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Research on the Vibration Characteristic of a Seawater Hydraulic Piston Pump System and Vibration Reduction Approach

ZUTI ZHANG[1], YUNZHI LIANG[1], KAI ZHAO[1], XINPING LONG[1], AND HANYANG LIU[2]
[1]School of Power and Mechanical Engineering, Wuhan University, Wuhan 430072, China
[2]Wuhan Maritime Communication Research Institute, Wuhan 430072, China

Corresponding author: Xinping Long (afanti_2012@hotmail.com)

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ABSTRACT Seawater hydraulic piston pump has been widely used in ocean engineering and become a key power component for underwater equipment due to its inherent characteristic such as energy conservation and environment friendly. As one of the most important properties of seawater hydraulic piston pump, vibration would further determine the stability, reliability and stealth of underwater equipment. In this paper, the dynamic of a novel crankshaft seawater piston pump with gearbox are analyzed theoretically, illustrating the dramatically effect of the gearbox on the vibration characteristic of the seawater hydraulic piston pump. Then, a new type crankshaft seawater hydraulic piston pump system driven by a permanent magnet synchronous motor instead of asynchronies motor is proposed in order to eliminate the gearbox and improve the vibration characteristics. In addition, the vibration characteristic of this new type crankshaft seawater hydraulic piston pump system driven by asynchronous motor and permanent magnet synchronous motor are studied and compared experimentally. The experiment results indicate that the seawater hydraulic piston pump system driven by permanent magnet synchronous motor eliminating the gearbox has lower vibration acceleration level in comparison with the pump system driven by asynchronous motor and gearbox. The vibration acceleration level can be reduced from 7dB to 4.3dB. Driving the seawater hydraulic piston pump by permanent magnet synchronous motor instead of asynchronous motor is of benefit to eliminating the gearbox and reducing the vibration excitation source. Consequently, the vibration characteristic of the seawater hydraulic piston pump system driven by a permanent magnet synchronous motor is significantly improved.

INDEX TERMS Seawater pump, vibration, gearbox, synchronous motor.

I. INTRODUCTION

Water hydraulics possesses the advantages of little environment pollution, nonflammable, lower operating cost and so on because it directly uses natural water as working medium instead of traditional mineral oil [1]–[3]. And it is more flexible in underwater application to build open-circuit system directly utilizing the seawater as working medium and discharging drainage to the open surroundings without needing return hose and reservoir. These special characteristics make it widely used in underwater equipment, such as variable ballast system (VBS), bilge drainage system, high pressure water mist fire suppression system, high pressure waterjet propulsion system, etc. [4]–[8]. With the missions of underwater equipment becoming increasingly complex and varied, such as military applications, inspection of underwater structures, exploration of unknown environments, oceanographic observations, submarine rescue, underwater animal behavior research, and so on [9], the underwater equipment with better stability, higher reliability, lower vibration and noise radiation characteristics is strongly required.

In general, seawater hydraulic piston pump system, as the core power component of water hydraulics system, is the main vibration excitation source of underwater

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equipment. Consequently, the vibration performance of seawater hydraulic piston pump system would directly affect the overall properties of water hydraulics system and even worse the invisibility, stability and reliability characteristic of the underwater equipment. Reducing the vibration of seawater hydraulic piston pump system has become the primary strategy to improve the overall performance of water hydraulics system and underwater equipment [10], [11]. Seawater hydraulic piston pump system is a special equipment which relates to the mechanic dynamic and hydrodynamic, and its vibration excitation source can be divided into the mechanical vibration and the fluid vibration [12]. In addition, the fluid vibration and mechanical vibration coexist in the actual working process [13]. Therefore, in order to reduce the vibration of seawater hydraulic piston pump, suppressing the mechanical vibration excitation source and reducing the fluid excitation source have become the two primary approaches. Researchers have explored many effective methods to control the mechanical and fluid vibration excitation source. Passive vibration reduction technology which utilizes vibration isolation device to break the vibration transmission path from the source to the vibration-sensitive unit of equipment and improve the performance of the connected objects is possibly the most widely used approach for vibration protection because it is economical and convenient, that there is no need to make any changes to the equipment structure itself [14]. Rubber isolator is widely applied in heavy equipment (such as hydraulic pump, etc.) due to its high load-bearing capacity, low cost and good performance of vibration reduction. Chen *et al.* [15] studied the vibration characteristic of the rubber isolation system and evidenced the significance of the uncertainty and nonlinearity in rubber isolation system. Wang *et al.* [16] studied the effect of rubber isolator's dynamic stiffness on the dynamic of seawater hydraulic piston pump. In conclusion, the vibration attenuation performance of rubber isolator directly depends on its dynamic and static characteristics, and the dynamic stiffness is the important characteristic of rubber isolator which has complex nonlinearity. Generally, rubber isolator has good robust but cannot be adaptive to the change of excitation frequency.

Floating raft isolation system is a special double-layer isolation system which utilizes the stiffness, damping and intermediate mass of the elastic element within the isolation system to reduce the vibration energy and further reduce the vibration of ship hell, this passive isolator could achieve good effects beyond the system resonance region and has been widely applied in ships. Li and Xu [17] presented the analytical model of a floating raft isolation system and investigated its vibration attenuation performance. In general, the floating raft isolation system has the characteristic of complex structure and high cost. In addition, the damping effect of the dampers within the floating raft isolation system is limited to a certain frequency band.

Optimizing the structure and modifying the stiffness and damping of the hydraulic piston pump to change its dynamic behavior is another significant method to reduce the vibration of hydraulic piston pump [10], [14], [18], [19]. Palmen [19] reduced the vibration of a hydraulic piston pump by optimizing the pump housing. Zhang *et al.* [20] utilized a kind of free-layer viscos-elastic damping material to regular the natural frequency of an axial hydraulic piston pump, and the experimental results showed that the vibration of the hydraulic piston pump at the second harmonic can be suppressed by using this damping material. Wu *et al.* [21] researched the influence of the port valve materials on the vibration of a seawater hydraulic piston pump. Meanwhile, extracting the vibration modes (such as natural frequencies, damping coefficients and mode shapes) is very important to analyze the dynamic

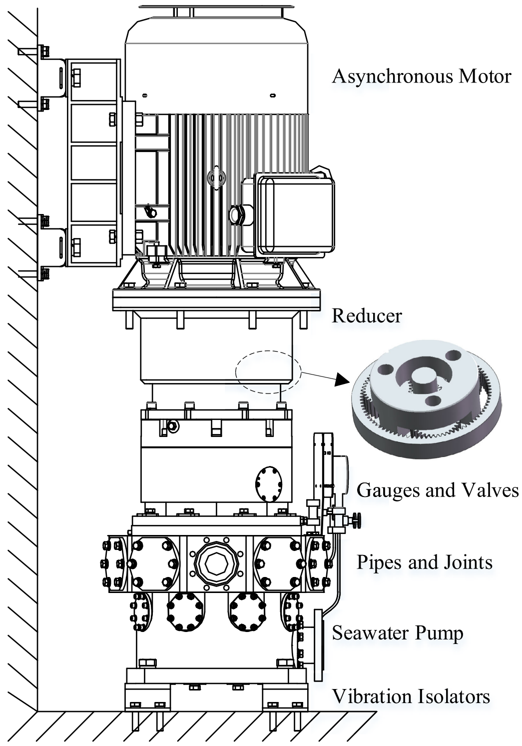

FIGURE 1. Configuration of seawater hydraulic piston pump system.

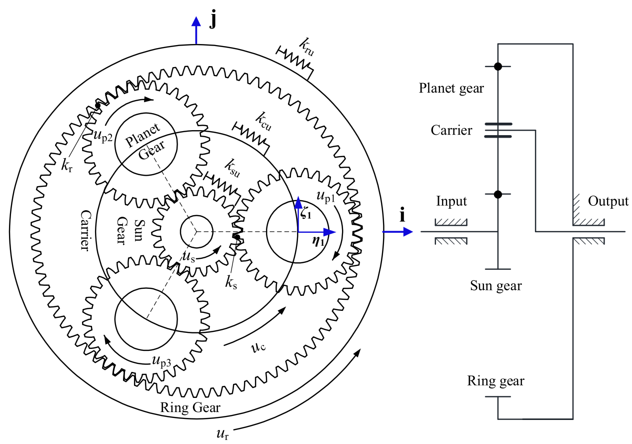

FIGURE 2. Diagram of the planetary gear train and system coordinate.

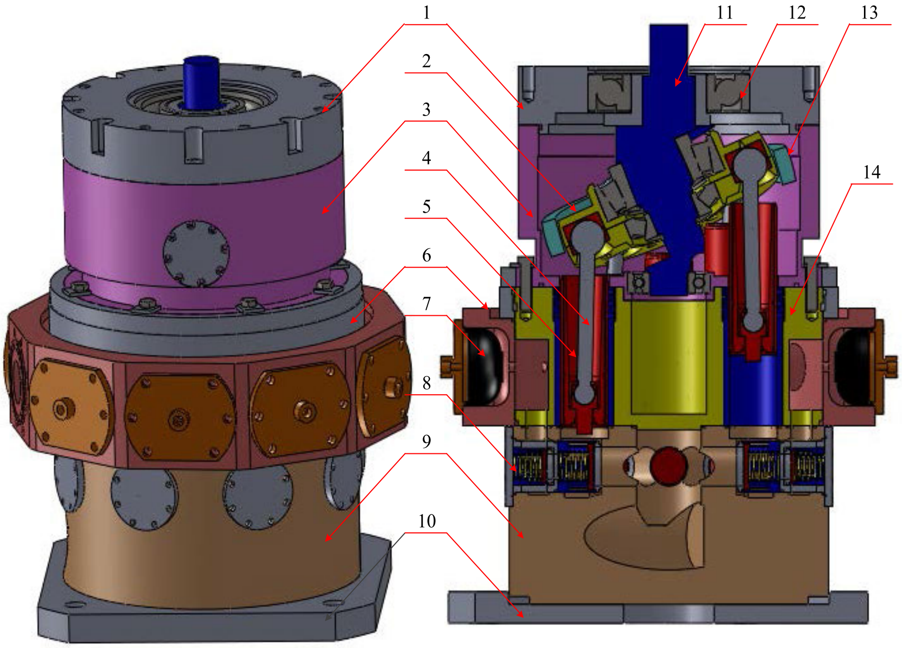

FIGURE 3. Configuration of water hydraulic piston pump.

TABLE 1. Parameters of planetary gear.

Item	Sun gear	Ring gear	Planetary gear	Carrier
Module	3	3	3	-
Teeth Number	22	86	32	-
Base Diameter	66	258	96	162

behavior and optimize the hydraulic piston pump structure. Modal analysis has become a key technology in structural dynamics analysis and structure optimization. Xu *et al.* [11] optimized the hydraulic piston pump housing using the EMA technique, and the results showed that the average sound pressure level was reduced about 2.0 dB(A) at the discharge pressure of 250 bar. Kunze and Berneke [18] used the EMA method to optimize the structure of a hydraulic piston pump, and the experimental results indicated that the EMA technique could effectively reduce the vibration of hydraulic piston pump. Wang *et al.* [22] used a transmissibility-based method to predict the modal parameters of a hydraulic piston pump under pressure excitation.

