## ARTICLE

# Development of Shape Memory Alloy Based Quarter Car Suspension System

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| ARTICLE INFO | ABSTRACT |
|---|---|
| | It is well-known that suspension systems plays a major role in automotive technology. Most of the today's vehicle applies a passive suspension systems consisting of a spring and damper. The design of automotive suspension have been a compromise between passenger comfort, suspension travel and road holding ability. This work aims in reducing the suspension travel alone by developing a quarter car model suspension for a passenger car to improve its performance by introducing shape memory alloy spring (Nitinol) instead of traditional spring. A two way shape memory alloy spring possesses two different stiffness in its two different phases (martensite and austenite). In this study, road profile is considered as a simple harmonic profile and vibration analysis of a miniature quarter car model suspension system has been carried out experimentally. Using theoretical method, the displacement of the sprung mass is also studied and discussed. The vibration analysis have been carried out for the suspension system at both phases of the spring and the results gives a significant improvement in reducing the displacement of sprung mass for various excitation frequencies. |

## 1. Introduction

Suspension comprises the system of springs, shock absorbers and their linkages that connects vehicle to the wheels. Suspension system serves for the following purposes: contributing to the vehicle on-road holding/handling, braking in order to provide good active safety, driving pleasure, keeping passengers comfortable and well isolated from road noise, bumps, vibrations, etc. These above mentioned goals needs to be balanced, hence the design of suspension system involves adequate compromise [1-2]. A two-degree-of-freedom "quarter-car" automotive suspension system is shown in Figure 1. The suspension itself is shown to consist of a spring ($K_s$) and a damper ($D_s$). The sprung mass ($M_s$) represents the quarter car equivalent of the vehicle body mass. The unsprung mass ($M_u$) represents the equivalent mass due to the axle and tire. The vertical stiffness of the tire is represented by the spring ($K_t$). The variables $Z_s$, $Z_u$ and $Z_r$ represent the vertical displacements from static equilibrium of the sprung mass, unsprung mass and the road respectively. Since it is difficult to perform analysis of a full car model, a single segment (quarter car model) has been studied [7].

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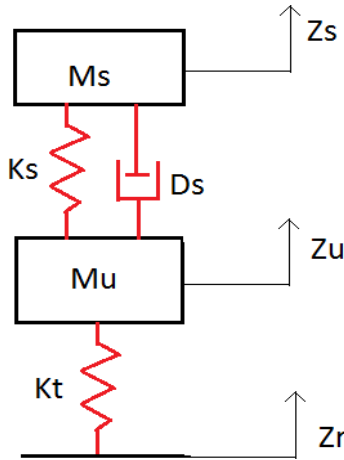*Email: sathishiitkgp@gmail.com*

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**Figure 1.** Quarter car suspension system

A shape memory alloy (SMA) is a material which re-members its original shape and when it is deformed within a limit, will return to its original shape when heated. This solid-to-solid phase transformation occurs when the material passes through a transformation temperature. If the transformation temperature is below 55oC, the material is in martensite phase and if it above 55oC, the material is in austenite phase [3-6]. Since heating an SMA spring causes change in elastic modulus of that material, a shape memory alloy spring possess two different stiffness at this two phases. Nitinol, nickel-titanium (Ni-Ti) alloys are the most important among all SMA's. These alloys typically are made of 55%-56% nickel and 44%-45% titanium [8-12].

The paper reports the vibrational effect of a miniature quarter car vehicle model when it is subjected to harmonic excitation by road profile. SMA spring is introduced into the suspension system and the amplitude of vibration is studied when the vehicle is moving with varying speeds on the harmonic profile.

