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Vibration control of FSAE quarter car suspension test rig using magnetorheological damper

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ABSTRACT

Recently MRF damper -which has a significant controllable damping force -used frequently in many active and semi-active suspension systems. However, MRF damper needs controller to estimate the desired force to dissipate the occurred vibration instantaneously. PID controller is one of the effective feedback controllers which shows robustness and simplicity in control MRF dampers, but still the parameters of the PID controller under study to find out the optimum values. This study focused on the vibration control using Magneto-rheological (MR) damper on a FSAE quarter car suspension test rig to study and obtain the optimum running condition. The test rig was designed, modified and then tested using a P-controller integrated with MR damper, unbalance mass used as disturbance and analyzed using LABVIEW software in time and frequency domains. The natural frequency obtained was 2.2 Hz were similar to the actual FSAE car natural frequency. Based on the acceleration against time graph with different proportional gain value the optimal value for proportional gain, Kp was 1. Hence, the experiment work could be used as the initial stage to study and develop a robust controller to suppress vibration on a car.

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1. INTRODUCTION

Commonly, suspension systems are divided into three classes, passive suspension system, active suspension system and semi-active suspension system. The conventional passive suspension limited to fixed mass-spring-damper parameters based on the design requirements, so it is limited to specific design and cannot controlled for other circumstances [1-2]. Active suspension system uses an actuator which generate a counter force to raise or lower the chassis independently at each wheel. Moreover, semi-active suspension system can only change the viscosity of the damping coefficient of the shock absorber to reduce the vibration [3-4].

MR dampers are a controllable damper by controlling yield stress of the MR fluid when a magnetic field exposed to the MR fluid this phenomenon is defined as the MR fluid impact [5-7]. The dampers utilize the benefits of MR fluid into semi-active control system which capable of generating a significant amount of pressure for large-scale applications, with only using battery for power [8-9]. Valve mode are usually used in damper and shock absorber application due to it able to produce a pressure drop when magnetic field are applied to the MRF and at the same time increase the viscosity of the fluid [10]. MR Damper have a lot of part, among of it are pairs of wire, housing, piston, magnetic coil and an accumulator. MR fluid are keep in the using within the cylinder and flow in and out through small orifice. The magnetic coil which produce the magnetic field are built in the piston, when a magnetic force are applied the fluid particle aligned and change of state occur from liquid state to semi-solid state within millisecond [11-12]. This phenomena make the damping force controllable and the force produced from the MR damper are depending on the magnetic field induced based on current flow in the magnetic coil and the velocity of the piston [13].

PID controller are used in many field in the industry due to its robustness and simplicity. The PID controller calculate the difference between measuring signal and the command signal or set a point as error value [14]. But due to the presences of non-linearity in the system PID controller have lower efficiency compare to other type of controller [15]. There are three main parameters in PID controller which are proportional (P), integral (I) and derivative (D). Each parameter are interpreted in term of time, where P depends on the present error; I on the accumulation of past errors and D is a prediction of future errors, based on current rate of change [16-17].

FSAE or Formula SAE cars are small light weight race cars designed according to the rules of Society of Automotive Engineers or SAE collegiate competition [18-19]. In this study we would like to concentrate on enhancing the suspension system of the FSAE cars by using MRF damper controlled by P-controller to achieve comfort driving for FSAE cars.

Quarter car test rig can help in study the effect of various road bumps/humps on a passive semi-active [20], and active suspension system and at the same time perform parametric study to obtain optimum running conditions. In other words, principles of competency- and research-based learning can be experienced [21].

In this study, a quarter car suspension test rig has been develop integrated with MR damper and P-controller system depend on the proportional gain and disabling the integral (I) and the derivative (D) values of the PID controller, the PID controller used here in this research is LABVIEW PID component which simulate the real PID calculations to investigate the performance of the P-controller in controlling the stiffness of the MR damper.

2. P-CONTROLLER SYSTEM

In this study the effect of different proportional gains (Kp) of the PID controller tested, however the Integral and the derivative factors disabled. The aim is to produce a P-Controller system capable of controlling the MRF damper damping force and find out the optimum Kp gain for best performance results.