As for the fluid-born vibration generated in hydraulic piston pump. Pressure pulsation and fluid field within the hydraulic piston pump channel are the main influence factors. Installing auxiliary attenuators to absorb pressure pulsation and optimizing the existing channel structure to improve the fluid field character are the two most reliable solutions to reduce the fluid-born vibration in hydraulic piston pump [17], [23], [24]. Generally, valve plate is considered as the main generation sources of vibration and noise [25], [26]. Optimizing the open timing delay, relief groove and pre-compression volume are the most commonly used methods to improve the vibration and noise performance of valve plate [27]–[31]. Manring [32] invested the effects of valve plates with three typical slot geometries on the piston chamber pressures, and demonstrated that the triangular slot geometry is to have the best performance. Mandal *et al.* [33], [34] built a mathematical model to analyze the effects of the length, width and height of triangular grooves on the flow ripple and then updated the model by taking inertia into consideration. Seeniraj and Ivantysynova [35] adopted a multi-parameter multi-objective optimization method using genetic algorithm to optimize the valve plate parameters considering their effects on noise and efficiency. Xu *et al.* [10], [31] optimized the valve-plate transition region, including relief grooves and timing to reduce the pressure pulsation, vibration and noise of hydraulic piston pump, and presented several new design methods of the valve plate. Ye *et al.* [26] investigated the reducing mechanism of a valve plate optimization method with damping holes. Chacon and Ivantysynova [36] presented a comprehensive design methodology of axial piston pumps, and a type of non-linear groove opening profile was adopted within the virtual prototyping optimization procedure and plausible results on maximizing volumetric efficiency and reducing flow ripples were received. Kim *et al.* [37] found that using an appropriate pre-compression angle and notch design could reduce the pump vibration and noise, and V-type notch valve plate is better than circular type notch in the characteristic of pressure and flow pulsation [38]. Passive attenuators are the mostly widely used devices to achieve the fluid pulsation suppression in hydraulics system [39]. Zuti *et al.* [40] integrated a

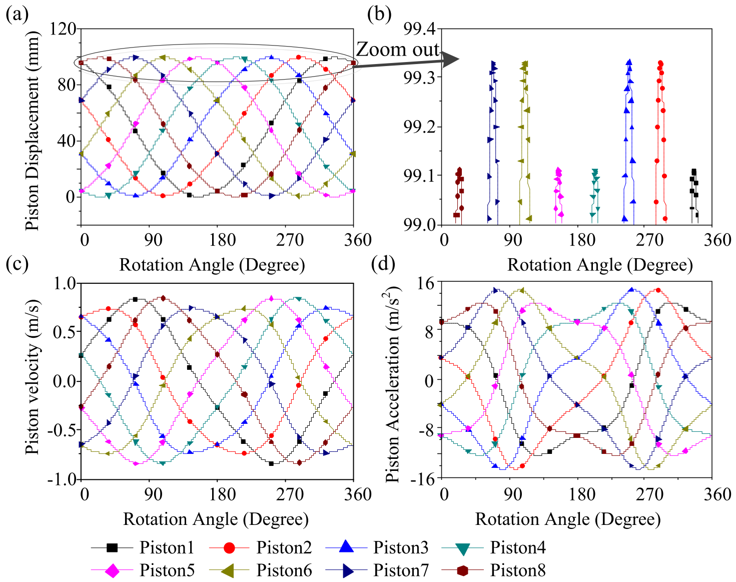

FIGURE 4. The kinematics of multiple pistons: (a) the displacement of the pistons, (b) the enlarged drawing of the displacements, (c) the velocity of the pistons, (d) the accelerations of the pistons.

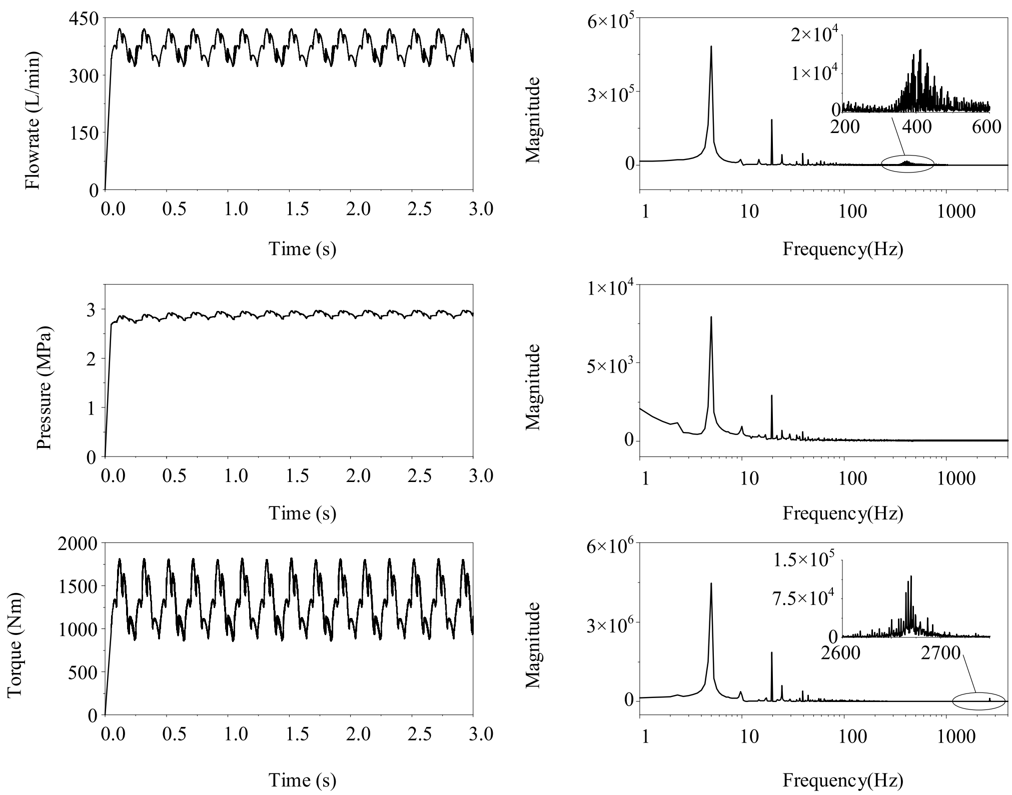

FIGURE 5. The operation characteristic of the seawater hydraulic piston pump.

group of accumulators into a seawater hydraulic piston pump and optimized the precharge pressure to reduce the pressure pulsation and vibration. Chen *et al.* [41] contributed an integral sliding mode control strategy to stabilize the pressure of hydrodynamic system with accumulator. Luo *et al.* [42] conducted a semi-active accumulator to match the requirement of parameters various under different working conditions.

However, the aforementioned methods mainly focus on the influence factors of the hydraulic piston pump structure itself

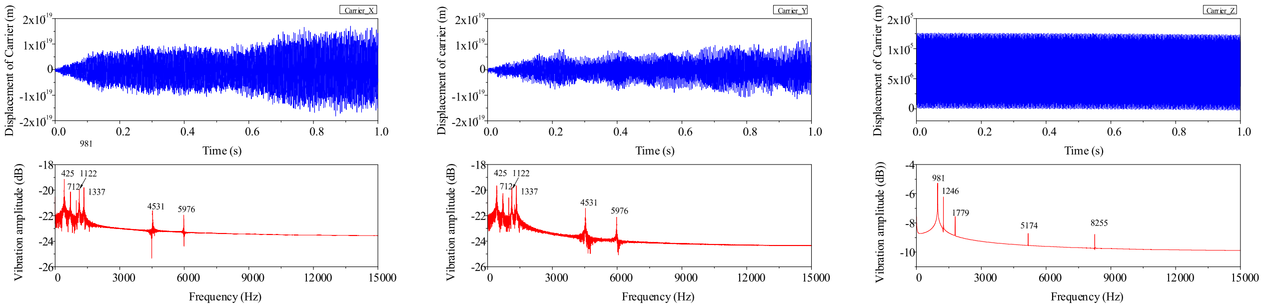

FIGURE 6. The vibration characteristic of the carrier gear.

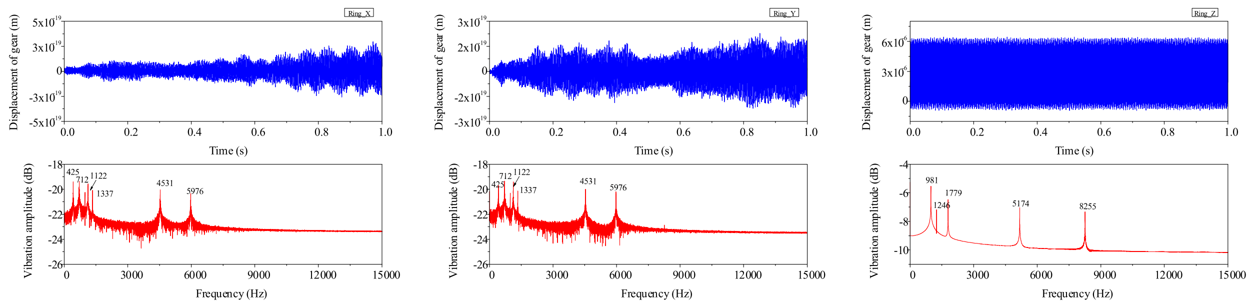

FIGURE 7. The vibration characteristic of the ring gear.

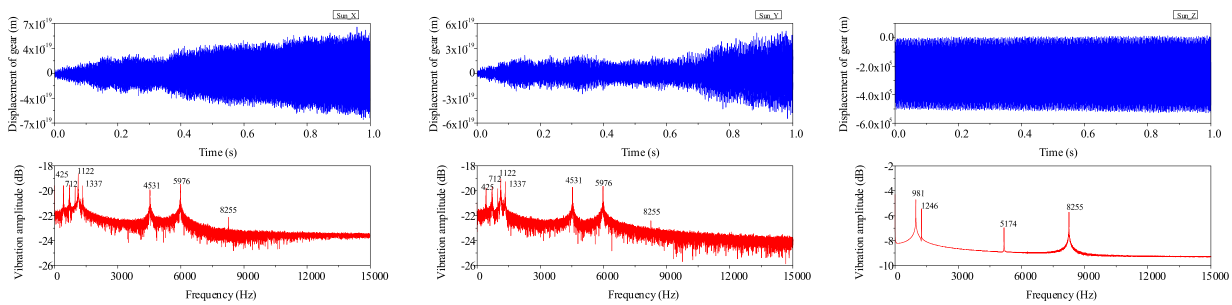

FIGURE 8. The vibration characteristic of the sun gear.

