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VIBRATION ISOLATION ANALYSIS OF A STABILIZED PLATFORM MOUNTED ON A SMALL OFF-ROAD VEHICLE

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Abstract: Stabilised platforms are regularly integrated with vehicles in various applications such as terrain mapping and surveillance. The equipment installed on the platform is often sensitive to motion and has to be isolated from unnecessary vibrations. In this paper the implementation of different platform suspension systems to improve the orientation and motion of the platform in the pitch degrees of freedom (DOF) is investigated. A one DOF platform model is merged with a validated, nonlinear, 12 DOF simulation model of a small off-road vehicle known as a Baja. The effectiveness of passive, semi-active and active suspension systems are investigated when the vehicle model is excited by a sinusoidal road profile input. Skyhook control is used to vary the damping in the semi-active system, and a PID controller is implemented in the active system. It is concluded that a passive suspension is ineffective due to conflicting spring stiffness and damping settings required for reduced pitch motion and level platform orientation. The semi-active system can improve the pitch orientation and motion as compared to the passive suspension without conflicting suspension or controller gain settings, but the best solution is obtained using an active suspension system.

Keywords: semi-active, active, platform stability control, vehicle model, skyhook, PID.

INTRODUCTION

Specialised small multi-terrain vehicles are often deployed when improved mobility is required. Depending on the application, these vehicles may be equipped with stabilized platforms in order to perform various tasks, including surveillance and terrain mapping. For optimal equipment functioning the vehicle mounted platform has to be isolated from road excitations transferred through the vehicle body, or the orientation has to be controlled. For example, equipment for continuous surveillance needs to be stabilized in order to maintain the field of view located on the observed object. Also, for terrain mapping and other sensing applications the movement that the sensors are subjected to must be limited to acceptable levels.

In this paper, the optimal vehicle mount suspension system that will minimize the pitch motion and incline of the platform, is investigated. A mass-spring-damper model of a vehicle mounted platform is developed. The platform consists of a rotational system with given mass and inertia and the pitch degree of freedom (DOF) is coupled to the vehicle body with a suspension system. Three suspension types are considered, namely passive, semi-active, and active suspensions. The passive suspension consists of a linear spring and damper of which the optimal spring stiffness and damping coefficient has to be determined. The damper in the semi-active suspension system is an adjustable magnetorheological (MR) damper, or the spring and damper can be replaced by a fully active system containing an actuator.

The platform model is merged with a 12 DOF simulation model of a small single seated off-road vehicle referred to as a Baja vehicle. The vehicle is 1.91m long and 1.48m high, with a track width of 1.35m and a wheel base of 1.55m. This vehicle simulation model has been developed using ADAMS (Automatic Dynamic Analysis of Mechanical Systems) software and has been adequately validated for the vertical and pitch DOF.

To determine the optimal passive suspension characteristics, and semi-active and active suspension control parameters, a sinusoidal displacement road input of increasing frequency is used. The road profile is defined in the simulation model.

VEHICLE MODELLING

Simulation Model

The vehicle simulation model is developed using ADAMS View software. It is a 13 DOF model consisting of 21 moving bodies connected by various joint types. The base of the stabilised platform is fixed to the front right upright bar of the frame, and the platform is only allowed to move relative to the vehicle frame in the pitch DOF by means of a revolute joint. A schematic layout of the vehicle model is shown in Figure 1, and a graphic representation of the vehicle with the added platform is shown in Figure 2.

The geometry of the vehicle and the properties of the unsprung mass components (mass and inertia) are obtained from a detailed CAD model. The unsprung mass consists of the hub-reduction gear boxes, brake disks and callipers, wheel hubs, double wishbone arms and steering rods.

Since the mass of the driver and testing equipment used for model validation adds to a significant mass as compared to the mass of the vehicle sprung mass, their effects on the dynamics of the vehicle should be taken into account in the development of the simulation model. Therefore the driver, stabilised platform base, and testing equipment are modelled as part of the sprung mass. The location of the centre of mass and the moments of inertia of the sprung mass about its three axes (roll, pitch and yaw) were determined experimentally, as described by Uys et al. [1]. The results are summarized in Table 1.

The mass and inertia properties of the stabilised platform depend on its intended application. For this case study the application is stabilizing of surveillance equipment with geometry, mass and inertia properties obtained from a CAD model.

