A LIGHT COMMERCIAL VEHICLE WHEEL DESIGN OPTIMIZATION for WEIGHT, NVH and DURABILITY CONSIDERATIONS

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KEYWORDS –

Modal Analysis, Weight Optimization of Wheel, FRF, Durability Simulations, Vibration

ABSTRACT –

In recent years, competition in the automobile market is getting increase with restpect to fuel economy, especially for the light commercial vehicles. Moreover, there is a significant necessity about reducing fuel consumption level for automobile companies. The weight of a vehicle is one of the most important factor that affecting the fuel economy.

The weight minimization of wheel has more effective than the weight minimization of elsewhere in a vehicle due to the rotational moment of inertia effect during motion(1). Therefore, the wheel design should be optimized by considering fundamental attributes of a light commercial vehicle such as NVH, Durability and Weight.

In this study, the modal correlation between CAE simulations and tests is performed. For this purpose, mode shapes and their natural frequencies obtained from CAE simulations are compared with experimental modal analysis results. After the correlation is provided, wheel design optimization proposals are given by considering NVH and Durability criteria.

TECHNICAL PAPER -

1. STRUCTURE OF WHEEL

Wheel and tire are a coupled which determine the direction of car and they insure that cars move. Wheel is a rotating load-carrying member between the tire and the hub. It usually consist of two major parts: the rim; the wheel disc. The rim and wheel disc may be integral, permanently attached or detachable. Wheel nomenclature is stated in SAEJ1982 DEC91(2) as below.

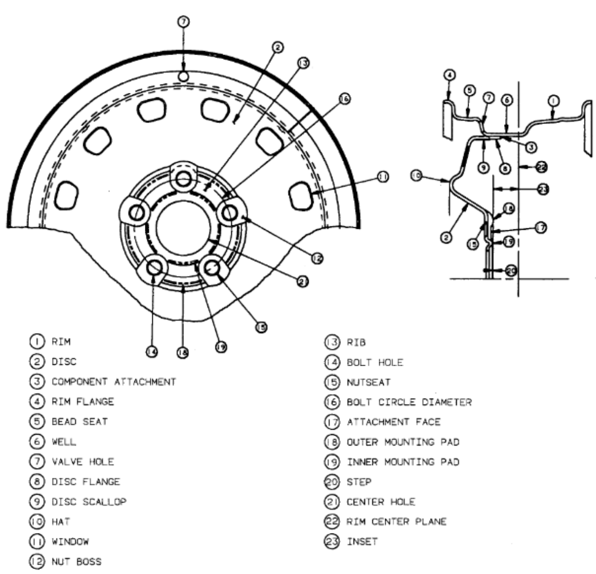


Figure 1 - Wheel nomenclature.

Material specification of the aluminium alloy wheel that will be analized in this work, are demostrated in Table 1.

Table 1 - Material specification of aluminium alloy wheel.

Modulus of Elasticity	70 GPa
Poisson Ratio	0.30
Density	2632 kg/m^3
Weight	8 kg

CAD model of the aluminium alloy wheel is demostrated in Figure 2.

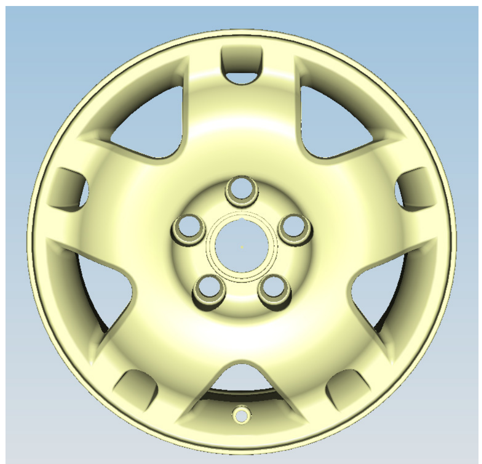


Figure 2 - CAD model of aluminium alloy wheel.

2. FINITE ELEMENT MODELLING OF WHEEL

During finite element modeling process , ANSA is used as pre-processor. Finite element model of wheel is composed using 34267 tetra finite elements. Moreover , second order tetra finite elements are used in order to represent all possible mode shapes. Modal analysis is going to be performed on free-free boundry condition. Created finite element model in ANSA can be seen in Figure 3.