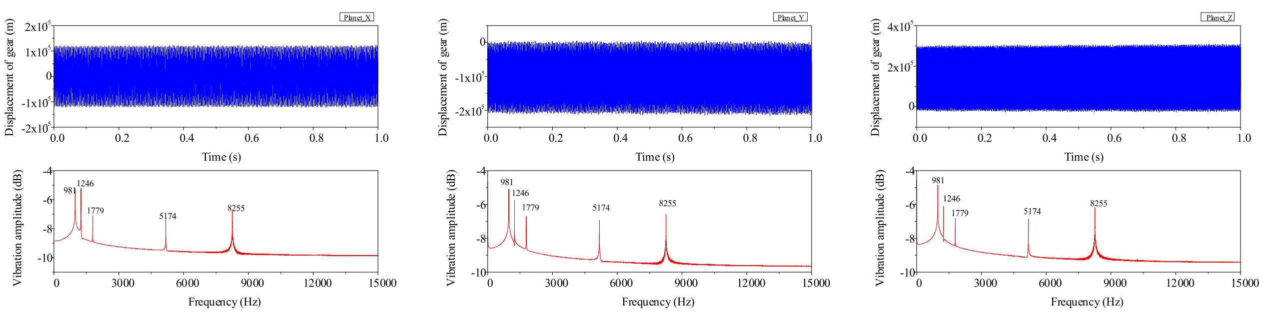

FIGURE 9. The vibration characteristic of the planetary gear.

and the solutions are just confined to utilize the optimal isolation system or optimize the pump structure. Actually, gearbox containing a large number of tooth meshing and hard contact also significantly contributes to the seawater hydraulic piston pump vibration characteristic, few literature research the vibration reduction approach from the system perspective considering the overall influence factors including the motor and gearbox. This paper analyze the vibration characteristic of a hydraulic piston pump system from system perspective. The dynamic model of the mechanic gear system was built and its influence on the vibration characteristic was analyzed and experimented. Then a new system integration scheme

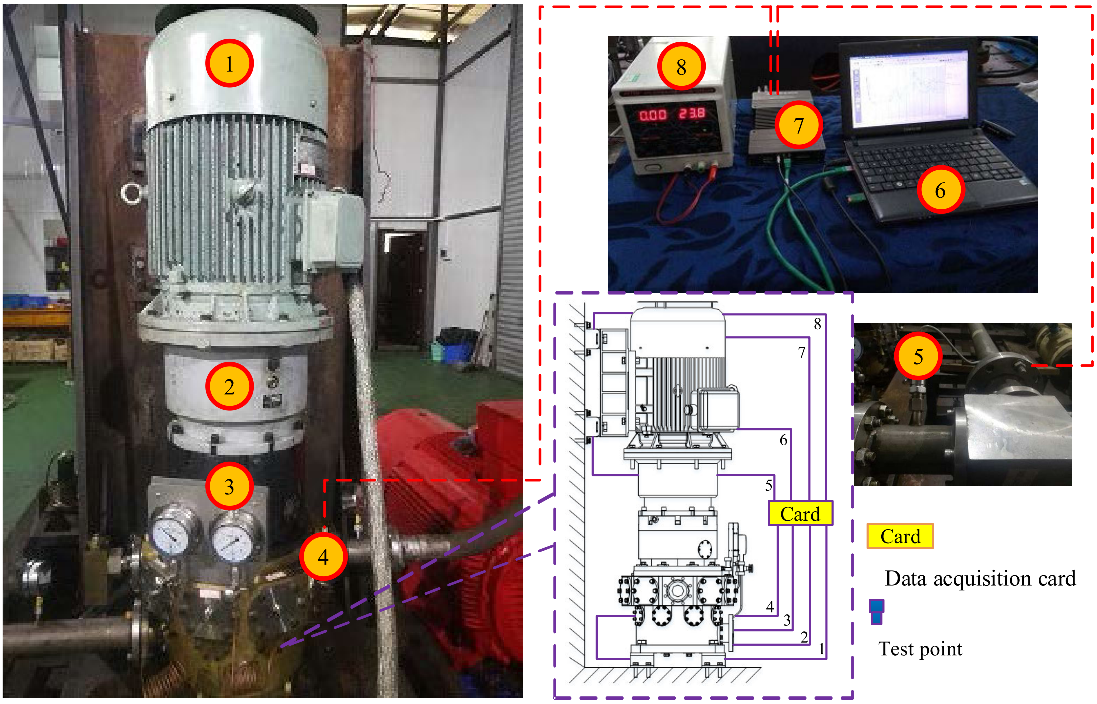

FIGURE 10. Diagram of test rig.

that a new type crankshaft seawater hydraulic piston pump driven by a permanent magnet synchronous motor instead of asynchronous motor was proposed, it eliminates the complex mechanic gear system and simplifies the system structure that would bring significant performance improvement. The comparison of the vibration characteristic between the pump system driven by permanent magnet synchronous motor and asynchronous motor was conducted to identify the advantage of the pump system driven by permanent magnet synchronous motor directly without gearbox.

II. CONFIGURATION OF SEAWATER HYDRAULIC PUMP SYSTEM

The seawater hydraulic piston pump system mainly consists of an asynchronous motor, a gearbox reducer, a seawater hydraulic piston pump and hydraulic auxiliaries such as vibration isolators, regulate valves, connect pipes, joints and monitor vogues as shown in **FIGURE 1**. The asynchronous motor is the power source component with the rated velocity of 750r/min. Because seawater has the properties of poor lubrication and low viscous in comparison with mineral oil, the friction and wear conditions of motion pairs in seawater hydraulic piston pump are very harsh. Reducing the operation velocity of the drive system within the seawater hydraulic piston pump is an important method to ensure the operation life of seawater hydraulic piston pump. In addition, lower operation velocity is of benefit to improving the vibration characteristic of the seawater hydraulic piston pump system. Installing reducer between the motor and the seawater hydraulic piston pump to reduce the operation velocity of the seawater hydraulic piston pump is a general solution in industry. The transmission ratio of the reducer is chose as 1:4.9, and it reduces the angular velocity of the asynchronous motor from 750r/min to 150r/min to adapt to the seawater hydraulic piston pump.

Considering the constrain of installing space, planetary gear trains are utilized due to its compact structure and the coaxial property of the output and input drive shafts. The planetary gear trains mainly consist of a ring gear fixing to the case, a sun gear connecting to the input shaft, 3 planet gears and a carrier connecting to the output shaft, the diagram of the planetary gear train is as shown in **FIGURE 2**, and its parameters are listed in Table 1.

III. OPERATION CHARACTERISTIC OF SEAWATER HYDRAULIC PISTON PUMP SYSTEM

The seawater hydraulic piston pump utilizes a new type transmissions mechanism to implement the motion conversation from the rotation of motor to the reciprocation of pistons, as shown in **FIGURE 3**. The crankshaft, swashplate and connecting rod are the key component of the transmission mechanism. When the crankshaft rotates, the swashplate is compelled to swing conically and the eight pistons annularly distributed in the cylinder block are driven to undergo periodical reciprocation movements. The special motion conversion mechanism makes the hydraulic piston pump couples' friction and wear condition improved to overcome the poor lubrication performance of the seawater medium. It has only one essential piston-cylinder tribology, and the lateral force between piston and cylinder is only 1/18 of traditional

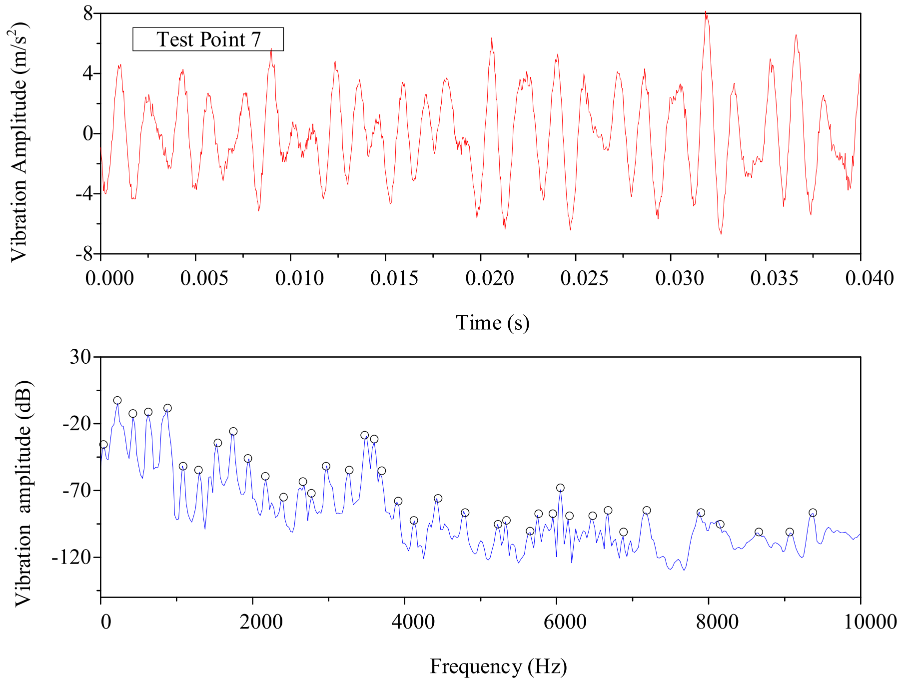

FIGURE 11. The vibration of pump system with gearbox at test point 7.

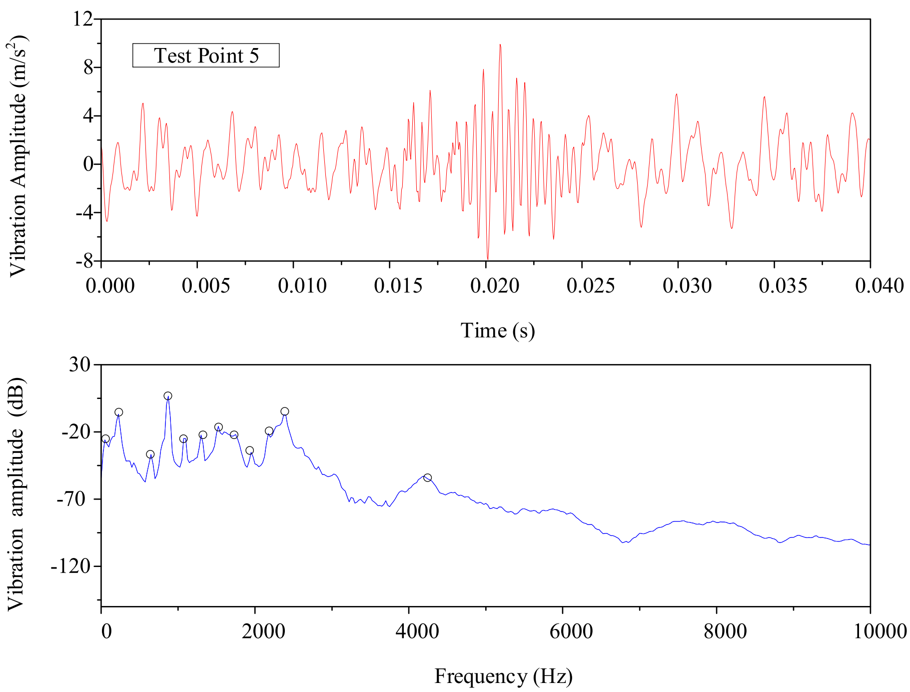

FIGURE 12. The vibration of pump system with gearbox at test point 5.

