DESIGN AN AUTOMOBILE SIMULATION TEST RIG

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ABSTRACT

This project report deals with dynamic behaviour of Car body with simulation of Sinusoidal wave and Random excitation wave and also modal analysis using theoretical and experimental analysis method, to study selection of test rig development. This project report is to study the design of an automobile simulation test rig using working model 2D analysis and also modal analysis, finally compare it. The structural three-dimensional solid modelling of car body was developed using the SOLIDWORK drawing software. The modal analysis was then performed using ANSYS 14.0 (Modal Analysis). Finally, the experimental modal analysis was performed using Impact Hammer Testing method. The natural frequency of the mode shape is determined and comparative study was done from both method results. The comparison between natural frequencies of modal analysis ANSYS modelling and model testing shows the closeness of the results. From the results, the percentage error had been determined and the limitation in the natural frequency of the car body is observed. The results of this project shown the mode shapes of car body simulation are generally is not in agreement with the experimental value and the frequencies of the experimental modal analysis are a bit different with the frequencies of the simulation. The percentage error is bit high because there are some errors occur during the experimental modal analysis. The experimental modal analysis is conducted with fix condition which effect test rig by using spring as a base of the plate is a factors as the higher percentage error. The comparison between two concept design also been done using working model 2D by sinusoidal wave and random wave excitation. As result, the best studies deformation were been conducted through concept solenoid Actuator Test rig, therefore that concept has been chosen according to the concept scoring method.

ABSTRAK

Laporan projek ini berkaitan dengan perilaku dinamik model kereta dengan menggunakan kaedah analisis teori dan eksperimen. Laporan ini adalah untuk mempelajari sifat dinamik dan perilaku model kereta dengan menggunakan analisis modal secara eksperimen dan membandingkannya dengan analisis elemen secara teori. Pemodelan struktur tiga-dimensi model kereta dilukis menggunakan perisian melukis SOLIDWORK. Analisis elemen modal kemudian dijalankan dengan menggunakan perisian ANSYS 14.0. Analisis di dalam perisian ini menggunakan pendekatan analisis linier modal. Kemudian, analisis modal secara eksperimen dilakukan dengan menggunakan kaedah kesan ketukan. Frekuensi dan bentuk mod ditentukan dan kajian perbandingan dilakukan dari kedua-dua keputusan kaedah. Perbandingan antara frekuensi dari pemodelan elemen secara teori dan ujian model secara eksperimen menunjukkan keputusan yang hampir sama. Dari hasil tersebut, peratus perbezaan antara kedua kaedah telah direkod dan had frekuensi asas. Keputusan projek ini telah menunjukkan bahawa bentuk mod model kereta bagi simulasi secara umumnya adalah tidak sama dengan nilai eksperimen dan frekuensi analisis ragaman eksperimen adalah agak berbeza dengan frekuensi simulasi. Peratus ralat agak tinggi kerana terdapat beberapa kesilapan berlaku semasa eksperimen. Eksperimen dijalankan dengan keadaan tetap kerana menggunakan spring sebagai pelapit model kereta dan memberi kesan redaman berlaku. Kajian perbandingan di antara konsep model dijalankan dengan mengunakan perisian Working Model 2D, dimana dua jenis eksperimen dijalankan untuk menentukan konsep model yang terbaik untuk proses permodelan sebenar iaitu arus sinusoidal dan juga arus secara rawak. Didapati bahawa konsep solenoid actuator memberikan respon terbaik terhadap keperluan eksperimen, maka konsep tersebut dipilih untuk permodelan sebenar menerusi kaedah skor konsep.

