	
Length X	0.5952 m
Length Y	0.2285 m
Length Z	0.5952 m
Properties	
Volume	3.4842e-002 m³
Mass	273.51 kg
Density	0.1786 g/L
Speed of Sound	972m/s

Figure 13 Experimental model for impact modal tests of the fixed mounted rim structure – measured frequency response amplitude curves in the directions of x and y (see online version for colours)

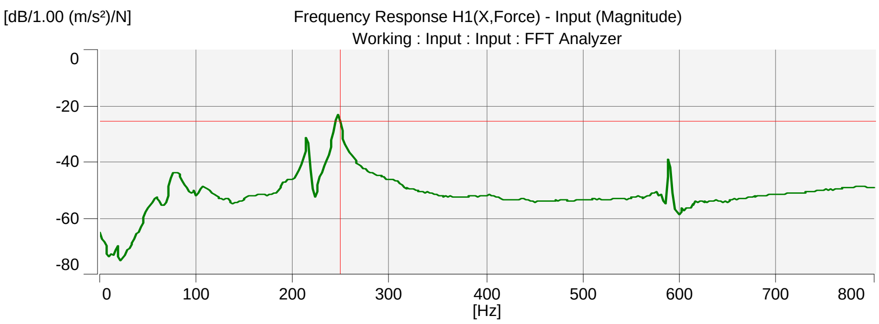


Figure 13 Experimental model for impact modal tests of the fixed mounted rim structure – measured frequency response amplitude curves in the directions of x and y (continued) (see online version for colours)

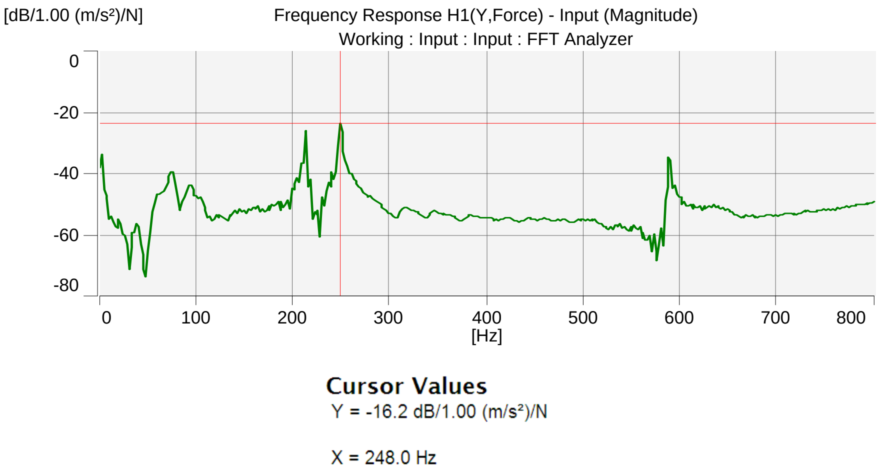
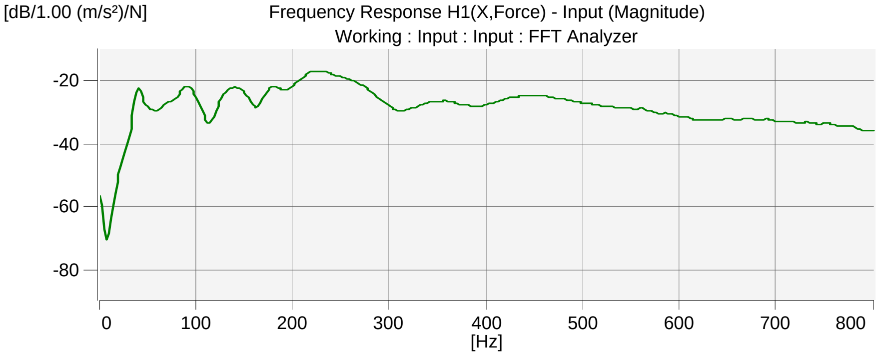


Figure 14 Experimental model for impact modal tests of the tyre structure - measured frequency response amplitude curves in the directions of x and y (see online version for colours)



