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# Research on Thermodynamic Characteristics of Hydraulic Power Take-Off System in Wave Energy Converter

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**Abstract:** Hydraulic power-take-off (PTO) systems which utilize high-pressure oil circuits to transmit energy are widely applied in wave energy generation. The properties of hydraulic oil are significantly influenced by environmental conditions, and its dynamic viscosity is sensitive to temperature, especially in relatively low-temperature cases. This paper studies the characteristics of the hydraulic PTO when started in different temperature conditions via numerical analysis and experimental verification. An improved numerical model of the hydraulic PTO system is proposed, in which the effects of temperature on the hydraulic oil viscosity and hydraulic motor efficiency are quantitatively investigated, and consequently, the thermal-hydraulic characteristics can be sufficiently considered. The performances of the hydraulic PTO in start-up processes with different initial temperatures and in long term operation are assessed. The results show that the presented model can reasonably describe the hydraulic PTO characteristics. The efficiency of hydraulic PTO degrades when it starts at low temperatures. The efficiency increases in relatively high temperature, while larger fluctuations of the flow rate and output power are observed. This study can provide guidance for enhancing the efficiency and consistency of hydraulic PTO operating in actual sea conditions.

**Keywords:** thermal-hydraulic characteristics; hydraulic oil viscosity; hydraulic PTO; wave energy converter



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## 1. Introduction

As a widespread renewable energy source, ocean wave energy has gradually received extensive attention from scientists since the oil crisis broke out in the 1870s [1,2]. Waves have irregular motion and their power is largely discontinuous, which makes it difficult to obtain wave energy. An important feature of ocean waves is that they have the highest energy density among all renewable energy sources [2]. If wave energy can be effectively used, it will make a great contribution to alleviating the world energy crisis [3].

Some European countries were the first to carry out the research of wave energy conversion technology. At present, a large number of wave energy conversion devices have been designed and manufactured, some of which have been tested in actual sea conditions [2]. The wave energy converters are usually composed of a wave energy capture (WEC) system and a power take-off (PTO) system. There are many ways to classify wave energy converters currently, according to the different working principles of wave energy captures. They can be mainly divided into three types, the oscillating water column, overtopping, and oscillating body [4]. The role of the WEC system is to absorb wave energy, while the role of the PTO system is to convert the energy captured by the WEC system into relatively consistent mechanical energy and then outputs electrical energy through the generator. Therefore, the PTO system is the core part of the wave energy converter. There is no doubt that it has become the research focus of many universities and energy institutions [5]. In general, the typical PTO system can be equipped with four types of drive mechanisms including air turbines, low-head hydraulic turbines, linear generators, and hydraulic oil circuits [4–6]. Air turbines are mainly adopted in oscillating water column WEC due their special working principles [4]. Their structures are reliable, however, the compressibility of air and the inevitable deviation of the actual rotation speed to the optimal value may easily degrade their efficiencies [7,8]. With high efficiency and reliability, low-head turbines are mainly applied to overtopping devices. Nevertheless, their large-scale applications are still impeded by the inconvenient constructions of the overtopping WECs. The linear generators are also broadly utilized in WECs, which primarily benefits from their low mechanical complexity. However, the linear generator structures are vulnerable to damages from vibrations and impacts, especially in real irregular waves and extreme waves. In addition, the power outputs from the linear generators fluctuated, and consequently, the quality of the converted electricity cannot satisfy the criteria. In contrast, hydraulic PTO systems are frequently employed in oscillating body WECs, and they use hydraulic oil circuits to transmit energy, which can cope with low frequency-large torque inputs of waves and overload protection. Meanwhile, their advantages of stabilizing power output are attractive for engineering applications. However, the performances, e.g., the pressure consistency, the flow rate consistency, and the efficiency of the hydraulic PTO, are sensitive to operating conditions, and the aforementioned performances may significantly deteriorate when the system states deviate from the optimal states. Consequently, we focus on hydraulic PTO in this paper and investigate the influences of several key elements to their performances [9,10].