The suspension of the vehicle consists of a double wishbone suspension system with hydro-pneumatic spring-damper units positioned between the wishbone arms and the vehicle body. Due to the nonlinearity of the spring-damper units the suspension forces are obtained using splines rather than defining spring stiffness and damping coefficients. Separate spring and damper characteristics are obtained experimentally by imposing displacement inputs to the spring-damper units and measuring the corresponding force. The spring-damper characteristics are shown in Figure 3. The effects of the rebound stop of the spring and friction in the damper are also included in the characteristics.

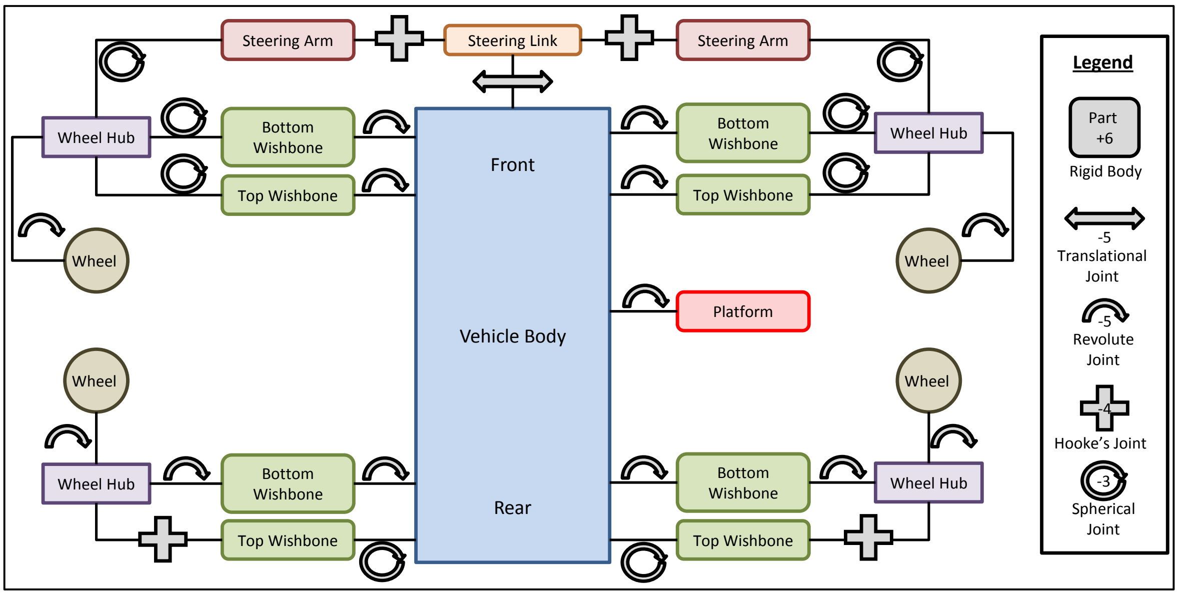

Figure 1 Schematic layout of vehicle model.

Table 1 Baja vehicle sprung mass properties.

Property	*Value*
Mass (including driver)	251.71kg
Centre of gravity position measured from vehicle body nose	1.195m
Centre of gravity position measured from vehicle body floor	0.3851m
Pitch moment of inertia	90.6kgm^2
Roll moment of inertia	53.7kgm^2
Yaw moment of inertia	69.2kgm^2

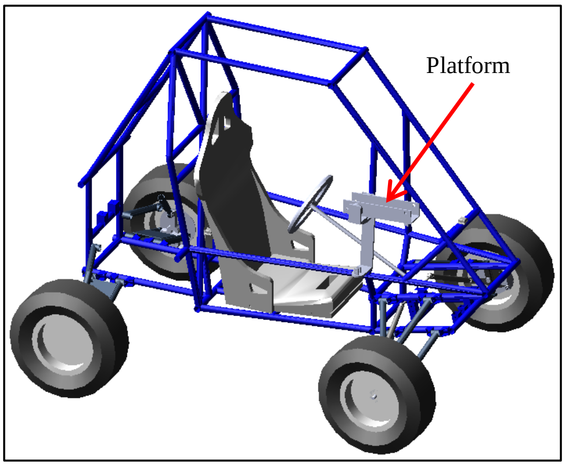

Figure 2 Graphic representation of vehicle model.