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LIST OF SYMBOLS

m	Meter
mm	Milimeter
min	Minute
N	Newton
%	Percent
Hz	Hertz
ω	Omega
π	Pi
s	Second
°	Degree
θ	Angle
$\ddot{\theta}$	Acceleration angle

LIST OF ABBREVIATIONS

NVH	Noide,Vibration And Hardness
DOF	Degree Of Freedom
SDOF	Single Degree Of Freedom
MDOF	Multi Degree Of Freedom
FRF	Frequency Response Function
DAS	Data Acquisition System
CAD	Computer Aided Diagram
IGES	Initial Graphics Exchange Specification
FFT	Fast Fourier Transform
2D	Two Dimensional
3D	Three Dimensional
SI	International System of Units
ODE	ordinary differential equations
RANDBETWEEN	Random in Between

CHAPTER 1

INTRODUCTION

1.1 General Introduction

In a road vehicle, mainly there are two types of vibrations. The first one is deterministic vibration which is caused by rotating parts of the vehicle. Vibration characteristic of this type can be predicted by analytical or numerical methods. At least approximate solutions can be found. The second type is random vibration which is caused by unpredicted loads such as road roughness and wind. Because of the unpredictable loads, future behaviour of the system cannot be precisely predicted.

Responses of randomly excited systems are usually treated using statistical or probabilistic approaches. Studying random vibrations is particularly important, because practically all real physical systems are subjected to random dynamic environments. Random vibration elements may have several damage due to the unpredictable loads, meanwhile fatigue is the most critical concept and vital in aeroplane design, the passengers comfort are very important and determine the design parameter of the suspension systems in road vehicles.

A comfortable ride is essential for a vehicle in order to obtain passenger satisfaction. In this view, vehicle manufacturers are continuously seeking to improve vibration comfort. The transmission of vibration to the human body will have a large influence on comfort, performance and health. Many factors influence the transmission of vibration to and through the body. Being a dynamic system, the transmission associated with it will depend on the frequency and direction of the input motion and the characteristics of the seat from which the vibration exposure is received.

Vibration within the frequency range up to 12 Hz affects the whole human organs, while the vibrations above 12 Hz will have a local effect. Low frequencies (4-6 Hz) cyclic motions like those caused by tyres rolling over an uneven road can put the body into resonance. Just one hour of seated vibration exposure can cause muscle fatigue and make a user more susceptible to back injury.

In this project, we will investigate the vibration response of a three-degree-of-freedom half car model under uncertain random road excitations.

1.2 Project Objective

The purpose of this research is to design an automobile test rig that creates various dynamic properties according to the random road excitation.

1.3 Project Scope

This project focuses on the following points:

i. Design an automobile simulation test rig
ii. Develop Sinosiodal wave and Random road exciation wave diagram.
iii. Study of mode shape and natural frequency of car module
iv. Study of Test rig characteristic in Working Model 2D.

1.4 Problem Statement

Although mathematical modelling tools for analysis have experienced a tremendous growth, most research in vehicle dynamics was based on the assumption that all parameters of vehicle systems are deterministic. Actually, the spring stiffness and damping rate may vary with respect to the nominal value due to production tolerances or wear, ageing. Hence, the problem of vehicle vibration subject to uncertain parameters is of great significance in realistic engineering applications. Therefore this simulation is develop to study every single vibration occurred in vehicle body.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

This chapter is based on the initial study on vibration vehicle by using random road profile excitation. The road noise is creating by different excitation from the actuators, simple indoor four post test rig was studied to understand the basic concept of automobile ground excitation simulation. Actuator studies also been done in term of types and specification to synchronize with the analysis method. The dynamic properties and behaviour of vehicle will be analysis through Dasy lab and so with the actuator is power up therefore initial studies on it was done to understand the functions in detail.

2.2 Basic Vibration Theory

Method and the type of vibration occurred in a vehicle is an important role for the Structural durability, structural integrity and NVH (noise, vibration and Harshness). These effects can cause an adverse effect on the car chassis. The structural durability, structural integrity, and NVH automotive structural performance often depended on the change in compliance with the various road or ground excitations such as uneven road profile, bumps or some taper cornering. Lots of vehicle suspension system contains many types such as passive, semi active and active suspension system which allows the changes of the road profile in compliance with road profile excitations. Vibration of uneven road profile creates damage can have a major impact on the vehicle’s comfort drive. (S. H. Sawant, Mrunalinee, 2010)

2.3 Sinusoidal Wave

Road is considered as an infinite cam with uneven or wavy profile of harmonic waves and wheel of quarter car model is considered as follower. As the road is considered as cam which will give harmonic road excitation to suspension system, in this paper an eccentric cam is used as exciter for suspension system. The road profile is approximated by a sine wave represented by $q(t)=A\sin\omega t$ as shown in Figure.2.1.