The system is simple 2 degree of freedom quarter car model using two degree of freedom as shown in Figure 1, contains sprung mass “m2”, un-sprung mass “m1” stiffness of the suspension system “k2” and stiffness of tire “k1”. The excitation of the road are “x0” which will affect the displacement of the un-sprung mass “x1” and the displacement of sprung mass “x2”.

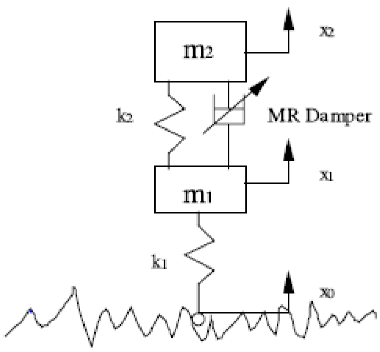

Figure 1. Simplified quarter car model [22]

The governing equation for this simplified quarter car model [23-24] are shown in (1) & (2):

$$m_1\ddot{x}_1 - c_0(\dot{x}_2 - \dot{x}_1) - k_2(x_2 - x_1) + k_1(x_1 - x_0) = \alpha z \quad (1)$$

$$m_2\ddot{x}_2 - c_0(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = -\alpha z \quad (2)$$

Where,
z = MRF damper damping force
$\ddot{x}_2$ = acceleration of sprung mass
$\dot{x}_2$ = velocity of sprung mass
x_2 = displacement of sprung mass
$\ddot{x}_1$ = acceleration of un-sprung mass
$\dot{x}_1$ = velocity of un-sprung mass
x_1 = displacement of un-sprung mass

Let $x = [x_1\ x_2]^T$, and it can written in matrix form as shown in (3):

$$\ddot{x} + C\dot{x} + Kx = Fy \tag{3}$$

Where,

$$C = \begin{bmatrix} \frac{c_0}{m_1} & -\frac{c_0}{m_1} & -\frac{c_0}{m_2} & \frac{c_0}{m_2} \end{bmatrix},\ K = \begin{bmatrix} \frac{k_1+k_2}{m_1} & -\frac{k_2}{m_1} & -\frac{k_2}{m_2} & \frac{k_2}{m_2} \end{bmatrix}$$

$$F = \begin{bmatrix} \frac{a}{m_1} & \frac{k_2}{m_1} & -\frac{a}{m_2} & 0 \end{bmatrix}\ and\ y = [z\ x_0]^T$$

Finally, let $X = [\dot{x}\ x]^T$ and the equation cam be written in state space form as shown (4):

$$\dot{X} = AX + Bu \tag{4}$$

Where,

$$A = [C\ K\ 1\ 0\ ], B = [F\ 0\ 0\ 0\ ]\ , and\ u = [y\ 0\ 0\ ]^T \tag{5}$$

In the next section will explain how the P-controller will be integrated to the suspension system to adjust the MR damper stiffness.

3. EXPERIMENTAL SETUP

Figure 2 shows the controller are design in close loop feedback system, hence the P-controller can adjust the stiffness of the MR damper continuously throughout the experiment. The P- controller adjust the MR damper stiffness proportionally to the error of data collected from accelerometer located on the sprung mass and un-sprung mass. All of the experimental data was recorded using LabVIEW software and then collected to be analyses and discuss. The DC motor was controlled by Arduino and Motor Driver to ensure able of simulate road bump excitation continuously. The manipulative variable in the experiment was the proportional gain value, Kp = 0, 1, 2, 3 & 4, these values feeded to the LABVIEW PID component. The sample rate, time taken, DC motor PWM value and road bump height was set at constant value at 1000, 10 second, 450 delay and 10 cm respectively. The responding variable in this experiment are the difference of sprung mass acceleration at different proportional gain value.