swashplate type pump [43], [44]. According to the operation principle of this piston pump, the mechanical kinematic of the pistons are derived based on the coordinate transformation theory [43], [44], as shown in Eq. (1).

$$(x_{ci}, y_{ci}, z_{ci}) = \left(x_x R\sin\theta_i + R\cos\theta_i\cos\varphi,\ y_y R\sin\theta_i,\ z_z R\sin\theta_i + R\cos\theta_i\sin\varphi\right) \quad (1)$$

where (x_{ci}, y_{ci}, z_{ci}) is the global coordinate of the connecting rods, $\theta_i = 2\pi i/8 + \pi/16$, $(i = 1, 2\ldots, 8)$. R represents the distribution circle radius of the connecting rod on the swashplate, ϕ is the crankshaft angle and $\phi = 25$, φ is the swinging angle of the swashplate referring to a fix coordinate plane, and $\sin\varphi = \cos\omega t\sin\phi \Big/ \sqrt{\cos^2\phi + \sin^2\phi\cos^2\omega t}$, $\cos\varphi = \cos\phi \Big/ \sqrt{\cos^2\phi + \sin^2\phi\cos^2\omega t}$. x_x, y_y, z_z are the first

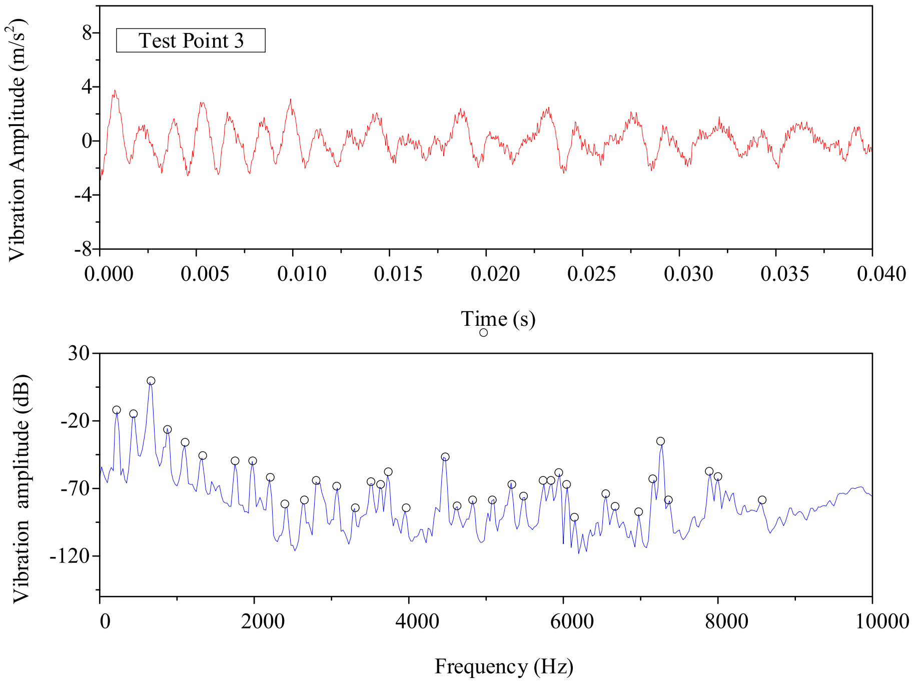

FIGURE 13. The vibration of pump system with gearbox at test point 3.

column elements of the coordinate transformation matric. $x_x = -\sin\phi\sin\varphi\sin\omega t$, $y_y = \cos\phi\cos\varphi + \sin\phi\sin\varphi\cos\omega t$, $z_z = \sin\phi\cos\varphi\sin\omega t$, ω denotes the crankshaft angular velocity.

Then, the displacements, the velocities and accelerations of the pistons are obtained. The results show that the discrete distribution of the piston would cause the periodic change of the piston motion, and the reciprocation movements of the pistons in the cylinder are different from each other, even though they are annually distributed in the cylinder block. The velocities and accelerations are also different accordingly, as shown in **FIGURE 4**. This difference in motion between the pistons is mainly caused by the motion conversion mechanism which converts the motion from the rotation of crankshaft to the reciprocation of the pistons.

The kinematic performance of the pistons would further affect the flow and pressure characteristic as well as the torque load characteristic of the seawater hydraulic piston pump, as shown in **FIGURE 5**. The flow rate, pressure and torque is fluctuant due to the periodic motion of the piston. In addition, the fluctuant amplitude at the frequency of 5Hz, 10Hz and 20Hz are relatively high, that are related to the shaft rotation velocity and the number of the pistons. The fluctuant characteristic at the frequency of 400Hz and 2700Hz is remarkable, that is the results of the nature property of the mechanical structure.

Consequently, the special motion conversion mechanism brings the risk of increasing the pressure pulsation and vibration because of the nonuniform motion of pistons. In addition, this nonuniform characteristic of the pressure pulsation and torque fluctuant would further affect the vibration performance of the planetary gear train and even the overall performance of the seawater hydraulic piston pump system.

Despite the planetary gear train reduces the operation speed of the seawater hydraulic piston pump and further improves the friction and wear condition of the hydraulic piston pump, the noise and vibration performance of the planetary gear train remains key concerns in applications. The gear mesh positions of multiple ring-planet gear pairs and sun-planet gear pairs change periodically as the planet carrier shaft rotating, which makes the vibration characteristics of the planetary gear train significantly complex. These special properties make the system integration of this novel seawater hydraulic piston pump and the planetary gearbox further increases the complexity of the vibration characteristic.

If all the components of the planetary gear train are appended with the angular frequency of the carrier $-f_{nc} = n/60$ as illustrated in Eq. (2), the exciting force consists of harmonics of the gear mesh frequency under the normal condition. Defining the rotation of the sun gear be clockwise, some feature frequencies can be calculated by following equations, and the mesh frequency would be 215Hz.

$$\begin{cases} f_{nsc} = f_{ns} - f_{nc} \\ f_{nrc} = f_{nr} + f_{nc} \\ f_{npc} = f_{np} + f_{nc} \\ f_z = z_s f_{nsc} = z_p f_{npc} = z_r f_{nrc} = z_s z_r f_{ns} / (z_s + z_r) \end{cases} \tag{2}$$

where, f_z is the mesh frequency. z_s, z_r and z_p are tooth numbers of the sun gear, ring gear and planet gear, respectively, $z_s = 22$, $z_r = 86$ and $z_p = 32$. f_{nsc}, f_{nrc} and f_{npc} are relative rotating frequencies of the sun gear, ring gear and planet gear to the carrier. f_{nc}, f_{ns}, f_{nr}, and f_{np} are absolute rotating frequencies of the carrier, sun gear, ring gear and planet gear, respectively, $f_{nc} = n/60$, $f_{ns} = (z_r/z_s + 1)f_{nc}$, $f_{nr} = 0$, and $f_{np} = (z_r/z_p - 1)f_{nc}$.

IV. THEORETICAL ANALYSIS OF THE GEARBOX VIBRATION

Despite the planetary gear possesses distinguishing advantages, its noise and vibration remain key concerns in these applications. In some engineering applications, planetary gear vibration is the primary source of cabin noise that can exceed 100 dB [45].

In a planetary gear train, the gear mesh positions of multiple ring-planet gear pairs and sun-planet gear pairs change periodically as the planet carrier shaft rotating, which makes vibration characteristics of the planetary gear train significantly differ from those of fixed-shaft gear trains.

In order to analyze the planetary gear vibration and its effect on the seawater hydraulic piston pump system, the dynamic model of the planetary gear was built. As shown in Eq. (3-9), each of the sun, ring, carrier and 3 planets are treated as rigid bodies. Component bearings are modeled by linear springs. Gear mesh interactions are represented by linear springs acting along the line of action. Each component has two translations and one rotation freedom degree. The carrier, ring and sun translations x_j, y_j (j = c, r, s) and the planet translations ζ_n, η_n (n = 1, 2, 3) are measured with respect to a rotating coordinate system $\{\mathbf{i}, \mathbf{j}, \mathbf{k}\}$ fixed to the carrier with origin O. The x_j, y_j (j = c, r, s) are directed towards the equilibrium position of planet 1, and ζ_n, η_n are the radial and tangential deflections of the n-th planet. The basis $\{\mathbf{i}, \mathbf{j}, \mathbf{k}\}$ rotates with the constant carrier angular speed ω_c. The rotational coordinates are $u_j = r_j\theta_j$ (j = c, r, s, 1, ..., n), θ_j is the component rotation; r_j is the base circle radius for the sun, ring and planet, and the radius of the circle passing through the planet centers for the carrier. Circumferential planet locations are specified by the fixed angles ψ_n, and ψ_n is measured relative to the rotating basis vector $\mathbf{i}$ so that $\psi_1 = 0$, and $\psi_n = 2\pi(\text{n-1})/n$. The position and acceleration are $r_G = x_G\mathbf{i} + y_G\mathbf{j}$, and $\ddot{r}_G = \left(\ddot{x}_G - 2\omega_c\dot{y}_G - \omega_c^2 x_G\right) i + \left(\ddot{y}_G - 2\omega_c\dot{x}_G - \omega_c^2 y_G\right) j$, (G = s, r, c, 1, 2 ..., n).

Then, the dynamic equations for the planetary gear system with masses m_c, m_r, m_s, m_p, and moments of inertia I_c, I_r, I_s, I_p are derived, and the dynamic equations for the carrier are given by:

$$\begin{cases} m_c\left(\ddot{x}_c - 2\omega_c\dot{y}_c - \omega_c^2 x_c\right) + \sum_{n=1}^{3} k_p\delta_{cnx} + k_c x_c = 0 \\ m_c\left(\ddot{y}_c + 2\omega_c\dot{x}_s - \omega_c^2 y_s\right) + \sum_{n=1}^{3} k_p\delta_{cny} + k_c y_c = 0 \\ \frac{I_c}{r_c^2}\ddot{u}_c - \sum_{n=1}^{3} k_p\delta_{cnu} + k_{ct}u_c = T_c/r_c \end{cases} \tag{3}$$

The dynamitic equations for the ring gear are obtained:

$$\begin{cases} m_r\left(\ddot{x}_r - 2\omega_c\dot{y}_r - \omega_c^2 x_r\right) - \sum_{n=1}^{3} k_{rn}\delta_{rn}\sin(\psi_{rn}) + k_r x_r = 0 \\ m_r\left(\ddot{y}_r + 2\omega_c\dot{x}_s - \omega_c^2 y_r\right) + \sum_{n=1}^{3} k_{rn}\delta_{rn}\cos(\psi_{rn}) + k_r y_r = 0 \\ \frac{I_r}{r_r^2}\ddot{u}_r + \sum_{n=1}^{3} k_{rn}\delta_{rn} + k_{rt}u_r = 0 \end{cases} \tag{4}$$