## 2. Methods and Materials

### Theoretical Analysis

The profile of the road is approximated to a line curve of amplitude 1.0 cm and a wavelength of 4.0 m. The sine wave is represented by $q = A\sin\omega t$.
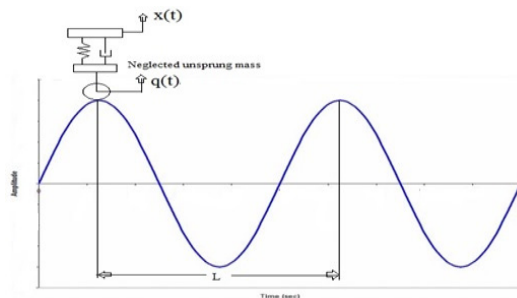


**Figure 2.** Harmonic road profile

q = road surface excitation in m.

A = amplitude             = 0.01 m.
L= wavelength of road surface  = 4 m.

**Dimension and properties of Nitinol:**
Spring outer diameter       = 40 mm
Wire diameter             = 6.5 mm
Number of coils            = 12
Young's modulus of SMA spring at martensite ($E_m$)
      = 38 GPa
Young's modulus of SMA spring at austenite ($E_A$)
= 80.95 GPa

Transformation temperature   = 55$^o$C
Stiffness at martensite ($K_m$)  = 7230 N/m
Stiffness at austenite ($K_A$)   = 15400 N/m
The natural frequency of the vehicle is given by
$\omega_n = \sqrt{(K/m)}$
m = sprung mass = 4.5 Kg,
Therefore,   $\omega_{n1}$ = 40 rad/sec, $\omega_{n2}$ = 58.50 rad/sec
Damping constant of damper (C) = 300 Ns/m
Damping constant, $C = \xi 2m\omega_n$
Where, $\xi$ = damping ratio. $\xi_1 = 0.83$ and $\xi_2 = 0.57$ are the damping ratio for the two different natural frequencies.

For the road profile mentioned and the velocity of the vehicle in Km/h, the excitation frequency is given by

$$\omega = V*(1/0.004)*(1/3600)*(2\pi) = 0.4363V \text{ rad/sec}$$

Where, V = velocity of vehicle in Km/h

The excitation frequencies ($\omega$) were calculated for different car velocities and corresponding to these frequencies, amplitude of the response (X) is calculated by the equation

$$X = A\sqrt{\frac{1+(2\xi r)^2}{(1-r^2)^2+(2\xi r)^2}}$$

Where, $r = (\omega/\omega n)$ = frequency ratio

The experimental set up consists of a test rig where an SMA spring and a damper is placed between an upper plate (sprung mass) and lower plate (unsprung mass). Both upper plate and lower plate are movable one which moves in a guide way. Figure 3 depicts the experimental set up. The excitation to the lower plate is given by cam rotation which is driven by a variable speed DC electric motor. The displacement of the upper plate due to cam rotation is measured by fixing an accelerometer at the upper plate, and measuring the signals with the help of DAQ card and DEWEsoft software.
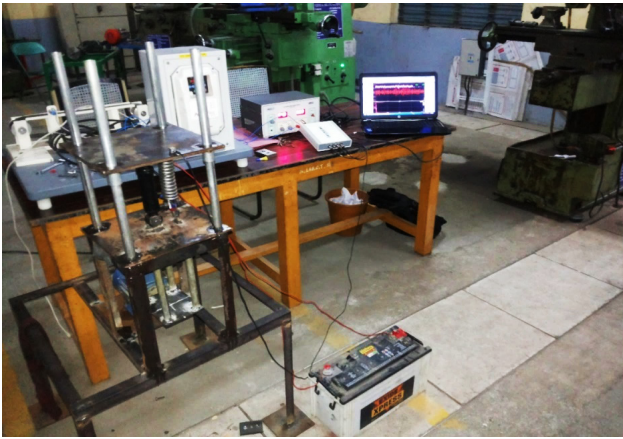
**Figure 3.** Experimental setup

## 3. Experimental Results and Discussion

Usually in a passenger car, the suspension system has the following parameters as unsprung mass = 45 kg, sprung mass = 320 kg, stiffness of spring = 45,000 N/m and damping constant of passive shock absorber = 3000 Ns/m. In the experimental setup, it is difficult to analyse the suspension system keeping the same sprung mass and unsprung mass. Also, the stiffness of the nitinol spring is not available upto 45,000 N/m. So, unsprung mass is taken as 4.5 kg and the commercially available SMA spring having stiffness in the range of 7230 N/m to 15400 N/m is taken for the experimental study.