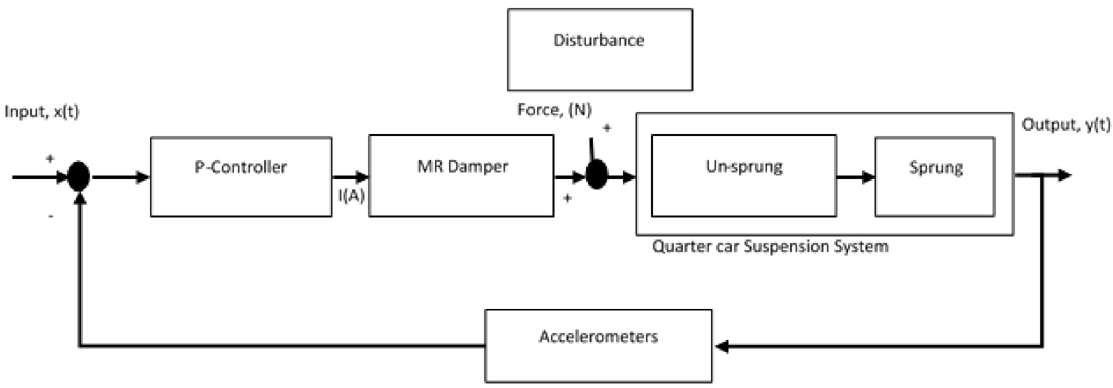

Figure 2. Block diagram of P-controller and MR damper of quarter car suspension test rig

Figures 3 and 4 shows the finished product of the FSAE quarter car test rig, after a couple pretest, the test rig able to function as it was design too. The only problem with the modified test rig shaker is it was unable to lift more than 40 kg, but the result still accepted since some of FSAE car weight are in between 150 kg to 250 kg.

