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Cross Coupling Effects of Modal Space Decoupling Control for Six Degree of freedom 6-DOF Parallel Mechanism (6 DOF PM)

P. O. Ogbobe [1, *], C. N. Okoye[2], N. S. Akonyi[3]

[1]*Department of Information and Communication Technology, National Board for Technology Incubation, Abuja, NIGERIA*
[2]*Federal Ministry of Education, Abuja, NIGERIA.*
[3]*Science Equipment Development Institute, Minna, Kaduna State, NIGERIA.*

Abstract

This study presented the performance of an alternative and effective control strategy through a decoupled controller by an input and an output transformation matrix, such that each Degrees of Freedom can be tuned independently with their bandwidth raised near to the eigenfrequencies. The simulation results of the Modal Space Decoupled Controller were analyzed in Matlab/Simulink environment and comparison made between the conventional PID controllers based on cross coupling effects. The results indicate that the conventional joint space conforms to the theoretical analysis when the compensation for the coupling effects was not considered. The results further showed that the Modal Space Decoupled Controller modified the dynamics characteristics, which can be attributed to reduction in the coupling effects between degrees of freedom motions.

Keywords: Six Degree of Freedom, Parallel Manipulator, Modal Space, Decoupled Control, Cross Coupling.

1.0 INTRODUCTION

There have been various studies on six degrees of freedom parallel mechanism (6 DOF PM), Stewart Platform due to its advantages over the serial mechanism. These advantages are: better accuracy, large mechanical advantage, higher payload to weight ratio, less sensitive to joint disturbances, higher force-to-weight, stiffness and positioning accuracy. They have been used in various fields such as space docking, armored tank, flight, submarine and earthquake simulation, and machine tools etc [1-3]. The mechanism has the disadvantages of smaller workspace, complex command, and lower dexterity due to high coupling effects. The coupling effects are caused by its highly interactive system where the elongation of one of the six hydraulic cylinders induces a reaction forces in all other cylinders. The position and attitude of the moving platform varies with different effective mass, thereby, changing the system characteristics in each DOF [1-5]. The coupling of the system makes motion planning and control difficult, because changing one input to control its corresponding output will also affect other. outputs. Hence, it is important to study coupling effects and decoupling control strategies for the improvement of the control performance.

Various authors have studied coupling effects and decoupled control strategies. Ogbobe et al [6-9] analyzed the coupling effects between DOF and actuators by applying a singular value decomposition to the properties of joint space inverse mass matrix using a transformation matrix, product of transposed Jacobian matrix and an orthogonal unitary matrix. They provided useful design information to mechanism and controller designers to assess from conception the coupling effects between DOF in respect of the requirement for a particular application. Decoupling the functional requirements of 6 DOF PM is becoming a basic design principle with certain advantages such as decoupled motion characteristics, selective actuation for different tasks etc [6-9]. In the past few years, various types of decoupling control strategies have been proposed in literatures [11–15]. McInroy and McInroy et al. [16-17] have proposed two decoupling algorithms by combining static input-output transformations with hexapod geometric design.

Chen and McInroy, [17] provided and formulated a modified model that loosened and removed severe constraints of prior decoupling methods on the allowable geometry and payload. Thus, they greatly expanded the applications, and then proposed a decoupled control method applicable to flexure-jointed hexapods for micro

***Corresponding author (Tel:** +234 (0) 8189677554**)**

Email addresses: *peter.ogbobe@nbti.gov.ng* (P. O. Ogbobe), *okoyetech@yahoo.com* (C. N. Okoye), *akonyi01@gmail.com* (N. S. Akonyi)

precision and vibration isolation. The flexure jointed hexapod has very limited stroke and no need to include typical position dependent nonlinearities. The above control methods also can be extended and be applied to controlling hydraulic controlled 6 DOF PM, with inclusion of pose dependent nonlinearities and dynamic of hydraulic actuators.

The work of Hoffman [18], has shown that the hydraulic controlled Stewart platform in any positions has six independent directions, a set of eigenvectors, with the characteristics as described for the one degree of freedom system, so it is possible to apply the methods and theories well suited for single DOF hydraulic mechanical systems to analyze and design the Stewart platform with the combination of decentralized feedback. but without consideration of the coupling effects between all the actuators, only the resonance peaks of the lower eigenfrequencies of six rigid modes can be attenuated. The peaking points of other relative higher eigenfrequencies will be over damped[18], resulting to a controlled system with a bandwidth corresponding to the lowest eigenfrequency [18]. This also exists in dynamic pressure feedback and is a significant drawback to its applications. The study by Ogbobe et al [9] investigated the use of dynamic pressure feedback for the control of hydraulic controlled 6 DOF PM considering the dynamic coupling effects. The approach was to design and tune the dynamic pressure feedback based on the modal space decoupling control strategies. The motivation of using modal space decoupling control strategy is that each degree of freedom can be almost tuned independently and their bandwidths can also be raised near to the eigenfrequencies [9]. The purpose of the present paper is to apply the cross-coupling effects of the modal space and conventional joint space controllers and use the figures plotted in frequency domain as a basis to appraise the performance of the two controllers.

The goal in the application of modal space decoupling strategy is to improve the control performance and with a view to strive for an increased and suitable bandwidth. This section will develop modal space decoupling control strategy and applied to a model of the hydraulically driven 6 DOF PM. The controller took into consideration the coupling effects of the system which usually are neglected in conventional joint space controller. The remaining part will evaluate the performance of the modal space decoupling control strategy based on the cross coupling effects between the degree of freedom. Through this, we will be able to see the cross coupling effects of the two controllers and compared with the conventional joint space controller.

2.0 PLANT MODELING

The 6 DOF parallel mechanism for our study is a hydraulic controlled 6 DOF closed kinematic chain mechanism with two bodies connected by the six extensible legs consisting of a fixed base {b} and a moveable platform {p} with six hydraulic actuators supporting it as shown in figure1, $\mathbf{a}_i$ denotes the 3×1 vector of upper joint center point in body axes, $\mathbf{b}_i$ represents 3×1 vector of the lower joint center point in fixed base frame , and the sub index *i* is the actuator number. At neutral pose the body axes {p} attached to the movable platform are parallel to and coincide with the inertial frame {b} fixed to the base with its origin at the geometric centre of the base platform.

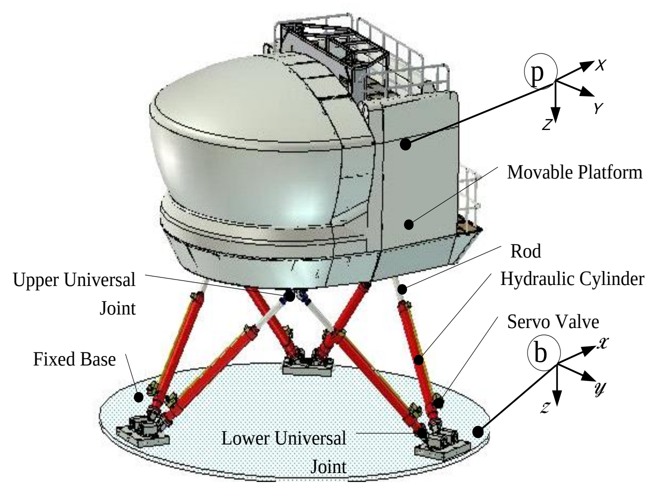


Figure1: Schematic view of a hydraulic controlled 6 DOF parallel mechanism

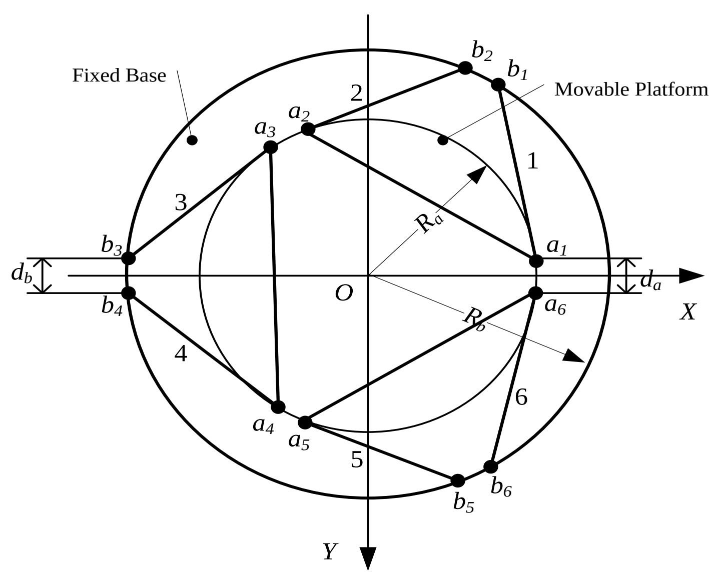


Figure 2: Top view of 6 DOF parallel mechanism

The equations of motion of a typical 6 DOF parallel mechanism considered as 14 rigid bodies are derived using a Kane's method [19] as

$$\mathbf{M}(\bar{q})\ddot{q}+\mathbf{C}(\bar{q},\dot{q})\dot{q}+\mathbf{G}(\bar{q})=\mathbf{J}_{\bar{q}}^{\mathrm{T}}(q)\mathbf{f}_a \qquad (1)$$

$\bar{\mathbf{q}}=[\mathbf{x}\quad \mathbf{y}\quad \mathbf{z}\quad \boldsymbol{\varphi}\quad \boldsymbol{\theta}\quad \boldsymbol{\psi}]^{\mathbf{T}}$, denotes the 6×1 vector of the platform position with respect to the fixed base frame, and contains translation and Euler angles. $\dot{\mathbf{q}}$ and $\ddot{\mathbf{q}}$ are the 6×1 platform velocity vector and acceleration vector respectively, and both contain translation and angular components. $\mathbf{f}_\mathbf{a}$ is the 6×1 vector of actuator output forces. $\mathbf{M}\bar{\mathbf{q}}$ represents the 6×6 mass matrix found in the base frame, considering the inertial effects of the actuators, $\mathbf{C}(\bar{\mathbf{q}},\dot{\mathbf{q}})$ is the 6×6 Coriolis/centripetal coefficients matrix, $\mathbf{G}(\bar{\mathbf{q}})$ is the gravity terms. Details of the above expressions have been described in detail by [2]. $\mathbf{J}_{\mathbf{l},\bar{\mathbf{q}}}$ is 6×6 Jacobian matrix relating the platform movements to the actuators length changes in joint space. Neglecting Coriolis/centripetal and gravity terms in Eq. (1), gravity will be compensated while Coriolis/centripetal will be treated as external disturbances in the Matlab/Simulink simulation model. Kinematics and dynamics analysis of the 6 DOF parallel mechanisms has been well established and can be found in several literatures [4,6-12].

3.0 DYNAMIC PRESSURE FEEDBACK FOR 1-DOF HYDRAULIC DRIVEN MECHANICAL SYSTEM

The linearised version from valve input to actuator position consists of a lightly damped second order system in series with an integrator. When controlled with proportional position feedback, only low performance can be obtained. In order to achieve higher bandwidth, the resonance of second order system has to be damped sufficiently. This can be done by pressure feedback [17-18]. But the inner loop with pressure difference feedback will decrease the rigidity of the system. To solve this problem, an alternative method can be applied using dynamic pressure feedback, in which pressure difference feedback signals are filtered by a one-order high pass band filter. The dynamic pressure feedback correction is equivalent to acceleration feedback without changing the undamped eigenfrequency ω_{hi} [17]. In this manner a closed loop bandwidth approximately equal to ω_{hi} can be attained.

Defining ξ_h as the original damping ratio and ξ_h'' is the desired damping ratio, and then the dynamic pressure feedback gain k_{dp} can be calculated by

$$k_{dp}=\frac{2(\xi_h''-\xi_h)\cdot\sigma_i A_p{}^2}{k_{a,i}k_q\omega_{hi}} \qquad (2)$$

The cutting frequency of the high pass filter is assigned with $\omega_{hi}/3$. A compromise of the damping ratio has to be achieved, because the damping ratio with higher value will also decrease the close loop bandwidth.

4.0 MODAL SPACE DECOUPLED CONTROLLER (MSDC)

The structure of a modal space controller is similar to the conventional joint space control, however, the signals including control errors, control outputs and pressure difference feedbacks are transformed into decoupled modal space by the unitary decoupling matrix, **U**, and so the coupling effects between all the actuators are fully considered and compensated. The proportional gains and dynamic pressure feedback functions are also tuned in modal space. The unitary decoupling matrix, **U**, will be chosen from the simplified linearised version at neutral position to the more complicated pose dependent one. The criterion of choosing is to ensure both good performance and facilitating the real time implementation. In steady state, the term of gravity compensation added at the controller output is written as $\frac{k_{ce}}{k_q A_p^2}\mathbf{J}_{lx}^{\mathrm{T}-1}\mathbf{G}$, the steady errors caused by gravity can be reduced to almost zero. So the control inputs of the servo valves can be expressed as

$$\begin{aligned}\bar{\mathbf{i}}_u &= \mathbf{U}\,\mathrm{diag}([\mathrm{k}_{a,1}\quad \mathrm{k}_{a,2}\quad \cdots\quad \mathrm{k}_{a,6}]^{\mathrm{T}})\mathbf{U}^{\mathrm{T}}\mathbf{e}\\ &\cdots+\mathbf{U}\,\mathrm{diag}\left(\begin{bmatrix}\mathrm{k}_{dp,1}\dfrac{\tau_{c,1}\mathrm{s}}{\tau_{c,1}\mathrm{s}+1} & \mathrm{k}_{dp,2}\dfrac{\tau_{c,2}\mathrm{s}}{\tau_{c,2}\mathrm{s}+1} & \cdots & \mathrm{k}_{dp,6}\dfrac{\tau_{c,6}\mathrm{s}}{\tau_{c,6}\mathrm{s}+1}\end{bmatrix}^{\mathrm{T}}\right)\mathbf{U}^{\mathrm{T}}\mathbf{P}_{\mathrm{L}}\\ &\cdots+\frac{k_{ce}}{k_q A_p^2}\mathbf{J}_{lx}^{\mathrm{T}-1}\mathbf{G}\end{aligned} \qquad (17)$$

Where, $\mathbf{e}=\bar{\mathbf{l}}_{com}-\bar{\mathbf{l}}$ is the position error, $\bar{\mathbf{l}}_{com}$ is the actuator length command in terms of inverse kinematics. For the ease of programming, each element of $\bar{\mathbf{i}}_u$ is given here as:

$$\bar{\mathrm{i}}_{u,i}=\sum_{j=1}^{6}\left(\left(\sum_{k=1}^{6}\mathrm{U}_{ik}\mathrm{U}_{jk}\mathrm{k}_{a,k}\right)\mathrm{e}_j+\left(\sum_{k=1}^{6}\mathrm{U}_{ik}\mathrm{U}_{jk}\mathrm{k}_{dp,k}\frac{\tau_{c,k}\mathrm{s}}{\tau_{c,k}\mathrm{s}+1}\right)\mathrm{P}_{\mathrm{L}j}+\frac{k_{ce}}{k_q A_p^2}\mathrm{J}_{ij}\mathrm{G}_j\right) \qquad (18)$$

Where, $\mathbf{e}=(e_j), \mathbf{U}=(U_{jk}), \mathbf{P}_L=(P_{Lj}), \mathbf{J}_{lx}^{T-1}=(J_{ij})$, $\mathbf{G}=(G_j)$ with i, j, k $=1,2\cdots6$.

Analyzing Eqs.(17,18), when $k_{a,1}=k_{a,2}=\cdots=k_{a,6}$ and $k_{dp,1}=k_{dp,2}=\cdots=k_{dp,6}$, the modal space controller degenerates into the conventional counterpart

5.0 CROSS COUPLING CHARACTERISTICS OF MODAL SPACE AND CONVENTIONAL CONTROLLERS

Various performance characteristics are used in the analysis of 6 DOF motion simulation platform. These characteristics are commonly called performance indexes and are usually peculiar to a given system architecture or control algorithm by using kinematic, static or dynamic models. They are defined as the system capability to achieve some specified criteria related to: motion and force transmission, repeatability, control performance etc.

To carry out the analysis, we acquired the cross coupling characteristics for both the modal space and conventional controller in frequency domain using Bode plots. Figures 1, 2 and 3 show the result of the cross coupling effects of modal space decoupling and the conventional controllers respectively; the interaction of each DOF with each other is evident.

