REDUCTION OF SEAT VIBRATION IN AN ATV THROUGH DESIGN MODIFICATION

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Abstract- In this paper, Transfer Path Analysis (TPA) method has been used to reduce the vibration at the seat of an All Terrain Vehicle (ATV). The vibration source considered is the engine. Modifications are made on an existing Baja roll cage and comparisons are made to the existing design to arrive at the optimized design.FE model of the roll cage is analyzed and experimental validation is done.Modifications in the Transfer Path change the vibration pattern at the receiver. The modifications include addition of new roll cage members and use of different cross sectional beams for the engine mount. These changes result in reduced transmission of vibration from the engine to the roll cage. The analysis has been done in three different RPM ranges of the engine in order to ensure that a particular modification gives good performance over the entire operating range of the engine.

Index Terms- TPA, ATV, Vibration Reduction, Roll cage.

I. INTRODUCTION

Vibration in an ATV is a persistent problem and exposure to such vibrations over long durations makes the driver susceptible to joint diseases. For rough roads, if the driver is exposed to more than 30 to 35 minutes, it would make the driver uncomfortable. [1].Driving comfort is essential for safety as well as better handling of the vehicle [2]. The sources of these vibrations are both external and internal. The contribution of the external sources is less at low speed whereas the internal sources contribute at both high and low speed conditions of the vehicle.Majority of these vibrationsaffects the driver through the seat. In this paper we have reduced the vibration level at the seat of an All Terrain Vehicle caused due to engine, an internal source

A. Transfer Path Analysis

Transfer path analysis (TPA) is an effective method for determining vibration contributions through different structural transmission paths. TPA and modal analysis can be used to determine how to reduce the seat vibrations at low frequencies. [2]Experimental transfer path analysis is generally a powerful tool for the diagnosis of complex transmission of vibration and air-borne sound via multiple paths [3]. It represents the structure as a source-path-receiver model. Source is the engine, seat mount is receiver and roll cage is the transfer path. In this paper, for a single source we have studied the vibration pattern at the receiver for various transfer paths. So by modifying the transfer path, i.e. modifying the roll cage, the vibration level at the receiver has been reduced.

B. Roll Cage

A roll cage is a space-frame type of chassis, generally used in sports vehicle for its low weight and high stiffness. The roll cage we used complies with the SAE Baja 2014 event's rules and regulations.

i. Chassis Specifications
Material: SAE 1018, 31.75mm OD, 1.65mm thickness
Weight:38 kg
Type of weld: SMAW

ii. Engine specifications
Briggs & Stratton 10HP OHV Engine
Idle RPM – 1750 Max RPM – 3020

II. FEA OF EXISTING DESIGN

Finite Element Analysis (FEA) was used to see the vibrational response of the chassis. The natural frequencieswerecalculated and harmonic analysis was done by giving a sinusoidal time varying function force at the engine mount to obtain the Displacement vs. Frequency graph. FEA was done using Ansys 14.5

The natural frequencies were calculated using the modal analysis, which gives all the natural frequencies of the model in a particular range of frequency.

The next step is to calculate the FRF at the seat mount for a harmonic load at the engine mount. The displacement across various frequency range was plotted. The displacement measured, is along the vertical axis, on the seat mount.

The frequency range for the FRF was 0-25 Hz, in order to accommodate more than first 100 natural frequencies.
This calculation of FRF was done for three different RPM of the engine for Idle RPM – 1750.