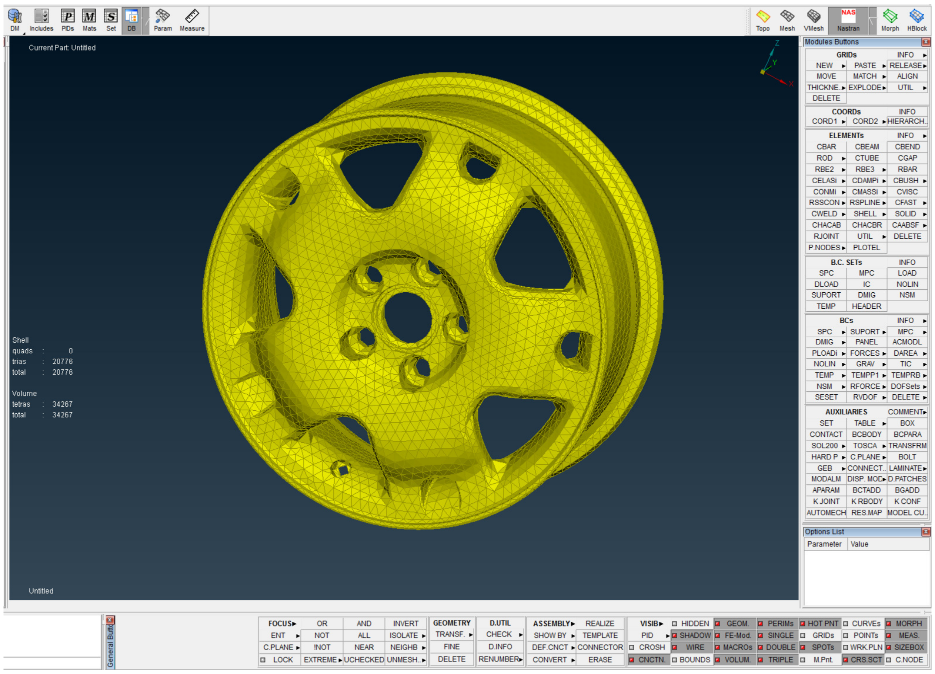


Figure 3 - Finite Element model of aluminium alloy wheel.

3. EXPERIMENTAL MODAL TEST SET-UP PREPERATION

In order to verify attained finite element model, experimental modal analysis has been performed on free-free boundry condition. Free-free boundry condition has been attained by using elastic string. Testing has been performed via using impact hammer and three axial accelerometer.

LMS software has been used during experimental modal analysis. LMS uses polymax method on determining mode shape and natural frequencies.

In order to represent all mode shapes 100 points has been determined and these points defined to LMS program. Using these points 100 FRF hase been obtained from structure. Free-free boundry condition and determined 100 points can be seen in Figure 4 and Figure 5.

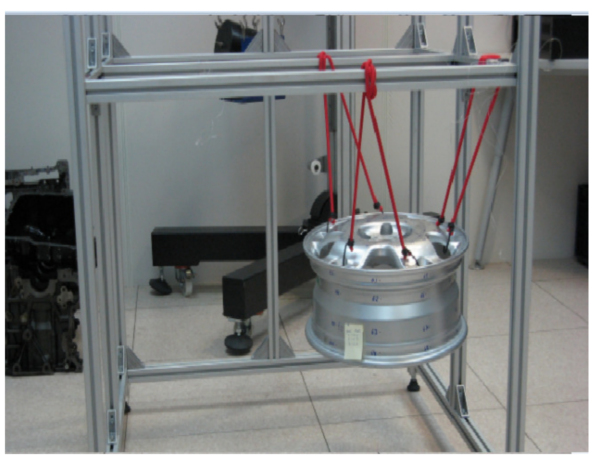


Figure 4 - Free-free boundry condition on test set-up.

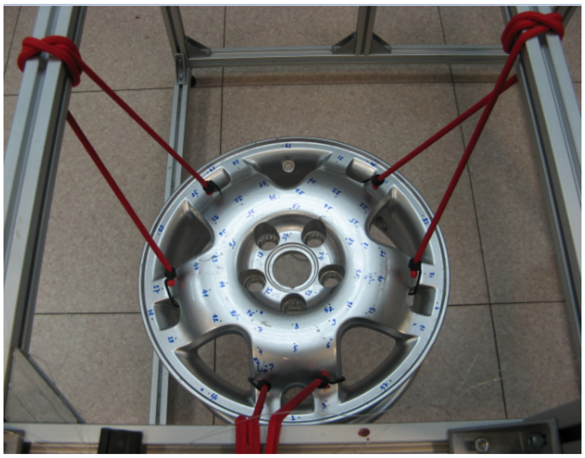


Figure 5 - Determined 100 points on free-free boundry condition simulated structure.