In the past few decades, universities and scientific research institutions have carried out a large amount of research work on hydraulic PTO systems, including structural design and system optimization [9]. Stephen Salter proposed a kind of wave energy generation device named the nodding duck [10]. A buoyant pendulum wave energy generation system was invented by Lancaster University in the United Kingdom [11]. Zhejiang University designed a kind of pendulum wave energy generation device [12]. The Guangzhou Energy Research Institute of the Chinese Academy of Sciences created the duck-type and eagle-type wave energy conversion devices [13]. The University of Edinburgh in the United Kingdom put forward a wave energy power generation device based on a new hydraulic transmission system [14,15]. Gaspar presented some incremental modifications to the PTO architecture, such as using more cylinder ports, installing more parallel cylinders, and assembling an oil bypass circuit [16]. The consistency and efficiency of the wave energy conversion device are two key factors that determine whether the wave energy can be effectively used. The PTO system in the Pelamis wave energy converter realizes the torque control of the hydraulic cylinder by improving the one-way valve into a control valve block, thereby achieving the purpose of high-efficiency energy capture [17,18]. Falcao introduced and implemented a latching control method to increase the captured mechanical energy by the hydraulic PTO [19]. Liu et al. improved the power capture ability of a two-raft-type WEC by optimizing several parameters of the hydraulic PTO, including the area of the piston, the displacement of the hydraulic motor, and the effective damping of the generator [20]. Cargo et al. realized the maximization of the generated power in different wave period situations by optimizing the size of the hydraulic motor [21]. Chen and Jiang verified that the parameter settings of the key components play an important role in the consistent operation of the hydraulic PTO system through simulation and experiments [9]. Yue proposed integrated characteristic curves in constant voltage conditions, which are used to optimize system design and improve the operating efficiency of the PTO system [22]. Geng proposed a novel hydraulic PTO power module consisting of a pressure compensator and throttle valve by adjusting the parameters to improve the consistency of hydraulic motor speed and output power [23]. The focus of the research on the consistency and efficiency of the hydraulic PTO system is focused on how to select pivotal components and optimize parameters and control strategies, which ignored the influence of the characteristics of the working medium on the consistency of the output power and efficiency when the wave energy converters are running.

From the perspective of hydraulic oil characteristics, this paper studies the influence of temperature on the performance of the hydraulic PTO system during operation. The viscosity of hydraulic oil is an important property to measure the performance of hydraulic oil. If the viscosity of the hydraulic oil is too high, the internal friction force of the hydraulic oil will be relatively large when it flows, which will generate more heat energy during the energy transmission process and increase the total energy consumption. If the viscosity of hydraulic oil is too low, the leakage will increase, leading to a decrease in system pressure and affecting the consistency of the system. The viscosity of hydraulic oil is significantly affected by temperature changes [24]. Therefore, it is of great significance to study the effect of viscosity on the consistent operation and efficient work of the PTO system when it starts at different temperatures. According to the results of simulated analysis, the viscosity of hydraulic oil varies significantly within the range of −20 °C to 20 °C [24], considering that the seawater temperature in the sea area where electricity can be generated is above 0 °C, this paper selects the ambient temperature within the range of 0~20 °C to research the impact on the performance of the hydraulic PTO system.

The rest of this paper is organized as follows. Section 2 introduces the working principle of the hydraulic PTO system for a pendulum wave energy converter and the digital model of a hydraulic motor and simulation of hydraulic PTO system. Section 3 presents the components of the prototype test platform. The experimental results and analysis are arranged in Section 4. Finally, the corresponding conclusions are showed in Section 5.