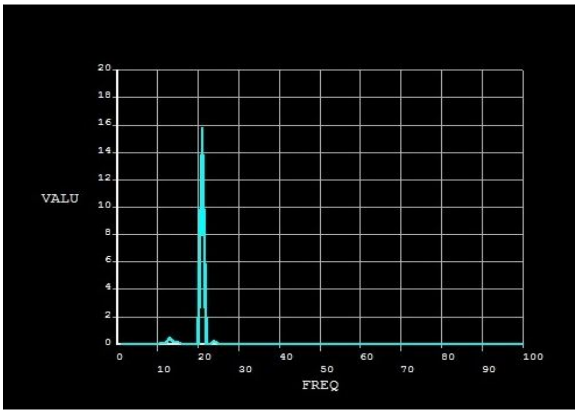

Figure 7: FRF of the roll cage without any change for 0-100 Hz

Modal Analysis Boundary Conditions: Free Free Run
Harmonic Analysis Boundary conditions:
Constraining points at the top of the roll cage, to simulate the hanging of the roll cage.
Force of 300 sin (29.167*2π*(time))
The amplitude of the force was determined by the weight of the engine which is about 30kg.
The idle RPM is 1750. That is, $\varpi = \frac{1750}{60} \times 2\pi$
The extracted frequencies were from 0 Hz – 25 Hz. The number of sub steps are 100.
Results of Analysis at Idle RPM for the Existing Roll Cage Design
Modal Analysis
First 5 natural frequencies are - 1.763 Hz, 2.002 Hz, 2.238Hz, 2.877 Hz, and 2.979 Hz
From 0-50 Hz, we get 215 Natural Frequencies.
Harmonic Analysis

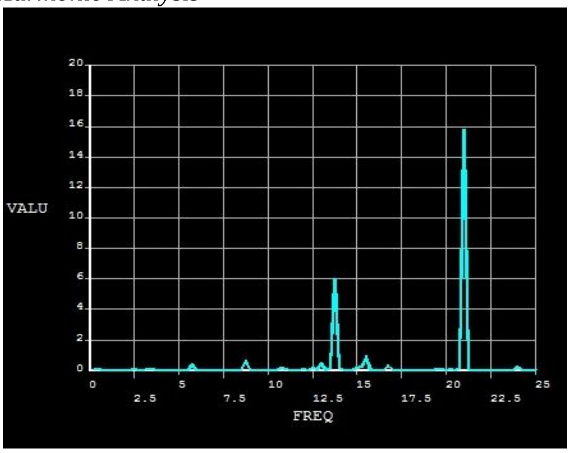

Figure 8: FRF of the roll cage without any change for 0-25 Hz at idle RPM

The peaks are Min: -6.044 mm and Max: 15.8126 mm.
The peaks occur at: 13.750 Hz and 21 Hz.
This is the vibration of the roll cage in the initial stage. The vibration has to be reduced by reducing the peaks, i.e. amplitude of the vibration. The peaks also have to be moved from the higher frequency to lower frequency range. This has to be done by optimizing the design. The earlier design was not made keeping vibration reduction as top priority, hence optimization is required.

III. DESIGN OPTIMIZATION

The design of the roll cage was modified in order to reduce the vibration transfer from the engine to the seat. This was achieved by changing the stiffness of the roll cage which was achieved by adding or removing members from the chassis. The main consideration was distributing the vibration from the source away from the seat.
Three changes were made in the existing roll cage.

A. Change 1

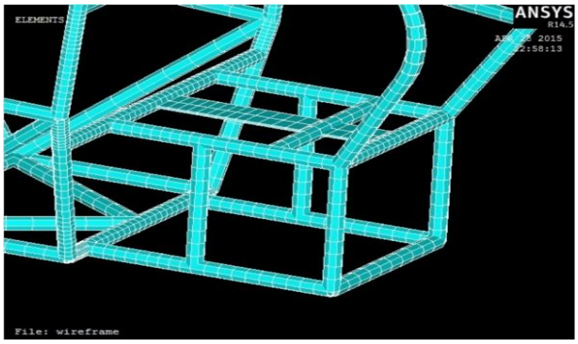

Figure 1: the member below the engine before the change

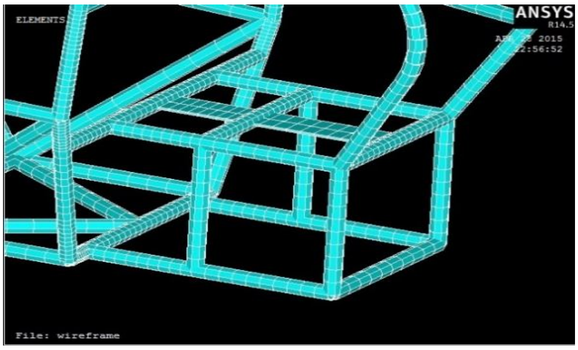

Figure 2: the member below the engine after the change

A member under the beam which supports the engine was removed, and another member of the same cross section was added between 2 parallel members. This reinforces the structure and hence the vibration transfer reduces.