4. MODAL ANALYSIS OF FINITE ELEMENT MODEL ON FREE-FREE BOUNDRY CONDITION

Created finite element model using ANSA has been solved using Nastran software's Lanczos method on free-free boundry condition. Lanczos method is widely used in the structural vibration aplications for solution of eigenvalue problem. It can be determine the required number of values and corresponding eigenvectors with minimum number of iterations. This method is a type of power methods in which iteration begins with an initial vector.

Obtained four mode shapes and corresponded natural frequencies is given in Figure 6, Figure 7, Figure 8, Figure 9 below.

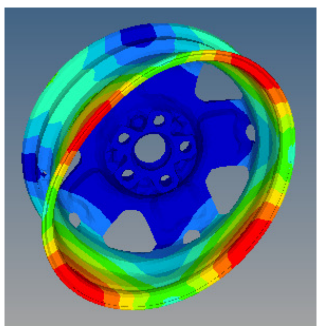


Figure 6 - 1. Mode, 425 Hz

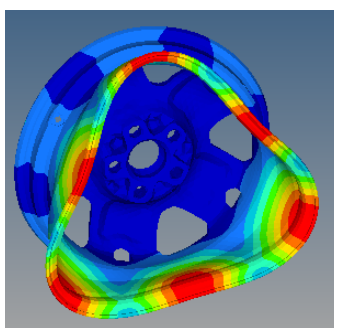


Figure 7 - 2. Mode, 908 Hz.

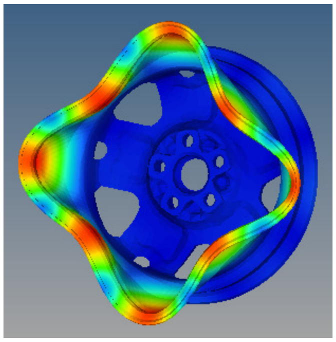


Figure 8 - 3. Mode, 1398 Hz.

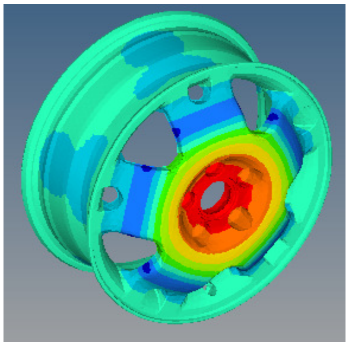


Figure 9 - 4. Mode, 1670 Hz.

5. EXPERIMENTAL MODAL ANALYSIS OF WHEEL ON FREE-FREE BOUNDRY CONDITION

Three axial accelerometer has been sticked to point 14. 100 different FRF acquired from each of these 100 points. Accelerometer location can be seen from Figure 10.

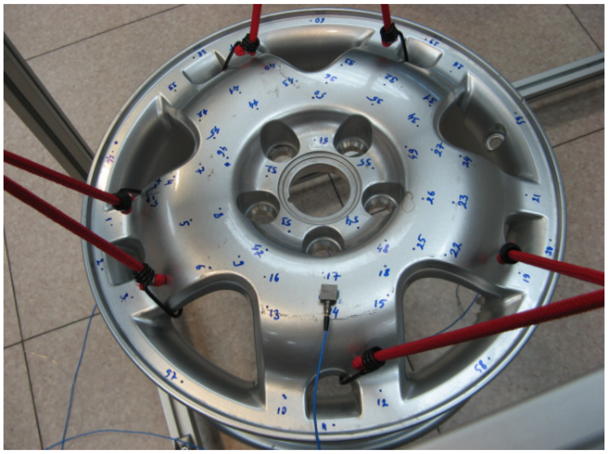


Figure 10 - Accelerometer position.

100 acquired FRF plot and average of these 100 FRF plot demonstrated in Figure 11 and Figure 12.