The tyre characteristics are determined experimentally and the measured tyre data is used to develop a Pacjeka ’89 tyre model [2]. The simulation model is driven by a prescribed rotational speed input at the rear wheels, while the steering input remains fixed to steer straight. A sinusoidal displacement road input of increasing frequency (up to 20Hz) with a constant amplitude of 0.04m is used. The vehicle is travelling at 16.2km/h, and the duration of road profile excitation is 30s.

Experimental Work

The vehicle test used to validate the simulation model in the vertical and pitch DOF is the discrete bump test. In this test the vehicle accelerates from a stationary position at a defined distance from a bump of known geometry, and comes to a halt after it passed over the bump. The test is performed at various bump entry speeds. During vehicle testing 22 channels are measured using an electronic data acquisition system (eDAQ), and data is sampled at 100Hz. String displacement sensors are used to measure suspension deflection and the steering rack displacement. Accelerometers mounted on the vehicle body are used measure the sprung mass acceleration in the vertical, lateral and longitudinal directions. The pitch, roll and yaw rates are measured using a gyroscope, and the rotational speed of one of the rear driving shafts is measured using an optical sensor.

Model Validation Results

To validate the vehicle model the outputs of the computer simulation model are compared to measured vehicle test results, as recommended by Heydinger et al. [3]. The measured vehicle bump test data are compared to simulation results in Figure 4. The pitch rate (Figure 4, top) simulation results correspond well

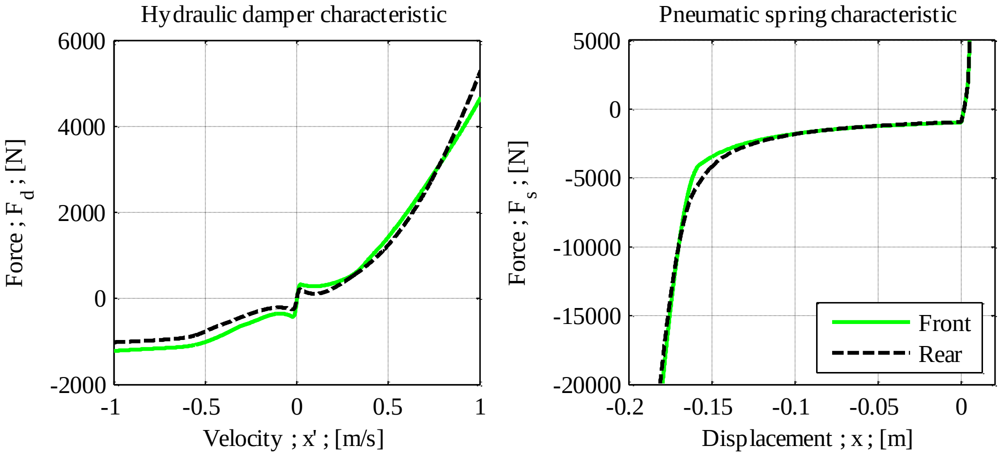

Figure 3 Experimentally determined spring and damper characteristics.

