ARPN Journal of Engineering and Applied Sciences

VIBRATION CONTROL OF A PASSENGER CAR ENGINE COMPARTMENT MODEL USING PASSIVE MOUNTS SYSTEMS

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ABSTRACT

Engine mounting is one of the devices that provide vibration isolation for unwanted vibration from engine to the driver. There are 3 types of engine mounting system which are passive, semi-active and active engine system. This study emphasizes on the validation of mathematical equation derived from Newton 2nd Law of Motion with real time experiment. The study of the characteristic of mounts using simulation the 3-Degree Of Freedom (DOF) mathematical modeling in Matlab Simulink software. Then, the mathematical model is verified by using experimental approach. By comparing the results from the experimental and simulation it shows that the model enables to give same response as in the experimental result.

Keywords: engine mounting, vibration control, frame structure.

INTRODUCTION

Noise vibration and harshness (NVH) issues have become vital aspect concerned in automotive industries. Unwanted vibration may come from many sources that may cause uncomfortable situation to the driver during maneuvers the vehicle. This unwanted vibration may caused by the road condition and also the engine itself. The vibrations produce by the engine cause by the unbalance mass on the engine component during running the engine. Engine mounts system was discovered to minimize the vibration transfer to the driver. There are three types of engine mounting system that had been discovered, namely passive, semi-active and active engine mounts [1-3]. Rubber engine mounting is the passive engine mounting, usually the passive engine mounting is only effective in isolation single frequency. The passive systems depending on the damping coefficient of the rubbers materials used. However, rubber mounts still been used usually for passenger cars due to the simplicity of the design and also in term of low manufacturing cost and also maintenances cost. Then, Hydraulic Engine Mounts (HEM) had been main focus for bigger engine such as diesel's engine vehicle. However, modern car and trucks become lighter from time to time whereas the performance of engine is increase, thus the vibration produces much higher [4]. Due to the drawback of passive system, semi active and active systems were discovered intentionally to provide better isolation vibration range. This had been by many previous publications, one of them was about that the comparison between the performance of passive and active engine mounts [5-8]. Active system is best vibration isolation because of, it provide external source to counter the vibration that produce from the engine. The complexity of the system and high cost is the main resistance to develop the active system. However, recent study focus on low cost active engine mounts. One of the researcher modified the actuator, so that the tunable damping and stiffness of the engine mounts can be optimize by controlling the input current to the actuator [9]. Therefore, less study covered on the active systems in automotive sectors.

Researchers more attracted to discover the Semi–active system, where it gives better performance than passive system with the same cost. Variable stiffness engine mount (VSEM) was proposed which a types semi active engine mounts system, where the system used simple ON-OFF strategy which based on changing the static stiffness [10-13]. Besides that, the design of semi-active is using the same design as HEM. The semi-active system worked using smart material such as Electrorheological (ER) fluid and also Magnetorheological (MR) fluid. The ER Fluid change its phase from liquid to solid-liquid phase when supplied with electric current. While Magnetorheological (MR) fluid will reacts with magnetic field that generated from current that flow into the iron coil. The MR and ER engine mounts that had been discussed by Elahinia [1] about the key parts of the mounts which are bottom and top rubber, top and bottom fluid chambers, flow passages and types of fluid itself. Basically, the flow of these smart materials classified into three basic flows which is flow mode, shear modes and squeeze mode. These modes can be combined or worked as single modes. In order to study the characteristic of these engine mounts, a frame structure system had been derived by using Newton 2nd Law of Motion. The frame structure proposed in this work is considered as a small-scaled intermediate structure which has the dominant elastic modes in the frequency range of ~20 Hz. The method used was referred from previous researcher in his publication [14-16]. The main contribution of this work is to validate the mathematical model to be used before developed the control structure of semi active engine mounts. In order to achieve this, the appropriate experiments on the frame structure was designed and fabricated, its vibration characteristic is filtered after the experiment to eliminate the noise and others disturbance. The passive system is used to the frame structure and subjected to vibration source which is up to ~15 Hz. After established the sensors on the right position, the body acceleration and force transmitted was formulated. The control response such as acceleration and transmitted force at each mount are presented.