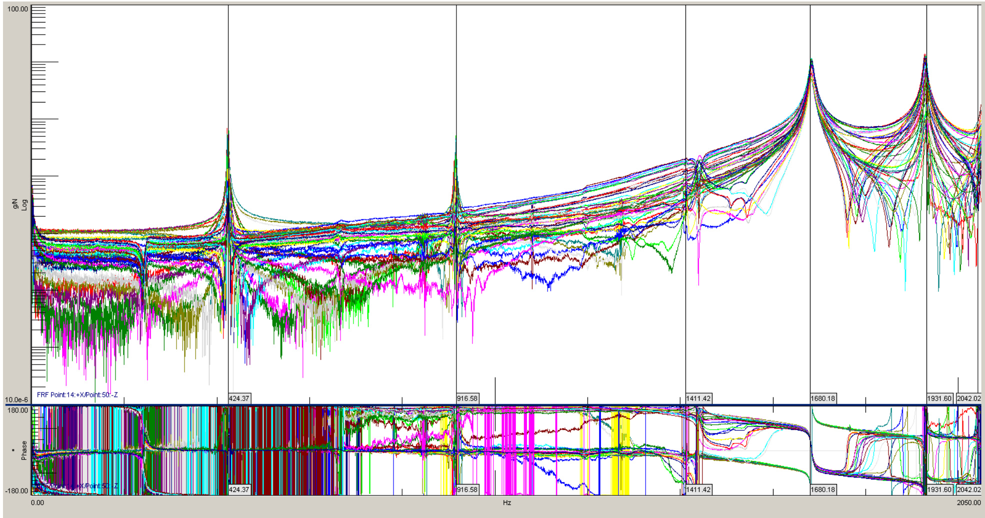


Figure 11 - 100 FRFs’ frequency-amplitude plot.

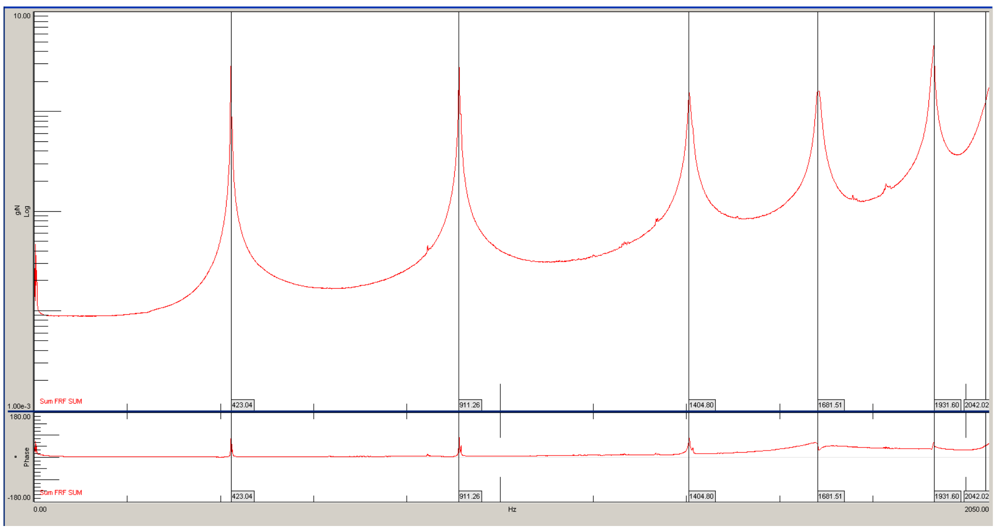


Figure 12 - Average of 100 FRFs’ frequency-amplitude plot.

As shown above, four natural frequency obtained at 425 Hz, 916Hz, 1413Hz and 1683Hz. Corresponded mode shapes are listed Figure 13, Figure 14, Figure 15 and Figure 16.

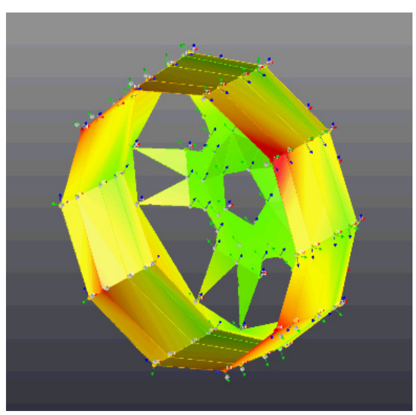


Figure 13 - 1. Mode, 425 Hz.

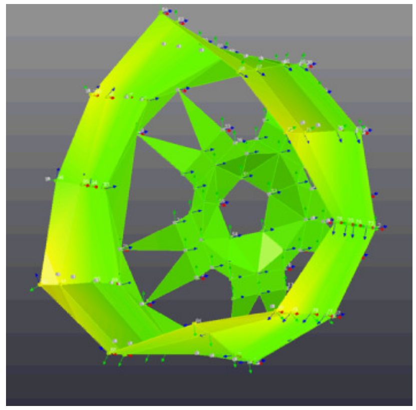


Figure 14 - 2. Mode, 916 Hz.