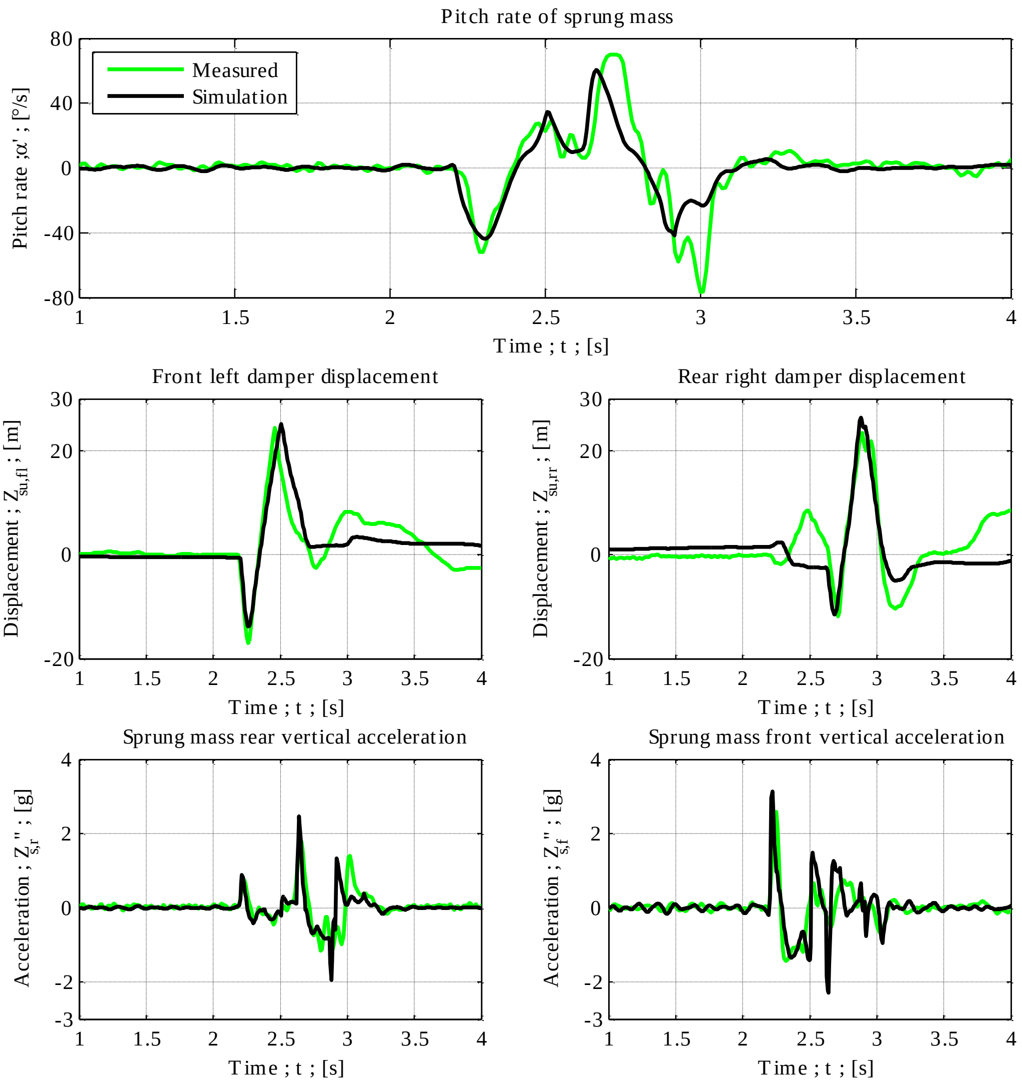

Figure 4 Vehicle model validation results (bump test).

to the measured data. The correlation is especially good when the vehicle nose lifts (indicated by a negative pitch rate) when the bump is hit by the front tyres at 2.2s. The simulation results of the damper displacements (Figure 4, middle) correspond well to measured results, especially at the compression (negative) and extension (positive) peaks. However, there is some discrepancy at the rear right damper where measured results show an extension of the damper at 2.5s not present in simulation results. After 3.5s the left damper compresses while the right damper extends, indicating the vehicle turning to the right. This turn is not visible in the simulation results as the steering input is fixed. The sprung mass vertical acceleration simulation results (Figure 4, bottom) correspond well to the measured data, with accurate peak acceleration predictions when the bump is hit at the front (bottom right figure, at 2.2s) or the rear (bottom left figure, at 2.6s). Positive quantities indicate upward acceleration. It is concluded that the vehicle simulation model is successfully validated in the vertical and pitch DOF.

PLATFORM STABILITY CONTROL

Three methods of platform stabilisation are investigated, namely passive, semi-active and active suspension systems. The classification of suspension systems are discussed by Housner et al. [4], Fischer and Isermann [5], and Savaresi et al. [6]. The optimal passive, semi-active and active stabilising system characteristics are determined in order to minimise the pitch rate of the platform and to keep it in a level position.

ADAMS/Simulink co-simulation is used to perform platform stability control. Simulink is a commercial software package developed by MathWorks and is used to simulate and analyse dynamic systems. The suspension characteristics and controller settings are defined in Simulink, and the dynamics of the vehicle model is solved by ADAMS. ADAMS exports several quantities, such as the platform pitch angle, to Simulink, which then use the model outputs to determine the suspension forces according to the suspension type and/or controller implemented. The suspension forces are then imported to ADAMS and used to solve the dynamic equations of the vehicle model.

Passive Suspension

The traditional passive suspension system consists of springs and hydraulic dampers of which the characteristics remain fixed during operation. Passive suspensions merely impart forces in response to motion, dissipating the vibratory energy of the platform, and cannot adapt to varying operating conditions. The ideal spring stiffness (k) and damping coefficient (c) for optimal platform stability are determined.