MATHEMATICAL MODELLING

The 3 DOF of modeled engine mount represent vertical displacement (Z_s), moment pitch (θ) and moment roll (α) respectively. Assuming that, all of parameter produced during the engine runs due to the unbalance mass on the rotating shaft. The force that generate by the unbalance mass noted as P*(t)*.

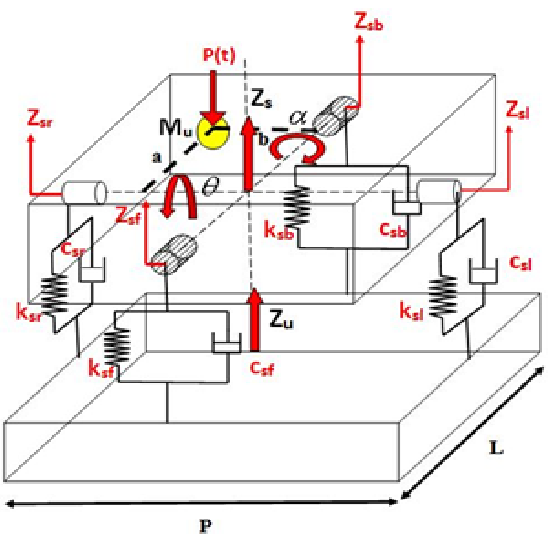

Figure-1. Free body diagram engine mounting system (EMS).

$$\sum F_V = (M_e + M_u)\ddot{Z}_S \tag{1}$$

$$P(t) - F_f - F_r - F_l - F_b = (M_e + M_u)\ddot{Z}_s \tag{2}$$

$$\begin{aligned}&P.(t) - c_{sf}(\dot{Z}_{uf} - \dot{Z}_{sf}) - k_{sf}(Z_{uf} - Z_{sf}) - c_{sr}(\dot{Z}_{ur} - \dot{Z}_{sr}) - k_{sr}(Z_{ur} - Z_{sr})\\ &- c_{sl}(\dot{Z}_{ul} - \dot{Z}_{sl}) - k_{sl}(Z_{ul} - Z_{sl}) - c_{sb}(\dot{Z}_{sb} - \dot{Z}_{sb}) - k_{sb}(Z_b - Z_b) = (M_e + M_u)\ddot{Z}_S\end{aligned} \tag{3}$$

Moment roll:

$$\sum M_r = I_r \ddot{\alpha} \tag{4}$$

$$P(t).a - F_r.\frac{P}{2} + F_l.\frac{P}{2} = I_r \ddot{\alpha} \tag{5}$$

$$P(t)a - \left[c_{sr}(\dot{Z}_{ur} - \dot{Z}_{sr}) + k_{sr}(Z_{ur} - Z_{sr})\right]\frac{P}{2} + \left[c_{sl}(\dot{Z}_{ul} - \dot{Z}_{sl}) + k_{sl}(Z_{ul} - Z_{sl})\right]\frac{P}{2} = I_p \ddot{\alpha} \tag{6}$$

Where,

Z_{sr} = vertical displacement at right side of the frame structure.

Z_{sl} = vertical displacement at left side of the frame structure.

Z_{sb} = vertical displacement at back side of the frame structure.

Z_{sf} = vertical displacement at front side of the frame structure.

$\dot{Z}_{sf}$ = velocity at front side of the frame structure.

$\dot{Z}_{sb}$ = velocity at back side of the frame structure.

$\dot{Z}_{sl}$ = velocity at left side of the frame structure.

$\dot{Z}_{sr}$ = velocity at right side of the frame structure.

$\ddot{Z}_{sf}$ = acceleration at front side of the frame structure.

$\ddot{Z}_{sb}$ = acceleration at back side of the frame structure.

$\ddot{Z}_{sl}$ = acceleration at left side of the frame structure.

$\ddot{Z}_{sr}$ = acceleration at right side of the frame structure.

