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Huang, Hai, High performance control of a multiple-DOF motion platform for driver seat vibration test in laboratory, Master of Philosophy thesis, School of Electrical, Computer and Telecommunications Engineering, University of Wollongong, 2016. https://ro.uow.edu.au/theses/4692

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School of Electrical, Computer and Telecommunication Engineering

High Performance Control of a Multiple-DOF Motion Platform for Driver Seat Vibration Test in Laboratory

Hai HUANG

This thesis is presented as part of the requirement for the Award of the Degree of Master of Philosophy of the

University of Wollongong

March 2016

ABSTRACT

Dynamic testing plays an important part in the vehicle seat suspension study. However, a large amount of research work on vibration control of vehicle seat suspension to date has been limited to simulations because the use of a full-size vehicle to test the device is an expensive and dangerous task. In order to decrease the product development time and cost as well as to improve the design quality, in this research, a vibration generation platform is developed for simulating the road induced vehicle vibration in laboratory. Different from existing driving simulation platforms, this research focuses on the vehicle chassis vibration simulation and the control of motion platform to make sure the platform can more accurately generate the actual vehicle vibration movement. A seven degree-of-freedom (DOF) full-vehicle model with varying road inputs is used to simulate the real vehicle vibration. Moreover, because the output vibration data of the vehicle model is all about the absolute heave, pitch and roll velocities of the sprung mass, in order to simulate the vibration in all dimensions, a Stewart multiple-DOF motion platform is designed to generate the required vibration. As a result, the whole vibration simulator becomes a hardware-in-the-loop (HIL) system. The hardware consists of a computer used to calculate the required vibration signals, a Stewart platform used to generate the real movement, and a controller used to control the movement of the platform and implemented by a National Instruments (NI) CompactRIO board. The data, which is from the vehicle model, can be converted into the length of the six legs of the Stewart platform. Therefore, the platform can transfer into the same posture as the real vehicle chassis at that moment. The success of the developed platform is demonstrated by HIL experiments of actuators. As there are six actuators installed in the motion platform, the signals from six encoders are used as the feedback signals for the control of the length of the actuators, and advanced control strategies are developed to control the movement of the platform to make sure the platform can accurately generate the required motion even in heavy load situations. Theoretical study is conducted on how to generate the reasonable vibration signals suitable for vehicle seat vibration tests in different situations and how to develop advanced control strategies for accurate control of the motion platform. Both simulation and experimental studies are conducted to validate the proposed approaches.

ACKNOWLEDGEMENTS

It is indeed a great pleasure to thank all those who have, directly or indirectly, helped me in successfully completing this thesis.

First I would like to express my deepest sense of gratitude and appreciation to my first supervisor Haiping Du and co-supervisor Weihua Li who had patiently and constantly inspired, encouraged and guided me throughout this program.

Particular thanks go to my senior group member Donghong Ning, who helped me to start the research, thank you very much for your assistance.

And, I want to express my deepest appreciation to my family in China who always understood and supported me during the elaboration and finalization of this thesis.

Thank you all.

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ABBREVIATIONS

cRIO	CompactRIO
DOF	Degree of freedom
HIL	Hardware-in-the-loop
IMU	Inertial Measurement Unit
I/O	Input/Output
NI	National Instruments
PRPS	P-Prismatic joint, R-Revolute joint, S-Spherical joint
R&D	Research and Development
SIL	Software-in-the-loop
SPC	S-Spherical joint, P-Prismatic joint, S-Spherical joint
VI	Virtual Instruments

PUBLICATION

- Hai Huang, Haiping Du, and Weihua Li. "Stability enhancement of magnetic levitation ball system with two controlled electromagnets." In Power Engineering Conference (AUPEC), 2015 Australasian Universities, pp. 1-6. IEEE, 2015.
- Ning, Donghong, James Coyte, Hai Huang, Haiping Du, and Weihua Li. "Experimental Vibration Simulation for Heavy Duty Vehicle Seat Suspension with a Multiple-DOF Motion Platform," SAE Technical Paper 2015-01-0613, 2015, doi:10.4271/2015-01-0613.

CONFERENCE PRESENTATIONS

Hai Huang, Haiping Du, and Weihua Li. "Stability enhancement of magnetic levitation ball system with two controlled electromagnets."; 25th Australasian Universities Power Engineering Conference 2015 (AUPEC 2015) on 27th -30th September 2015, Wollongong, New South Wales, Australia. (Oral Presentation)

1 INTRODUCTION

1.1 Background

With the rapid development of modern technology and the development of heavy manufacturing and the mining industry, vehicle construction and performance during driving have become more complex and people's expectations for vehicles have increased. Heavy duty vehicles are required for some industries and these vehicles must meet increased demands. For these vehicles, the requirements of driver's working environment and safety have also increased [1]. For the vehicle manufacturing industry, the comfort and safety testing of vehicles has become an important component of the vehicle research and development process. This testing is required to assure the dynamic performance and reliability of the vehicle, measure of comfort and minimize the influence of vehicle vibration on the driver.

Early vibration testing was done on the road [3] [4]. However, this type of test was time-consuming and costly and posed potential safety concerns. For these reasons, this kind of vehicle test method was gradually eliminated. Several companies established different special testing environments to experimentally measure vibration conditions of vehicles. These testing methods can reduce experimental cost and gather accurate vibration data for a driving vehicle. However, this type of outdoor test requires use of a real vehicle, thus the cost of the experiment is still expensive and safety cannot be guaranteed. Therefore, some vehicle manufacturers prefer to use indoor or laboratory testing.

Compared with outdoor testing, laboratory testing uses a dynamic vehicle simulator test rig or virtual vehicle simulation on a computer in place of real vehicles. Especially for the driver seat vibration test, laboratory vehicle vibration simulation allows the replication in the laboratory of the conditions experienced by a vehicle in service. The objectives of these simulations are to assess the performance of the vehicle seat in terms of ride comfort or vibration isolation. The advantage of laboratory vibration simulation is that it allows accurate measurement and the tests can be carried out in a controlled environment to enhance repeatability. Additionally, this approach can reduce testing time and cost and maintains safety. Therefore, for vehicle seat vibration isolation testing, simulators have proved popular for durability testing.

However, much research of vehicle vibration control of seat has been limited to simulation because it is expensive and dangerous to use a complete vehicle in the test devices. Although some early vehicle seat test platforms can generate vibrations, they cannot perfectly simulate the situation when a vehicle is moving off-road, because these vehicle vibrations are normally generated in multiple degrees of freedom [2]. Therefore, in order to reduce development time and research cost and improve design effectiveness, establishment of an effective vehicle vibration simulation platform in the laboratory is required to simulate road-induced vehicle vibration.

1.2 Historical Overview

The previous development of a vehicle test platform corresponded with the two stage servo-value designed by Moog in the 1950's [6]. The servo-value in a simple position feedback loop connected the flexibility of electronic control and the power of hydraulic actuation. Closed-loop control of hydraulic actuators in an acceptable frequency range became a reality, and engineers recommended test platforms as a strategy to reduce the time and cost of vehicle development.

The 4-Posters were the early vehicle test platforms, and were designed to input vertical displacements through the contact patch at each wheel. These were used to investigate vehicle ride. However, there were several flaws with this system [7]: there were significant differences between the rolling and unrolling in terms of the stiffness and damping characteristics of the unsprung mass and there was a total lack of lateral or fore and aft inputs to the vehicle. Thus the simulation quality and accuracy were insufficient.