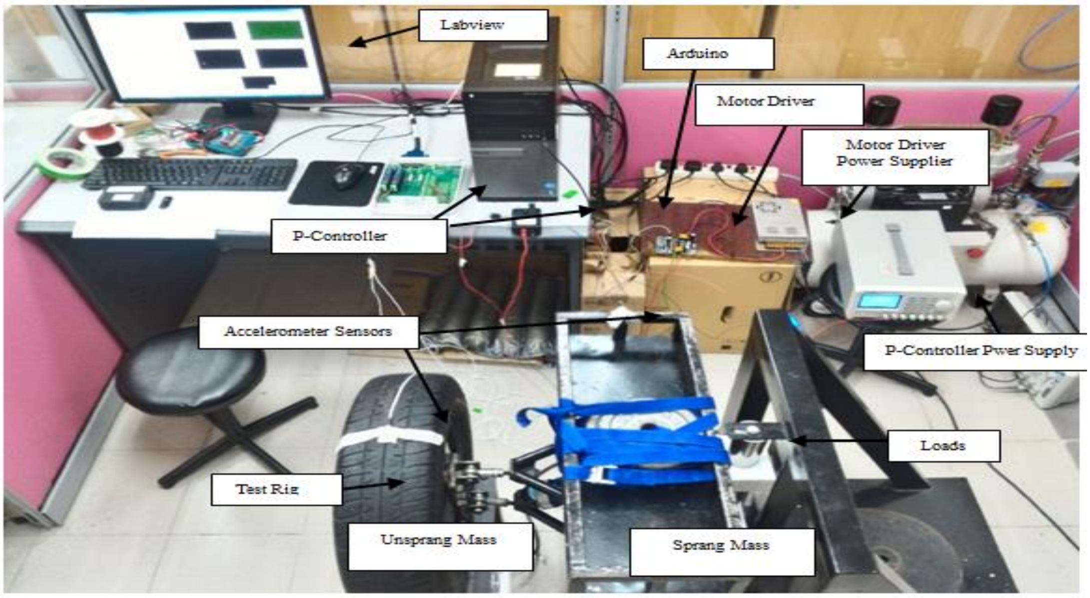

Figure 3. Actual experiment setup

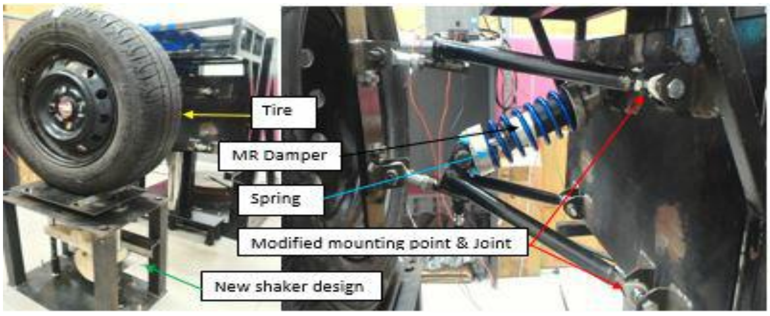

Figure 4. Actual shaker setup

4. RESULTS AND ANALYSIS

MATLAB software was used to generate the graph based on thousands of data collected from the LabVIEW software. The MATLAB software was used as a coding platform to generate graph of acceleration (ms-2) against time (s), transmissibility against frequency (Hz) and power ((ms-2)2/Hz) against frequency (Hz). The graph generated from the coding based on Fast Fourier Transform (FFT) method which were commonly used to analyses vibration. The shaker input signal as shown in Figure 5, was only set up to be 50% of the duty cycle, which mean average of 12 volt are used to power the DC motor to produce it is highest torque to push the load on the test rig.

The response of sprung mass with and without controller in time domain measured. When Kp equal to 1 were the optimize value for P-controller. Hence, the MR damper able to reduce the vibration at the sprung mass effectively. When the gain of Kp more than 1 which make the P-controller becomes unstable thus causing the P-controller to overshot. As shown in Figure 6 the amplitude of acceleration for Kp =1 was lower than Kp =2, 3 & 4. The graphs timeline were not exactly same due to the position of the DC motor cam and starting time were not synchronous for each experiment run. But the result were still accepted due to the timeline of each experiment were not too far apart.

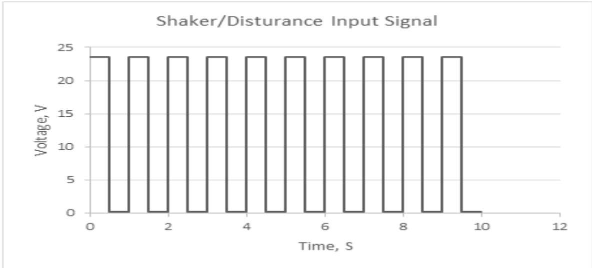

Figure 5. Shaker input signal

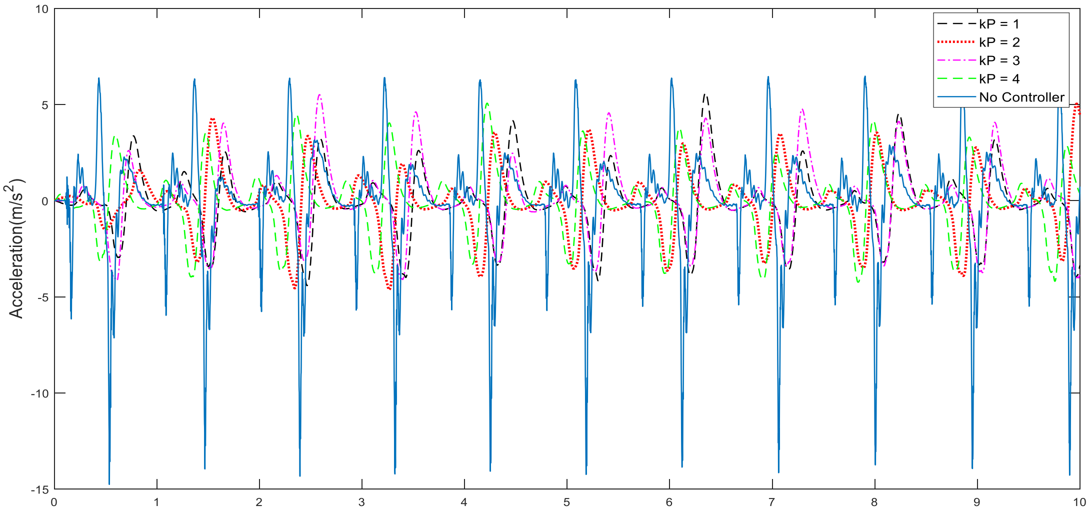

Figure 6. Acceleration amplitude in time domain when kp = 0 (No Controller), 1, 2, 3 & 4

Based on the Figures 7 and 8 the highest peak of amplitude occur when frequency at 2.2 Hz and most of the graph were stacked at the same frequency range, thus the natural frequency for the sprung mass were 2.2 Hz. Typical FSAE car suspension system has natural frequency in range of 2 Hz to 3 Hz, so the experiment were verified with actual FSAE car natural frequency [25-26]. Furthermore, amplitude were highest when the P-controller were not applied to the system and decreasing as the Kp increase, but for Kp = 3 & 4 the amplitude reading increases because the P-controller start to overshoot due to un-stability of the system.