The dynamitic equations for the sun gear are given by:

$$\begin{cases} m_s\left(\ddot{x}_s - 2\omega_c\dot{y}_s - \omega_c^2 x_s\right) - \sum_{n=1}^{3} k_{sn}\delta_{sn}\sin(\psi_{sn}) + k_s x_s = 0 \\ m_s\left(\ddot{y}_s + 2\omega_c\dot{x}_s - \omega_c^2 y_s\right) + \sum_{n=1}^{3} k_{sn}\delta_{sn}\cos(\psi_{sn}) + k_s y_s = 0 \\ \frac{I_s}{r_s^2}\ddot{u}_s - \sum_{n=1}^{3} k_{sn}\delta_{sn} + k_{st}u_s = T_s/\mathrm{r_s} \end{cases} \tag{5}$$

The dynamitic equations for the planetary gears are derived:

$$\begin{cases} m_p\left(\ddot{x}_n - 2\omega_c\dot{y}_n - \omega_c^2 x_n\right) + k_{sn}\delta_{sn}\sin(\psi_{sn}) \\ \quad + k_{rn}\delta_{rn}\sin(\psi_{rn}) - k_p\delta_{cnx} = 0 \\ m_p\left(\ddot{y}_n + 2\omega_c\dot{x}_n - \omega_c^2 y_n\right) + k_{sn}\delta_{sn}\cos(\psi_{sn}) \\ \quad - k_{rn}\delta_{rn}\cos(\psi_{rn})\, k_s y_s = 0 \\ \frac{I_p}{r_p^2}\ddot{u}_n + k_{sn}\delta_{sn} - k_{rn}\delta_{rn} = 0 \end{cases} \tag{6}$$

where, $\psi_{sn} = \psi_n$- α_s, and α_s is the pressure angle of the sun-planet mesh and α_r is the pressure angle of the ring-planet mesh and $\psi_{rn} = \psi_n + \alpha_r$.

The following expressions are also provided for solving the dynamitic equation.

$$\begin{cases} \delta_{sn} = (x_n - x_s)\sin(\psi_{sn}) + (y_s - y_n)\cos(\psi_{sn}) \\ \quad + u_s + u_n + e_{sn}(t) \\ \delta_{rn} = (x_n - x_r)\sin(\psi_{rn}) + (y_r - y_n)\cos(\psi_{rn}) \\ \quad + u_r - u_n + e_{rn}(t) \end{cases} \tag{7}$$

$$\begin{cases} \delta_{cnx} = x_c - x_n - u_c\sin(\psi_n) \\ \delta_{cny} = y_c - y_n + u_c\cos(\psi_n) \\ \delta_{cnu} = (x_n - x_c)\sin(\psi_n) + (y_c - y_n)\cos(\psi_n) + u_c \end{cases} \tag{8}$$

$$\begin{cases} \psi_{sn} = \psi_n - \alpha_s \\ \psi_{rn} = \psi_n + \alpha_r \end{cases} \tag{9}$$

Assembling the system dynamic equations in matrix form yields:

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{C}\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{F}(t) \tag{10}$$

TABLE 2. Parameters of the planetary gear train.

	Sun gear	Ring gear	Planetary gear	Carrier
Mass(kg)	1.955	5.533	1.579	18.909
I/r^2(kg)	0.826	6.516	1.001	19.818
Base Radius(mm)	33	129	48	81
Mesh Stiffness(N/m)	$k_{sp} = k_{rp} = 5\times10^8$			
Bearing Stiffness(N/m)	$k_c = k_r = k_s = k_p = 10^8$			
Torsional Stiffness(N/m)	$k_{rt} = 10^9$ $k_{st} = k_{ct} = 0$			
Pressure Angle	$\alpha_s = \alpha_r = 24.6°$			

$$\mathbf{q} = [\underbrace{x_c, y_c, u_c}_{Carrier}, \underbrace{x_r, y_r, u_r}_{Ring}, \underbrace{x_s, y_s, r_s}_{Sun}, \underbrace{x_1, y_1, u_1}_{Planet\,1} \ldots \underbrace{x_n, y_n, u_n}_{Planet\ n}] \tag{11}$$

Based on the dynamitic equations, the free vibration of the linear, time-invariant representation are considered. The planets are assumed identically and equally spaced. All bearings have equal stiffness in all directions (isotropic). All planet bearing stiffnesses are equal ($k_s = k_p$), all sun-planet mesh stiffnesses are equal ($k_{sp} = k_{sp}$), and all ring-planet mesh stiffnesses are equal ($k_{rp} = k_{rp}$). With these specifications, the planetary gears are cyclically symmetric structures that can be divided into n ($n = 3$) identical sectors. Each sector has a central angle $\psi = 2\pi/n$. The specifications of the planetary gear parameter are shown in TABLE 2.

Then, the cyclic symmetry of the planetary gears would lead to the distinctive properties of natural frequency and vibration mode that will be demonstrated analytically. According to the system dynamitic equations, the Eigensolution properties of the planetary gear train are illustrated through numerical calculation with the parameters shown in TABLE 2.

The motion of the carrier, ring gear, sun gear and planetary gear are obtained and the multiplicity of the natural frequencies are derived as shown in TABLE 3. The Eigensolution properties identified in the planetary gear train are analytically shown to be true for general planetary gears. The results show there are eighteen natural frequencies derived from eighteen equations. There are six distinct nature frequency associated with the torsion vibration of the carrier, ring gear and sun gear, and the zero nature frequency refers to the rigid motion. In addition, there are six natural frequency with multiplicity 2 that relate to the transverse vibration of the carrier, ring gear and sun gear at the X-axis and Y-axis direction.

FIGURE 6 shows the vibration characteristic of the carrier in time and frequency domain. It illustrates that the transverse vibration of the carrier at X-axis and Y-axis direction is axisymmetric in time domain and they have a little difference between each other. The transverse vibration frequency is similarly because of the cyclic symmetry property of the carrier, and they both has the two-multiplicity nature frequency mode. The torsion vibration characteristic of the carrier is asymmetric due to the effect of torque load, and the torsion vibration frequency is distinct.

TABLE 3. Nature frequencies with 3 planet gears.

N	Nature Frequencies (Hz)					
m=1	0	980.8	1246	1779.1	5174.2	8255.3
m=2	424.8	711.5	1122.3	1336.8	4532.1	5976.1

The transverse vibration characteristic of the ring gear is similar to of the carrier, as shown in **FIGURE 7**. The transverse vibration at X-axis and Y-axis direction is axisymmetric in time domain, and they have a little difference between each other. The transverse vibration frequency has the two-multiplicity nature frequency mode. The torsion vibration is asymmetric in time domain and the frequency is distinct.

Same as the transverse vibration characteristic of the carrier, the transverse vibration of the sun gear at X-axis and Y-axis direction is also axisymmetric in time domain and have two-multiplicity nature frequency mode in frequency domain, as shown in **FIGURE 8**. In comparison with the carrier, the transverse vibration includes the distinct nature frequency of 8255Hz, however, the distinct torsion frequency of 1779Hz vanishes, because of the cyclic symmetry and the load sharing among planets. The torsion vibration of sun gear at the frequency of 8255Hz appears at X-direction and Y-direction. And the cyclic structure suppresses the torsion vibration at frequency of 1779Hz.

The vibration characteristic of the planet gear is different from the sun gear, ring gear and carrier. The planetary gears only have the six distinct nature frequency related to the torsion vibration, as shown in **FIGURE 9**. That is, the planetary gears only have the torsion vibration without the transverse vibration. The transverse vibration at X-axis and Y-axis direction in time domain are the projection of the torsion vibration of the planetary gears.

Consequently, the vibration characteristic of the planetary gear train is very complex due to its complex structure even the cyclic symmetric structure is of benefit to load sharing and vibration suppression. The planetary gears have both the

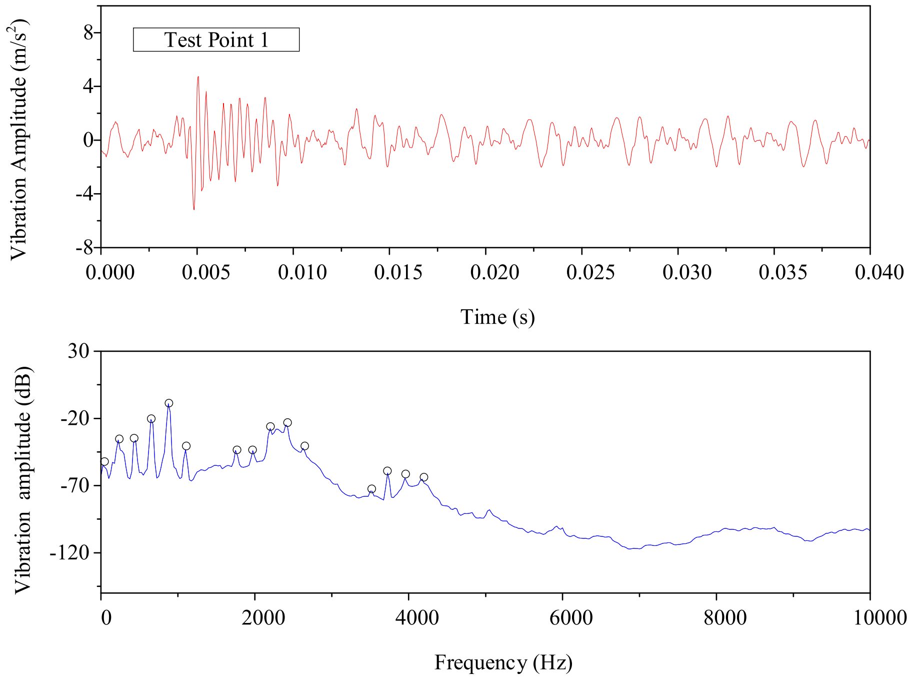

FIGURE 14. The vibration of pump system with gearbox at test point 1.

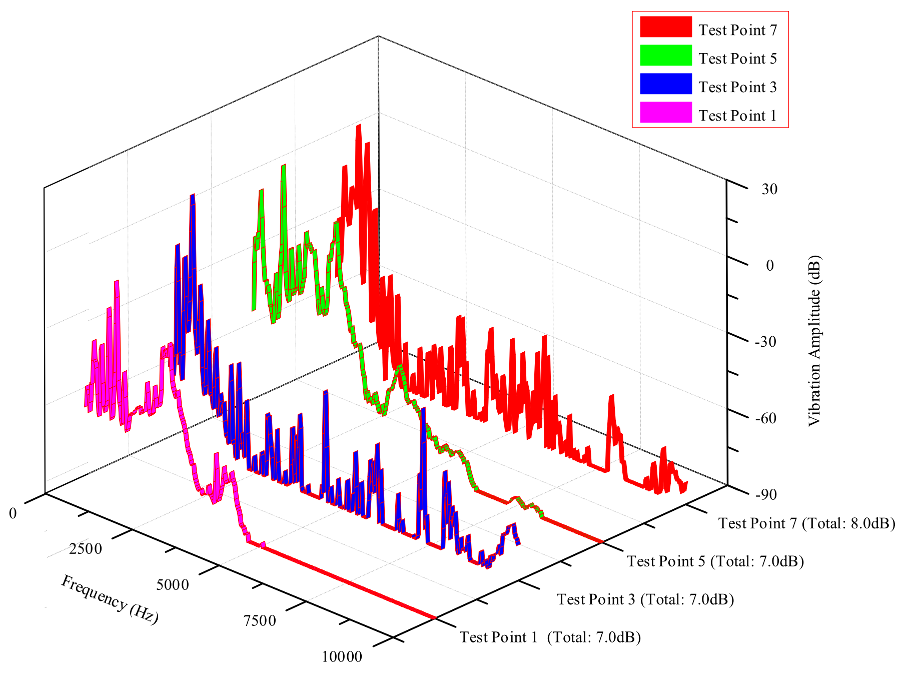

FIGURE 15. The vibration characteristic of the pump system with gearbox.

transverse vibration and torsion vibration, and the natural vibration frequency range is from the low frequency of 425Hz to the high frequency of 8255Hz. The complex vibration characteristic of the planetary gear train would further influence on the overall performance of the seawater hydraulic piston pump system when coupled with the mechanic vibration and fluid vibration excited by the seawater hydraulic piston pump in engineering application.