In the 1970's, research and development was focused on improving simulation quality and concentrated on minimising the effects of the non-rolling type [8][9]. Paper [10] improved the test platform using a rig with a 0.7m endless steel band around each vertical actuator. The vertical motion was imparted to the rolling tyre using an air cushion and horizontal excitation was added using electro-dynamic shakers to the wheel centre-line. For continued development of excitation simulation, the reproduction of measured road profiles through the four vertical actuators upon which the vehicle rested was performed using non-rolling tyres. Tyre pressures were adjusted to allow for stiffness changes between the road and laboratory conditions [11].

Subsequently, many simulation test systems were designed and applied in automotive plants and research laboratories. The early laboratory vehicle test platforms mainly used complete vehicles for ride comfort and vibration performance testing. However, as the range of vehicle application and the requirement of driving comfort increased, researchers focused both on improving the vibration isolation ability of the suspensions and on seat vibration isolation. A complete vehicle is not required for vehicle seat vibration isolation testing, as long as the platform can generate vibrations that mimic the characteristics of the whole vehicle vibration. Most vehicle seat testing platforms did not fully reflect vehicle vibrations, but only set the value of the vibration frequency to that most harmful to human body and exerted vibration only in the vertical direction [13]. Although some seat test platforms were designed based on the ISO 7096-82 International Standard, which specifies the preconditions and the procedure for laboratory seat testing [12], the vibrations generated by the platform did not fully approximate the vibrations produced by a real vehicle, especially an off-road heavy duty vehicle.

1.3 Types of Vibration Generate Platform Structure

Various methods have been proposed to improve the accuracy of vehicle chassis vibration simulation. In [18], a vibration simulator was designed to closely simulate the responses of a particular vehicle under driving conditions. As shown in Figure 1.3-1, the platform is supported by springs and there are 4 cylinders to drive the platform movement about the X, Y, Z axes. This means the simulator only has 3-DOF, and therefore cannot simulate the complex vibration in the 6-DOF.

However, the advantage of this simulator is that it simulated the driver's cab completely, including a seat, a steering wheel, and footplates. These objects all are able to transmit vibrations from chassis to the driver; therefore, this platform is welldesigned for a more comprehensive study of vibration on drivers. However, when a driver is operating the vehicle, his centre of gravity is on the seat and the main vibrations are transmitted from the seat to the driver's body. Thus, for the vehicle seat vibration isolation study, the other components of the driver's cab are not considered.



Figure 1.3-1 Schematic diagram of a vehicle vibration simulator [18]

In [21], the authors aimed to design a low-cost PC based simulator. Using a PC to control a vehicle seat test platform allows advantages over other general controllers. The powerful processing and computing PC can improve the response of the system and provides real-time process control [22]. The authors presented a new structure of the vehicle simulator, as shown in Figure 1.3-2. This vehicle simulator consists of a 3-DOF parallel mechanism platform and a 2-DOF motion platform. The advantage of this structure is that it is easy to be controlled, because there is little effect of the low cooperation degree between motors when testing several motors concurrently. However, compared with other parallel platforms, like the Stewart platform, the load-carrying capacity of the platform is less but the performance requirements of the motors are higher. This means that the use of motors with inferior performance will reduce the response competence of the system and reduce the accuracy of the simulation. Thus, one proposal is to use springs to improve the performance of the system [21].



Figure 1.3-2 Low cost 5-DOF platform [21]

Comparing the work reported in [18] and [21], both used springs to support the platforms, considering that the load-carrying capacity of the platform is an important specification for the vehicle test platform and the springs can reduce the influence of gravity [23]. However, the springs only can work at a specific vibration frequency, otherwise the springs could interfere with the simulation. For this reason, for a seat test platform designed to mimic that of a heavy-duty vehicle, it is necessary to avoid the use of springs in the system.

Multiple-DOF parallel mechanisms are suitable to describe the vehicle seat vibration test [24][25][26] and must include high accuracy position, high dexterity, and high force-to-weight ratio [30]. There is a 6-DOF parallel cable-driven mechanism used on a virtual sports machine and on an ultrahigh speed manipulator [27]. In [24], authors applied this mechanism to a motion platform, as shown in Figure 1.3-3. Comparing with the Stewart platforms described in [25][26], the 6-DOF parallel cable-driven mechanism has a larger motion range and shows a higher simulation accuracy because no spherical joints are used. However, a large motion range is unnecessary for a vehicle seat vibration test platform [25][28]. Additionally, this kind of structure is less convenient for test seat installation and removal than the Stewart platform. There are 4 principal measurements that must be considered for the vehicle seat vibration test platform, magnitude, frequency, direction (axis) and duration [29].



Figure 1.3-3 Photo of the prototype motion base developed [24]

The Stewart platform is a classical 6-DOF parallel motion mechanism that is widely used for driving simulators, large spherical radio telescopes, manipulators, and others [31][32][33][34]. A Stewart platform works effectively in a vehicle chassis vibration simulator. Although the Stewart platform increases the complexity of kinematic analysis, dynamics analysis, and control compared to a conventional serial mechanism due to its inclusion of several close-loop structures, it remains the best structure for vehicle seat vibration test platform.

1.4 A Review of Stewart Platform

1.4.1 Historical Overview

A parallel mechanism was designed by Gough and Whitehall and used in tire testing [36]. In 1965, Stewart proposed a parallel mechanism with 6-DOF motion competence [35]. Stewart, Gough and Whitehall proposed improvements to the parallel mechanism, resulting in the Stewart Platform. The classical structure of Stewart platform includes an up and down platform (load platform and base station) and six flexible supporting legs. In this structure, the motion platform and supporting legs are connected by six spherical hinges and the down platform and supporting legs are connected by six Hooke joints, therefore it is also described as a 6-SPS (S-Spherical joint, P-Prismatic joint, S-Spherical joint) mechanism structure. Since then, engineers have developed a typical six freedom parallel platform. In 1988, Behi proposed a three leg PRPS (Prismatic joint, Revolute joint, Spherical joint) parallel platform mechanism [37]. That same year, Hudgens and Tesar proposed six flexible legs, each leg with four-bar linkage mechanism [38].

To solve the motion problem of the Stewart platform, Mcinroy et al. developed a Stewart platform with Jacobi orthogonal matrix for decoupling. In 1993, Geng and Hanes incorporated a Cubic configuration model in the development of a 6-DOF vibration isolation Stewart platform prototype [39]. In 2003, Jafari and McInroy demonstrated the orthogonal Stewart platform in their published thesis [40].

The Stewart platform is a parallel mechanism and has advantages over serial mechanisms. The Stewart platform shows high precision, high stiffness, stable structure, strong bearing capacity, small movement inertia and excellent dynamic characteristics. Therefore, since the development of the Stewart platform, it has received widespread attention and has demonstrated effective in the field of automotive testing with a wide range of applications:

(1) Tire testing machine

The tire testing machine is the earliest application of a Stewart platform. Before Stewart pointed out the parallel mechanism of flight simulator with 6-DOF motion competence, Gough invented a parallel mechanism used for tire testing in 1947 [35], as shown in Figure 1.4-1.