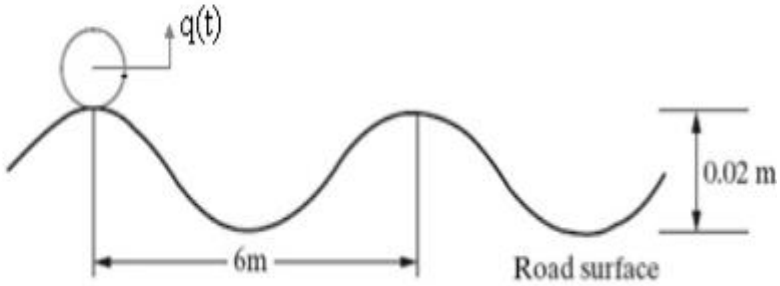

Figure 2.1: Sinusoidal Wave as Represented The Road Profile

Source: S. H. Sawant (2010)

Multi body systems are modelled to study mechanical structures of vehicles using the system approach (Schiehlen 1997). Then, the resulting equations of motion are usually ordinary differential equations (ODEs). There are ideal and non-ideal systems, the first ones are characterized by applied forces and torques independent of any constraint or reaction forces respectively, while the second show such a dependency.

Within the class of ideal systems, ordinary and general multi body systems are distinguished. Ordinary multi body systems are due to holonomic constraints and applied forces depending only on position and velocity quantities, and can be always represented by a system of differential equations of the second order. For non-holonomic constraints or general force laws we get general multi body systems (see also Eberhard & Schiehlen 2004).

2.4 State of Equations

Automobile simulation with ground excitation has several parameters to be determined in order to find the natural frequency and mode shape. Figure 2.2 shows a simple diagram of automobile simulation-ground excitation which explains in detail about the undamped system of half car model with some rotational angle. There are several equation stated as motion direction occurs in car model and the ground excitation are assumed to be random always.

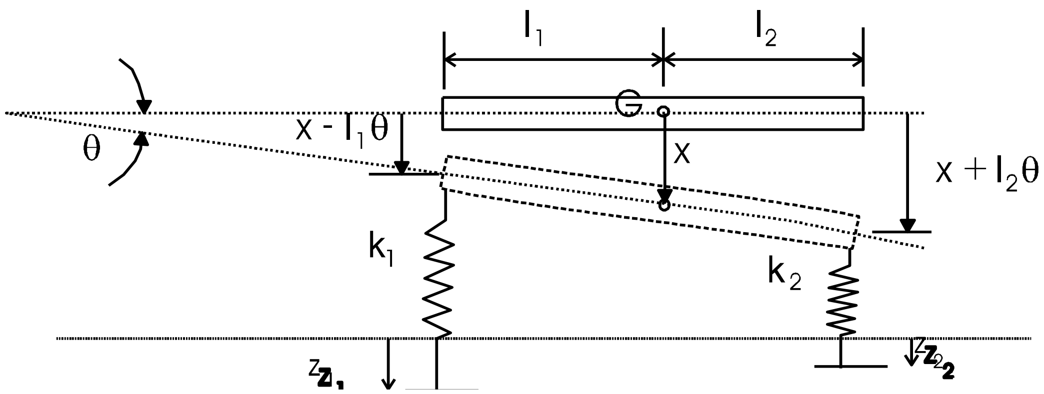

Figure 2.2: Automobile Simulation with Ground Excitation

Source: A.G.A. Rahman (2010)

For motion in the vertical direction:

$$m\ddot{x} + k_1(x - l_1\theta - z_1) + k_2(x + l_2\theta - z_2) = 0 \qquad (2.1)$$