## 2. Modeling and Simulation of Thermal Characteristics of Hydraulic PTO System

The hydraulic PTO system of a wave energy converter is mainly composed of a gear and rack mechanism, a single-acting hydraulic cylinder with single rod, four check valves, a relief valve, a high-pressure accumulator, a flow control valve, a permanent magnet synchronous generator, a hydraulic motor, resistor loads, and an oil tank [20]. The working principle is shown in Figure 1. The floating pendulum converts wave energy into its own mechanical energy and moves around the fulcrum [4]. The mechanical energy of the floating pendulum is transmitted to the single-acting horizontal bar hydraulic system by using the meshing gear and rack mechanism. Then the piston rod drives the piston to reciprocate to perform work on the hydraulic oil, which completes the conversion from mechanical energy to hydraulic energy. The four check valves integrate the two-way flow of hydraulic oil at the rod-less cavity oil port of the hydraulic cylinders into a one-way flow in the main oil circuit through rectification. The variable displacement hydraulic motor converts the hydraulic energy into rotating mechanical energy, which is then converted into electrical energy output by permanent magnet synchronous generator [3,25].

The main functions of the accumulator are to store energy, stabilize pressure, eliminate periodic fluctuations of the system pressure and flow generated when the cylinder is reversing, and enhance the consistency of the system. The overflow valve can avoid the high pressure in the pipeline, and the function of the flow control valve is to control the flow rate and participate in the control of the power; the oil tank is used to compensate for the leakage and short-term negative pressure that may occur in the system and to supplement the oil pressure in time [9,22,23,26].

### 2.1. The Temperature-Hydraulic Oil Viscosity-Efficiency Model of Hydraulic Motors

Hydraulic oil is most commonly used in hydraulic transmission, and most of the published literatures about hydraulic oil are selected in combination with specific engineering machinery. There are few theoretical studies on the external characteristics of hydraulic oil. The main physical properties of hydraulic oil include density, viscosity, elastic modulus, surface tension, etc. From the perspective of fluid movement and transmission force, viscosity is the main factor to prevent the flow of hydraulic oil. Therefore, it is of great significance to study the influence of fluid viscosity on the performance of the PTO system.

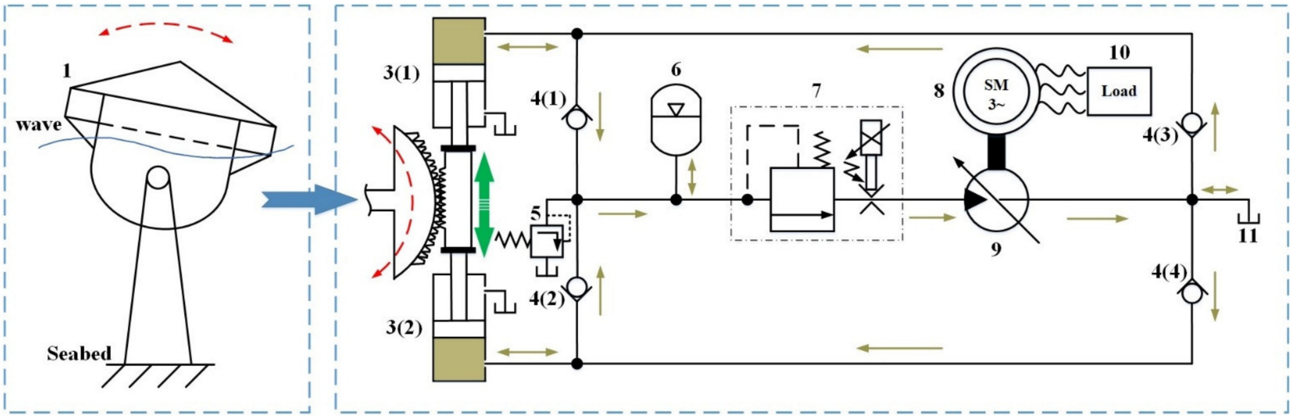


**Figure 1.** Schematic diagram of a hydraulic PTO system with a pendulum (**1**) wave energy capture device (**2**) rack and pinion mechanism (**3**) single-acting hydraulic cylinder with single rod (**4**) check valve (**5**) relief valve (**6**) high pressure accumulator (**7**) flow control valve (**8**) permanent magnet synchronous generator (**9**) hydraulic motor (**10**) resistor load (**11**) oil tank.