The quarter car suspension system is operated with two natural frequencies because of the two spring stiffness of the SMA spring. When the spring is in its martensite phase (cold), (i.e., the system at its first natural frequency) the excitation is given to the lower plate through the cam and the response of the upper plate is measured using accelerometer in the quarter car setup. Further, when the spring transforms to austenite phase (hot), (i.e., the system at its second natural frequency) the same excitation frequency is given and the response of the upper plate is measured. The phase transformation of the spring from its cold phase to hot can be done by applying direct current to the spring from the battery.

The road profile taken in the theoretical analysis is reproduced by designing an eccentric cam and excitation to the system is applied through this latter.

Excitation frequency through cam ($\omega$) = $2\pi N/60$

The different excitation frequencies taken in the theoretical method can be achieved in the experimental method by varying the speed of the motor which is connected to the cam. The amplitude responses for different excitation frequencies which are calculated theoretically and experimentally are tabulated as shown in table 1.

**Table 1.** Amplitude response values for different excitation frequencies

| Vehicle speed (Km/h) | Speed of the motor (rpm) | Excitation frequency (Hz) | Amplitude X (m) at $\omega n1$ | | Amplitude X (m) at $\omega n2$ | |
|---|---|---|---|---|---|---|
| | | | Theoretical | Experimental | Theoretical | Experimental |
| 10 | 42 | 0.6944 | 0.010117 | 0.011121 | 0.010056 | 0.011034 |
| 20 | 83 | 1.3890 | 0.010438 | 0.011343 | 0.010221 | 0.011123 |
| 30 | 125 | 2.0832 | 0.010888 | 0.011811 | 0.010492 | 0.011523 |
| 40 | 167 | 2.7776 | 0.011366 | 0.012621 | 0.010862 | 0.011956 |
| 50 | 208 | 3.472 | 0.011771 | 0.013012 | 0.011317 | 0.012672 |
| 60 | 250 | 4.1664 | 0.012028 | 0.013632 | 0.011834 | 0.013452 |
| 70 | 292 | 4.8608 | 0.012100 | 0.013741 | 0.012381 | 0.014023 |
| 80 | 333 | 5.5551 | 0.011990 | 0.013573 | 0.012911 | 0.014256 |
| 90 | 375 | 6.2495 | 0.011730 | 0.013543 | 0.013367 | 0.014712 |
| 100 | 417 | 6.9439 | 0.011362 | 0.013122 | 0.013686 | 0.014802 |
| 110 | 458 | 7.6383 | 0.010927 | 0.012891 | 0.013820 | 0.015043 |
| 120 | 500 | 8.3327 | 0.010459 | 0.012543 | 0.013743 | 0.014846 |
| 130 | 542 | 9.0271 | 0.009984 | 0.011523 | 0.013466 | 0.014243 |
| 140 | 583 | 9.7215 | 0.009516 | 0.011231 | 0.013023 | 0.014067 |
| 150 | 625 | 10.4159 | 0.009067 | 0.010976 | 0.012467 | 0.013425 |

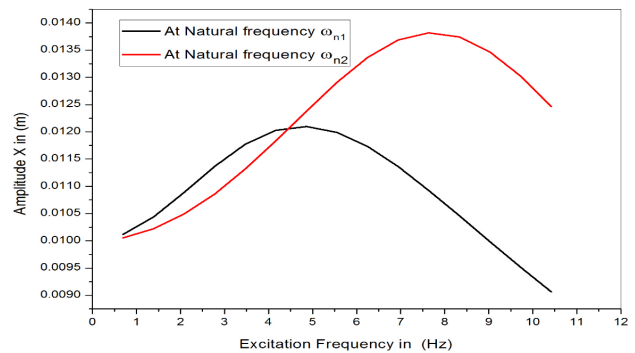The graph for theoretical method is shown in figure 4.