B. Change 2

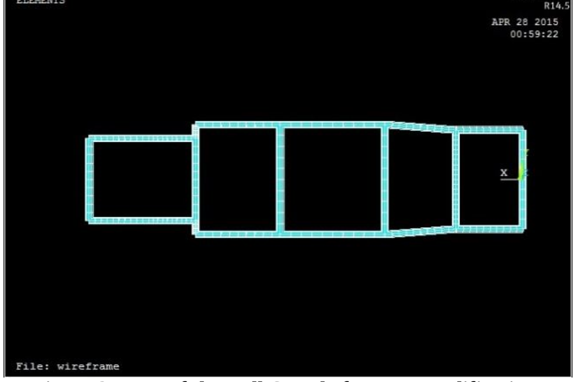

Figure 3: Base of the Roll Cage before any modification

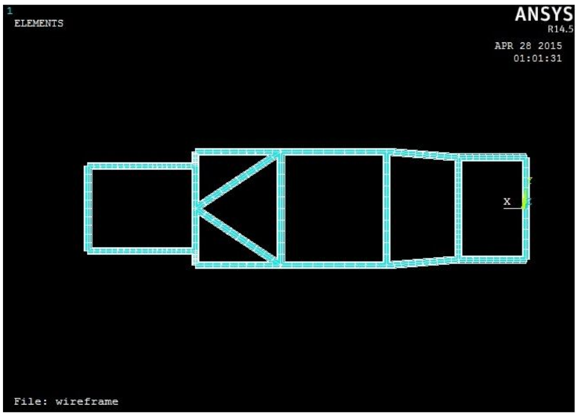

Figure 4: Base of the Roll Cage after the modified design

The design of the base was changed, where 2 members were added to distribute the vibration which would be going to the seat. A ‘V’ type of design is made in the base.

C. Change 3

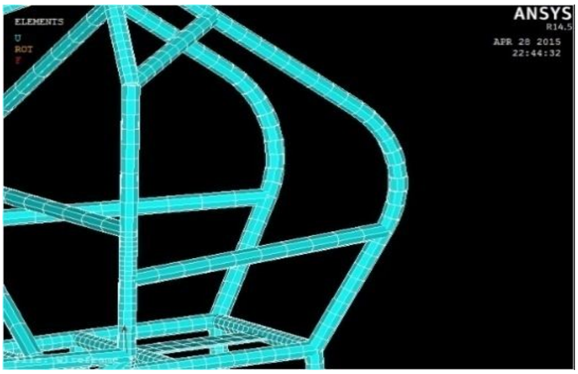

Figure: 5 Rear of the Roll Cage before modification

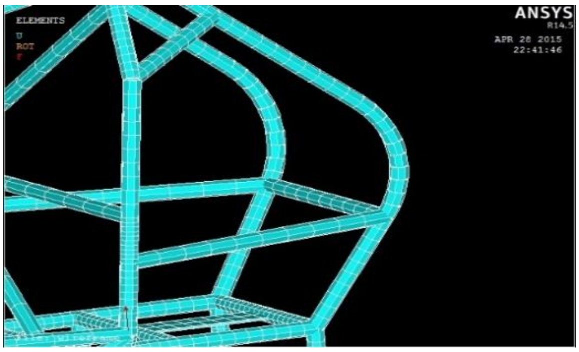

Figure 6: Rear of the Roll Cage after the modification

The rear of the roll cage was joined together so as to again change the stiffness to lower the transferring vibration.

IV. FEA OF CHANGED DESIGN

A. Roll Cage Design Change 1

Modal Analysis

First 5 natural frequencies are - 1.7632 Hz, 2.0042 Hz, 2.2397 Hz, 2.8792 Hz, and 2.9841 Hz.

From 0-50 Hz, we get 213 Natural Frequencies.

Harmonic Analysis

The peaks are Max: 2.0693 mm and Min: -0.3784 mm.

The peaks occur at: 3.5 Hz and 17.750 Hz.

The amplitude has decreased after this design change 1. The peaks in the FRF also moved from 13 Hz to 3.5 Hz.

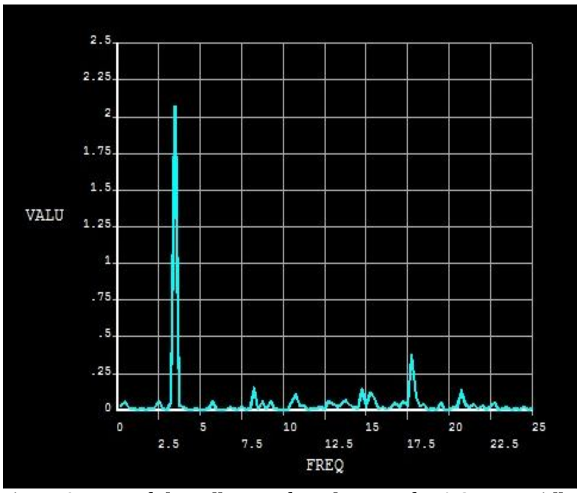

Figure 9: FRF of the roll cage after change 1 for 0-25 Hz at idle RPM

B. Roll Cage Design Change 2

Modal Analysis

First 5 natural frequencies are 1.7826 Hz, 2.0024 Hz, 2.2486 Hz, 3.0146 Hz and 3.1882 Hz

From 0-50 Hz, we get 215 Natural Frequencies.

Harmonic Analysis

The peaks are Max: 0.6365mm and Min: -0.7 mm.

The peaks occur at: 3.5 Hz and 15.5 Hz.

The amplitude has decreased after the design change 2. The peak at 17.75 Hz has reduced to 15.5 Hz

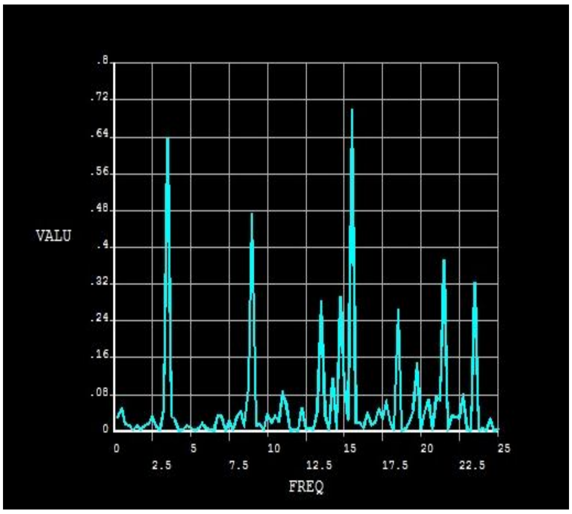

Figure 10: FRF of the roll cage after change 2 for 0-25 Hz at idle RPM

C. Roll Cage Design Change 3

Modal Analysis

First 5 natural frequencies are -1.8024 Hz, 2.0785 Hz, 2.3091 Hz, 3.0709 Hz, and 3.1851Hz

From 0-50 Hz, we get 215 Natural Frequencies.

Harmonic Analysis

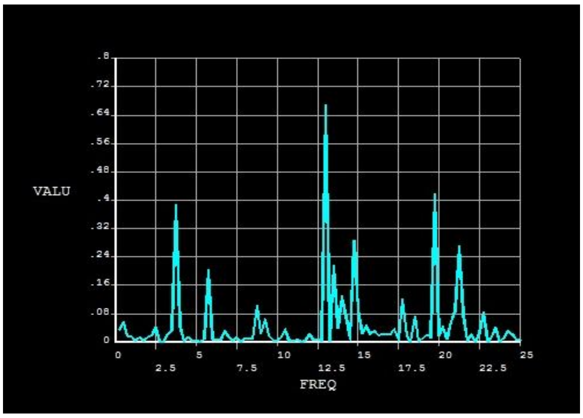

Figure 11: FRF of the roll cage after change 3 for 0-25 Hz at idle RPM

The peaks are Max: 0.668mm and Min: -0.417 mm.
The peaks occur at: 13 Hz and 19.750 Hz.
The amplitude has reduced from 15.816 mm to 0.668 mm for the idle RPM.
The Finite Element Analysis has shown that the three design changes have reduced the vibration transfer from the engine to the seat. The concept of the changes was to distribute the vibration from the engine to the whole chassis avoiding the concentration at the seat mount.