The law of internal friction proposed by Newton believes that the relative movement of two adjacent layers inside the flowing fluid produces internal friction, and its magnitude is proportional to the viscosity of the fluid, the velocity gradient of the relative movement, and the contact area. The mathematical expression is:

$$F = \mu A \frac{du}{dy} \tag{1}$$

The corresponding shear stress can be expressed as:

$$\tau = \frac{F}{A} = \mu \frac{du}{dy} \tag{2}$$

where $\mu$ refers to the dynamic viscosity of hydraulic oil, $Pa \cdot s$, $\tau$ is the internal friction shear stress of hydraulic oil, $m^2/s$, $A$ means the contact area of hydraulic oil between two layers, $m^2$, and $du/dy$ presents the velocity gradient of hydraulic oil, $s^{-1}$.

The greater the dynamic viscosity of the fluid is, the stronger the stickiness generated when it moves. The normal pressure has little effect on the viscosity of the fluid and can be almost ignored, while the temperature has a great influence on the viscosity of the fluid. The reason is that the viscosity of the fluid mainly comes from the attraction of molecules. The hydraulic oil belongs to the Newtonian fluid, so it conforms to the Newton internal friction law. In a constant temperature condition, the viscosity coefficient of this kind of fluid does not change, which is an oblique straight line through the origin of the coordinate in the $\tau \sim du/dy$ coordinate system. When the temperature increases, the molecular distance increases, the attractive force decreases, and the shear stress generated by the same shear deformation rate decreases; therefore, the dynamic viscosity decreases. The relationship between hydraulic oil viscosity and temperature can be expressed as:

$$\mu = \mu_{T_0} e^{-\lambda(T - T_0)} \tag{3}$$

where $\mu_{T_0}$ is the dynamic viscosity of hydraulic oil when $T$ is equal to $T_0$, and $Pa \cdot s$, $\lambda$ refers to the hydraulic oil viscosity coefficient.

According to the literature, the relationship between hydraulic motor torque loss and hydraulic oil viscosity [27] is:

$$M_L = K_\mu \mu n + K_P \Delta p + K_{\rho PSP} n^3 + \frac{K_{\rho VN} \Delta p \sqrt{\Delta p}}{n} + K_{P2} \Delta p^2 + M_{Lo} \tag{4}$$

where $K_\mu$, $K_P$, $K_{\rho PSP}$, $K_{\rho VN}$, $K_{P2}$, and $M_{Lo}$ are moment-loss-related performance characteristic coefficients, $n$ represents speed of the hydraulic motor, $r/min$, $\Delta p$ represents the pressure difference between inlet and outlet of hydraulic motor, *Pa*, $\rho$ stands for hydraulic fluid density, and $kg/m^3$, $M_L$, and $M_{Lo}$ delegate the torque loss and fixed torque loss of hydraulic motor, $N{\cdot}m$.

The relationship between hydraulic motor leakage loss and viscosity [27] is:

$$Q_L = C_\mu \frac{\Delta p}{\mu} + C_v n + C_{VN}\sqrt{\frac{\Delta p}{\rho}} + C_{PP}\frac{\rho n^3}{\Delta p} + C_C\frac{n\Delta p}{\beta} + Q_{Lo} \tag{5}$$

where $C_\mu$, $C_v$, $C_{VN}$, $C_{PP}$, $C_C$, and $Q_{Lo}$ are leakage-related performance characteristic coefficients, $n$ represents the speed of hydraulic motor, $r/min$, $\Delta p$ represents the pressure difference between inlet and outlet of hydraulic motor, *Pa*, $\rho$ stands for hydraulic fluid density, and $kg/m^3$, $Q_L$, and $Q_{Lo}$ delegate respectively leakage loss and fixed leakage loss of hydraulic motor, *L/min*.

The mechanical efficiency of a hydraulic motor can be expressed as:

$$\eta_m = \frac{M - M_L}{M} = 1 - \frac{M_L}{M} \tag{6}$$

where $\eta_m$ stands for the hydro-mechanical efficiency and $M$ is the theoretical output torque of a hydraulic motor, $N{\cdot}m$.