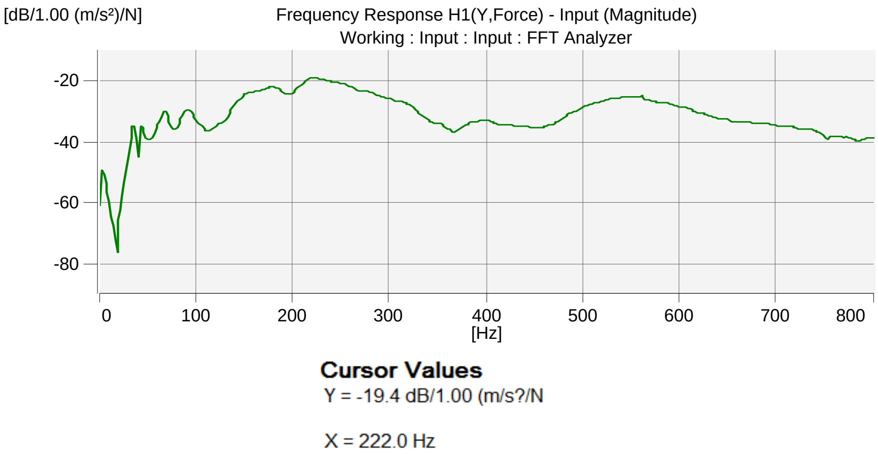
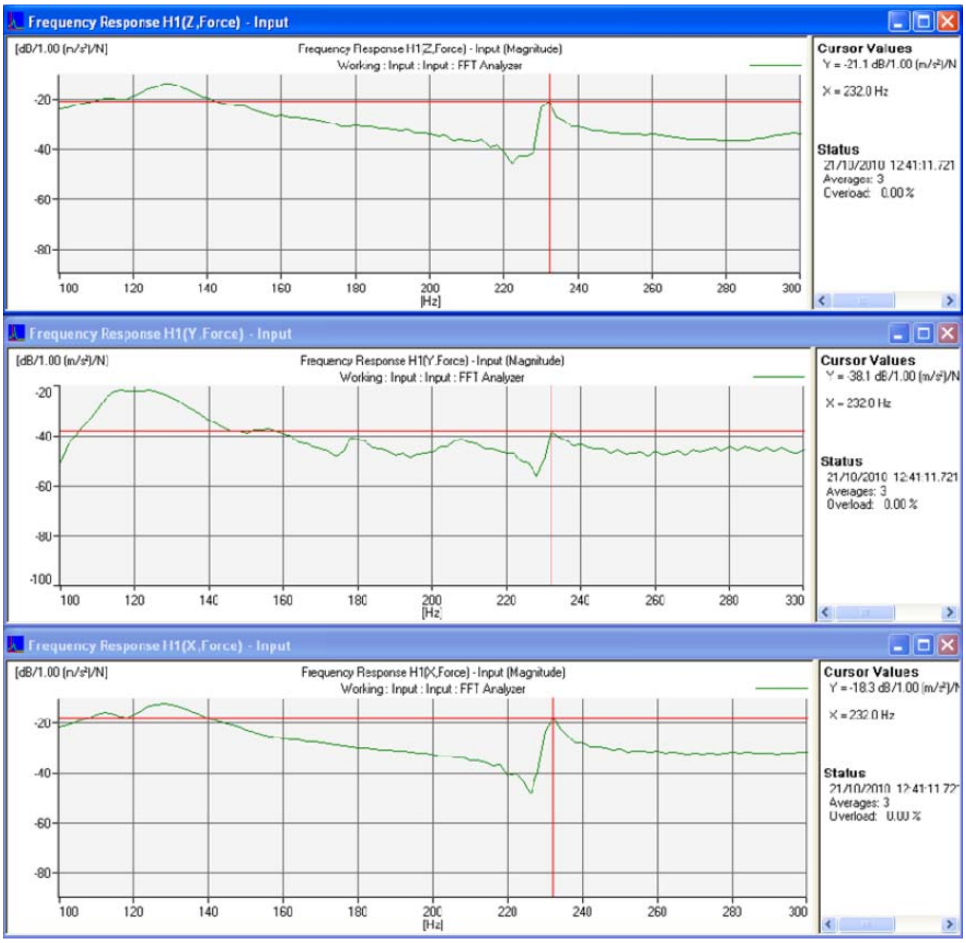


Figure 15 Experimental model for impact modal tests of the tyre-wheel assembly – measured frequency response function amplitude curves in the directions of x, y and z for tyre cavity filled with the air or nitrogen gas (see online version for colours)



The introduction of internal cavities within the wheel itself may create a larger volume for the cavity and hence would decrease the pair of the second and third resonant frequencies to acceptable values and avoid the coupling resonant problem. But this is entirely dependent on the wheel size and available volume of the wheel material which can be removed safely without compromising the integrity of the wheel structure. Another discussed solution is to add a rubber band around the rim circumference surface where four flaps are attached to the band which, under the centrifugal force when the tyre spins, will erect outwards in the radial direction, as shown in Figure 16. This effectively creates four separate volumes within the tyre cavity. The quarter tyre air cavity was simulated where the pair of the second and third modal frequencies are shown to move from 229.23 Hz and 231.32 Hz to 452.45 Hz, again far away from the rim resonant frequency pair of 234.4 Hz and 236.6 Hz and the tyre structure resonant frequency pair of 222.66 Hz and 224.03 Hz as shown in Figure 17. Similar to the absorptive band solution, implementation of this quarter cavity solution with separator flaps is subject to resolution of the issues such as the durability and manufacturing assembly.

In addition to shifting the resonant frequencies of the tyre cavity, the resonant frequencies of the tyre and rim structures can also be shifted away from the resonant frequency of the tyre cavity by design changes of the tyre and rim structures, which will be investigated in future work.

Figure 16 Tyre air cavity with a built in rubber band and four separator flaps (see online version for colours)

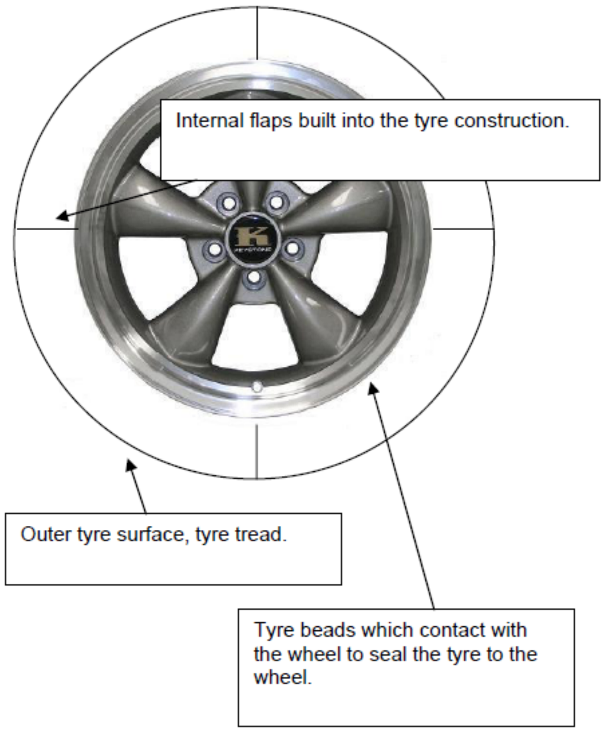
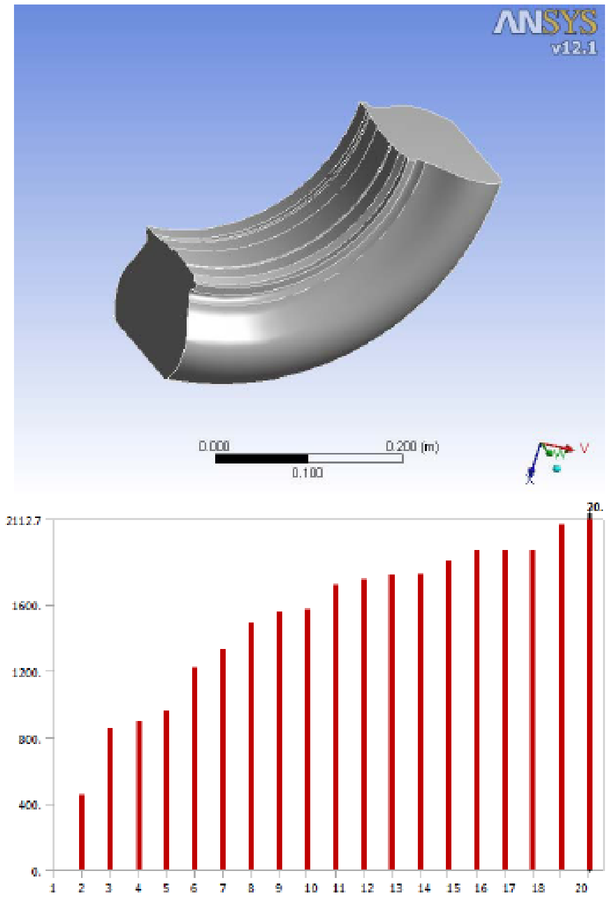


Figure 17 CAE simulation model – modal analysis: resonant frequencies of a quarter tyre air cavity (see online version for colours)