Figure 1.4-1 Gough universal tire-testing machine [35]

(2) Driving simulator

The use of a Stewart platform as a driving simulator is one of the most important applications in the automotive industry. It can provide the experience of transient overload dynamic to the driver during vehicle movement. It provides a continuous sense of gravity component and the information of component convulse impact allowing realistic vehicle simulation. Figure 1.4-2 shows a driving simulator designed by SHERPA. In [41], a SHERPA driving simulator was used to study driving comfort with hydraulic suspensions and continuous semi active dampers.



Figure 1.4-2 Stewart platform based the SHERPA driving simulator [41]

(3) Damper test equipment

Compared with a serial-link mechanism, the Stewart platform can maintain uniformity in six directions with respect to frequency performance and rigidity. In [42], a Stewart platform was used as universal damper test equipment. With this equipment, test dampers can be installed in two directions for testing.



Figure 1.4-3 Vehicle damper test equipment [42]

1.4.2 Theoretical Research

Previous studies of Stewart platforms have focused on kinematics including inverse and forward kinematics solutions, workspace analysis, and singularity analysis. For the vehicle seat vibration test platform, the inverse kinematics problem is an important issue and has been discussed extensively. Compared with the forward kinematics, the inverse kinematics problem of the Stewart platform can be solved relatively easily. The forward kinematics solution of the Stewart platform requires the lengths of the six legs to obtain the posture of the motion platform. However, with closed-form solutions, there are multiple possibilities [47].

In the Hardware-in-the-Loop (HIL) vehicle seat vibration test platform study described below, the posture of the motion platform is provided, so only the inverse kinematics problem is considered. Although there are many reports of kinematics analysis of the Stewart platform, fewer studies have focused on dynamics and control [45].

There is a method to model the dynamics of Stewart platform using the Euler-Lagrange equation [44]. Although it has a clear expression, a large amount of symbolic calculations are required, thus this is not suitable for HIL real-time computation.

The Newton-Euler approach is another method for deriving closed-form dynamic equations [46]. The Newton-Euler formulation does not require evaluation of derivatives of any functional, so it obviates a large number of complex calculations.

For the HIL vehicle seat vibration test platform study, a complete linear dynamic model of motion platform using Newton-Euler method is suggested.

1.5 Hardware-In-The-Loop (HIL) System

The Hardware-in-the-Loop (HIL) system is extensively applied in engineering R & D (Research and Development), especially for vehicle design, control tests, and electronics systems [9]. Compare with a virtual prototyping simulation, HIL simulation used both real physical and virtual system components. HIL techniques play an important role in validating ideal models in an early development phase, and for vehicular on-board devices testing, a HIL test approach is cheaper and more efficient than a prototype vehicle test [15] [16].

Real-time control approaches can be divided into three categories, as shown in Figure 1.5-1:

- 1. Control prototyping: Simulating and testing the controller during control of the real process.
- 2. Software-in-the-loop (SIL): Using a virtual controller to control the virtual prototype. This is usually used for design and feasibility verification of the devices during the pre-exploitation period.
- 3. Hard-ware-in-the-loop (HIL): The simulated process can be controlled by a real controller.



Figure 1.5-1 Classification of real-time control system

The first applications of the HIL system were for flight simulation, where the early aim was to test the instruments using a cockpit, and then to control the cockpit to simulate aircraft motions for pilot training [17]. The simulator uses a real cockpit and motions generated by a platform [17] [18].

The HIL motion simulation platforms, with hydraulic or electrical actuators, were designed for vehicle dynamic tests, such as suspension performance, ABS, and other applications. One type of HIL simulation system is found in a driving simulator [19].

As digital control systems continue to develop, the HIL simulation system is constantly being improved. The simulation models became more and more complex, and computers were unable to meet the requirements for control speed. The application of Field Programmable Gate Array (FPGA) technology allowed for real-time simulation systems of a parallel mechanism system composed of sensors, actuators, and suspensions. FPGA facilitated the further development of Programmable Array Logic (PAL), Generic Array Logic (GAL) and programmable logic devices (PLD) programmable devices. The logic functions are performed by an internal regular array of logic cell arrays. Logic cell arrays are composed of three parts: a configurable logic block, an input output block, and interconnection of these two units. Engineers can reconfigure the internal logic module and IO module of the FPGA to achieve a desired function. The advantages of PFGA include customized FIFO, hardware timer, high reliability, and digital signal processing and analysis. These advantages provide a flexible and low-cost solution for the vehicle electric testing technology.

FPGA can achieve splitting of closed-loop control loop rates on hardware level. FPGA programming allows rapid response of the simulation to input signals. With the parallelism of FPGA, multiple fast control loops can be integrated into one system. For example, Drivven used the reconfigurable application performance of FPGA to establish a Yamaha YZF-R6 engine control system and reduced costs by eliminating the need to purchase custom hardware during the design process [48]. MicroNova also uses the highly reliable and customizable logic functionality of FPGA hardware platform in world's first V12 fuel engine hardware-in-loop simulation [49].

1.6 Research Objectives

Although the desirability of laboratory simulation is clear, there are significant pitfalls. The loading conditions and practical limitations of the test rig hardware to

mimic external loading are not fully understood. Additionally, the capital cost of the test equipment and control technology may be a limiting factor.

As technology develops, vehicle design and dynamical driving properties become more complex. In driver seat vibration isolation studies, more research is devoted to improvements of vehicle driving comfort and safety. However, current vibration control of seat suspension study has been limited to simulations because the use of a full-size vehicle would be expensive and dangerous. Currently, two methods can be used for laboratory driver seat vibration testing. The first one is a physical system where excitation signals are applied via four vertical actuators into the tyres of the vehicle. However, this system suffers from high complexity and high cost. Additionally, use of a particular vehicle as the laboratory simulation platform will restrict the generalization of the findings. For example, different vibrations of the vehicle may be detected due to the use of different damper or chassis structures on the same road. In these kinds of systems, these components may be difficult to replace. Therefore, this system can only simulate one kind of vehicle. Another method is a virtual prototyping system. Although this method is highly reconfigurable, it can only be used for virtual testing, meaning that it only can simulate vehicle dynamics on the computer.

In this study, the goal is to find a better approach to combine the computer vehicle vibration mathematical model and the Stewart platform together as a HIL system and to control the platform to simulate the vibrations in a close approximation of real vehicle movement dynamics. Especially for heavy-duty vehicles, the main range of the chassis vibration frequency is from 0.5Hz to 5.0 Hz. Because the vibrations of off-road vehicles occur in multiple directions, the vibration platform must possess a sufficient degree of freedom to approximate the real vehicle vibrating situation. The necessary problems to address are how to process the output data of mathematical model and convert the data into the Stewart platform controller input data.

Additionally, it is necessary to determine the accuracy of the system when the platform is in a heavy load situation, the maximum vibration frequency of the platform, and the range of the platform movement.

1.7 Thesis Layout

This thesis is structured to provide a chapter-by-chapter view of the progressive work done to achieve these research objectives. Chapter 2 focuses on the relevant theories and simulations, including 7-DOF full car and Stewart platform modelling. Chapter 3 outlines the experimental design and achievement. Chapter 4 details the results of the experiment. Chapter 5 presents the conclusions and directions for future work.

2 RELEVANT THEORY AND SIMULATION

In this section, the 7-DOF full-car model is used to study the response of the vehicle to a bump road input. Furthermore, a mathematical model of the Stewart platform is designed and inverse kinematics task is implemented in relation to the model of the platform.