The volumetric efficiency of a hydraulic motor can be expressed as:

$$\eta_v = \frac{Q}{Q + Q_L} = \frac{1}{1 + \frac{Q_L}{Q}} \tag{7}$$

where $\eta_v$ stands for the volumetric efficiency and $Q$ represents the theoretical flow rate of a hydraulic motor, *L/min*.

The mechanical power loss of the hydraulic motor can be expressed as:

$$\varnothing_1 = 2\pi n M_L \tag{8}$$

The volumetric power loss of the hydraulic motor can be expressed as:

$$\varnothing_2 = \Delta p Q_L \tag{9}$$

When the temperature rises, the viscosity of the hydraulic oil decreases according to Formulas (3), (4), (6) and (8). Consequently, the torque loss of the hydraulic motor will decrease, which will increase the mechanical efficiency of the hydraulic motor. The mechanical power loss of the hydraulic motor is correspondingly reduced. It can be seen from Formulas (3), (5), (7) and (9) when the temperature rises, the viscosity of the hydraulic oil decreases, which will cause the leakage loss of the hydraulic motor to increase, thus the volumetric efficiency of the hydraulic motor decreases, and volumetric power loss of the hydraulic motor increases accordingly. The influence of viscosity on the efficiency of hydraulic motors mainly includes the influence of its mechanical efficiency and volumetric efficiency. It is very necessary to control the viscosity of hydraulic oil within a suitable range for the efficient operation of the PTO system. Therefore, this paper focuses on the influence of the viscosity of the hydraulic oil with the rise of temperature on mechanical loss, volumetric loss of the hydraulic motor, and system efficiency.

### *2.2. Simulation Model of Hydraulic PTO System*

The hydraulic PTO system simulation model was established in AMESim, a software used for the modeling, simulation, and dynamic analysis of hydraulic systems, it is shown in Figure 2. The parameters for the simulation model are based on the design of the experiment platform and are shown in Table A1 of Appendix A.