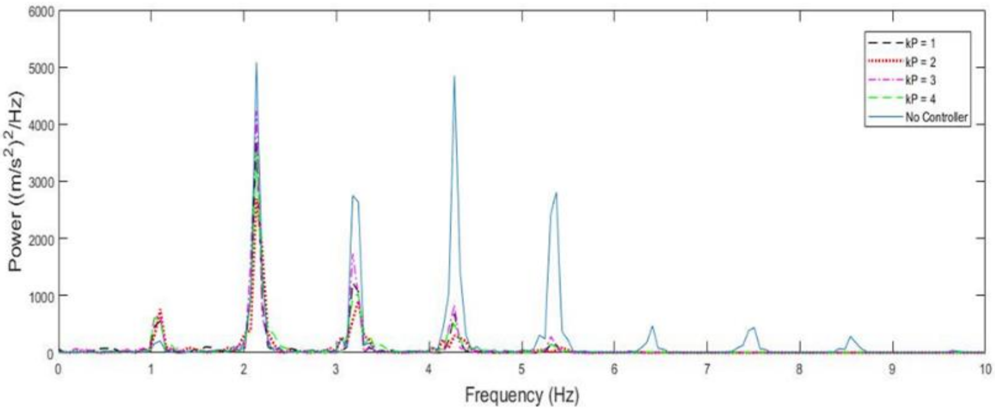

Figure 7. Sprung mass magnitude in frequency domain when kp = 0(No Controller), 1, 2, 3 & 4

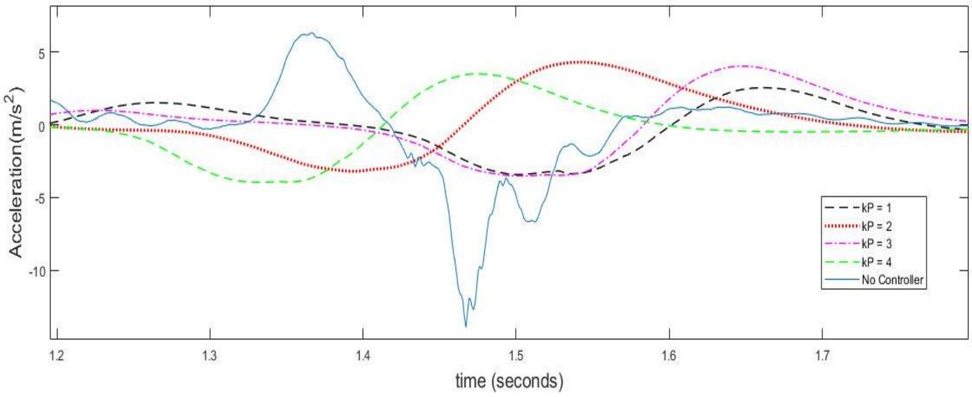

Figure 8. Zoom image acceleration signal in time domain at 1.2 seconds to 1.8 seconds when kp = 0 (No Controller), 1, 2, 3 & 4

Amplitude comparison among the Kp values summarized on the following Table 1. In Table 1 it is clear that Kp=1 has the most optimum results and can be recommended to chosen for further tests on PID controllers as the best parameter value.

Table 1. Amplitude Comparison between Kp Values from 0 to 4

Kp	Time domain (m/s2)	Frequency domain (Hz)
0 (Co Controller)	6 - (-14.9)	0 - 5000
1	1 – (-1)	0 - 2600
2	4- (-4.9)	0 - 2800
3	5.1 – (-4.9)	0 - 4100
4	3- (-4.9)	0 - 4100

5. CONCLUSION

In summary, the most optimum proportional gain value to be set in P-controller were Kp = 1 with the amount of acceleration reduction up to 50% for jounced and 76.66% for bounced compare to system without P-controller. The natural frequency for the FSAE quarter car was found out from amplitude spectrum graph to be 2.2 Hz, where it was verified with the theoretical data which was in the range of 2 Hz to 3 Hz.

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