Semi-Active Suspension

The characteristics of semi-active systems can be adjusted with time, but as with passive devices, semi-active devices cannot supply energy to the controlled system, they can only dissipate energy. The characteristics of semi-active devices can be changed in fixed steps, or continuously, depending on the technology implemented. Semi-active systems have become increasingly popular due to their simplicity, adaptability, ease of implementation, low cost, and low power requirements [7]. Furthermore, semi-active systems are inherently stable and considered to be fail-safe as they merely revert to a passive system in the event of control failure [8].

In this work the platform is stabilised using a continuously adjustable magnetorheological (MR) damper and a fixed spring. A MR damper is very similar to a conventional viscous damper, the main differences being the MR fluid and an electromagnet embedded inside the damper piston which delivers a magnetic field in the orifices. MR fluids consist of micron-sized iron particles suspended in a carrier fluid such as silicone oil. When a magnetic field is applied to the fluid, the particles become arranged in chains, causing the free-flowing liquid to undergo a change in viscosity [4, 6]. Thus by controlling the electromagnet current of the MR damper, continuously variable damping can be produced.

The damping delivered by the MR damper is governed by skyhook control. Skyhook control was originally developed by Crosby and Karnopp [9] and has been implemented in several studies to reduce the vertical motion of the vehicle sprung mass [10, 11]. The principle of skyhook control is to connect the suspended object to a hypothetical reference in the sky using a damper. This principle requires a fictional reference in the sky and is realised in practice by placing an adjustable damper between the base and the suspended object. The adjustable damper is then controlled as to reproduce the forces that would be delivered by the skyhook damper. Since the adjustable damper can only deliver forces in response to motion, the damping is set to a minimum when active forces are required.

$$\dot{Z}_p \dot{X}_d > 0: \quad F_{SA} = G\dot{Z}_p \tag{1}$$

$$\dot{Z}_p \dot{X}_d \leq 0: \quad F_{SA} = 0 \tag{2}$$

In equations (1) and (2) $\dot{Z}_p$ is the vertical velocity of the platform, $\dot{X}_d$ is the relative velocity of the MR damper, F_{SA} is the prescribed semi-active damping force, and *G* is the control gain.

Active Suspension

In contrast to passive and semi-active systems, active systems provide additional force inputs by active devices, such as hydraulic actuators, and are therefore used to aid and resist motion. Full pitch control, and ideal skyhook control, can be performed by active systems. Although active suspensions can be far more effective than semi-active and passive suspensions, there are many obstacles associated with active systems that obstruct commercialisation, including high cost, system complexity, substantial power requirements, and potentially dangerous failure modes [5, 8].

The controller implemented for active platform stability control is a PID controller [12]. The pitch angle of the platform is used as the error signal as it is desired that the platform remain level throughout operation. The optimal proportional (K_P), integral (K_I), and differential (K_D) term coefficients are determined.

RESULTS

Baseline Results

For the baseline setup the platform is connected to the vehicle body using a fixed joint instead of a revolute joint (with reference to Figure 1). The platform and the vehicle body thus form a single rigid body, and the results reflect the vibration transmission to an unsuspended platform. The Root Mean Square (RMS) pitch angle (α) and pitch rate ($\dot{\alpha}$) of the platform are used as measures to determine the effectiveness of the implemented platform stabilisation method. The pitch angle indicates the orientation accuracy of the equipment on the platform, and can for instance be used to determine how accurately surveillance equipment is located on the target. The pitch rate indicates the degree of relative movement of the platform allowed, where for terrain mapping applications a still platform is required. The platform baseline pitch angle and pitch rate results are:

RMS Pitch Angle: $\alpha_{RMS} = 4.94°$
RMS Pitch Rate: $\dot{\alpha}_{RMS} = 11.88\ °/s$

In Figures 5 to 10 the baseline results are indicated by a flat surface outlined in red.

Passive Suspension Results

The RMS pitch angle and RMS pitch rate of the stabilised platform as the damping and stiffness of the passive suspension varies are shown in Figures 5 and 6. The spring stiffness is increased from 40N/m to 1600N/m, while the damping coefficient is increased from 5Ns/m to 800Ns/m. The minimum values are selected in order to avoid platform orientations reaching beyond ±90° (which is the vertical position).