2.1 Full Car Model Analysis and Synthesis

2.1.1 7-DOF Full-car Model

The vehicle suspension fork is an important component as it connects the vehicle body and axle and transmits the interaction forces and moments of the vehicle body and wheel. The main function is to reduce the vibration and shock caused by uneven surface, and to provide a comfortable and safe driving environment.

Many designs of a control system for a full car system used the well-known 7-DOF model. In this study, we considered 3-DOF for the sprung mass and 4-DOF for the front and rear unsprung masses for their vertical motion. A goal was to determine a set of equivalent parameters that makes the simple model as close to the original system as possible.

The full car model, which is shown in Figure 2.1-1, was used to approximate the behaviour of a complete vehicle. The variables are defined as follows: The $Z_{r1} \sim Z_{r4}$ are the road displacement inputs. It is assumed that the rear tires travel over the same path as the front tires. Therefore, Z_{r2} is a delayed version of Z_{r1} , $Z_{r2} = Z_{r1}(t - \tau)$, and $Z_{r4} = Z_{r3}(t - \tau)$, where $\tau = v/(a + b)$, v is the forward vehicle speed, and a + b is the vehicle wheelbase. The variables $M_{u1} \sim M_{u4}$ are the under sprung mass displacements. The four actuators placed between the sprung mass, Ms, and the unsprung mass, $M_{u1} \sim M_{u4}$, produce control forces $u_1 \sim u_4$. The variables $Z_{u1} \sim Z_{u4}$ are the sprung mass displacement, and φ is the roll angular displacement. The tire stiffness are denoted by $k_{u1} \sim k_{u4}$. The damping effect of the tires is negligible. The $k_{s1} \sim k_{s4}$ and $c_{s1} \sim c_{s4}$ denote the stiffness and damping ratios of passive suspension elements for the assemblies.



Figure 2.1-1 A 7-DOF full car model

The roll axis defined by the suspension kinematics is assumed to be horizontal. The pitch axis is assumed to intersect the roll axis directly under the centre of the sprung mass. The characteristics of all passive suspension elements are linear and the equations of motion can be derived by application of Newton's second law and its simplification as Euler's equation of motion for a rigid body as follows: For the bouncing of the sprung mass:

$$\begin{split} M_{S}\ddot{Z}_{S} &= -(c_{s1} + c_{s2} + c_{s3} + c_{s4})\dot{Z}_{s} - (k_{s1} + k_{s2} + k_{s3} + k_{s4})Z_{s} \\ &- (-ac_{s1} + bc_{s2} - ac_{s3} + bc_{s4})\dot{\theta}_{s} \\ &- (-ak_{s1} + bk_{s2} - ak_{s3} + bk_{s4})\theta_{s} + c_{s1}\dot{Z}_{u1} + c_{s2}\dot{Z}_{u2} \\ &+ c_{s3}\dot{Z}_{u3} + c_{s4}\dot{Z}_{u4} + k_{s1}Z_{u1} + k_{s2}Z_{u2} + k_{s3}Z_{u3} \\ &+ k_{s4}Z_{u4} + u_{1} + u_{2} + u_{3} + u_{4} \end{split}$$
(1)

Pitching of the sprung mass:

$$I_{P}\ddot{\theta}_{S} = -(-ac_{s1} + bc_{s2} - ac_{s3} + bc_{s4})\dot{\theta}_{s}$$

$$-(-ak_{s1} + bk_{s2} - ak_{s3} + bk_{s4})\theta_{s}$$

$$-(a^{2}c_{s1} + b^{2}c_{s2} + a^{2}c_{s3} + b^{2}c_{s4})\dot{\theta}_{s}$$

$$-(a^{2}k_{s1} + b^{2}k_{s2} + a^{2}k_{s3} + b^{2}k_{s4})\theta_{s} - ac_{s1}\dot{Z}_{u1}$$

$$+ bc_{s2}\dot{Z}_{u2} - ac_{s3}\dot{Z}_{u3} + b_{s4}\dot{Z}_{u4} - ak_{s1}Z_{u1} + bk_{s2}Z_{u2}$$

$$- ak_{s3}Z_{u3} + bk_{s4}Z_{u4} - au_{1} + bu_{2} - au_{3} + bu_{4}$$
(2)

Rolling of the sprung mass:

$$I_{r}\ddot{\varphi}_{s} = -\frac{1}{4} \left(t^{2}{}_{f}c_{s1} + t^{2}{}_{r}c_{s2} + t^{2}{}_{f}c_{s3} + t^{2}{}_{r}c_{s4} \right) \dot{\varphi}_{s} - \frac{1}{4} \left(t^{2}{}_{f}k_{s1} + t^{2}{}_{r}k_{s2} + t^{2}{}_{f}k_{s3} + t^{2}{}_{r}k_{s4} \right) \varphi_{s} - \frac{1}{2} \left(-t_{f}c_{s1}\dot{Z}_{u1} - t_{r}c_{s2}\dot{Z}_{u2} + t_{f}a^{2}c_{s3}\dot{Z}_{u3} + t_{r}c_{s4}\dot{Z}_{u4} \right) - \frac{1}{2} \left(-t_{f}k_{s1}Z_{u1} - t_{r}k_{s2}Z_{u2} + t_{f}a^{2}k_{s3}Z_{u3} + t_{r}k_{s4}Z_{u4} \right) - \frac{1}{2} \left(-t_{f}u_{1} - t_{r}u_{2} + t_{f}u_{3} + t_{r}u_{4} \right)$$

$$(3)$$

Vertical direction for each wheel:

$$M_{u1}\ddot{Z}_{u1} = -c_{s1}\dot{Z}_{u1} - (k_{u1} + k_{s1})Z_{u1} + k_{u1}Z_{r1} + c_{s1}(\dot{Z}_s - a\dot{\theta}_s + \frac{1}{2}t_f\dot{\phi}_s) + k_{s1}(Z_s - a\theta_s + \frac{1}{2}t_f\phi_s) - u_1$$
(4)

$$M_{u2}\ddot{Z}_{u2} = -c_{s2}\dot{Z}_{u2} - (k_{u2} + k_{s2})Z_{u2} + k_{u2}Z_{r2} + c_{s2}(\dot{Z}_{s} - a\dot{\theta}_{s} + \frac{1}{2}t_{f}\dot{\phi}_{s}) + k_{s2}(Z_{s} - a\theta_{s} + \frac{1}{2}t_{f}\phi_{s}) - u_{2}$$
(5)

$$M_{u3}\ddot{Z}_{u3} = -c_{s3}\dot{Z}_{u3} - (k_{u3} + k_{s3})Z_{u3} + k_{u3}Z_{r3} + c_{s3}(\dot{Z}_s - a\dot{\theta}_s + \frac{1}{2}t_f\dot{\phi}_s) + k_{s3}(Z_s - a\theta_s + \frac{1}{2}t_f\phi_s) - u_3$$
(6)

$$M_{u4}\ddot{Z}_{u4} = -c_{s4}\dot{Z}_{u4} - (k_{u4} + k_{s4})Z_{u4} + k_{u4}Z_{r4} + c_{s4}(\dot{Z}_s - a\dot{\theta}_s + \frac{1}{2}t_f\dot{\phi}_s) + k_{s4}(Z_s - a\theta_s + \frac{1}{2}t_f\phi_s) - u_4$$
(7)