P and *L* = length of the frame structure.

a and *b* = distance of the unbalance mass from center of gravity.

M_e and M_u = mass of engine and mass of unbalance mass, respectively.

EXPERIMENTAL SETUP

Model validation test as shown in Figure-2 contain 2 parts that represent the engine and passenger compartment of passenger car which the steel frame structure represent the vehicle chassis, the electric motor is the engine which provide the vibration source. The motor power up by the power supply to give the unbalance rotation create based on the set frequency. There were 2 types of sensors used that embedded together with data acquisition which ware accelerometer and gyro sensor. Gyro sensor used to collect moment pitch and roll data while the accelerometer used to save vertical acceleration data. The data acquisition device used was LEGO Mindstorm EV-3, converts the data from analog to digital data which will be restore in the computer. The experimental setup is shown in Figure-2. The position of embedded gyro sensor and accelerometer was placed under the motor electric where the center of gravity located. The electric motor generates 0-50 Hz frequency range. However since this validation test using the passive system as the mounts in order to study whether the pattern of the vibration graph made on the test rig will similarly as the result in the simulation model therefore the frequency that was choose was at idle frequency which is 9 Hz. Low frequency produce large force excitation [17], this is the

VOL. 10, NO. 17, SEPTEMBER 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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reason of selection 9 Hz for electric motor frequency input. The parameter of the experimental setup will be used in the simulation test was tabulated on Table-1.

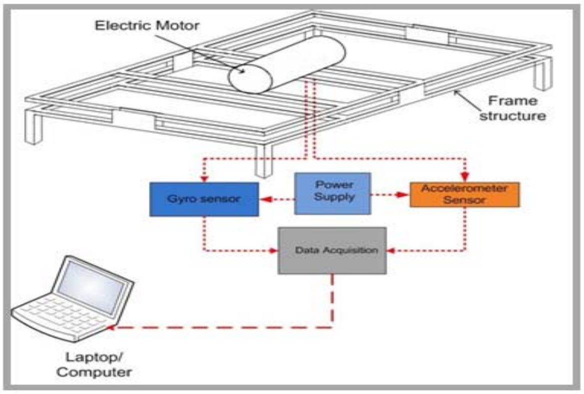

Figure-2. Experimental setup description.

The experiment conducted for various time to ensure the optimization calibration method for each sensors. The results data from the experiment test was filtered using Kalman filter block in the Matlab simulink software in order to remove disturbance data such as noise.

Table-1. Experimental parameter.

Parameter	Value
Electric Motor Frequency	9 Hz
Total Mass	15 Kg
Spring constant	12000 K/N
Damping Constant	1200 K/N
Length of 'a' and 'b'	0.15m
Length of 'P'	0.55m
Length of 'L'	0.36m

RESULTS AND DISCUSSIONS

This section discussed about the comparison between simulation data with experimental data results. Both test using frequency input 9 Hz and the physical parameter for the experiment test such as length, mass and height also had been used for the simulation test. From the comparison, it decides the acceptance of using this simulation model for future works which to build control structures for semi active engine mounting. Figure-3 shows that the comparison of vertical acceleration. The amplitude for the experimental results shows slightly lower than the simulation results. However, it clearly shows that, the experiment enable to follow the trends output results as the simulation. The differences due to the mass of sensors does not included for the simulation test, therefore the lower acceleration result affected from increasing mass.

Meanwhile, the comparisons for moment pitch acceleration have shown in Figure-4 shows that moment pitch acceleration for experiment slightly higher than simulation results. The experimental value slightly higher than simulation results because of the affect of mass and length of the frame structure. When the length increase the moment inertia of the structure also increase thus moment pitch acceleration also increases. This also can be clearly shows in Figure-5 where the moment roll acceleration for experiment is higher than simulation results.