In Figure 5 it is shown that as the damping coefficient and spring stiffness increase from $c = 200Ns/m$ and $k = 400N/m$, the improvement in pitch angle is as little as 0.5° up to the optimal point at $c = 800Ns/m$ and $k = 1600N/m$ (2.749°). As the stiffness decreases the pitch angle increases. Simulation results showed that as the vehicle drives over the sine sweep road profile the vehicle body starts to lean forward. This is due to the asymmetric damping characteristic, as shown in Figure 3, indicating that the vehicle spring-damper units are easier compressed than extended, resulting in the vehicle lowering on its suspension. As a result a zero RMS pitch angle is difficult to obtain using a passive suspension.

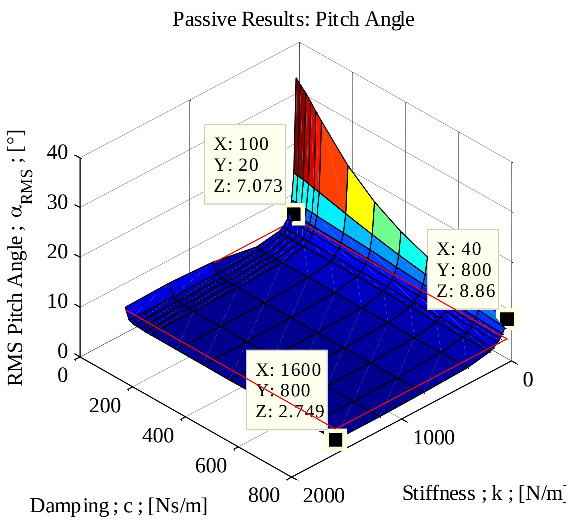

Figure 5 Passive results: RMS pitch angle of stabilised platform.

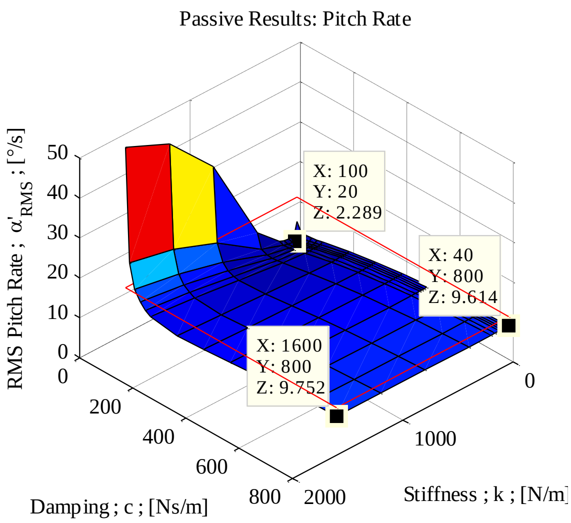

Figure 6 Passive results: RMS pitch rate of stabilised platform.

While the best pitch angle results are obtained for high stiffness and damping values, for the best pitch rate results much lower stiffness and damping are required. In Figure 6 it is indicated that the optimal suspension characteristics for reduced pitch rates are a spring stiffness of 100N/m and a damping coefficient of 20Ns/m (2.289°/s). As the damping is decreased below the optimal point the RMS pitch rate increases significantly. Also, at high damping levels (800Ns/m) the spring stiffness has little effect on the pitch rate of the platform. Reducing the stiffness from 1600N/m to 40N/m reduces the RMS pitch rate by merely 0.14°/s.

It is concluded that the optimal passive suspension system is capable of yielding a 44% improvement in RMS pitch angle and a 81% improvement in RMS pitch rate as compared to the baseline suspension system. The suspension characteristics for optimal pitch angle and pitch rate are however in conflict, and will result in a trade-off in platform orientation and platform motion.

Semi-Active Suspension Results

The RMS pitch angle and pitch rate of the stabilised platform as the skyhook control gain and stiffness of the semi-active suspension varies are shown in Figures 7 and 8. The spring stiffness is increased from 50N/m to 1600N/m, while the skyhook gain is increased from 5Ns/m to 2000Ns/m. It should be noted that the damping forces resulting from the control gain are ideal forces and may not always be deliverable. For instance, required zero damping forces (refer to equation 2) can't be delivered as MR dampers (and other semi-active devices) deliver some degree of passive damping in its minimum damping level state. There also exists a maximum deliverable damping force under specific input conditions such as the piston velocity and damper current input.