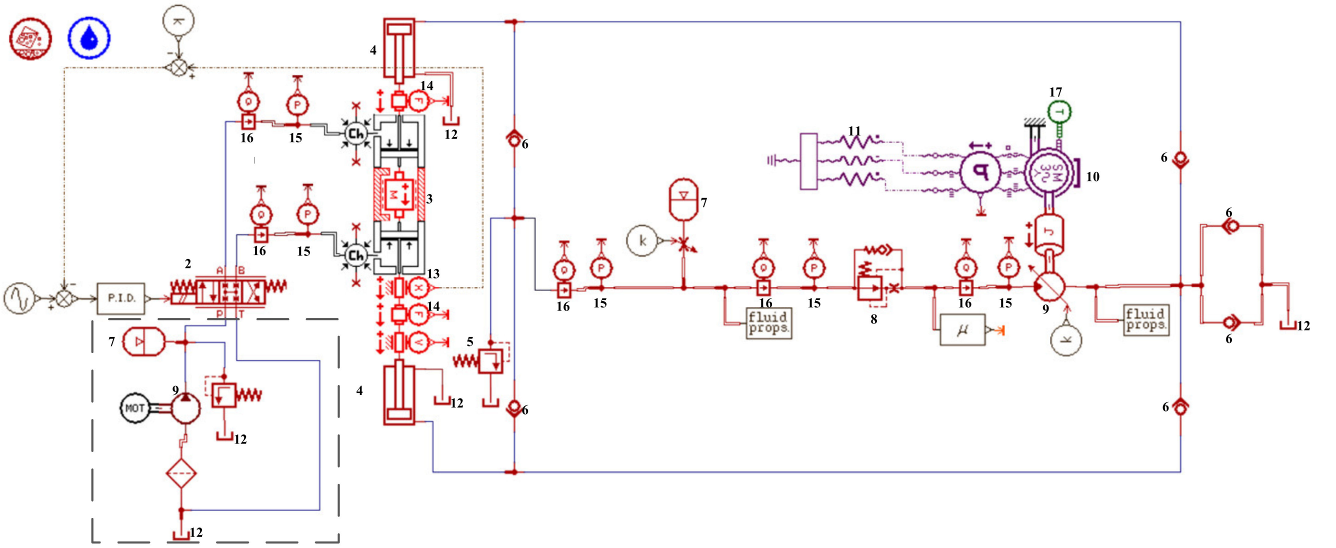

**Figure 2.** Schematic diagram of the experimental platform. (**1**) A pump station, (**2**) three-position and four-way reversing valve, (**3**) double-acting hydraulic cylinder with double rods, (**4**) single-acting hydraulic cylinder with single rod, (**5**) relief valve, (**6**) check valve, (**7**) high pressure accumulator, (**8**) flow control value, (**9**) hydraulic motor, (**10**) permanent magnet synchronous generator, (**11**) resistor loads, (**12**) oil tank, (**13**) a linear variable displacement transducer, (**14**) bidirectional force transducers, (**15**) pressure gauge, (**16**) flow rate sensor, (**17**) Torque tachometer.

Because the range of wave energy period is 6–8 s and the significant wave height distributes between 1.5 m and 3.5 m [23], we chose the typical sea state where the period was 8s, and the wave height was 1.5 m to carry out the simulation and experiments.; then controlling the pressure of the system kept in 7 MPa, and another sea state whose period was 6s and wave height was 1.5 m. The designed output powers of the hydraulic system were respectively about 1.2 KW and 1.5 KW.

## 3. Prototype Test Platform of Hydraulic Power Take-Off System

The verification test was completed by building a hydraulic PTO system prototype platform, shown in Figure 3, which consists of a drive system, a hydraulic PTO system, a data acquisition and monitoring system, and loads. The drive system, which simulates the motion of the wave energy capture device, realizes the reciprocating motion of the piston in the double-hydraulic-chamber double-rod cylinder through a hydraulic pump station and controls the direction of the piston by adjusting the three-position and four-way reversing valve, which guides the high-pressure oil into different chambers, and finally achieves the input of wave energy at different periods. The hydraulic PTO system utilizes the reciprocating movement of two single-acting hydraulic cylinders with single rods that are solidly connected to the double-acting hydraulic cylinder with double rods to discharge high-pressure oil from the rod-less cavity of the cylinders. The check valves integrate the two-way flow of hydraulic oil into a one-way flow, and then hydraulic oil enters the hydraulic motor through the flow control valve to drive the generator, which is rigidly connected with the motor to rotate. This process realizes the conversion of mechanical energy to electrical energy. The monitoring and control system was developed based on the LabVIEW software. A human-computer interaction interface was established in LabVIEW, and the serial communication technology of the computer and the programmable controller was applied to realize the condition monitoring of the PTO system and the real-time collection of data such as tubing pressure, system flow, hydraulic oil viscosity, and hydraulic motor speed and system while realizing the storage of data by using the database system.