V. EXPERIMENTAL VERIFICATION

The experimental verification of the FRF was performed using RT Pro Software Version.

The experimental setup is shown below

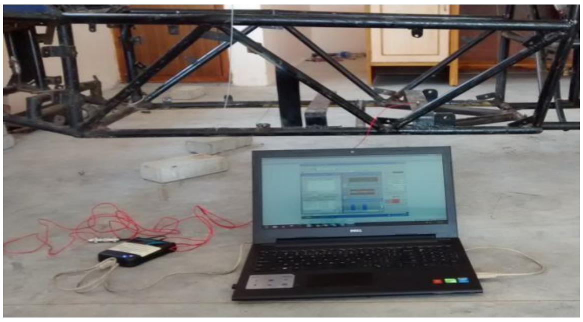

Figure 12: Experimental Setup

Figure 13: Experimental Setup

In order to achieve the correct FRF, the accelerometers were fixed at equally spaced 5 locations on the seat mount.

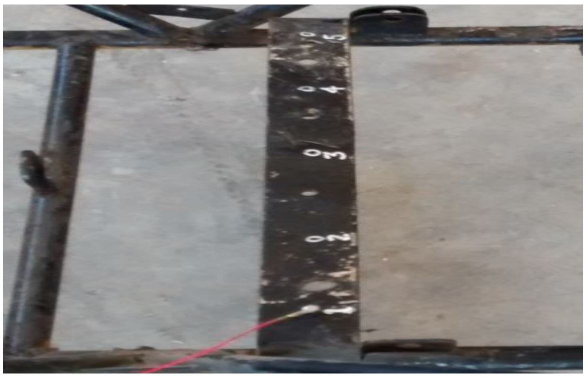

Figure 14: Different locations on seat mount for taking reading

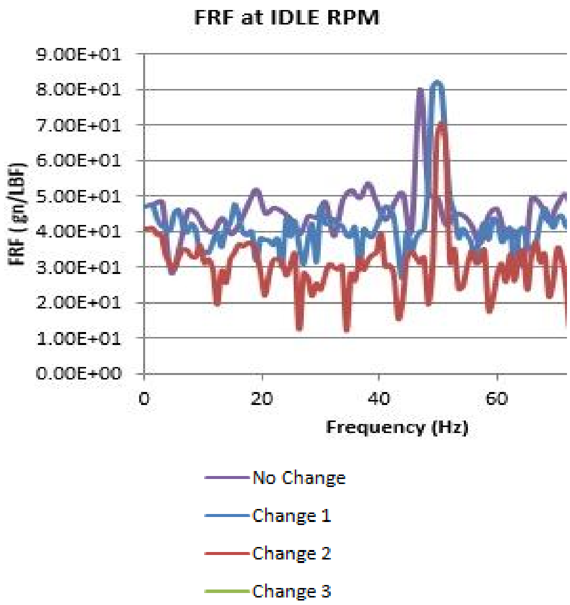

Figure 15: FRF obtained in the experimental verification

The experimental results confirm the results from the FE Analysis. The vibration amplitude has reduced by 18.75% at resonance. And has a more stable response in the frequency range 0-25 Hz.

CONCLUSION

We have used Transfer Path Analysis (TPA) to reduce the vibration at the seat. This was done by using Passive vibration reduction techniques to reduce the vibration caused by the internal source, i.e., engine. Design of the roll cage was modified to disperse the vibrational load going to the seat. Three changes were done and the vibration at the seat was experimentally found out to reduce.
The first change was to have the member's joints at the node rather than at the middle of another member. This would improve the stiffness of the member. The second change was done to disperse the vibration away from the seat mounts. The third change was to

remove the presence of long unsupported members, which would increase vibration. After these changes were done, the vibration reduced 18.75% compared to the initial design. Thereby, the driver comfort and the vehicle handling are improved.

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