V. EXPERIMENTAL VIBRATION OF PUMP SYSTEM WITH GEARBOX

In order to study the overall vibration performance of the seawater hydraulic piston pump system, the experiment rig was established as shown in **FIGURE 10**. The rated speed of the AC motor is 750r/min, and it is reduced to 150r/min through the planetary gearbox with the reduction ratio of 1:4.9 to meet the requirement of the seawater hydraulic piston pump.

TABLE 4. The vibration frequency of pump system with gearbox at the test point 7.

Nature Frequencies (Hz)													
50	225	425	625	875	1100	1300	1425	1525	1725	1950	2025	2375	2625
2675	2775	2975	3250	3500	3575	3700	3900	4125	4425	4750	5000	5225	5325
5425	5650	5725	5950	6050	6150	6375	6675	7175	7875	8025	8600	8875	9375

TABLE 5. The vibration frequency of pump system with gearbox at test point 5.

Nature Frequencies (Hz)												
50	225	425	650	875	1100	1300	1525	1750	1950	2375	2175	4200

TABLE 6. The vibration frequency of pump system with gearbox at test point 3.

Nature Frequencies (Hz)												
25	225	650	875	1100	1325	1750	1975	2200	2425	2800	3300	3075
3475	3625	3725	3950	4450	4625	4825	5100	5325	5500	5750	5850	5950
6050	6150	6550	6975	7175	7275	7375	7925	7675	8000	8550		

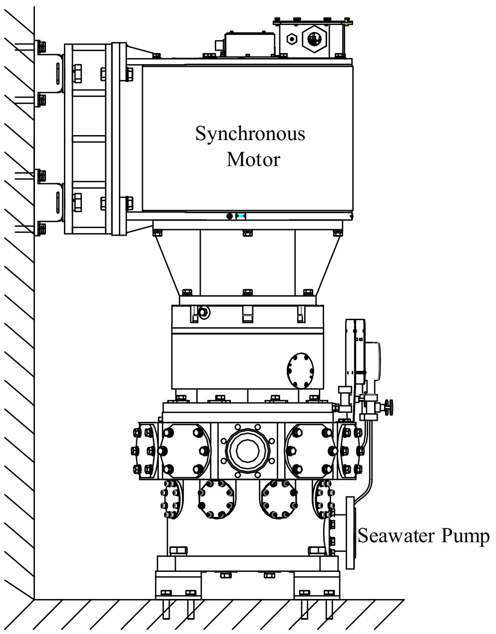

FIGURE 16. Configuration of the seawater hydraulic piston pump system driven by synchronous motor without gearbox.

The delivery pressure of the seawater hydraulic piston pump is regulated through a throttle valve. The pressure is measured through pressure transducers, and the measurement signals are collected by the data acquisition card (LAN-XI Type: 3050-B-0606ch, Brüel & Kjær) and processed through the analysis software PULSE LabShop 14.1. At the same time, 8 test points are selected on the base foundation fixing the seawater hydraulic piston pump and the AC motor to obtain the vibration acceleration signals. These test points are locating on the connection points of the pump and ship shell which are the most important vibration transmission paths from pump to ship. The accelerometers (Type: 4514-B-001, Brüel & Kjær) are also connected to the data acquisition system and processed by the analysis software PULSE LabShop 14.1. All the date are finally saved as txt-type documents and processed through MATABLE software.

FIGURE 11 shows the experiment results of the seawater hydraulic piston pump driven by an asynchronous motor and planetary gearbox at the test point 7 which is on the lateral foundation base near to the motor. The maximum vibration acceleration is below $8m/s^2$, and its vibration frequency is shown in TABLE 4. The vibration frequency contains 50Hz which is highly relative to the shaft frequency of the pump that is 5Hz, 10Hz and 20Hz as analyzed in section 2. And the 225Hz is near to the tooth mesh frequency 215Hz. In addition, the vibration amplitude at the frequency of 425Hz, 625Hz, 875Hz, 1100Hz, 1300Hz, 1425Hz, 1525Hz have relatively strong intensity that is consistent with the theoretical results of the planetary gearbox about vibration characteristic. The other frequency modes are related with the other mechanic devices such as motor and so on. The experiment result indicates that the vibration of the planetary gearbox would further dramatically influence on the vibration characteristic of the overall pump system. The vibration of the planetary gearbox is one of the main vibration excitation source of the pump system. And reducing the vibration of the planetary gearbox is an important method to improve the whole performance of the pump system.

FIGURE 12 shows the experiment results of the seawater hydraulic pump driven by an asynchronous motor and planetary gearbox at the test point 5 which is on the lateral foundation base near to the planetary gearbox. The maximum vibration acceleration is below $10m/s^2$ that is higher than the

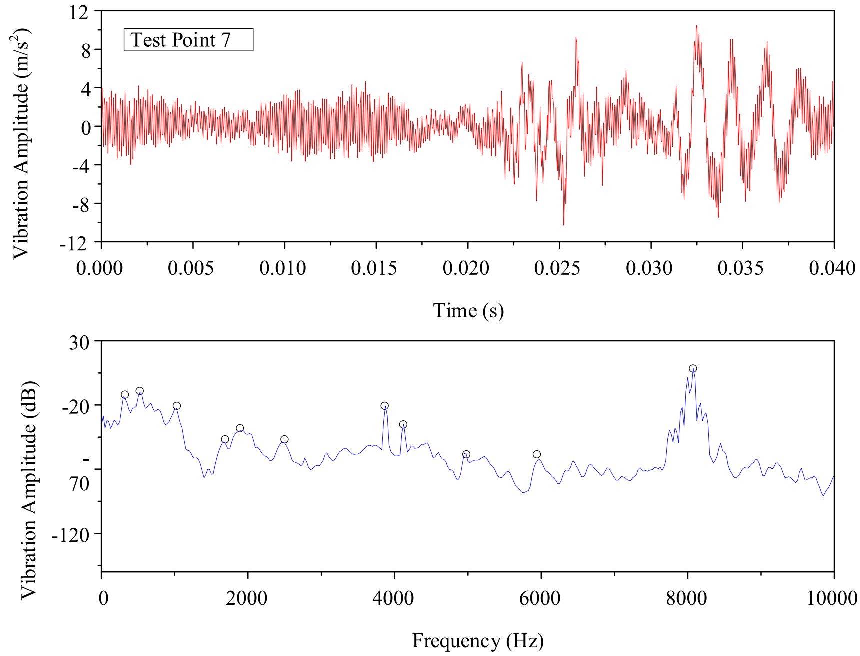

FIGURE 17. The vibration characteristic of pump system without gearbox at test point 7.

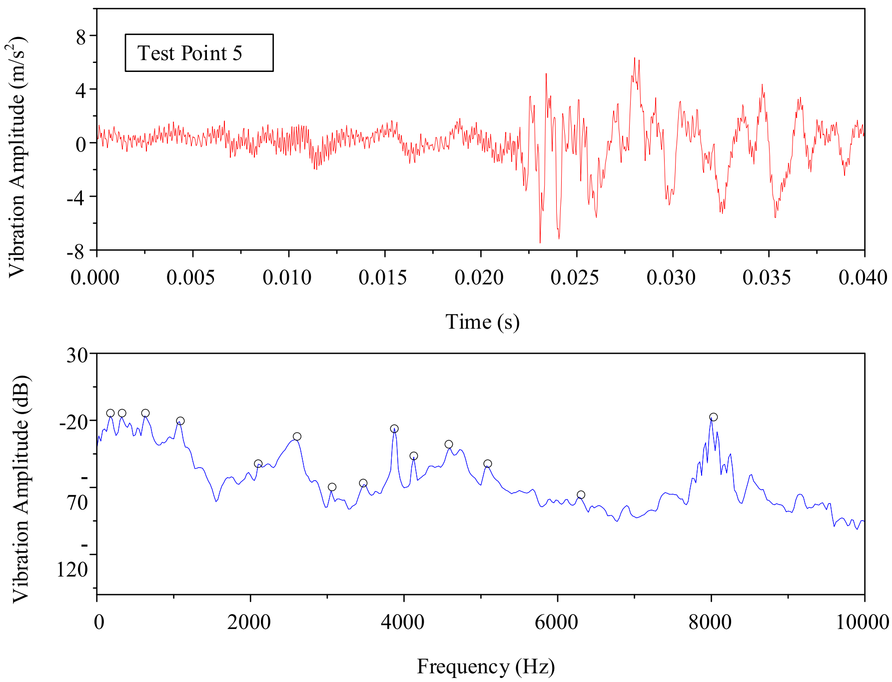

FIGURE 18. The vibration characteristic of pump system without gearbox at test point 5.

aforementioned results, however, the number of the vibration frequency sequence is less than the previous as shown in TABLE 5. The vibration frequency sequence is more close to the theoretical results of the gearbox, and it also contains the frequency mode highly related to the shaft frequency and tooth mesh frequency. The vibration of the planetary gearbox contributes more to overall pump system because this test point is more close to the planetary gearbox.