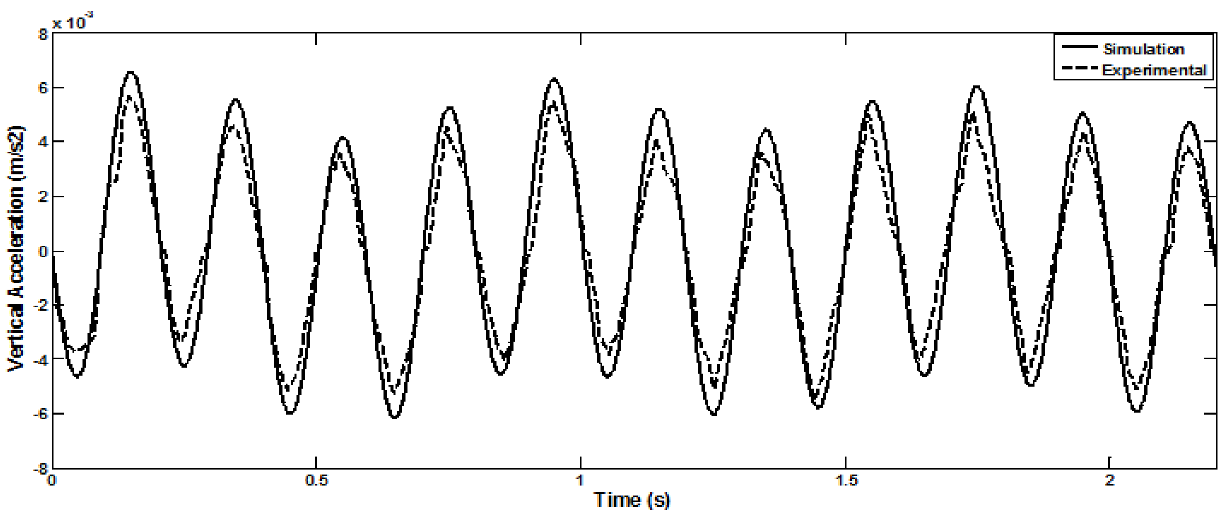

Figure-3. Comparison between the simulation results with the experiment results for the vertical acceleration with frequency input 9 Hz.

VOL. 10, NO. 17, SEPTEMBER 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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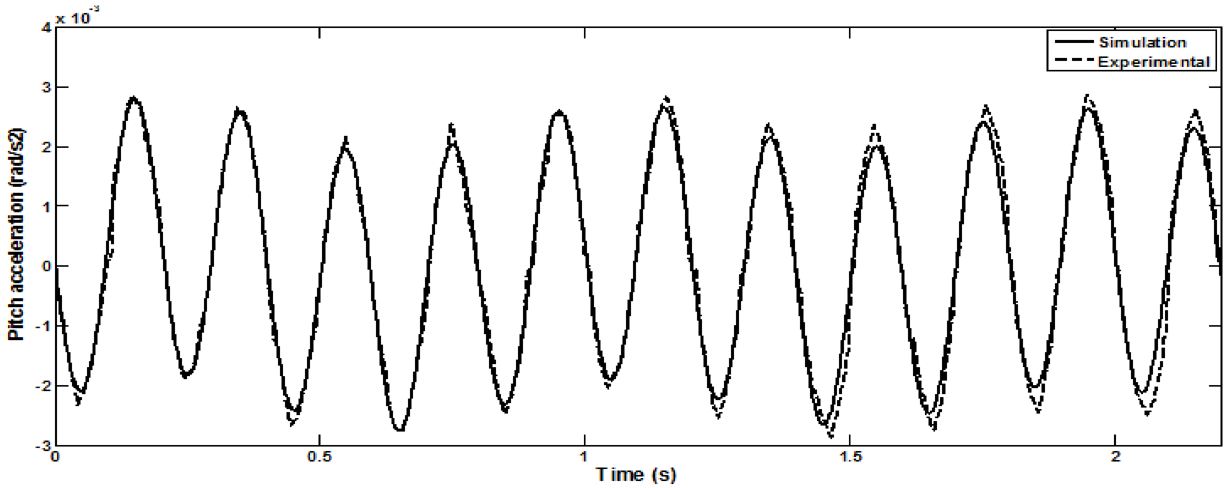

Figure-4. Comparison between the simulation results with the experimental results for the pitch moment acceleration with frequency input 9 Hz.