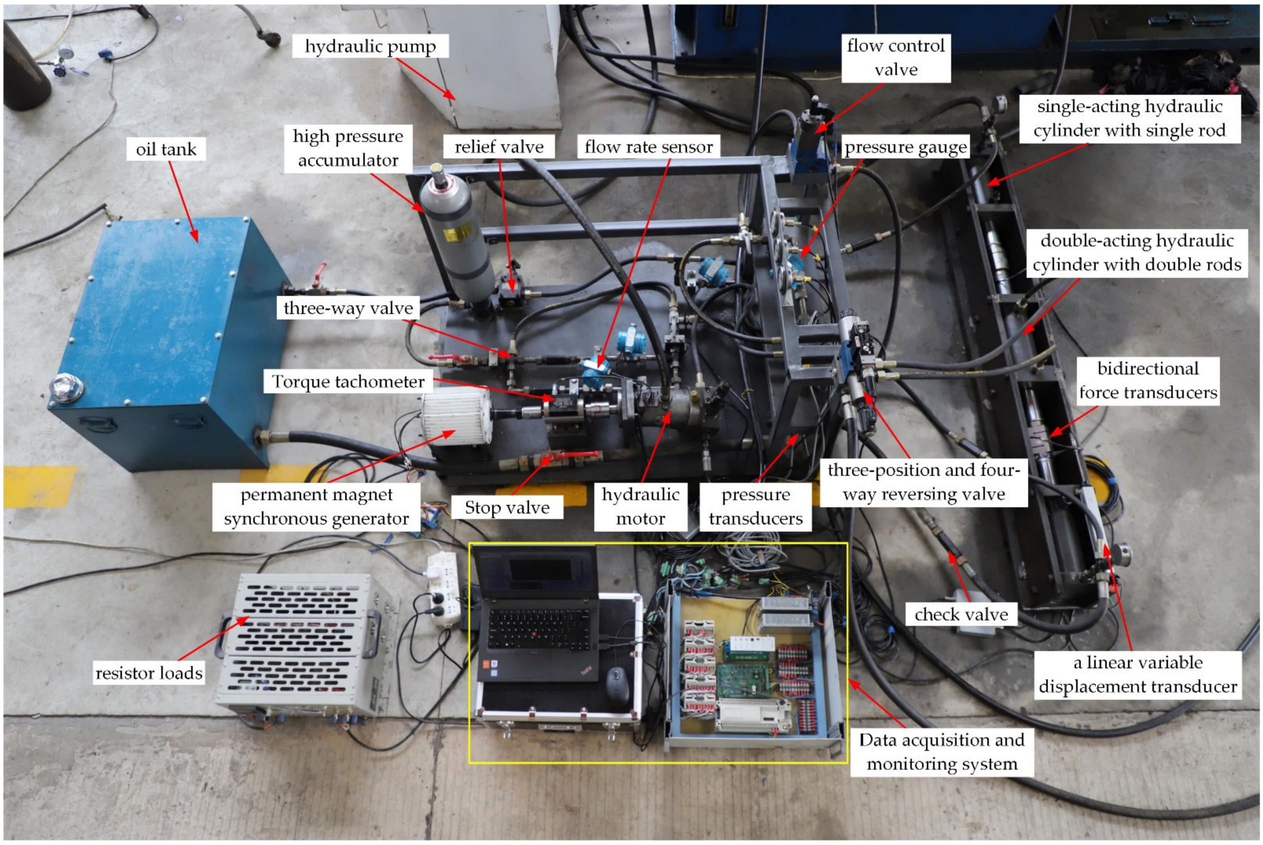


**Figure 3.** The composition of the experimental platform.

The research procedure was arranged as follows. Firstly, a prescribed piston motion with a period of 8 s and an amplitude of 0.075 m was used to validate the numerical model in this paper, and a comparison in system pressure was made, flow at variable temperatures and output power at 18 °C. Then the wave conditions with a period of 6 s and an amplitude of 0.075 m was used numerically to study the consistency of the PTO system. Finally, a long-running test was carried out at the temperature of 6 °C. Its purpose was to study the effect of temperature rise on the consistency and efficiency of the hydraulic PTO system.

The main component parameters are determined according to the designed working conditions, as shown in Table 1.

**Table 1.** Parameters of main components.

| Components | Parameters | Value | Unit |
|---|---|---|---|
| Three-position and four-way reversing valve | Maximum pressure | 31.5 | MPa |
| | Maximum flow-rate | 180 | L/min |
| Double-acting double-rod cylinder | Maximum pressure | 31.5 | MPa |
| | Piston diameter | 90 | mm |
| | Rod diameter | 63 | mm |
| | Stroke | 200 | mm |
| Single-acting single-rod cylinder | Maximum pressure | 31.5 | MPa |
| | Piston diameter | 80 | mm |
| | Piston area | 0.005 | $m^2$ |

**Table 1.** *Cont.*

| Components | Parameters | Value | Unit |
|---|---|---|---|
| | