The 10th JFPS International Symposium on Fluid Power FUKUOKA 2017 Oct. 24-27, 2017 The Japan Fluid Power System Society

ACTIVE VIBRATION COMPENSATION FOR CATWALK BY HYDRAULIC PARALLEL MECHANISM

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Abstract. In field of marine constructions, traffic ships are used to board/go ashore to floating structures. The traffic ship is fluctuated in tidal waves, and workers face a risk of accidents such as falling in the sea and being pinched between the traffic ship and the floating structure. It is a great issue to solve these problems. We focus attention on the possibility to contribute safety and workability out of consideration of the ship fluctuation if anywhere on the fluctuated working vessels will be kept in a horizontal position. In this paper, an active vibration compensator with Stewart Platform type of hydraulically driven parallel kinematic mechanism is proposed and developed. The analytical results show that the design of the dynamic behavior for the actual active vibration compensation system has been confirmed by simulation models using the simulation software, MATLAB+Simulink. Trial results show that 66% to 84% of the heave, roll and pitch motion of the main hull is absorbed.

Keywords: Active Vibration Compensation, Marine Construction, Parallel Kinematic Mechanism, Inverse Kinematic, Six Degrees of Freedom

INTRODUCTION

Japan has the sixth 30000 km marine shoreline in the world and 447 million km^2 with territorial waters and exclusive economic water area. From viewpoint of resource development and utilization of marine energy, it is important to carry out maritime construction in the waters of Japan. Especially, offshore wind power industry is growing fast in Japan. With this trend, it is important to maintain a good maintenance and repair strategy. Current offshore access is used a traffic ship which is fluctuated in tidal waves. Then workers face an increased risk of accidents such as falls in the sea and pinched between the traffic ship and floating structures.

There are many researches for anti-rolling systems for ships [1]. The anti-rolling systems for ships are categorized by three types, such as passive, active and hybrid type. In the passive type of the anti-rolling system, performance of vibration compensation depends on the natural frequency of a body of the ship. An active fin stabilized system has good performance of the vibration compensation under way of the ship, but has no effect on being berthed and floating conditions. An anti-rolling gyro system has been developed in practical use for yachts and sailboats [2]. Technical feasibility of a motion-stabilized cabin which is mechanically supported by the main hull by hydraulic cylinders has been investigated in practical use for a high-speed passenger ship [3]. Hybrid type of anti-rolling system by using passive moving mass and active servo motor drive has been developed [4] and applied to the Japanese oceanographic vessel [5].

A ship motion as shown in Figure 1 has six-degrees of freedom (6-DOF). There are three spatial axes in ship

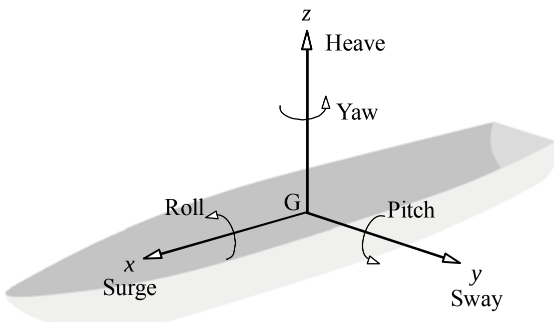


FIGURE 1. Six degrees of freedom for ship motion

and the movements around three axes are defined as roll, pitch and yaw. Pitch is an up/down rotation around lateral axis, roll is a tilting rotation around longitudinal axis, and yaw is a turning rotation around vertical axis. Heave is a linear vertical motion, sway is a linear lateral motion, and surge is a linear longitudinal motion. The motion of the ship has a complex movement combining 6-DOF. Actual waves are not regular waveforms, they are irregular and wave height is also different. In order to compensate the vibration on the ship according to the ship motion, we must introduce the anti-motion system with wide movable range and 6-DOF.
In this study, we focus attention on the possibility to contribute safety and workability out of consideration of the ship fluctuation if anywhere on the fluctuated working vessels is kept on a horizontal position. In this paper, an active vibration compensator with parallel kinematic mechanism is proposed and developed. The vibration compensator with the parallel kinematic mechanism has hydraulically driven. A new active vibration compensation system on the traffic ships is proposed and developed to provide safe ship-based access to the floating structure such as the offshore wind power system.