The system states are assigned as

- $x_1 = Z_{u1}$, vertical displacement of the left, front unsprung mass;
- $x_2 = Z_{u2}$, vertical displacement of the left, rear unsprung mass;
- $x_3 = Z_{u3}$, vertical displacement of the right, front unsprung mass;
- $x_4 = Z_{u4}$, vertical displacement of the right, rear unsprung mass;
- $x_5 = Z_s$, vertical displacement of the centre of gravity of sprung mass;
- $x_6 = \theta_s$, pitch of the sprung mass;
- $x_7 = \varphi_s$, roll of the sprung mass;

Introducing the following state, the control input, and disturbance input vectors:

$$x = [x_1, x_2, x_3, \cdots, x_{14}]^T$$
(8)

$$u = [u_1, u_2, u_3, u_4]^T$$
(9)

$$w = [Z_{r1}, Z_{r2}, Z_{r3}, Z_{r4}]^T$$
(10)

Defining x_8 to x_{14} as the derivatives of x_1 to x_7 , equations can be represented in the form of the state equations:

$$\dot{x} = Ax + Bu + Ew \tag{11}$$

where $A \in R^{14 \times 14}$ is the system matrix; $B \in R^{14 \times 14}$ is the input matrix and $E \in R^{14 \times 14}$ is the disturbance matrix.

Sprung mass (M _s)	2000kg
Unsprung mass (Mu1,Mu2)	150kg
Rear mass of the wheel (M _{u3} ,M _{u4})	150kg
Pitch of moment of inertia (I _p)	5800kg-m ²
Roll of moment of inertia (I _r)	850 kg-m ²
Stiffness of car body spring for front (k_{s1}, k_{s3})	20000N/m
Stiffness of car body spring for rear (k _{s2} , k _{s4})	17500N/m
Front width (t_f) , rear width (t_r)	2000mm
Font and rear tire stiffness $(k_{u1}-k_{u4})$	175000N/m
Front damping (c_{s1}, c_{s3})	7000N
Rear damping (c_{s2}, c_{s4})	4000N
Cockpit to front wheel (a)	1300mm
Cockpit to real wheel (b)	4000mm

Table 2.1-1 Car Parameters

The numerical data used for the pickup truck model simulation are given in Table 2.1-1.

For running stability and easy handling, the tyre deflection, which is proportional to the dynamic tire-road contact force, should be small. In a full car model, all four wheels are considered. Control forces are usually supplied by hydraulic actuators. It is better to use lower power consumption for active control, so the control forces are also considered in the performance index.

•

The design of a vehicle suspension involves a compromise among conflicting goals. Optimizing ride comfort, suspension rattle space and road holding, the performance index to be minimized can be written as follows:

$$J = \lim \frac{1}{2T} \int_{0}^{T} \begin{bmatrix} 1\\ \ddot{z}_{c}\\ \ddot{\theta}\\ \ddot{\theta}\\ \ddot{\theta} \end{bmatrix}^{T} \begin{bmatrix} 0 & 0 & 0 & 0\\ 0 & \rho_{1} & 0 & 0\\ 0 & 0 & \rho_{2} & 0\\ 0 & 0 & 0 & \rho_{3} \end{bmatrix} \begin{bmatrix} 1\\ \ddot{z}_{c}\\ \ddot{\theta}\\ \ddot{\theta}\\ \ddot{\theta} \end{bmatrix}$$

$$+ \begin{bmatrix} s_{1}\\ s_{2}\\ s_{3}\\ s_{4} \end{bmatrix}^{T} \begin{bmatrix} \rho_{4} & 0 & 0 & 0\\ 0 & \rho_{5} & 0 & 0\\ 0 & 0 & \rho_{6} & 0\\ 0 & 0 & 0 & \rho_{7} \end{bmatrix} \begin{bmatrix} s_{1}\\ s_{2}\\ s_{3}\\ s_{4} \end{bmatrix}$$

$$+ \begin{bmatrix} t_{1}\\ t_{2}\\ t_{3}\\ t_{4} \end{bmatrix}^{T} \begin{bmatrix} \rho_{8} & 0 & 0 & 0\\ 0 & \rho_{9} & 0 & 0\\ 0 & 0 & \rho_{10} & 0\\ 0 & 0 & \rho_{10} & 0\\ 0 & 0 & \rho_{11} \end{bmatrix} \begin{bmatrix} t_{1}\\ t_{2}\\ t_{3}\\ t_{4} \end{bmatrix}$$

$$+ \begin{bmatrix} u_{1}\\ u_{2}\\ u_{3}\\ u_{4} \end{bmatrix}^{T} \begin{bmatrix} \rho_{12} & 0 & 0 & 0\\ 0 & \rho_{14} & 0\\ 0 & 0 & \rho_{14} & 0\\ 0 & 0 & 0 & \rho_{15} \end{bmatrix} \begin{bmatrix} u_{1}\\ u_{2}\\ u_{3}\\ u_{4} \end{bmatrix}$$

$$(12)$$

where:

$$\ddot{z} = \ddot{x}_5 = \dot{x}_{12} \tag{13}$$

$$\ddot{\theta} = \ddot{x}_6 = \dot{x}_{13} \tag{14}$$

$$\ddot{\varphi} = \ddot{x}_7 = \dot{x}_{14} \tag{15}$$

And:

$$s_1 = x_5 - x_1 + \frac{1}{2}t_f x_7 - a x_6 \tag{16}$$

$$s_2 = x_5 - x_2 + \frac{1}{2}t_r x_7 + bx_6 \tag{17}$$

$$s_3 = x_5 - x_3 - \frac{1}{2}t_f x_7 - a x_6 \tag{18}$$

$$s_4 = x_5 - x_4 - \frac{1}{2}t_f x_7 + bx_6 \tag{19}$$

$$t_1 = x_1 - Z_{r1}$$
(20)

$$t_2 = x_2 - Z_{r2}$$
(21)

$$t_3 = x_3 - Z_{r3}$$
(22)

$$t_4 = x_4 - Z_{r4}$$
(23)

where $\rho_1 \sim \rho_{15}$ are the weighting constants; several sets of weights can be used depending on the conditions of motion, including velocity, road quality, vibration level acceleration, or other factors. In addition, with regard to unit parameters that are used in performance index (10), the values of weighting constants should be balanced. Two more generally used sets are road holding and ride comfort, which will be considered in this study. For expressing the performance index in a form that is quadratic in the state and input vectors, it is required to substitute the acceleration \ddot{z} , $\ddot{\theta}$ and $\ddot{\phi}$ (12). Using (11) and (13) ~ (15) to have:

$$J = \lim \frac{1}{2T} \int_0^T (x^T Q_1 x + 2x^T N_u + u^T R_u x + 2x^T Q_{12} \omega + \omega^T Q_2 \omega) dt$$
(24)

Where Q_1 , R and Q_2 are symmetric, time-invariant weighting matrices and R is also positive definite.

2.1.2 Full Vehicle Model Simulation Results

We used a 3.5cm \times 38cm \times 300cm bump to simulate a speed bump, as shown in Figure 2.1-2.



Figure 2.1-2 Bump road model

The vibrations of the vehicle when the vehicle passed the speed bump at 20km/h and 40km/h are shown in Figure 2.1-3. Although the result is contrary to the instinct that high speed usually leads to large vibration amplitude, the value of the stiffness of car body spring and sprung mass effect the suspension performance. Therefore, different vehicles may lead to different vibration amplitudes when passing the speed bump at a same speed. Usually passing a speed bump at a high speed lead to a good comfortable drive, however it may damage the suspension system.