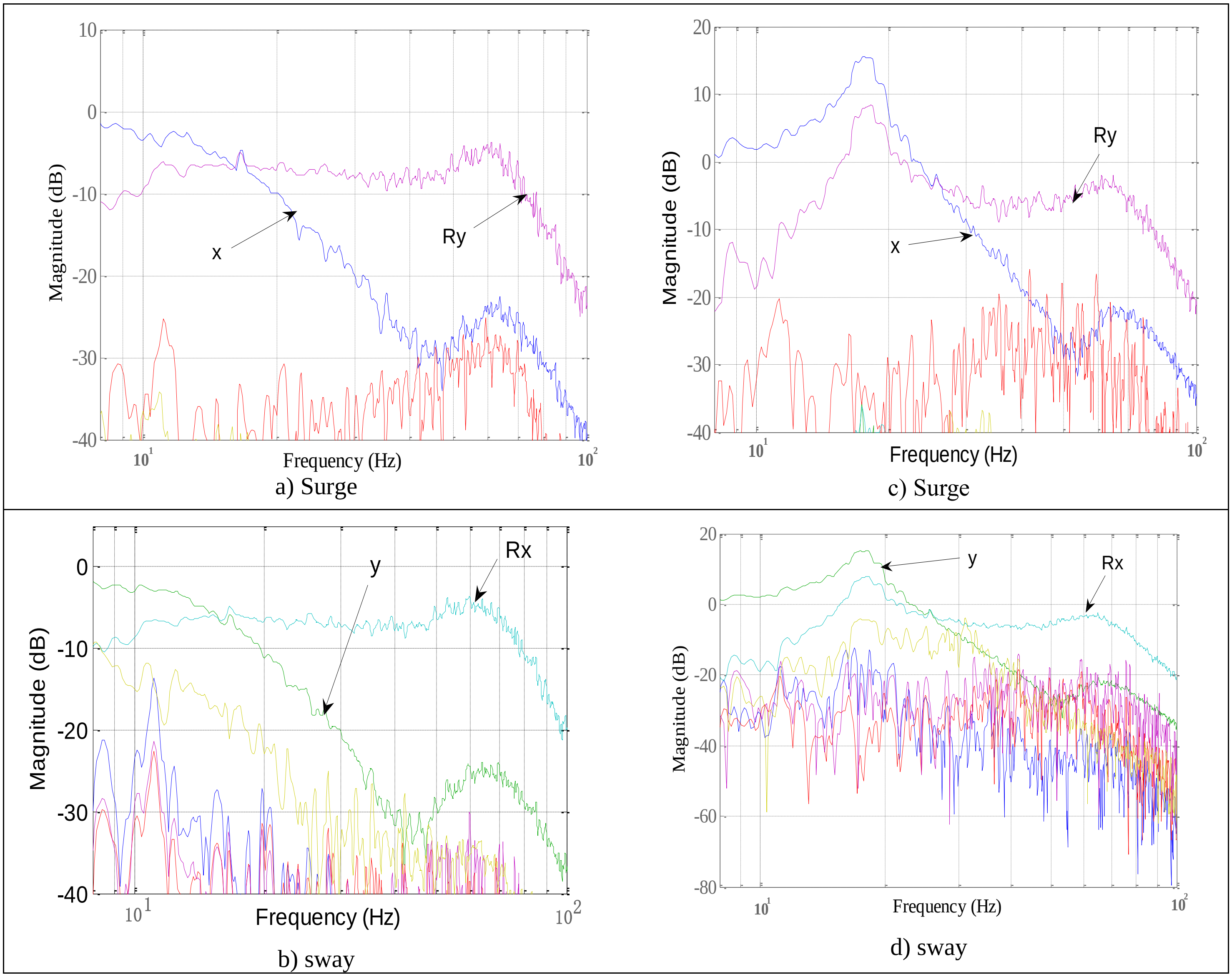


Figure 1: Simulation results of cross coupling characteristics for the modal space (right column, a-b) and conventional joint space controller (left column, c-d)