SYSTEM CONFIGURATION

A block diagram of the active vibration compensation system is shown in Figure 2. The system consists of a motion stabilized platform with Stewart Platform type of parallel kinematic mechanism [6] and a catwalk on the platform. The motion platform is supported on the main hull and kept in a horizontal position by means of the six-degrees of freedom hydraulic parallel mechanism which control to absorb the motion of the main hull in accordance with the control signal from an on-board computer and motion sensors.

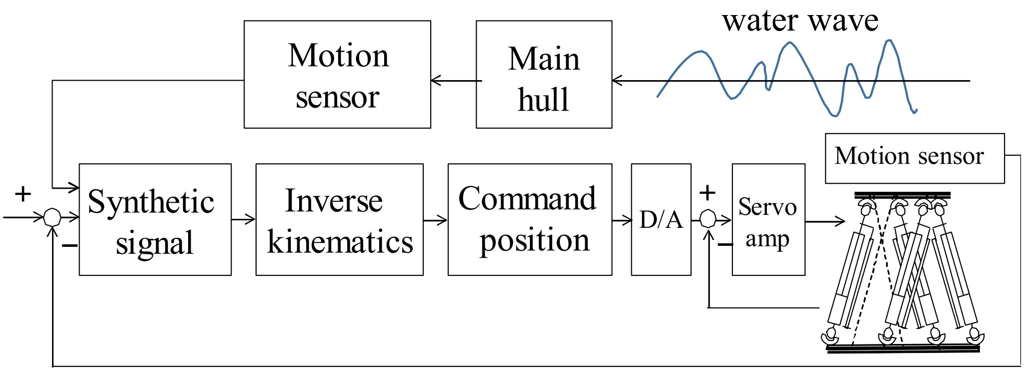


FIGURE 2. Configuration of the active vibration compensation system

PARALLEL KINEMATIC MECHANISM

In the parallel kinematic mechanism, two plates (base and upper) are connected through multi articulated links in which a linear actuator enables to change the link length. The links are connected through universal joints. The position and orientation of the upper plate with respect to the base plate are controlled by the length of links. The external forces applied to the upper plate are distributed among all parallel links and actuators [7].
Figure 3 shows illustrations of the Stewart type of parallel kinematic mechanism. The parallel mechanism, which achieves 6-DOF motion by the coordinated movement of six linear actuators, has many advantages compared with the conventional serial link mechanism as follows [8];

- Higher payload-to-weight ratio since the payload is carried by six cylinders in parallel;
- Higher accuracy owing to noncumulative joint error;
- Simpler solution to the inverse kinematic equation.

The choice of parallel mechanism for robot manipulators, machine tools and motion simulators is justified by its obvious advantages. Structure of parallel mechanisms has been known for a long time [9]. In 1965 Stewart illustrated use of the parallel structure for building an active flight simulator.
In our research, the hydraulically driven parallel mechanism is the foundational mechanism and functions as a compensation for stabilization. This system acts to absorb motion of the main hull with the 6-DOF motion base.

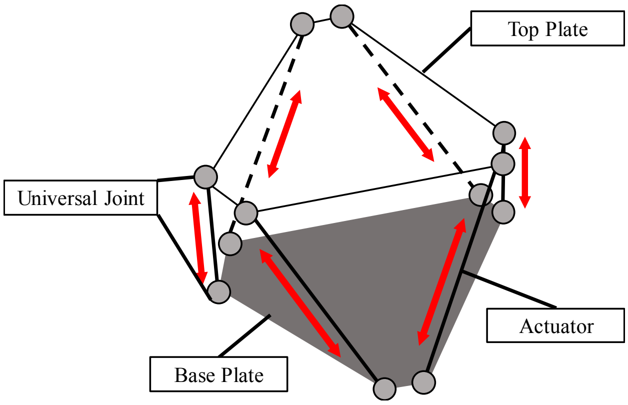


FIGURE 3. Stewart Platform type of parallel kinematic mechanism

INVERSE KINEMATICS

Figure 4 shows a vector diagram of Stewart Platform type of parallel kinematic mechanism. The upper rotund plate is the top plate which is motion platform and the lower plate is the base plate. The black dots are ends of the legs. a_i (i = 1,2,3,4,5,6) and b_i (i = 1,2,3,4,5,6) are constrained on each horizontal plane. This system has two reference frames, $(\mathbf{i}, \mathbf{j}, \mathbf{k})^{\mathrm{T}}$ and $(\mathbf{e_1}, \mathbf{e_2}, \mathbf{e_3})^{\mathrm{T}}$. One is the motion frame which is located at the centroid of the platform. The other is the world reference frame. Sign O' denotes the origin of the motion frame, and sign O denotes the origin of the world reference frame. $\mathbf{A}_i = (a_{ix}, a_{iy}, a_{iz})\ (\mathbf{e_1}, \mathbf{e_2}, \mathbf{e_3})^{\mathrm{T}}$ (i = 1,2,3,4,5,6) is the position vector of the bottom end of the link, Vector $\mathbf{B}_i = (b_{ix}, b_{iy}, b_{iz})\ (\mathbf{e_1}, \mathbf{e_2}, \mathbf{e_3})^{\mathrm{T}}$ (i = 1,2,3,4,5,6) denotes the universal joint which connects the motion platform and the fixed length link, and $\mathbf{P} = (p_x, p_y, p_z)\ (\mathbf{e_1}, \mathbf{e_2}, \mathbf{e_3})^{\mathrm{T}}$ is the position vector of centroid O' with respect to the world reference frame. For these vectors and the fixed length link vector $\mathbf{L}_i = (l_{ix}, l_{iy}, l_{iz})\ (\mathbf{e_1}, \mathbf{e_2}, \mathbf{e_3})^{\mathrm{T}}$ (i = 1,2,3,4,5,6), the following relation is derived [9] :

$$\mathbf{L}_i = \mathbf{P} + \mathbf{R}\mathbf{B}_i - \mathbf{A}_i \quad (i = 1,2,3,4,5,6) \quad , \tag{1}$$

where the coordinate transform matrix **R** is expressed as follow:

$$\mathbf{R} = \begin{bmatrix} \cos\varphi\cos\theta & -\sin\psi\cos\varphi + \cos\psi\sin\theta\sin\varphi & \sin\psi\sin\varphi + \cos\psi\sin\theta\cos\varphi \\ \sin\psi\cos\theta & \cos\psi\cos\varphi + \sin\psi\sin\theta\sin\varphi & -\cos\psi\sin\varphi + \sin\psi\sin\theta\cos\varphi \\ -\sin\theta & \cos\theta\sin\varphi & \cos\theta\cos\varphi \end{bmatrix} \tag{2}$$

Equation (1) denotes the relationship between 6-DOF attitude of the motion platform and the leg vector. The leg vector can be calculated by Eq. (1)and (2).