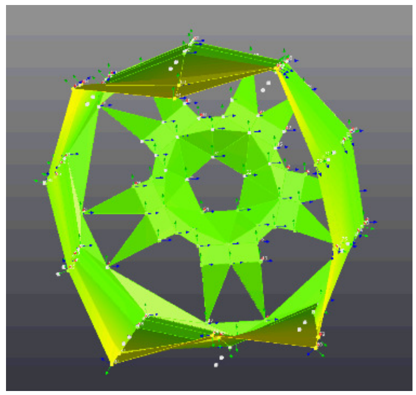


Figure 15 - 3. Mode, 1413 Hz.

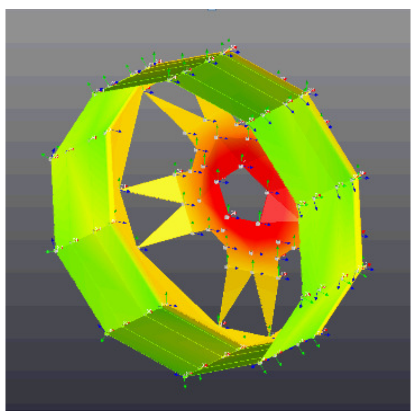


Figure 16 - 4. Mode, 1683 Hz.

6. COMPARISON OF FINITE ELEMENT MODEL AND EXPERIMENTAL MODEL OF WHEEL

In this section the first four mode shapes and their natural frequencies will be compared. Comparison of first four mode shapes and their natural frequencies can be seen in Figure 17, Figure 18, Figure 19 and Figure 20 and overall comparison can be seen in Table 2.

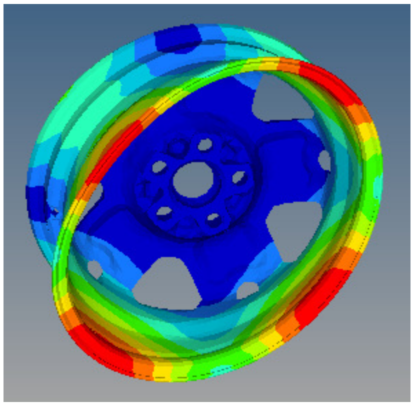
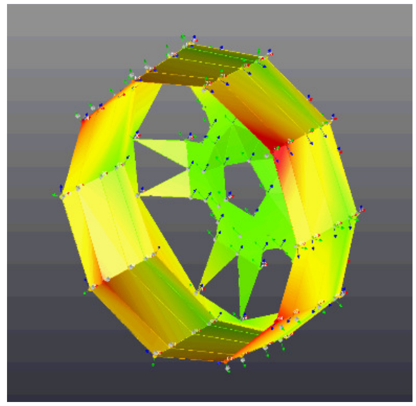
	
425 Hz	**425 Hz**

Figure 17 - 1. Mode comparison.

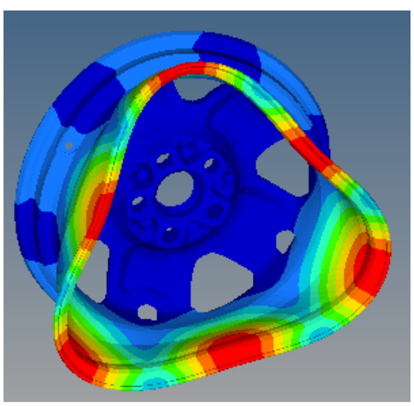
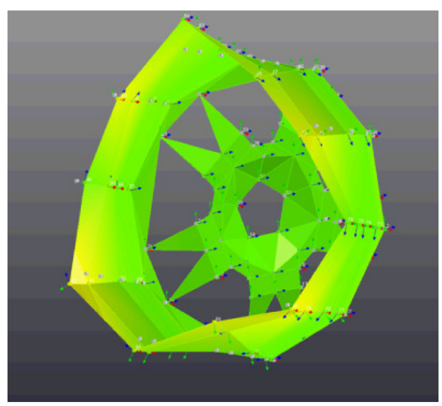
	
908 Hz	**916 Hz**

Figure 18 - 2. Mode comparison.

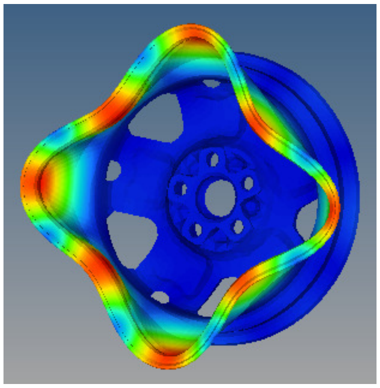
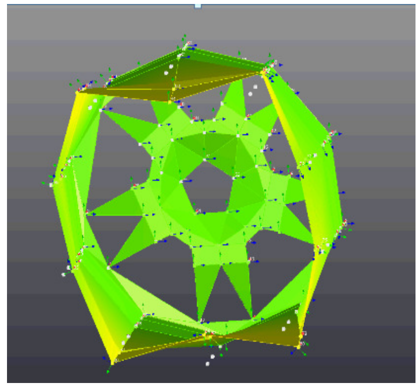
	
1398 Hz	**1413 Hz**

Figure 19 - 3. Mode comparison.

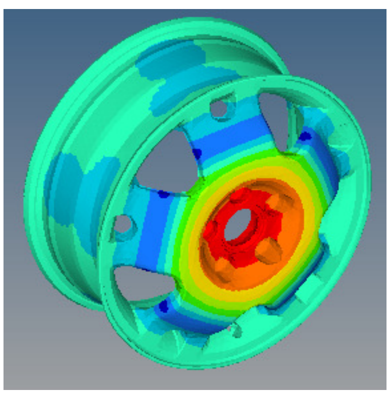
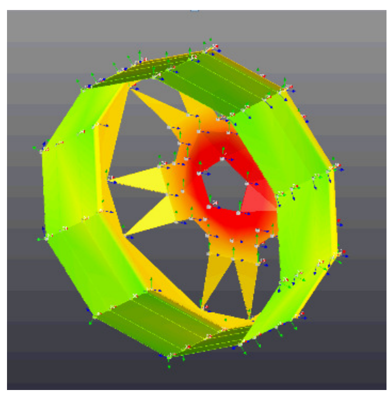
	
1670 Hz	**1683 Hz**

Figure 20 - 4. Mode comparison.

Table 2 : Natural frequency comparison on free-free boundry condition.

	Finite Element Analysis	Experimental Modal Analysis
1. Mode	425 Hz	425 Hz
2. Mode	908 Hz	916 Hz
3. Mode	1398 Hz	1413 Hz
4. Mode	1670 Hz	1683 Hz

The results obtained from finite element and experimental modal analysis have been compared. The natural frequencies and mode shapes have properly been correlated. Therefore, generated finite element model of a light commercial vehicle wheel is validated by experimental modal analysis on free-free boundry condition . As a result of this, a finite element model which is representing the dynamic characteristic of the real system is generated. Thanks to this model the effect of any design modification can be easily observed.

7. DESIGN CHANGE PROPOSAL IN ORDER TO REDUCE WEIGHT OF WHEEL

In this section, design change proposals will be given in order to reduce wheel weight and these proposals will be investigated for durability and NVH considerations.

In order to avoid undesired interior noise the natural frequency of the wheel should be higher than 350 Hz since this value is considered as critical frequency. Moreover, durability availability will be checked on defined condition in SAE J328.

Weight Reduction Proposal

2 mm metal removing has been applied to the areas that white coloured and marked with 1 in the Figure 21.

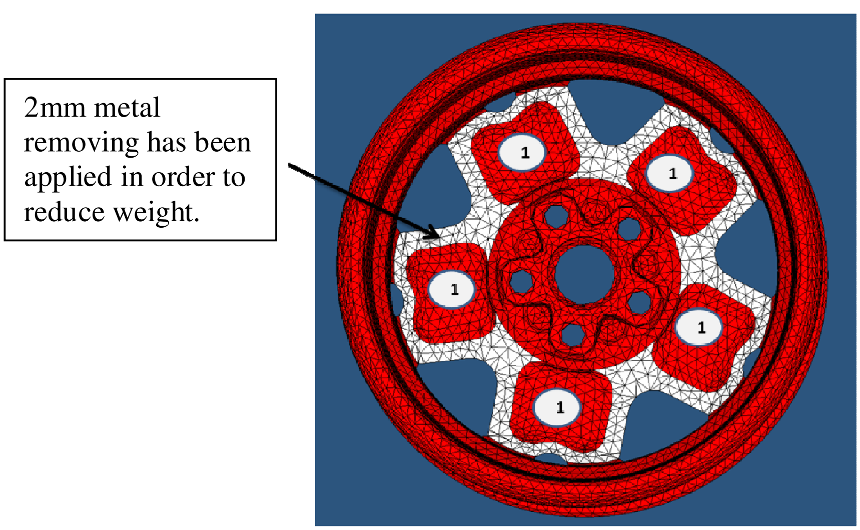


Figure 21 - 4. Weight reduction proposal.

NVH Evaluation for Weight Reduction Proposal

Modal analysis has been performed in this section again for modified model.

Natural frequency comparison between modified and base model can be seen in Table 3 below.

Table 3 : Natural frequency comparison between base and modified model.

	Base Model	Modified Model
1. Mode	425 Hz	410 Hz
2. Mode	908 Hz	905 Hz
3. Mode	1398 Hz	1388 Hz
4. Mode	1670 Hz	1563 Hz

As shown above, 15 Hz frequency has been sacrificed from first natural frequency. Since first natural frequency decreased 410 Hz, still greater than 350 Hz which is interior noise limit, this proposal is acceptable from NVH point of view.

Durability Evaluation for Weight Reduction Proposal

Second of all, the effect of weight reduction should be investigated in terms of durability criterias. To evaluate fatique performance wheels when subjected to bending (cornering) loads, a long cylinder is mounted to the center of wheel and a force is applied to the normal to the wheel axis. This force is calculated according to SAE J328 test.

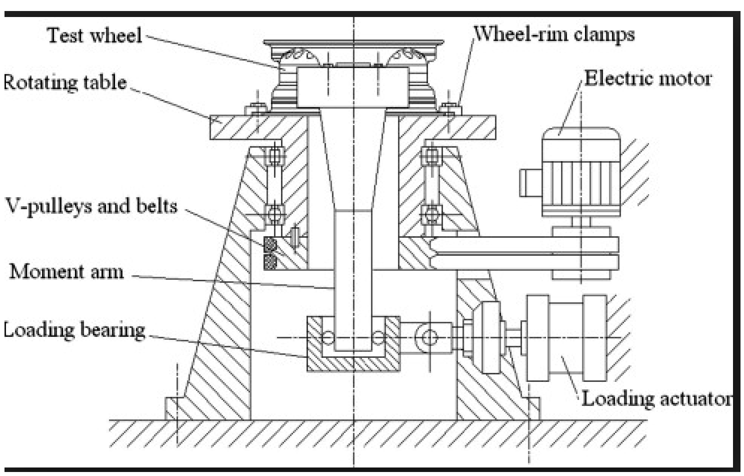


Figure 22 - Wheel cornering fatique test

After the applied force is calculated and boundary condition is given via ANSA 13.1.5, the static analysis is performed. Maximum Von Mises Stress for wheel is presented with Metapost 6.7 shown in Figure 23.

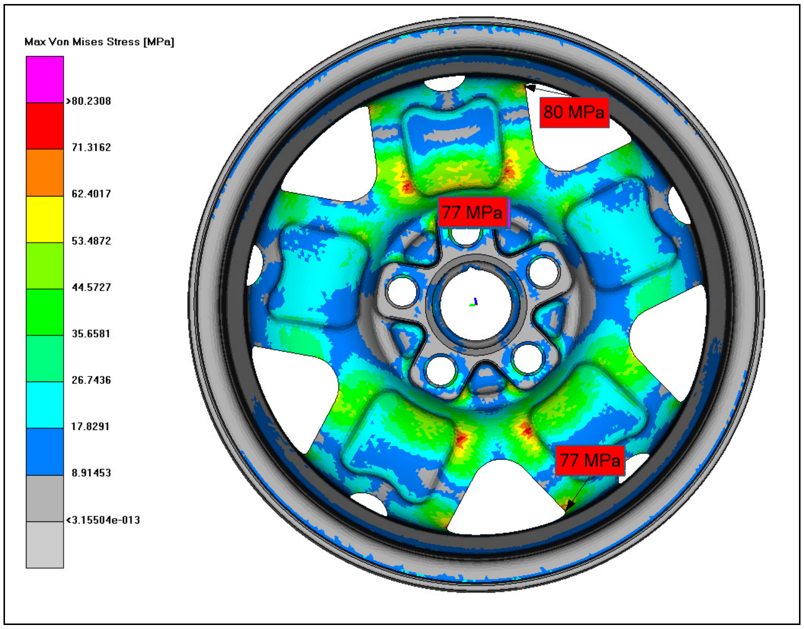


Figure 23 - Maximum von mises stress

According to Figure 23, maximum Von Mises Stress value is 80 MPa. This value is lower than the endurance limit of aluminum alloy (95 MPa). Furthermore, the loading condition is repeated 100000 cycle and this test is simulated via nCode. According to results, no damage occurs after 100000 cycle as given Figure 24 and Figure 25.