For rotation about G:

$$I_G\ddot{\theta} - k_1(x - l_1\theta - z_1)l_1 + k_2(x + l_2\theta - z_2)l_2 = 0 \qquad (2.2)$$

Thus

$$-(k_1l_1 - k_2l_2)x + I_G\ddot{\theta} + (k_1l_1^2 + k_2l_2^2)\theta = l_2k_2z_2 - l_1k_1z_1 \qquad (2.3)$$

$$m\ddot{x} + (k_1 + k_2)x - (k_1l_1 - k_2l_2)\theta = k_1z_1 + k_2z_2 \qquad (2.4)$$

Decoupling by making:

$$k_1 l_1 = k_2 l_2 \tag{2.5}$$

Then,

The heaving mode,

$$m\ddot{x} + (k_1 + k_2)x = k_1 z_1 + k_2 z_2 \tag{2.6}$$

and pitching mode,

$$I_G \ddot{\theta} + k_1 l_1 (l_1 + l_2)\theta = l_2 k_2 (z_2 - z_1) \tag{2.7}$$

Natural frequencies

$$\omega_1 = \sqrt{\frac{k_1 + k_2}{m}} \tag{2.8}$$

For heaving (x) mode,

$$\omega_2 = \sqrt{\frac{k_1 \ell_1 (l_1 + \ell_2)}{I_G}} \tag{2.9}$$

For pitching (θ) mode,

2.5 Automobile Simulation – Ground Excitation

A four-degree-of-freedom model, shown in Figure. 2.3, was developed for the study. The car was approximated as a flat plate with mass equal to the car, the suspension was represented by four spring-and-damper systems attached to the four corners of the plate, and the driver was approximated as a block mass supported by another spring-and damper system. The forces resulting from the unbalanced inertia force of the engine were taken into consideration as well. The car was assumed to have three degrees of freedom; one for rolling (x1), one for pitching (x2, assumed to be positive opposite the sense of Fig. 2.3), and one for heaving (x3). In addition, the driver was assumed free to move vertically (x4), giving the model its fourth and final degree of freedom.

The analysis required derivation of equations of motion, calculation of natural frequencies and mode shapes, state space analysis, and graphical depictions of system responses over different input and constraint conditions. The system was then optimized to minimize the vertical steady state vibration of the driver, keeping realistic constraints in mind. (Pai, P. F, 2005)