FIGURE 13 shows the experiment results of the seawater hydraulic pump driven by an asynchronous motor and planetary gearbox at the test point 3 which is on the bottom foundation base near to the seawater hydraulic pump. The maximum vibration acceleration is below 4m/s^2 that is lower than the previous results due to the effect of the pump system mass itself on the vibration reduction. However, its frequency characteristic is relative complex because of the synthetically

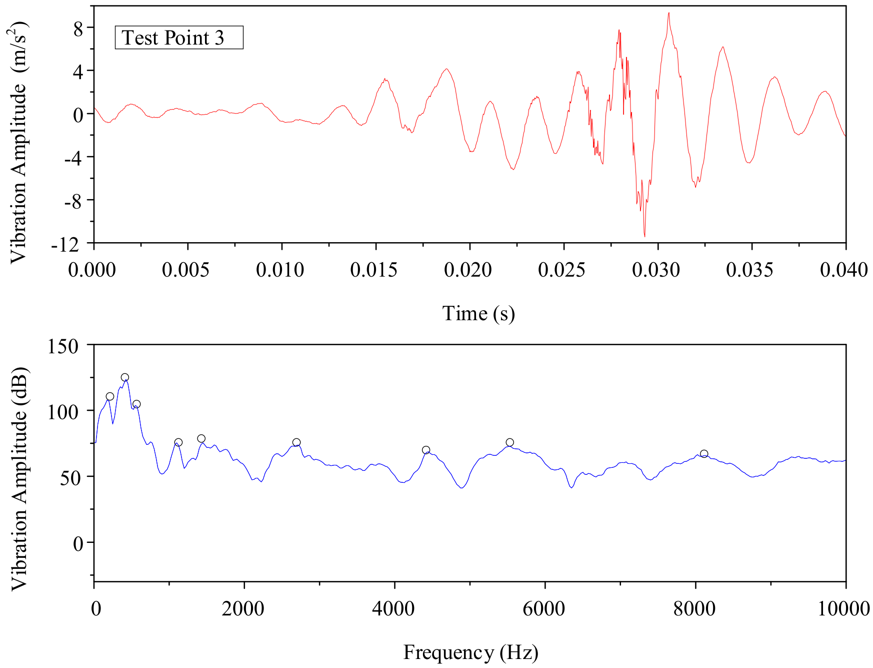

FIGURE 19. The vibration characteristic of pump system without gearbox at test point 3.

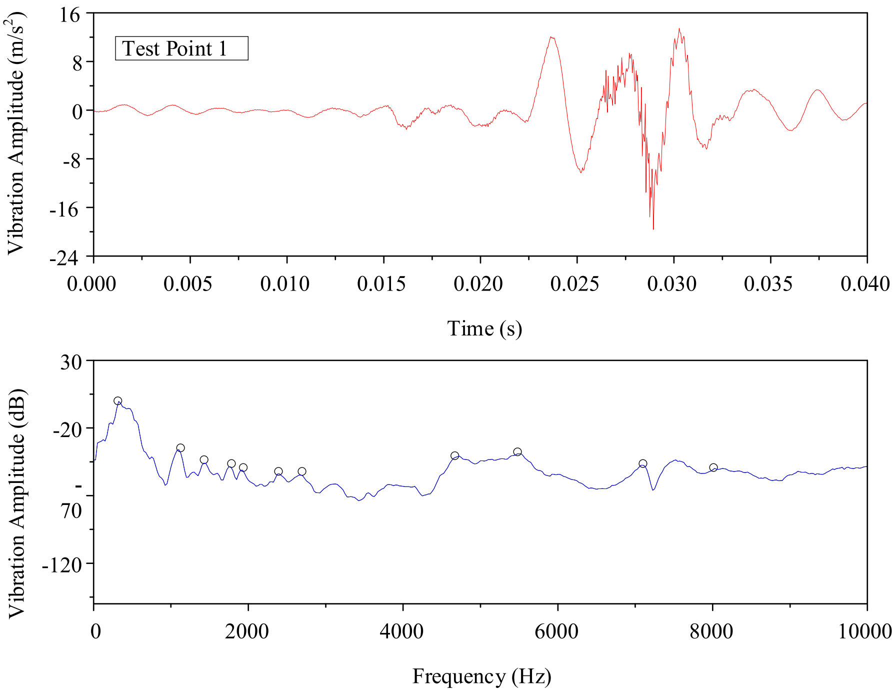

FIGURE 20. The vibration characteristic of pump system without gearbox at test point 1.

influence of the fluid vibration and the mechanical vibration of the pump system, as shown in TABLE 6. The vibration frequency of 25Hz is highly correlated with the shaft frequency, and the vibration at high frequency range is much remarkable. The experiment results of the vibration frequency presents strong relevance with the planetary gearbox.

FIGURE 14 shows the experiment results of the seawater hydraulic pump driven by an asynchronous motor and planetary gear box at the test point 1 which is on the bottom foundation base near to the seawater hydraulic pump. The maximum vibration acceleration is below 4m/s^2 and its vibration frequency is relative simply as shown in TABLE 7. The vibration is also strongly correlated with the planetary gearbox.

Furthermore, the acceleration signals at the test points are obtained and processed, the average vibration acceleration

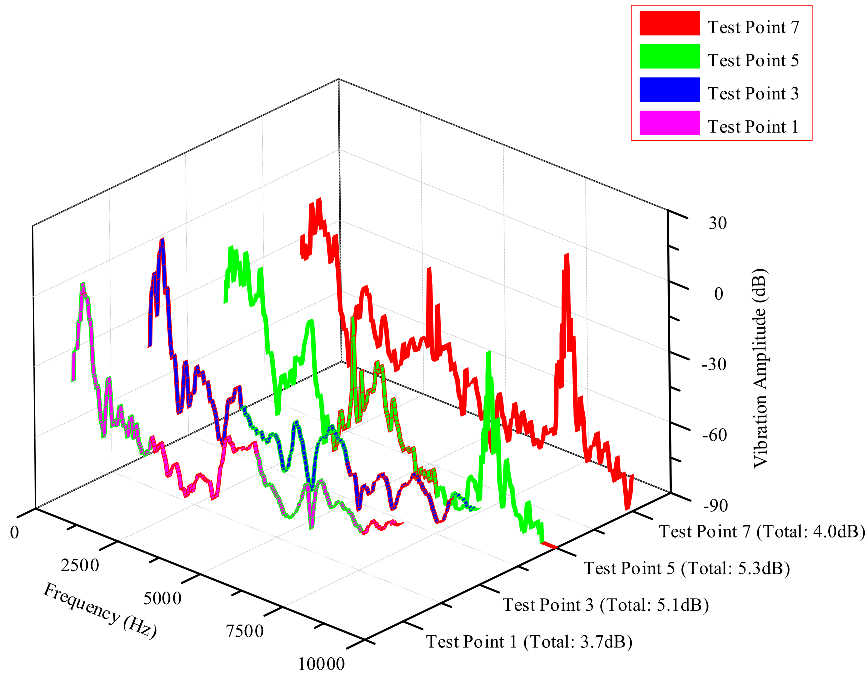

FIGURE 21. The vibration characteristic of pump system without gearbox.

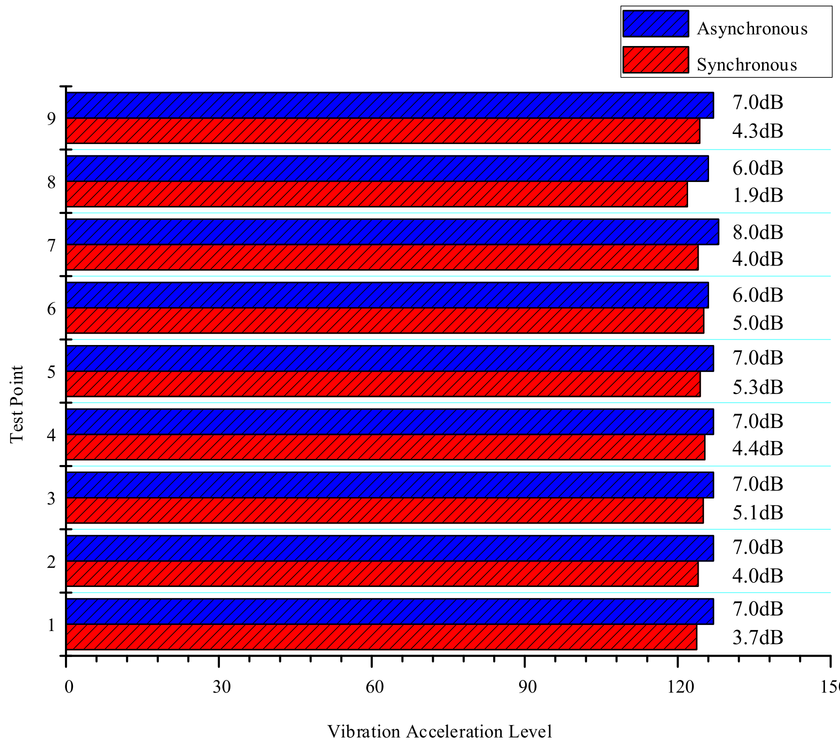

FIGURE 22. The comparison of the average vibration accelerate level.

levels is derived according to the power average method.

$$L = 10\log_{10}\left[\frac{1}{8}\sum_{i=1}^{8}10^{L_i/10}\right] \tag{12}$$

where, L is the average of the vibration acceleration levels, L_{i} is the vibration acceleration at i test point.

FIGURE 15 illustrates the average vibration acceleration level of the seawater hydraulic pump system at the rated pressure. The vibration frequency range is from 1Hz to 10000Hz, and the vibration at low frequency range (1Hz~2000Hz) is dominant. The vibration amplitude at high frequency range is also relatively remarkable because of the complex structure of the pump system. The average vibration acceleration at

TABLE 7. The vibration frequency of pump system with gearbox at test point 1.

Nature Frequencies (Hz)														
50	225	650	875	1100	1300	4175	1750	3950	1975	2200	2400	2625	3500	3725

TABLE 8. Parameters of the synchronize motor.

Item	Parameters
Working Frequency	35~60Hz
Carrier Frequency	8000Hz
Rotary velocity	130~225r/min
Torque	0~2000N.m
Poles	16

different test point have little difference, and the average vibration acceleration is about 7dB, that is relatively high for the engineering application especially for underwater equipment. Reducing the vibration acceleration level of the seawater hydraulic pump system is an impotent requirement to promote the application of the seawater hydraulic pump system.

VI. EXPERIMENTAL VIBRATION OF PUMP SYSTEM WITHOUT GEARBOX

Due to the tremendous contribution of the planetary gearbox on the vibration accelerate of the pump system, eliminating the gearbox has become an important method on reducing the vibration of pump system. A pump system driven by a permanent magnet synchronous motor eliminating the gearbox is conducted as shown in **FIGURE 16**. The pump system is directly driven by a multipole permanent magnet synchronous motor which has the properties of low rotary velocity and high torque output. The detail parameters are listed in TABLE 18, the permanent magnet synchronous motor can operate at the low velocity range of 130∼225r/min. consequently, the seawater hydraulic piston pump can be driven by the permanent magnet synchronous motor without the gearbox. Furthermore, its vibration performance is researched through experiment.

FIGURE 17 shows the experiment results of the seawater hydraulic piston pump driven by a permanent magnet synchronous motor directly at the test point 7 where is close to the motor. The maximum vibration acceleration is below 10m/s^2, and its vibration frequency is shown in TABLE 9. The vibration frequency is very different from the characteristic of the pump system with a planetary gearbox. The frequency sequence is much simple and the frequency characteristic related to the gearbox is disappear. The vibration component mainly comes from the mechanical structure of the permanent magnet synchronous motor and seawater hydraulic piston pump dismissing the effect of the gearbox.

Particularly, the frequency component of 4000Hz and 8000Hz are special which is coincide with the carrier frequency of the driver. That is, the carrier frequency of the driver amplifies the vibration accelerate of the seawater hydraulic piston pump system at the component of 8000Hz. The carrier frequency of the driver mainly affects the characteristic of the permanent magnet synchronous motor and further influence the whole vibration characteristic of the pump system.