Modal Analysis

Mode	Frequency [Hz]
1.	5.859e-006
2.	452.45
3.	851.23
4.	897.34
5.	958.38
6.	1221.5
7.	1330.
8.	1491.2
9.	1557.8
10.	1569.2
11.	1719.1
12.	1750.6
13.	1772.6
14.	1783.2
15.	1857.1
16.	1923.7
17.	1924.8
18.	1926.3
19.	2078.8
20.	2112.7
Bounding Box	
Length X	0.2976 m
Length Y	0.2285 m
Length Z	0.2976 m
Properties	
Volume	8.7099e-003 m³
Mass	68.372 kg
Density	1.20 kg/m³
Speed of Sound	343 m/s



5 Conclusions

In this paper, the ANSYS FEA method was first validated by comparing the calculated modal frequencies of an air cavity prism using the finite element analysis method with those using an analytical approach. In the analytical approach, the tyre cavity mode shape is assumed to be symmetric about at least one nodal plane passing the centre. The whole tyre cavity toroid is divided by the nodal plane into two symmetric half cavity toroids which have the same natural frequencies as those of the whole tyre cavity toroid. The symmetric half cavity toroid is treated as a rectangular air cavity prism for calculation of natural frequencies. This assumption has been validated by a comparison of the analytically calculated and ANSYS FEA simulated tyre cavity resonant frequencies.

The ANSYS FEA models for the tyre cavity, tyre and rim structures are then established. The FEA model of the tyre cavity is validated by comparing the calculated modal frequencies of the tyre air cavity using the FEA method with those using the analytical approach and the Rayleigh energy method. The FEA model is also validated by comparing the calculated modal frequencies of the tyre air cavity using the FEA method with the measured modal frequency from the impact modal analysis tests.

It is found the focused modal frequencies of the tyre cavity are close to those of the rim and tyre structures, which may generate a coupled resonance of the tyre cavity, tyre and rim structures which could cause the cabin noise concern. The focused modal frequencies of the tyre cavity are closer to those of the rim structure than those of the tyre structure. The coupling resonance induced noise is more likely caused by a coupling mode of the tyre cavity and the rim structure than caused by that of the tyre cavity and the tyre structure. Elimination of the coupling resonant mode can be achieved by shifting the focused tyre cavity modal frequencies far away from the modal frequencies of the rim structure. Design concepts of adding textures onto the inner tyre surface and adding ribs and slots onto the wheel rim circumference surface have been simulated and tested using the ANSYS finite element model of the tyre cavity, it is found that no enough frequency shift has been achieved by the design concepts. Nitrogen gas was used to replace the air inside the tyre cavity where very little frequency shift was found by the FEA of the tyre cavity and the impact modal analysis tests. Design proposals of inserting the helium gas in place of the air inside the tyre cavity and generating the quarter tyre air cavity by use of a rubber band with four evenly distributed light weight thin polymer flaps attached on the rim circumference surface are simulated and tested using the ANSYS finite element model. It is found both the proposals are effective in shifting the focused modal frequencies of the tyre cavity from those of the rim and tyre structures. Implementation of the Helium gas insert inside tyre cavity is subject to cost, and adoption of the rubber band and the flaps is subject to their durability. Change of the rim and tyre structure design leading to a substantial shift of the focused modal frequencies of the rim and tyre structures or reduction of the wheel scrub radius may be another alternative for further investigations.

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References

Fernandez, E.T. (2006) *The Influence of Tyre Air Cavities on Vehicle Acoustics*, KTH, Superseded Departments, Aeronautical and Vehicle Engineering, Doctoral thesis.

Hayashi, T. (2007) *Experimental Analysis of Acoustic Coupling Vibration of Wheel and Suspension on Tyre Cavity Resonance*, SAE Paper 2007-01-2345.

Molisani, L.R., Burdisso, R.A. and Tsihlas, D. (2003) 'A coupled tire structure/acoustic cavity model', *International Journal of Solids and Structures*, Vol. 40, No. 19, pp.5125–5138.

Nakanishi, E.Y. and Gerges, S.N.Y. (1995) *Acoustic Modal Analysis for Vehicle Cabin*, SAE Paper 952246.

Scavuzzo, R.W., Charek, L.T. and Sandy, P.M. (1994) *Influence of Wheel Resonance on Tire Acoustic Cavity Noise*, SAE Paper 940533.

Waisanen, A.S. and Blough, R. (2009) *Road Noise TPA Simplification for Improving Vehicle Sensitivity to Tire Cavity Resonance Using Helium Gas*, SAE Paper 2009-01-2092.