From Figure 7 it is seen that the semi-active suspension yield similar pitch angle results to the passive suspension (with reference to Figure 5). As the stiffness decreases the pitch angle increases, and at high stiffness levels the control gain has little effect on the RMS pitch angle. The best pitch angle results (2.71°) are obtained at high stiffness and control gain settings, with very little improvement when the stiffness is increased from $k = 400N/m$ and the control gain is increased from $G = 200Ns/m$.

As shown in Figure 8 the pitch rate reduces with increasing control gains, as opposed to the passive suspension results shown in Figure 6. Control gains above 400Ns/m can improve the pitch rate by up to only 1°/s. The lowest pitch rate (1.075°/s) is obtained for high control gains and low stiffness levels, as opposed to a high stiffness required for improved pitch angles. The pitch rate improvement obtained when reducing the stiffness is however negligible and the optimal solution may therefore contain a stiff spring.

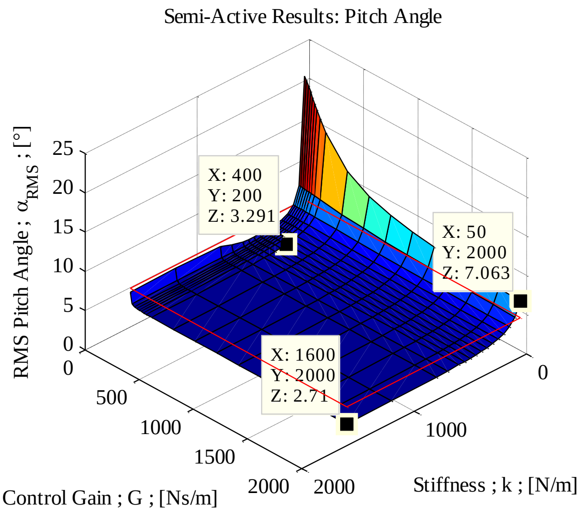

Figure 7 Semi-active results: RMS pitch angle of stabilised platform.

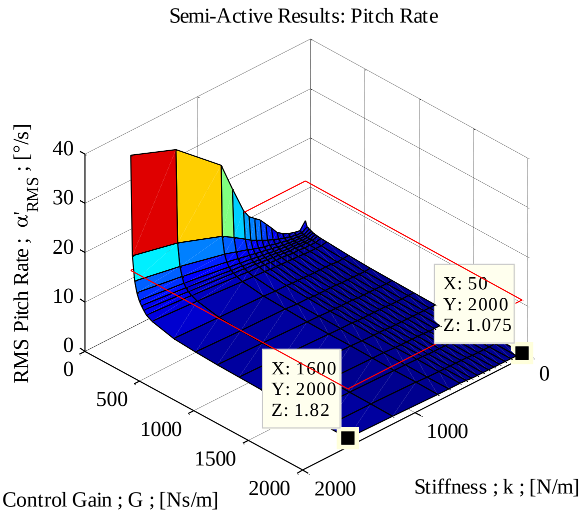

Figure 8 Semi-active results: RMS pitch rate of stabilised platform.

For optimal RMS pitch angle and pitch rates a spring stiffness of 1600N/m and a skyhook gain of 2000Ns/m are required. The improvement in platform orientation as compared to the passive suspension systems is marginal, only 1.4%. The RMS pitch rate obtained using the recommended suspension settings is 1.82°/s, an improvement of 20% as compared to the passive suspension system, and 84% as compared to the baseline results.

Active Suspension Results

The RMS pitch angle and pitch rate of the active stabilised platform are shown in Figures 9 and 10. The proportional and differential (K_P and K_D) coefficients are increased from 50 to 1000, and the integral coefficient (K_I) remains constant at 400. In Figures 9 and 10 it can be seen that the implementation of a PID controlled active suspension yields a large improvement in pitch angle and pitch rate over the passive suspension (indicated by the surface formed by the red lines).

In Figure 9 it is shown that as the proportional and differential coefficients are increased, the RMS pitch angle is also reduced. The best RMS pitch angle result (0.02°) is obtained for the highest proportional and differential coefficients, provided that the actuator implemented in the active suspension is capable of delivering the required forces under specific input conditions. The active suspension yields a significant improvement in RMS pitch angles as compared to the semi-active and passive suspensions (approximately 2.7°, as shown in Figures 5 and 7).

As with the pitch angle results, the best pitch rate results are obtained using the highest proportional and differential coefficients, as shown in Figure 10. As the differential coefficient is increased the RMS pitch rate is decreased to 0.225°/s. Increasing the proportional coefficient however has a very small effect on the pitch rate.