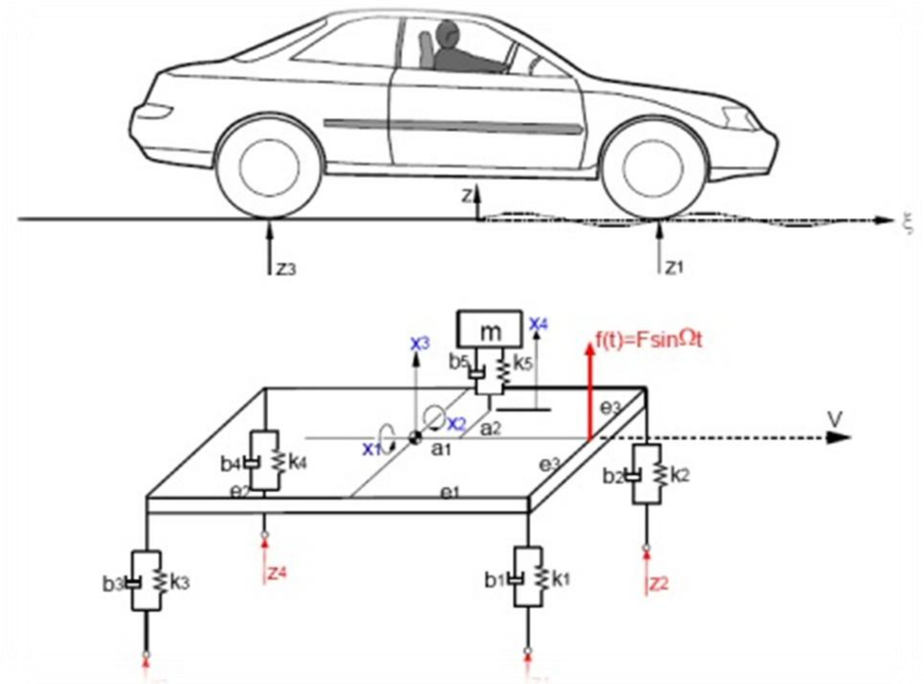

Figure 2.3: Car Model Development

Source: Pai, P. F 2005

The natural frequencies of the system are significant because when they are within the range of a human being's natural frequency (4-8 Hz) resonance will occur causing the motion experienced by the driver to be both exaggerated and uncomfortable. When optimizing the system, the natural frequencies should be made to lie outside this range. Each row gives the system's response in a particular degree of freedom.

Because the highest value for the first mode occurs in the row corresponding to DOF x4, the first mode results primarily in vertical displacement of the driver.

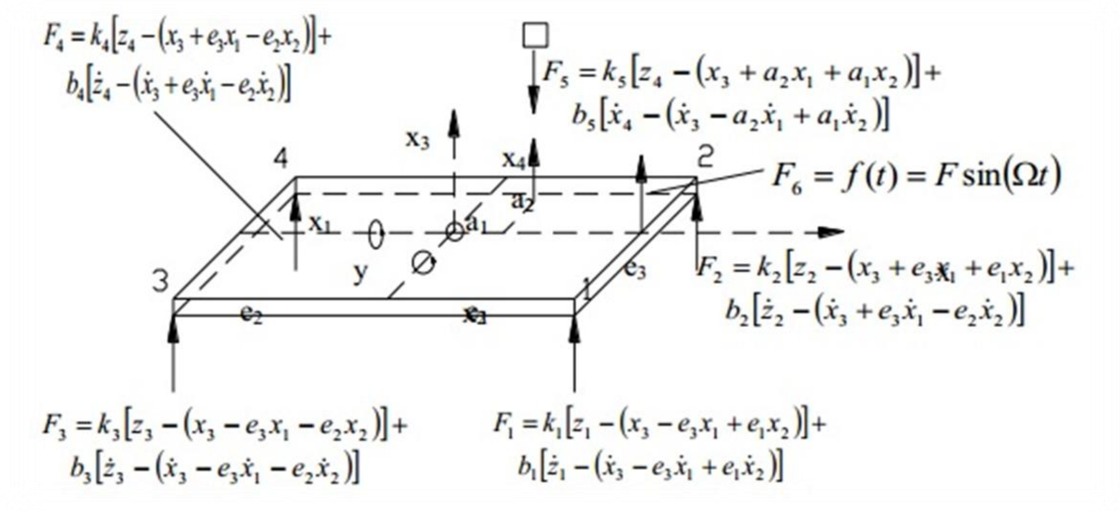

Figure 2.4: Free Body Diagram of a Car

Source: Pai, P. F 2005

2.6 Four Post Test Rig

Indoor testing sessions on two or four wheels vehicles can been carried on using a testing rig called four-poster as shows in figure 2.5. It is constituted by four servo hydrauilic actuators simultaneously controlled in position, which support four vibrating boards. These can be driven by an opportune signal in the range of frequency 0-100 Hz. Different kinds of testing are possible: sine test (sweep in frequency); simple wave form on each actuator; open loop test (driven by an external tension signal); a particular testing method called ICS control, which allows to reproduce on a car an actual service environment, starting from those data coming from the outdoor acquisition sessions. The four actuators are installed on a basement made of cast-iron.

This is isolated from the ground by six pneumatic springs. The testing rig is completed by a power hydraulic central and a control console, including a computer based controller (DCS2000) and a data acquisition unit. Each actuator is equipped with one displacement LVDT sensor, which is located in the rod. The displacement control loop is made possible by these transducers.