FIGURE 18 shows the vibration characteristic of the seawater hydraulic piston pump system at test point 5 where is closed to the seawater hydraulic piston pump. The maximum vibration acceleration is below 10m/s^2. The vibration frequency has more vibration frequency component as shown in TABLE 10 in comparison with the test point 7 because of the complex structure of the seawater piston pump. These vibration frequency components are mainly excited by the permanent magnet synchronous motor and seawater hydraulic piston pump without the effect of the gearbox. The vibration excitation of the hydraulic piston pump is very important and the frequency of 8000Hz is remarkable at test point 5.

The vibration frequency characteristic at point 3 is relative simple than that at point 7 and point 5 due to the effect of the pump material on the vibration absorption, as shown in **FIGURE 19** and TABLE 11. The test point 3 at the bottom foundation is relative far away from the permanent magnet synchronous motor and the vibration energy with high frequency could dissipate during the vibration transferring. In addition, the test point 3 is more close to the seawater hydraulic piston pump, so the fluid vibration and mechanical vibration of the seawater hydraulic piston has more significantly effect on the vibration characteristic at test point 3. In addition, the mass of the pump system is high and the vibration at the bottom could be somewhat suppressed. Consequently, the vibration component of 8000Hz is unremarkable at point 3. The vibration frequency characteristic at test point 1 similar to at the test point 3, as shown in **FIGURE 20**, and the vibration frequency is shown in TABLE 12. The vibration component of 8000Hz is also not remarkable.

The vibration characteristic of the pump system at the test points are shown in **FIGURE 21**. The total vibration accelerate level is respectively 3.7dB, 5.1dB, 5.3dB and 4.0dB. The vibration amplitude at the frequency of 8000Hz is remarkable at the point 5 and 7 where is close to the motor. The carrier frequency of the driver mainly affects the characteristic of the permanent magnet synchronous motor and further influences on the whole vibration characteristic of the seawater hydraulic piston pump system. Consequently, the vibration component at the frequency of 8000 Hz is remarkable at the test point near to the permanent magnet synchronous motor. The property of the permanent magnet synchronous motor has more influence on the vibration results at the test point 7 and 5. The vibration component at the 8000 Hz is unremarkable at the test point 1 and 3 located on the bottom

TABLE 9. The vibration frequency of pump system without gearbox at test point 7.

Nature Frequencies (Hz)										
300	525	1025	1675	1925	2575	3875	4125	4975	5975	8000

TABLE 10. The vibration frequency of the pump system without gearbox at test point 5.

Nature Frequencies (Hz)													
175	325	625	1075	1975	2575	4875	3050	3475	3875	4125	4600	6275	8000

TABLE 11. The vibration frequency of pump system without gearbox at test point 3.

Nature Frequencies (Hz)								
175	425	550	1075	1450	2725	4450	5525	8000

TABLE 12. The vibration frequency of pump system without gearbox at test point 1.

Nature Frequencies (Hz)										
325	1100	1425	1750	1900	2425	2675	4725	5525	7100	8000

foundation base close to the seawater hydraulic piston pump because of the dissipation of the vibration energy. The fluid vibration and mechanic vibration of the seawater hydraulic piston pump has significant effect on the vibration characteristic at the test point 1 and 3.

Comparatively, the frequency component of the pump system driven by a permanent magnet synchronous motor directly is much simple than that driven by an asynchronous motor and a planetary gearbox. The vibration characteristic frequency of the pump related to the planetary gearbox is disappeared. It means that the significant vibration source excited by the planetary gearbox is effectively eliminated. Consequently, the vibration performance of the pump system could be greatly improved.

Furthermore, the vibration amplitude of the pump system would significantly decrease when utilizing a permanent magnet synchronous motor as the power source eliminating the planetary gearbox. Both the total vibration acceleration level at the test point and the average vibration are reduced. As shown in **FIGURE 22**, the total vibration acceleration level at the test point is decreased from 7.0dB, 7.0dB, 7.0dB, 7.0dB, 7.0dB, 6.0dB, 6.0dB, 7.0dB to 3.7dB, 4.0dB, 5.1dB, 4.4dB, 5.3dB, 5.0dB, 4.0dB, 1.9dB, and the average vibration acceleration level decrease from 7.0dB to 4.3dB. The vibration performance is significantly improved.

One of the essential reasons for the significant improvement of the vibration performance of the crank shaft seawater hydraulic pump system utilizing synchronous motors in comparison with the asynchronous motors lies in the simplifies of the mechanical system and the elimination of the planetary gearbox. Through using the synchronous motor, the dramatic influence of planetary gearbox on the pump system vibration is dismissing and the vibration excitation source is reduced. So the frequency component becomes much simpler and the vibration amplitude is significantly decreased. Evidently, the utilization of permanent synchronous motor could also bring some other significant advantages, Such as, the synchronous motor is able to work in closed loops speed control mode and closed loops torsion control mode to provide more stable and non-slip power input which is of benefit to improving the vibration performance. The controllable capability of the permanent magnet synchronous motor could provide more potential valuable method for vibration reduction.

VII. CONCLUSION

As one of the most important properties, vibration would directly influence the performance of seawater hydraulic pump and further determine the stability, reliability and stealth of underwater equipment. This paper proposes a new system integration concept that a new type crankshaft seawater hydraulic pump driven by a permanent magnet synchronous motor instead of asynchronous motor considering the dramatic effect of planetary gearbox on the vibration characteristic from systemically perspective. The dynamic model of the pump and seawater hydraulic piston pump and gearbox are built and the effect of the planetary gearbox on the vibration characteristic is analyzed. The theoretical analysis results indicate that there are eighteen natural frequencies from the low frequency of 425Hz to the high frequency of 8255Hz. And the planetary gears have both the transverse vibration and torsion vibration. There are six single nature frequency associated with the torsion vibration of the carrier, ring gear and sun gear. And there are six natural frequency with multiplicity 2 related to the transverse vibration of the carrier, ring gear and sun gear at the X-axis and Y-axis direction. That would greatly enlarge the vibration of the seawater hydraulic piston pump. Furthermore, the experiment of the seawater hydraulic piston pump system driven by asynchronous motor and gearbox is accomplished. The vibration frequency of the experiment results contains the frequency component of the gearbox, the asynchronous motor and the seawater hydraulic piston pump. And the vibration frequency component of the theoretical results are all obtained in experiment. The average vibration acceleration is as high as 7dB. The experiment results identify the dramatically influence of the gearbox on the vibration performance of the pump system. Therefore, the seawater hydraulic piston pump system driven by synchronous motor without gearbox is proposed and the experiment studies is achieved. The experiment results indicate that the total vibration acceleration level at the test point and the average vibration are reduced. The total vibration acceleration level at the test point is decreased from 7.0dB, 7.0dB, 7.0dB, 7.0dB, 7.0dB, 6.0dB, 6.0dB, 7.0dB to 3.7dB, 4.0dB, 5.1dB, 4.4dB, 5.3dB, 5.0dB, 4.0dB, 1.9dB, and the average vibration acceleration level decrease from 7.0dB to 4.3dB. The frequency component of the pump system driven by a permanent magnet synchronous motor directly without gearbox is much simple than that driven by an asynchronous motor and a planetary gearbox, even the frequency component of 8000Hz is remarkable due to the effect of the carrier frequency of the driver. Eliminating the planetary gearbox and simplifying the mechanical system of the crank shaft seawater hydraulic pump system is one of the essential reasons for the significant improvement of the vibration performance by utilizing synchronous motors instead of the asynchronous motors. Through using the synchronous motor, the vibration source excited by the planetary gearbox is reduced. So the frequency component becomes much simpler and the vibration amplitude is significantly decreased. In conclusion, utilizing permanent magnet synchronous motor as power source is an effective method to reduce the vibration of the crankshaft seawater hydraulic piston pump. This paper provides a novel universal strategies to improve the vibration performance of the pump system. It would promote the application of seawater hydraulic technology in underwater equipment. At the same time, the synchronous motor possessing the controllable capabilities can provide more stable and non-slip power input which could lead to some potential method to further improve the vibration performance. Optimizing the control strategies and control parameters to improve the vibration performance of the new type crankshaft seawater hydraulic pump system is our research priorities in future.

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ZUTI ZHANG was born in China, in 1986. He received the B.S. degree in mechanical engineering from Northwest A&F University, Xi'an, China, in 2010, and the Ph.D. degree in mechanical engineering from the Huazhong University of Science and Technology, Wuhan, China, in 2017.

From 2017 to 2019, he was a Postdoctoral Researcher with the School of Naval Architecture and Ocean Engineering, Huazhong University of Science and Technology. Since 2020, he has been an Associate Professor with the School of Power and Mechanical Engineering, Wuhan University. His research interests include water hydraulic piston pump, water hydraulic technology, high-pressure waterjet propulsion technology, cavitation, fluid mechanical, and vibration and sound control.

YUNZHI LIANG was born in China, in 1996. He received the B.S. degree in energy and power engineering from Wuhan University, Wuhan, China, in 2019, where he is currently pursuing the master's and Ph.D. degrees in energy and power engineering with the School of Water Conservancy and Hydropower.

His research interests include the CFD simulation of waterjet and cavitation, jet pump, jet theory and application, and vibration and sound control.

KAI ZHAO was born in China, in 1997. He received the bachelor's degree in mechanical engineering from Wuhan University, Wuhan, China, in 2019, where he is currently pursuing the master's degree with the School of Water Resources and Hydropower.

His research interests include water jet technology, jet pump and hydraulic cavitation, fluid mechanical, and vibration and sound control.

XINPING LONG was born in China, in 1967. He received the B.S., M.S., and Ph.D. degrees in energy and power engineering from the Wuhan University of Hydraulic and Electric Engineering, Wuhan, China, in 1995.

From 1994 to 1996, he was a Lecturer with the Wuhan University of Hydraulic and Electric Engineering. From 1996 to 1998, he was a Visiting Scholar with the University of Minnesota, USA. From 1998 to 2000, he was a Visiting Scholar with The University of Manchester, U.K. Since 2004, he has been a Professor with the School of Power and Mechanical Engineering, Wuhan University. His research interests include water hydraulic piston pump, water hydraulic technology, high-pressure waterjet propulsion technology, cavitation, fluid mechanical, and vibration and sound control. He is an Editorial Board of *Journal of Drainage and Irrigation Machinery Engineering*.

HANYANG LIU was born in China, in 1989. He received the B.S. degree in electromechanical engineering from Huazhong Agricultural University, Wuhan, China, in 2012, and the master's degree in mechatronic engineering from the Huazhong University of Science and Technology, Wuhan, in 2015.

From 2015 to 2018, he was a Mechanical Engineer with SEC-KSB Nuclear Pumps and Valves Company Ltd., Shanghai, China. Since 2018, he has been a Mechanical Engineer with the Wuhan Maritime Communication Research Institute, Wuhan. His research interests include water hydraulic technology, fluid mechanical, high-pressure water jet cutting technology, and flow-induced vibration.