It is concluded that an active system has the potential to significantly improve both the RMS pitch angle and RMS pitch rate of the stabilised platform without resulting in trade-off situation. The RMS pitch angle improvement obtained is 99.6%, 99.2%, and 99.2% as compared to the baseline, passive and semi-active suspension systems, respectively, where the RMS pitch rate improvement is 98.1%, 90.1%, and 87.6%.

CONCLUSION

In this paper the development and validation of a 13 DOF simulation model of a Baja vehicle with a stabilised platform is presented. The vehicle model is nonlinear and contains experimentally determined parameters, such as centre of mass location and moments of inertia. The tyre model and characteristics of the hydro-pneumatic spring-damper units have also been developed using experimental data.

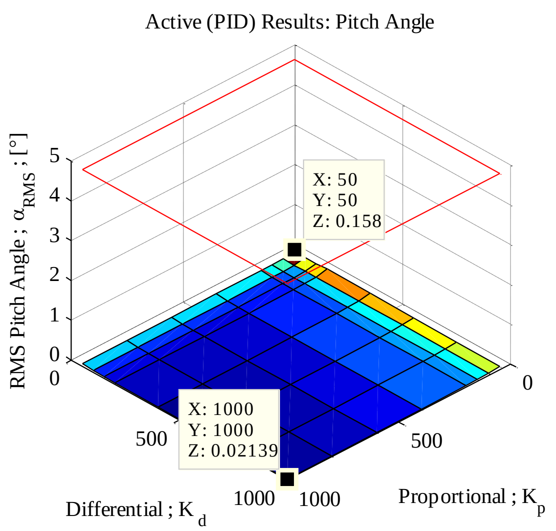

Figure 9 Active results: RMS pitch angle of stabilised platform.

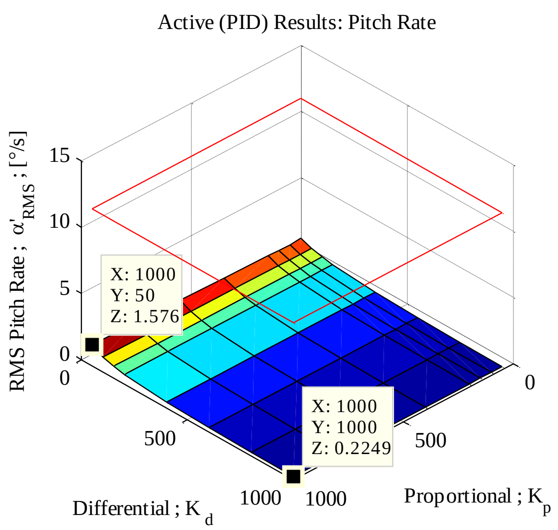

Figure 10 Active results: RMS pitch rate of stabilised platform.

The ability of three platform suspension types (passive, semi-active and active suspensions) to limit the pitch orientation and motion of the platform is investigated. It is demonstrated that the passive suspension characteristics for optimal pitch orientation are in conflict with the characteristics required for reduced pitch motion. Therefore a single passive suspension solution that satisfies both criteria does not exist. The semi-active suspension system, containing an adjustable MR damper, yield similar pitch orientation results as compared to the passive system, but an improvement in pitch motion is obtained, without conflicting suspension or controller requirements. The active suspension yield the most favourable results as the capabilities of the suspension allow full control over the pitch motion of the platform. The RMS pitch angle is improved by 99.6%, and the RMS pitch rate is improved by 98.1% as compared to the baseline setup.

The results as discussed in this paper can only be obtained if the semi-active MR damper or active system is capable of delivering the required forces, as prescribed by the skyhook control algorithm or the PID controller, under the specific input conditions. Since relatively small semi-active damping forces are prescribed by the skyhook control algorithm (up to ±20N at velocities below ±0.15m/s), it is likely that a special MR damper will have to be developed. The forces to be delivered by the active system are also very small (up to ±15N at velocities below ±0.15m/s), resulting in a very low energy demand (only 5W).

As part of future work the joint connecting the stabilised platform to the Baja vehicle will be revised and extended to include the yaw degree of freedom. An active platform stability control system will be developed and tested using various terrain profiles, such as a rough Belgian paving track [13], and different vehicle speeds.

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