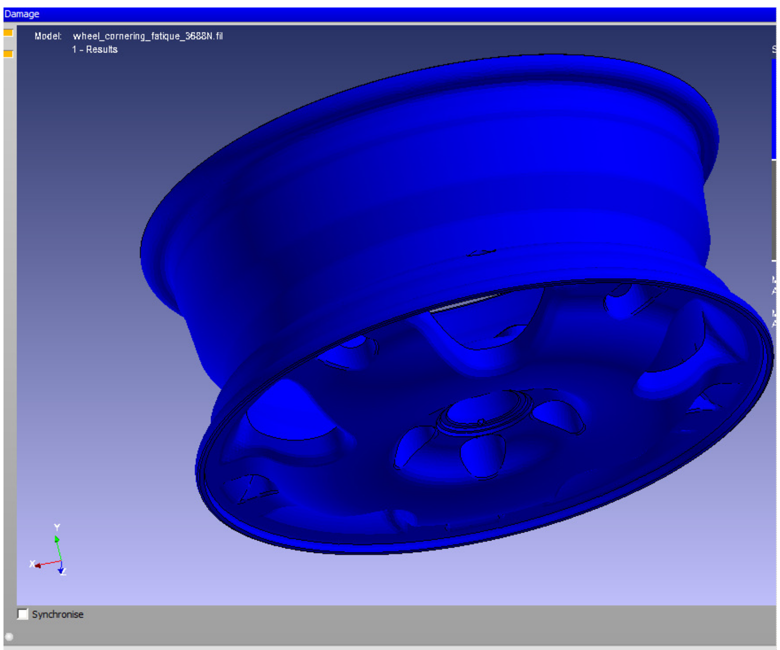
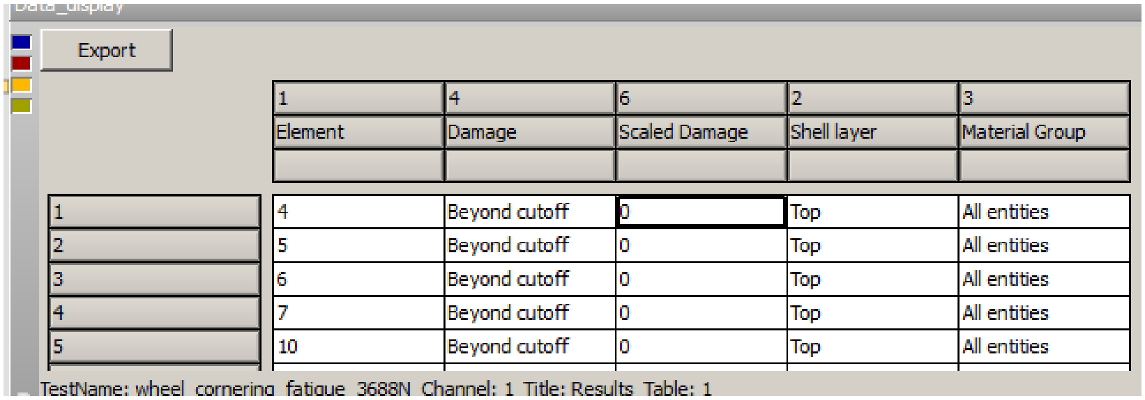


Figure 24 - Damage contours after 100000 cycles



	1	4	6	2	3
	Element	Damage	Scaled Damage	Shell layer	Material Group
1	4	Beyond cutoff	0	Top	All entities
2	5	Beyond cutoff	0	Top	All entities
3	6	Beyond cutoff	0	Top	All entities
4	7	Beyond cutoff	0	Top	All entities
5	10	Beyond cutoff	0	Top	All entities

Figure 25 - Damage values (no damage)

It can be seen that there is no risk in terms of durability due to weight reduction.

8. CONCLUSIONS

The fundamental natural frequency of light commercial vehicle wheel representing the dynamic behaviour of real structure is 425 Hz. 15 Hz frequency has been sacrificed from fundemental natural frequency for new design modification suggestion but new fundamental frequency is still greater than 350 Hz which is interior noise limit, so this proposal is acceptable from NVH point of view. Also, the proposal is acceptable durabilty point of view.

By using this opportunitiy 200 gr per wheel and 1 kg per vehicle weight reduction have been gained in this study.

REFERENCES

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(2) SAEJ1982 DEC91, Wheels for passenger cars, light trucks and multipurpose vehicles, 31, 2-6, 1991