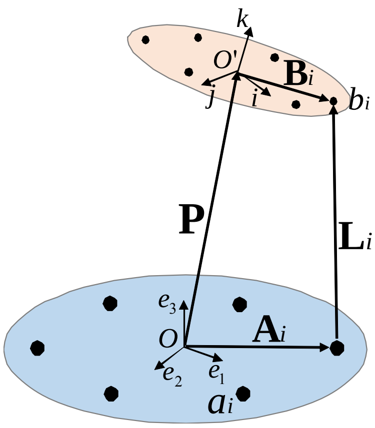


FIGURE 4. Vector diagram of parallel kinematic mechanism

PROTOTYPE TEST ON OFFSHORE

In order to confirm the performance of the active vibration compensation by the parallel mechanism, a prototype 3/10 scale model of the actual system used for the offshore tests on the sea near the Moji port of Yamaguchi Japan. Figure 5 shows a photograph showing the details of the prototype system. In our offshore experiments, the prototype of the parallel mechanism was fixed on the deck of 496 tons vessel, and motion sensors were fixed on the main hull and the top plate of the parallel mechanism. The motion of the top plate is measured and recorded to confirm whether or not the parallel mechanism absorb the motion of the main hull in accordance.
For comparison of the motions, RTK-GPS sensors were mounted on the top plate of the parallel mechanism and the main hull and measured azimuth angle and motion of sway.
Figure 6 shows a typical example of experimental results for motion of heave and roll of the main hull and the top plate. Maximum values and standard deviations between the measured amplitude peaks from the maximum and minimum values for the experimental results are tabulated in Table 1. These results are compared with turning on and off the fluctuation absorber of the parallel mechanism. These experimental results show that the motion of the top plate is greatly reduced with respect to the fluctuation of the main hull. In the case of heave motion, the vibration of the platform reduced about 82% on the amplitude peak and about 74% by the standard deviation. Also, the fluctuations are significantly reduced even at the roll angle about 84% by the amplitude peak and about 78% by the standard deviation. Therefore, by using this prototype system, the motions of heave, roll and pitch were greatly reduced on the top plate, and the performance of the hydraulic parallel mechanism absorber was experimentally confirmed.

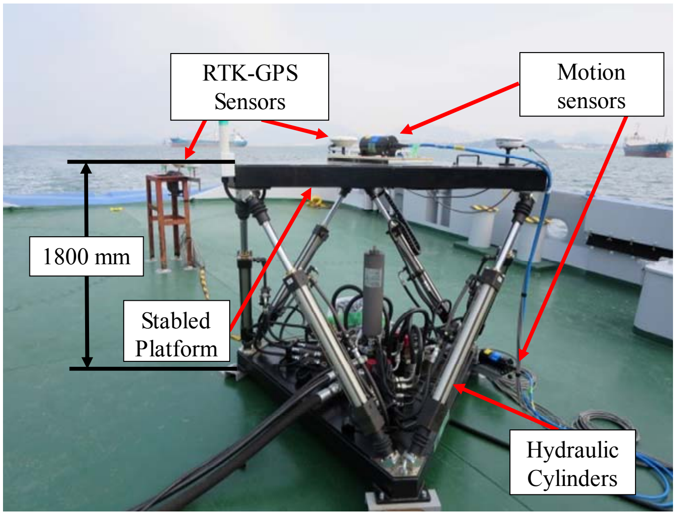
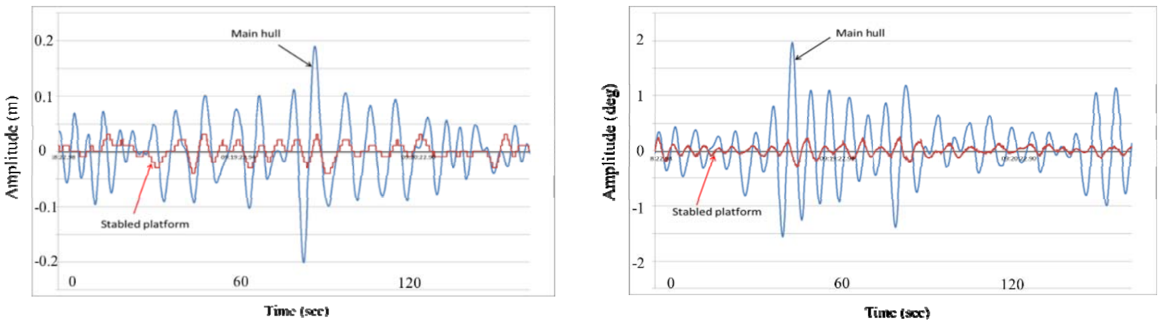