Figure 2.5: Four Post Test Rig

Source: Andrea Magalini (2006)

2.7 Suspension System

The suspension system can be categorized into passive, semi-active and active suspension system according to external power input to the system and/or a control bandwidth (Appleyard and Wellstead, 1995). A passive suspension system is a conventional suspension system consists of a non-controlled spring and shock-absorbing damper as shown in figure 2.6. The semi-active suspension as shown in figure 2.7 has the same elements but the damper has two or more selectable damping rate. An active suspension is one in which the passive components are augmented by actuators that supply additional force. Besides these three types of suspension systems, a skyhook type damper has been considered in the early design of the active suspension system. In the skyhook damper suspension system, an imaginary damper is placed between the sprung mass and the sky as shown in figure 2.8. The imaginary damper provides a force on the vehicle body proportional to the sprung mass absolute velocity. As a result, the sprung mass movements can be reduced without improving the tire deflections. However, this design concept is not feasible to be realized (Hrovat, 1988). Therefore, the actuator has to be placed between the sprung mass and the unsprung mass instead of the sky.

2.7.1 Passive Suspension System

The commercial vehicles today use passive suspension system to control the dynamics of a vehicle's vertical motion as well as pitch and roll. Passive indicates that the suspension elements cannot supply energy to the suspension system. The passive suspension system controls the motion of the body and wheel by limiting their relative velocities to a rate that gives the desired ride characteristics.

This is achieved by using some type of damping element placed between the body and the wheels of the vehicle, such as hydraulic shock absorber. Properties of the conventional shock absorber establish the trade off between minimizing the body vertical acceleration and maintaining good tire-road contact force. These parameters are coupled. That is, for a comfortable ride, it is desirable to limit the body acceleration by using a soft absorber, but this allows more variation in the tire-road contact force that in turn reduces the handling performance. Also, the suspension travel, commonly called the suspension displacement, limits allowable deflection, which in turn limits the amount of relative velocity of the absorber that can be permitted.

By comparison, it is desirable to reduce the relative velocity to improve handling by designing a stiffer or higher rate shock absorber. This stiffness decreases the ride quality performance at the same time increases the body acceleration, detract what is considered being good ride characteristics.

An early design for automobile suspension systems focused on unconstrained optimizations for passive suspension system which indicate the desirability of low suspension stiffness, reduced unsprung mass, and an optimum damping ratio for the best controllability (Thompson, 1971).

Thus the passive suspension systems, which approach optimal characteristics, had offered an attractive choice for a vehicle suspension system and had been widely used for car. However, the suspension spring and damper do not provide energy to the suspension system and control only the motion of the car body and wheel by limiting the suspension velocity according to the rate determined by the designer. Hence, the performance of a passive suspension system is variable subject to the road profiles.

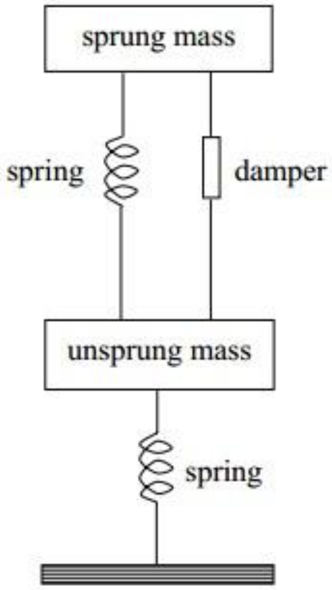

Figure 2.6: Passive Suspension System

Source: Yahaya Md. Sam (2006).

2.7.2 Semi Active Suspension System

In early semi-active suspension system, the regulating of the damping force can be achieved by utilizing the controlled dampers under closed loop control, and such is only capable of dissipating energy (Williams, 1994). Two types of dampers are used in the semi- active suspension namely the two state dampers and the continuous variable dampers. The two state dampers switched rapidly between states under closed-loop control. In order to damp the body motion, it is necessary to apply a force that is proportional to the body velocity. Therefore, when the body velocity is in the same direction as the damper velocity, the damper is switched to the high state. When the body velocity is in the opposite direction to the damper velocity, it is switched to the low state as the damper is transmitting the input force rather than dissipating energy.