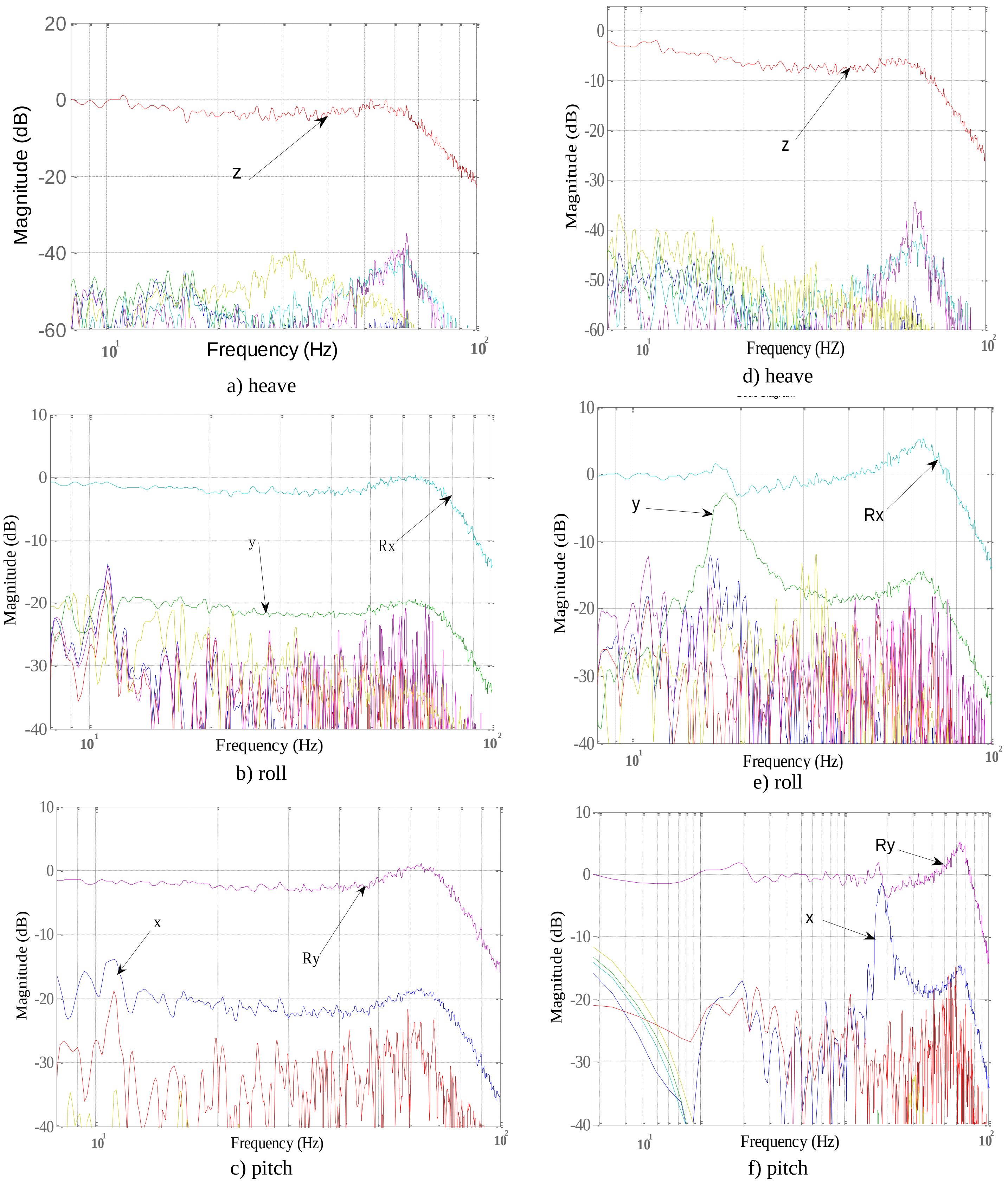
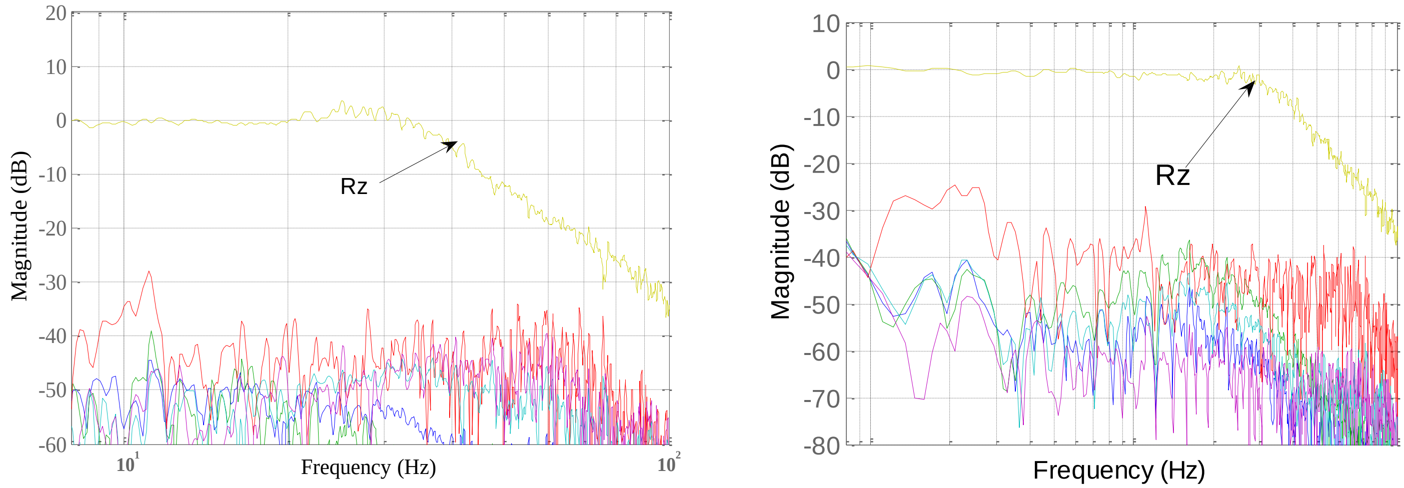


Figure 2: Simulation results of cross coupling characteristics for the modal space (right column, a-e) and conventional joint space controller (left column, d-f)



a) yaw b) yaw

Figure 3: Simulation results cross of coupling characteristics for the modal space (right column, a) and conventional joint space controller (left column, b)

6.0 CONCLUSION AND DISCUSSION

Comparing Figs 1, 2 and 3, the control performance of two controller based on cross coupling effects can be evaluated. The results indicate that the conventional joint space conforms to the theoretical analysis when the compensation for the coupling effects was not considered. It is evident from the figures that the MSDC modified the dynamics characteristics, which can be attributed to reduction in the coupling effects between degrees of freedom motions. Considering the figures in the right and left column, at low frequency, the amplitude of displacement is governed principally by the stiffness. In this region the dynamic characteristics level may be described as stiffness controlled. At a higher frequency (and also close to the natural frequency) the dynamic characteristics can be described as being damping-controlled. This is purely a measure of the capacity of the system to dissipate energy. The amplitude depends on the amount of damping in the system (a measure purely of damping) and the natural frequency, a function of the stiffness and the mass). The dynamic response system is therefore affected by the mass, stiffness and damping of the system, with the damping being the most important [19]. This explains the behavior of the system at low and high frequencies with respect to the cross coupling effects.

Although, modal space decoupling controller considerably reduced the coupling effects and enhanced the control performance, by sufficiently damping the system and increasing the bandwidth, it is also susceptible to cross coupling effects. We should bear in mind that the analyses have been based on light damping and rigid-body assumptions. However, it is considered from the results that the improved control performance of the modal space decoupling control strategy outweighed the inherent coupling effects that may not have been totally eliminated. These effects may be due to the light damping assumptions.

Damping management is not yet a mature design principle. The reason for this is that damping mechanisms, like friction, are notorious sources of inaccuracy and need to be further investigated; especially the passive joints frictional effects. The proportionality of the damping system need also further investigating with regards to motion coupling effects. This is with a view to justify the assumption of lightly or proportionally damped.

Several possibilities for future work have been identified during the course of this study. The proposed decoupling approach may still be susceptible to some coupling effects. In order to realize the complete decoupling of the system an implementable decoupling control law, by which complete decoupling could be achieved, can be further, investigated. The suggested approach is the use of velocity and acceleration feedback (VAF) control. This can allow the transformation of non-proportional to proportional damping by modifying the damping and stiffness matrices. It is our opinion that the areas highlighted for future areas of development represent significant areas of interest and possible weaknesses of this paper. Despite these weaknesses, the authors believe that a usable and exciting contribution has been made to the field of coupling problems and of decoupling control of 6 DOF motion simulation platform.

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