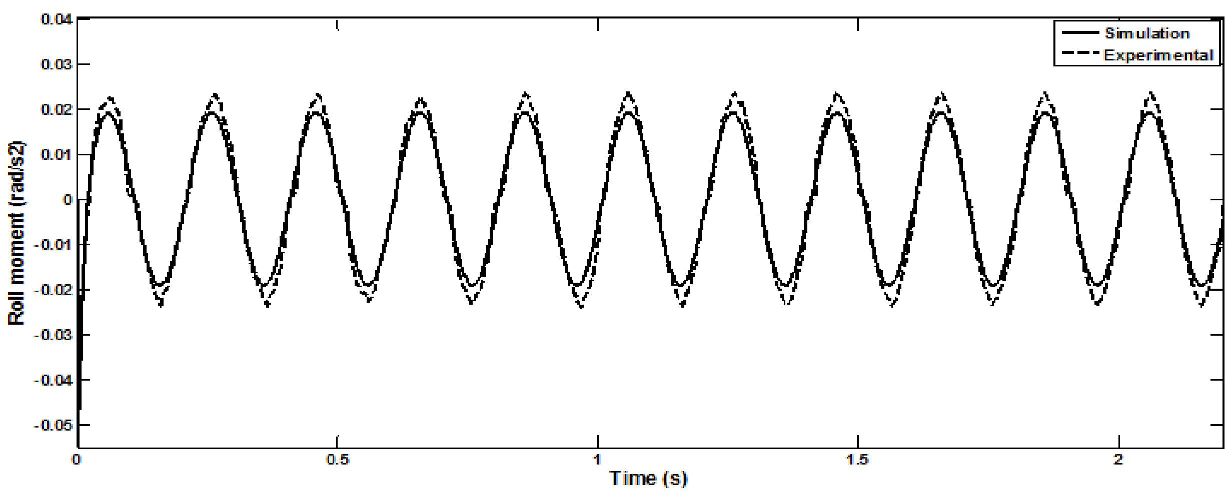

Figure-5. Comparison between the simulation results with the experiment results for the roll moment acceleration with frequency input 9 Hz.

From the experiment conducted, the experimental data was able to follow the trend from the simulation data with slightly difference. This problem ignites due to some problems that occur during the experiment conducted. The first problem discussed was noised disturbance cause by the motor electric. There were two difference sources of noised had been discovered that effected the results. They were through the surrounding environment and also from the mechanical link between unbalance disk and the electric motor shaft. The unbalance disk caused the rotation unstable and produced noised due to friction on the connection between shaft and the disc. Besides that, the mechanical linkage between the bolt and rubber, where the rubber had achieved stretch limits therefore it cannot achieve the desired displacement. This experiment was using the rubber bush fixed by bolt and nut as linkage between the upper frame and bottom frame. However, the limitation for the experiment was the drawback of the rubber bushing material's characteristic. From the experiment, results shows that the rubber bushing unable to achieve the desired amplitude at equal frequency.

CONCLUSIONS

As the conclusion, 3 engine mounts located on the designed test rig for engine mounts able to represents the passenger vehicle engine's compartment. The engine was represents by the Direct Current (DC) motor while the chassis of the vehicle represent by the structural steel frame. The simulink modeled from 3-DOF equation also had been done by using Newton First Law of motion. The parameter for the simulink model was used the actual value of the conventional engine mounts for damping (Cs) and its stiffness (Ks). The others parameter such as the physical measurement was synchronize with the fabricated test rigs data such as the dimension of the test rig. The experiments on the fabricated test rig was validated the derived 3-DOF Mathematical Model since the vertical acceleration, moment pitch acceleration and moment roll acceleration results both tests was similar.

ACKNOWLEDGEMENTS

The authors would like to be obliged to Faculty of Mechanical Engineering and Faculty of Engineering Technology, Universiti Teknikal Malaysia Melaka (UTeM) for providing laboratory facilities and financial assistance under this project.

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