FIGURE 5. Scale model of the active vibration compensation system on the ship



(a) Heave (b) Roll

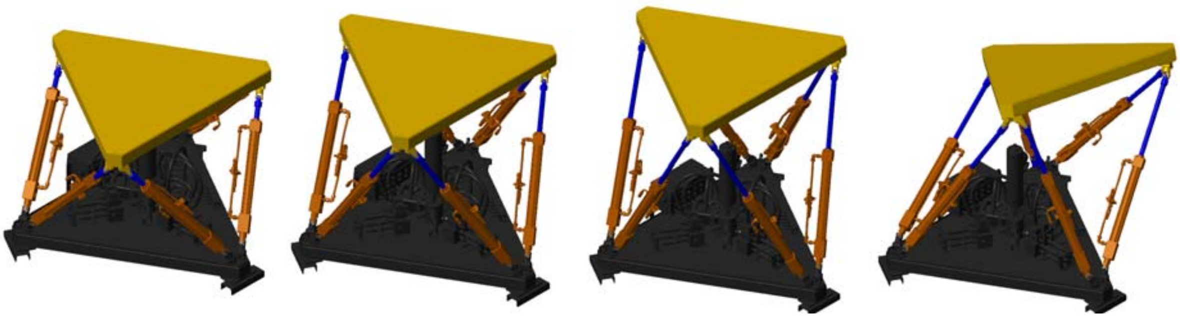
FIGURE 6. Measured heave and roll motions of the main hull and top plate

TABLE 1. Measured heave and roll amplitude and absorbing ratio

	Heave				Roll			
Compensator	Max(m)	Mix(m)	Peak to Peak	σ	Max(deg)	Mix(deg)	Peak to Peak	σ
Off	0.190	-0.201	0.391	0.057	1.970	-1.560	3.530	0.541
On	0.030	-0.040	0.070	0.015	0.250	-0.300	0.550	0.002
Absobing ratio			82.1%	74.1%			84.4%	78.3%

MOTION SIMULATION

This section gives the simulation results based on our analysis by simulation models using the simulation software, MATLAB+Simulink based SimMechanics. The purposes of the simulation are to verify experimentally the performance of the motion stabilized platform and to manufacture an actual scale system efficiently and exactly. For this simulation, the 3/10 scale model of the actual system which is a prototype was brought back into the simulation model. Dimensions of the prototype are about 1800 mm×1800 mm×1400 mm. The total weight is 450 kg and the stroke of actuators is 500 mm. Each stroke of six actuators is calculated by the inverse kinematics model. Figure 7 shows typical examples of the motion of parallel mechanism model. From the motion simulation using the model, the dynamic behavior of the parallel kinematic mechanism is numerically confirmed.



(a) Minimum stroke (b) Nutral stroke (c) Maximum stroke (d) Tilt position

FIGURE 7. Typical example of motion simulation

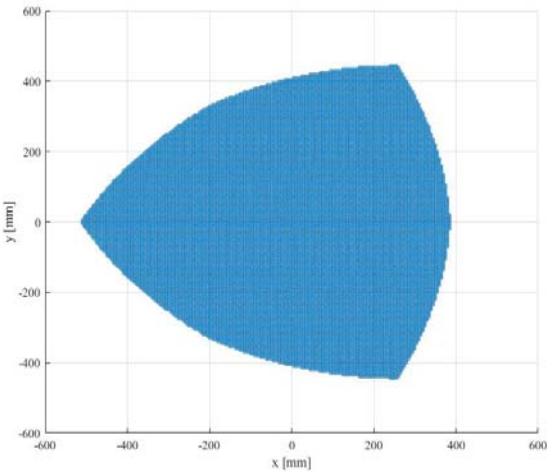
WORKSPACE

A workspace of the parallel mechanism greatly effects on the performance of the vibration compensation of the main hull. A numerical simulation model to analyze the performance of the workspace by repeated calculation has been developed by using mathematical software, MATLAB in a virtual motion simulator. The repeated calculations are carried out under consideration of a movable range of each joint.