The performance of the vehicle was affected by many factors including the sprung mass and the stiffness of spring. Compared with the results when the vehicle passed the speed bump at a high speed, the amplitude of vibration was smaller than at a slow speed. However, the results can differ when the parameters of the vehicle model are changed. This model enables testing and prediction of handling characteristics of the physical prototype including vehicle chassis roll, ride comfort, ride safety and suspension performance parameters. Road parameters such as surface conditions and driving environment, which vary widely for real vehicle test, also can be controlled conveniently for virtual vehicle test.



Figure 2.1-3 Vibrations generated by the vehicle model

2.2 Stewart Platform

Accurate control of the Stewart platform to simulate the vibration of a moving vehicle is required to achieve an effective HIL vehicle seat vibration test platform. There are two problems that must be addressed:

- 1. For the given position and orientation of the end effector, what are the lengths of the actuators?
- 2. For the given velocity, position and orientation of the end effector, what are the velocities of the actuators?

2.2.1 Inverse Kinematics

The performance of the motion platform designed for the simulation of vehicle seat suspension vibration is determined by the accuracy of the displacements of the electrical cylinders. The length of the actuator can be altered by extending or contracting movements to change the electrical cylinders.

The Stewart platform schematic view is shown in Figure 2.2-1. The inertial frame O is fixed at the centre of the base platform with a perpendicular O-axis. Another coordinate system P, which is a movable platform frame, is fixed on the center of the upper surface on the top of the platform with a P^{*}-axis perpendicular to the top surface platform.



Figure 2.2-1 Stewart platform schematic

The single leg schematic view is shown in Figure 2.2-2. The positon of the Stewart platform P can be described as:

$$\mathbf{P} = [\mathbf{x}, \mathbf{y}, \mathbf{z}, \boldsymbol{\alpha}, \boldsymbol{\beta}, \boldsymbol{\gamma}]^T$$
(25)

where $t = [x \ y \ z]^T$ is a vector of frame O to P, and α , β and γ are the Euler angles about the x, y and z axes. So the rotation matrix can be described as:

$$R$$

$$= \begin{pmatrix} \cos\beta\cos\gamma & \sin\alpha\sin\beta\sin\gamma - \cos\alpha\sin\gamma & \sin\alpha\sin\gamma + \cos\alpha\sin\beta\cos\gamma \\ \cos\beta\sin\gamma & \cos\alpha\cos\gamma + \sin\alpha\sin\beta\sin\gamma & \cos\alpha\sin\beta\sin\gamma - \sin\alpha\sin\gamma \\ -\sin\beta & \sin\alpha\cos\beta & \cos\alpha\cos\beta \end{pmatrix}$$
(26)

where, R is the rotation matrix.



Figure 2.2-2 Single leg schematic

First, the Coordinate System of the Stewart platform was built, as shown in Figure 2.2-1. P-X`Y`Z` designates the moving platform and O-XYZ is for the base platform. When the moving platform is at the centre position, the X`Y`Z` coordinate axes are in the same directions as XYZ. The coordinate point of the origin P of the coordinate system P-X`Y`Z` is (x_p, y_p, z_p) , so the homogeneous transformation matrix is

$$T = \begin{bmatrix} x_P \\ R & y_P \\ & z_P \\ 0 & 0 & 0 & 1 \end{bmatrix}$$
(27)

From the homogeneous transformation matrix, the coordinates of point B_i on the coordinate system, P-X`Y`Z`, can be changed to O-XYZ. If the coordinates of point B_i (i = 1... 6) are (X_{Bi}, Y_{Bi}, Z_{Bi}), then:

$$\begin{bmatrix} x_{Bi} \\ y_{Bi} \\ z_{Bi} \\ 1 \end{bmatrix} = T \begin{bmatrix} x \\ y \\ Bi \\ z \\ Bi \\ 1 \end{bmatrix}, i = 1, 2, 3, 4, 5, 6$$
(28)

The length of the hinge point between the moving platform and base platform are

$$l_{i} = A_{i}B_{i} = \sqrt{(x_{Ai} - x_{Bi})^{2} + (y_{Ai} - y_{Bi})^{2} + (z_{Ai} - z_{Bi})^{2}}, \qquad (29)$$
$$i = 1, 2, 3, 4, 5, 6$$

where l_i is the leg length that can allow the top platform to reach the expected pose q. The displacement that each electrical cylinder can extend is described as:

$$D_i = l_i - l_0, \qquad i = 1, 2, 3, 4, 5, 6$$
 (30)

where l_0 is the leg length when the electrical cylinders are in the original position.

2.2.2 Platform Movement Speed Analysis

It can be seen from Figure 2.2-2 that if the unit vector e_i is of the same direction as the l_i ; R_{bi} is the vector of the hinge point in the P-X`Y`Z`; and ω_p , V_p are the angular

velocity and linear velocity vector at the origin P of the movable platform in coordinate system O-XYZ.

Therefore, the velocity vector of point B_i is:

$$V_{Bi} = \omega_p \times R_{bi} + V_p = \begin{bmatrix} I & -\bar{R}_{bi} \end{bmatrix} \begin{bmatrix} V_p \\ \omega_p \end{bmatrix}$$
(31)

And the relative speed v_i of the leg is:

$$v_i = e_i \cdot V_{Bi} = e_i \cdot (\omega_p \times R_{bi} + V_p) \tag{32}$$

$$\begin{bmatrix} V_p \\ \omega_p \end{bmatrix}^T = \begin{bmatrix} V_{px} & V_{py} & V_{pz} & \omega_{px} & \omega_{py} & \omega_{pz} \end{bmatrix}^T$$
(33)

Therefore,

$$V = \begin{bmatrix} V_1 \\ V_2 \\ V_3 \\ V_4 \\ V_5 \\ V_6 \end{bmatrix} = \begin{bmatrix} e^{T_1} & (R_{b1} \times e_1)^T \\ e^{T_2} & (R_{b1} \times e_2)^T \\ e^{T_3} & (R_{b1} \times e_3)^T \\ e^{T_4} & (R_{b1} \times e_4)^T \\ e^{T_5} & (R_{b1} \times e_5)^T \\ e^{T_6} & (R_{b1} \times e_6)^T \end{bmatrix} \begin{bmatrix} V_p \\ \omega_p \end{bmatrix} = Jq$$
(34)

where J is the Jacobian matrix of the platform and q is velocity mapping from the P point to each leg.

2.3 Summary

From the simulation result it can be seen that the 7-DOF car model can be used to simulate the vibrations of the vehicle with bump road input. And according to the mathematical models of the Stewart platform, the simulation results of the car model can be used as inputs of the Stewart platform control. In the next chapter, a real Stewart platform will be built and controlled as a vehicle seat test platform. The data, which is from the 7-DOF car simulation, will be converted into the length of the six legs of the real Stewart platform.

3 EXPERIMENTAL DESIGN

According to the results of the chapter 2, this section outlines the experimental design. In the experiment, the hardware design, software design and control plan will be presented individually. Because the output vibration data of the vehicle model in last chapter is all about the absolute heave, pitch and roll velocities of the sprung mass, in order to simulate the vibration in all dimensions, a Stewart multiple-DOF motion platform will be designed to generate the required vibration.