**Figure 4.** Theoretical sprung mass displacement for different excitation frequencies

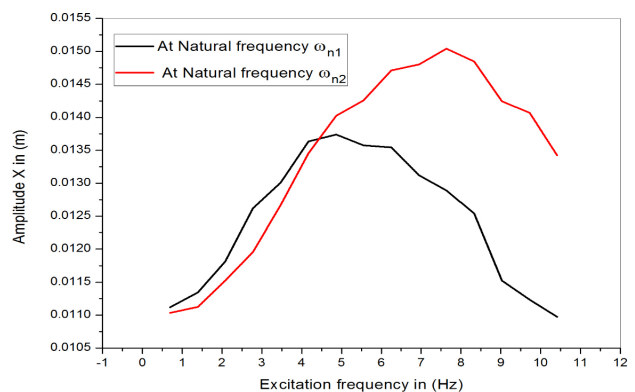The graph for the experimental method is shown in figure 5.



**Figure 5.** Experimental sprung mass displacement for different excitation frequencies

It is observed from the figure 4 that the amplitude increases upto an excitation frequency of 4.86 Hz and later drops for the suspension system operated with a natural frequency $\omega_{n1}$ (6.37 Hz). The maximum amplitude (0.012100 m) is achieved at excitation frequency of 4.86 Hz and this is due to the occurrence of resonance. The suspension system operating with natural frequency $\omega_{n2}$ (9.31 Hz) shows maximum amplitude of 0.013820 m at an excitation frequency of 7.63 Hz and later starts to decrease. This maximum amplitude is obtained because of resonance. The difference in maximum amplitude between the two natural frequencies is due to the fact that both are having different stiffness. It is noticed from the figure 4 that both the natural frequencies intersects at an excitation frequency of 4.5 Hz.

Figure 5 shows the experimental result of variation in amplitude. It is evident from figure 5 that the amplitude increases to an excitation frequency of 4.86 Hz and later it starts decreasing for the system ($\omega_{n1}$). The maximum amplitude of 0.013741 m is attained at 4.86 Hz. Secondly for the natural frequency ($\omega_{n2}$), the maximum amplitude of 0.015043 m is achieved at an excitation frequency of 7.63 Hz. The maximum amplitude achieved for both the natural frequencies are due to the existence of resonance and also it is clear that both the natural frequencies coincides at an excitation frequency of 4.5 Hz. Car suspension parameters considered for the analysis proves the attainment of constant excitation frequency of 4.5 Hz, when the vehicle moves at a speed of 65 Km/h.

## 4. Conclusion

Spring stiffness and damper rate are the two parameters which need to be controlled in designing a suspension system. Theoretical and experimental results conducted depicts difference in amplitude at the respective excitation frequencies, which may be due to non-linearity in the suspension parameters. It is clearly evident that upto an excitation frequency of 4.5 Hz, the amplitude is less for $\omega_{n2}$ system (9.31 Hz) and beyond excitation frequency of 4.5 Hz, the amplitude is less for $\omega_{n1}$ system at natural frequency of 6.37 Hz. Therefore, it is proposed that maintaining the natural frequency of 9.31 Hz until the car reaches 65 Km/h and retaining the natural frequency of 6.37 Hz beyond 65 Km/h will provide less vertical oscillations, when the vehicle is moving on the mentioned road profile. Further research will extend in reducing the reaction time of SMA spring during its transformation from martensite to austenite phase. Therefore, SMA springs with its property of variable stiffness can be best suited in the field of suspension systems in terms of vibration control applications.

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