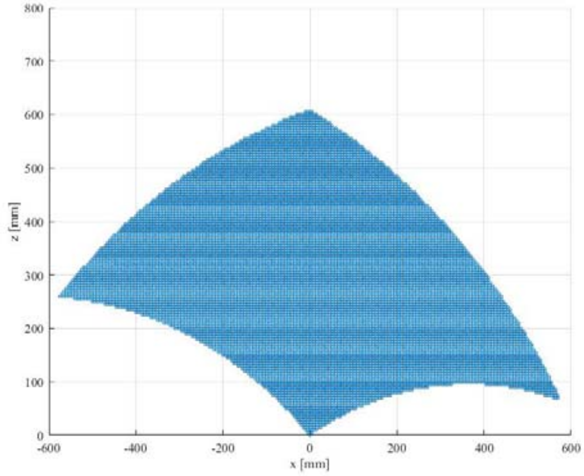
Figure 8 shows calculated workspaces for the *x*-direction (surge), *y*-direction (sway) and *z*-direction (heave). The stroke of each actuator is 500 mm. According to the results of the workspace analysis, the prototype of the parallel mechanism can move form -575 mm to 570 mm in the range of *x*-direction, from -495 mm to 495 mm in the range of *y*-direction and from 0 to 605 mm in the range of *z*-direction. The translational workspace of the prototype of the parallel mechanism is numerically confirmed through the calculations.

CONCLUSIONS

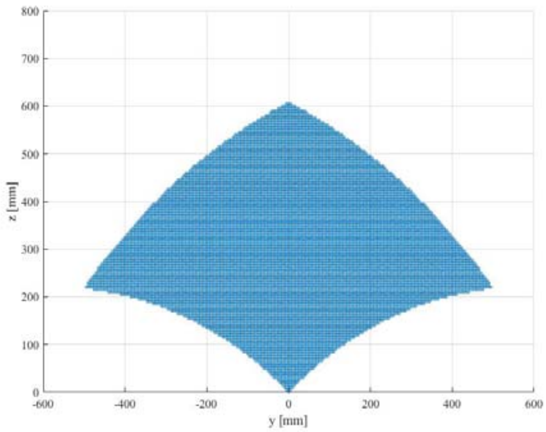
The active vibration compensation with Stewart platform type of the parallel kinematic mechanism has been proposed and developed. The 3/10 scale model of the actual system has been developed and stabilized performance of the platform was experimentally verified. The compensated ratio of 74 % to 84 % to the heave and roll motions of the main hull have been performed and the top plate has been almost horizontaly kept. From the



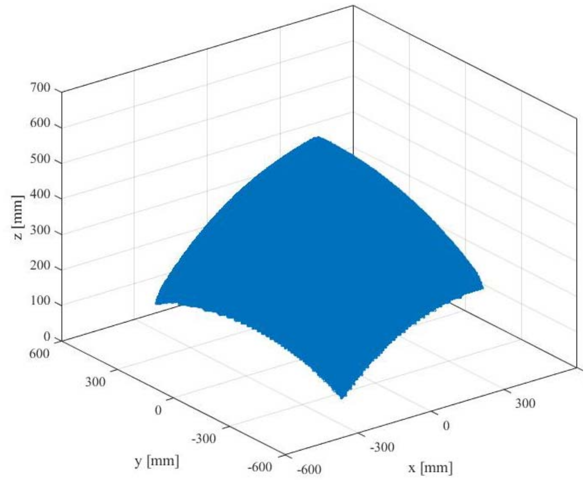
(a) Workspace of *x* and *y* direction



(b) Workspace of *x* and *z* direction



(c) Workspace of *y* and *z* direction



(d) Workspace of *x, y* and *z* direction

FIGURE 8. Workspace of the prototype of the parallel mechanism

result of the numerical simulation, the dynamic behavior for the actual active vibration compensation system has been confirmed.

ACKNOWLEDGMENTS

This work was supported by Toa Corporation and Koenn Co., Ltd., Japan. We also thank to Mr. Daichi Kojima for his valuable assistance for software the parallel mechanism and the hydraulic system. This research was supported and carried out by Hosei University Research Institute for Advanced Motion Simulators (HAMS).

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