3.1 Experimental Plan

Generally, laboratory road simulation applies to the reproduction of service loads on complete vehicles in the laboratory. However, regarding the vehicle seat vibration isolation study, it only focuses on the vibration of the vehicle chassis, and hence other parts of the vehicle can be excluded. The vehicle vibration simulation platform consists two parts. The structure of the system is shown in Figure 3.1-1.

Firstly, a linear 7-DOF full-car model will be built to study the response of the vehicle with on-road and off-road inputs. The response is analysed by treating them as input of the hardware part. And the response data will be used to calculate the length of each actuator of the Stewart platform via LabVIEW. Finally, a Stewart platform was designed to connect to the system and simulate the vibrations of the virtual vehicle.

The output vibrations data of the vehicle model are related to the pitch, roll and the displacements of sprung mass. In order to simulate the vibrations in all dimensions and to gain a better control of the platform, a CompactRIO will be used to connect the Stewart platform and the LabVIEW.

A well-defined Stewart platform in combination with a mathematical model of the vehicle-road is used to replace the conventional vehicle test system. Different vehicle-road mathematical model combinations can be easily simulated with a system by changing the parameters of models. As a result, the mathematic simulation model can be transformed into a real platform movement. The whole vibration simulator platform becomes a hardware-in-the-loop (HIL) system.



Figure 3.1-1 Hardware-in-the-loop System Structure

3.2 Hardware Design and Control Plan

The hardware devices consist of the Stewart platform and the CompactRIO control system. And in order to reduce the cost, only the cRIO-9076 modules are used.

3.2.1 The Stewart Platform

The 6-DOF Stewart platform consists of a fixed base platform, a movable top platform and six linear actuators, which is shown in Figure 3.2-1. Square steel tubes are used to respectively fabricate the top and base platforms which are regular hexagons with the lengths of sides 350mm and 500mm. Six electrical cylinders are applied as actuators of motion platform, and they are connected the top and base platform with spherical joints.



Figure 3.2-1 Stewart Platform Design

In general, Stewart platform can be driven by actuators and electrical cylinders or hydraulic cylinders are the frequently used actuators. The actuator dynamics may be neglected in many cases because of the fast time constants of electric cylinders. However comparing with the electrical cylinders, hydraulic cylinders cannot accurately apply torques over a great dynamic range. The dynamics of hydraulic system are highly nonlinear because of the fluid compressibility or the characteristics of nonlinear servo valve flow pressure.

Figure 3.2-2 shows the real Stewart platform. The six electric cylinders (actuators) of the platform are respectively driven by six AC servo drivers.



Figure 3.2-2 Stewart platform hardware structure

The electric cylinder consists of two parts: the telescopic actuator and AC servo motor. The parameters of the servo motor are shown in Table 3.2-1. Because the servo motor uses the rotating dynamics to drive the telescopic actuator to move, and by changing the length of the actuator, the servo motor operation must be operated by dedicated servo drive. Therefore, six servo motor drives respectively drive the six servo motors to operate. The servo motor drivers are shown in Figure 3.2-3. And the parameters of the motor driver are shown in Table 3.2-2.

Table 3.2-1 Servo Motor Parameters

Model	80MB-R7530A23F-0MF2
Input	4.4A
Rated Output:	0.75KW
Rated Rev	3000rpm



Figure 3.2-3 Servo motor drivers

The position control mode is used to control the servo motor in order to ensure the motion accuracy of the platform. The position control is based on the positional command (pulse train) from the host controller (cRIO-9076).

There are two input signals that must be set before sending the control commands to a servo driver, which is shown in Figure 3.2-4. The input 1 is pulse train, the frequency of the pulse determines the rotational speed of the motor. And the input 2 is directional control signal.



Figure 3.2-4 Pulse train and sign mode

For example, in the case if sending 2000 pulse signals to the servo motor drive, the servo motor can be rotated 360°, and the actuator can elongate or shorten 4mm, and rotate. The rotational speed of motors depends on the frequency of the pulse signal.

In order to ensure the reliability of the signal transmission, the input signals are all differential signal. Therefore <u>AM26C32</u> ICs are used to convert the TTL signals into differential signals, which is shown in Figure 3.2-5.



Figure 3.2-5 Convert TTL signal to differential signal

Table 3.2-2	AC Motor	Driver	Parameters
-------------	----------	--------	------------

Model	DMKE LS-022-H33B-M-B
Source	3PH AC380V 50/60HZ
Output	22KW 45A 0-9000rpm

In order to well control the platform, each servo motor has an encoder to feedback the movement position of current electric cylinder. The feedback is a differential signal, through the differential signal to TTL signal converter module, which is shown in Figure 3.2-6, to convert the signals into TTL signals, and then the signals can be received by the host controller.



Figure 3.2-6 Differential signal to TTL signal converters

3.2.2 National Instruments (NI) CompactRIO (cRIO) Control System

National Instruments CompactRIO is compatible with LabVIEW graphical programming language. LabVIEW is a user friendly software package for rapid and convenient development of complex measurement and control systems. Additionally, CompactRIO modules are less expensive with a wide variety of features. In this case, the model used in the CompactRIO is cRIO-9076, which is shown in Figure 3.2-7.



Figure 3.2-7 NI CompactRIO control system

The cRIO-9076 combines a real-time processor and a reconfigurable FPGA for embedded system control and measuring applications. It integrates a 400 MHz industrial real-time processor with an LX45 FPGA and has four slots for I/O modules [51].

There are four NI 9401 modules used as input/output interface ports. The NI 9401 is an 8-channel 100ns bidirectional digital module. And with reconfigurable I/O technology, it is easily to be used via the LabVIEW FPGA for I/O control, pulse generation. Each channel is compatible with 5V/TTL signals and features 1000 Vrms transient isolation between the I/O channels and the backplane [51].

3.3 Software Design and Control System Design

In order to control the Stewart platform to simulate the vibrations of the virtual vehicle, there are 3 steps should be considered:

- 1. Send the vibrations data to the CompactRIO host controller.
- 2. Convert the vibrations data to the lengths of the actuators and the movement speed in the CompactRIO FPGA.
- Convert the length and motion speed data to the control signals to the servo drivers to control the actuators via IO port of the NI-9401 module.

After that, servo motors will drive the actuator to move, and the platform can generate the same vibrations according to the computer model.

3.3.1 LabVIEW VI Design

Most controllers are serial systems; they only sent one control command at a time. When the first control command is executed, the next control command task can be processed. However, for controlling a parallel mechanism, like Stewart platform, serial system is unsuitable. Using FPGA can resolve this problem, because the FPGA can send all of the control commands at the same time.

The CompactRIO controller includes a processor and reconfigurable FPGA. With the user-programmable FPGA, it can implement the hardware high-speed control, and data processing. The six linear actuators of the platform must be controlled at the same time. Moreover, there are six high resolution encoders sending the signal back to the CompactRIO. Thus, the controller and data processing are all programmed in FPGA module, which is shown in Figure 3.3-1.



Figure 3.3-1 LabVIEW VI structure

The real-time layer is used to build the human-computer interface and monitor operational status.

3.4 Summery

This section presented the details of the HIL driver seat vibration test platform, including the real Stewart platform design, the system structure and the real Stewart platform controlled via the LabVIEW FPGA. In the next section, the HIL driver seat test platform will be used to simulate the vehicle seat vibrations with on-road and off-road inputs, and then the output data of the system will be analysed.

4 THE EXPERIMENTAL RESULTS AND DISCUSSION

In this section, the HIL system with varying road inputs will be used to simulate the real vehicle vibration. The simulation results and the real Stewart platform results will be compared and analysed. Moreover, the performance of the platform will be discussed.

4.1 Simulation Results

When a sine sweep road signal, which is shown in Figure 4.1-1, was set as the input of the virtual prototyping vehicle model, and set the speed of the vehicle model to 20 km/h, the vehicle chassis mostly produces 2-DOF vibration, pitch and vertical acceleration. It can be seen from the Figure 4.1-2 that the value of the pitch acceleration decreased to the resonance point and then increased gradually.



Figure 4.1-1 Sine sweep road model



Figure 4.1-2 Pitch acceleration generated by the vehicle model

And the value of the vertical acceleration was increasing all the times, which is shown in Figure 4.1-3. These vibrations were just the simulation results on the computer.



Figure 4.1-3 Vertical acceleration generated by the vehicle model

The Stewart platform generated the virbrations which can be measured by a 6-DOF IMU sensor. Figure 4.1-4 and Figure 4.1-5 shown the real vibrations are similar to the simulation result through the camparison. Although the sensor generated a lot of clutter signal on vertical acceleration, because of the accelerometer noise, it still can be seen that the curve is similar to the simulation reasult.



Figure 4.1-4 Pitch acceleration generated by the Stewart platform



Figure 4.1-5 Vertical acceleration generated by the Stewart platform

When the car model is running on the off-road at the same speed, the simulation result can be seen from the Figure 4.1-6 and the Figure 4.1-7.



Figure 4.1-6 Pitch acceleration generated by the vehicle model



Figure 4.1-7 Vertical acceleration generated by the vehicle model

The real vibrations which generated by the Stewart platform is shown in Figure 4.1-8 and Figure 4.1-9. With comparison of the simulation results, the actual data match the simulation result.



Figure 4.1-8 Pitch acceleration generated by the Stewart platform



Figure 4.1-9 Vertical acceleration generated by the Stewart platform

It means the Stewart platform can generate the corresponding vibrations according to the input data (vehicle model simulation result). In other words, whether the Stewart platform can accurately simulate the vibrations of the vehicles, it not only depends on the quality of the control system, but also depends on the accuracy of the vehicle model simulation.

4.2 Loading Platform Load Reactions and Load Capacities

Whether the platform is able to meet the need of seat testing to generate the required vibrations in the situation of heavy load is an important problem.



Figure 4.2-1 In the process of seat testing

In the process of seat testing, which is shown in Figure 4.2-1, the weight of the seats and human were within 100 kg. After with 100 kg loading and given 1 Hz, 7 mm

amplitude of sin signal as input, the vibration (encoder output) of the platform and input signal were overlapping, which is shown in Figure 4.2-2.



Figure 4.2-2 Results of the loading platform comparing

When the input signal changed into 5 HZ, 2mm sin signal, and the results can be seen in Figure 4.2-3, the curves were still overlapping.



Figure 4.2-3 Results of the loading platform comparing with 5 HZ and 2 mm sine signal input

However, increase the values of the sine input signal to the 7 Hz 3 mm, it can be seen from the Figure 4.2-4, the two curves were not overlapping. And the vibration generated by the platform was lag behind the input signal.



Figure 4.2-4 Results of the loading platform comparing with 7 HZ and 3 mm sin signal input

The weight of the loading has a great influence on the performance of the platform. That is because the electric cylinders of the platform had exceeded the rated power.



Figure 4.2-5 Comparison of the input signal and output result in frequency domain

Although the result lagged behand the input signal in the frequency domain, which is shown in Figure 4.2-5, the frequencies were the same. However, the amplitude of the vibration generated by the platform was slightly lower than the input signal.

5 CONCLUSIONS AND RECOMMENDATIONS

This thesis has aimed to build a vehicle seat vibration test platform, and the purpose was to test the seat vibration damping performance in the seat vibration research as well as to reduce the development costs and achieve better research data. The experiment and results proved the platform can generate the required vibrations, and the vibrations were in accordance with the vehicle mathematical model simulation results.

The cRIO host controller can well control the movement of the Stewart platform. Moreover, the complex code programming was replaced by clear and understandable graphical programming via the LabVIEW. The six electric cylinders can be controlled at the same time in the FPGA layer, therefore accuracy of the parallel mechanism can be guaranteed.

The quality of building a vehicle model is one of the key factors to accurately simulate the vibrations of vehicle in the platform. Although in this thesis, a simple 7-DOF vehicle mathematical model is built for simulation, from the obtained results it can be seen, vibrations generated by the platform were closer to the vibration generated by the vehicle simulation. And higher accuracy of vehicle model, like 11-DOF or more, also can make the results of the platform closer to the real vehicle. Moreover, via changing the parameters of the vehicle mathematical model, the system can be used for testing the seat damping performance on different vehicles. Therefore, it can greatly reduce the difficulty of seat testing. Furthermore, the platform has six degrees of freedom, so there is a lot of room for further improvements to meet the requirement of vehicle seat test.

The load weight of the test platform design is 90 kg, and in the case of 90 kg load the running accuracy of the platform can be guaranteed. If an excessive load was applied, the frequency of the vibrations was unchanged, however, the amplitude was reduced. Because the performance of the platform is restricted by the servo motor, high power servo motors can provide a better platform motion performance. If the seat test requires the load of the platform larger than 90kg, bigger power servo motors must be used.

With these milestones completed, this work contributes to provide an ideal test environment for the vehicle seat test:

- Combine the computer vehicle vibration mathematical model and the Stewart platform together to control the platform to simulate the vibrations in a close approximation of real vehicle movement dynamics as well as improved the vehicle seat testing environment.
- The platform expansibility can adapt to the requirement of the seat test and at the same time to reduce the time and cost of the vehicle seat testing.

With the goals achieved, and the results obtained, several avenues exist for future work.

- Improve the accuracy of the vehicle mathematical model, so that the platform can generate vibrations more closely to the actual vehicle vibrations in different degrees of freedom.
- On the real-time layer, the human-machine interface will continue to be improved, for example achieve real-time switching to a different vibration.
- Achieve the human-in-the loop test by improving the safety of the platform. More over through monitoring the temperature of the hardware devices, therefore the safety of the platform can be increased in continuous heavy load operation.

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APPENDIX A: THE LABVIEW VI PROGRAM

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Structure of project:



InverseKinematics.vi:



GenerateWaveForm.vi:



FPGA_M_Mode.vi:



FPGA_Leg_Driver.vi:







FPGA_Pluses_Timer.vi:



RealTime_Main.vi:



RealTime_Main.vi (front panel):



Fast Fourier transform Matlab code:

Fs = 128;	% sampling frequency
T = 1/Fs;	% sampling time
L = 256;	% Signal length
t = (0:L-1)*T;	% Time
$N = 2^{nextpow2(L)};$	% Sampling points
f = Fs/N*(0:1:N-1);	% Frequency

Yi = fft(input,N)/N*2; Ai = abs(Yi); Yo = fft(output,N)/N*2; Ao = abs(Yo);

figure; subplot(211); plot(t,input);hold on; plot(t,output); title('Input and Output Signal') xlabel('Time(s)') ylabel('Amount of pulse')

```
subplot(212);
plot(f(1:N/2),Ai(1:N/2)); hold on;
plot(f(1:N/2),Ao(1:N/2));
title('FFT Spectrum')
xlabel('Frequency(Hz)')
ylabel('Amplitude ')
```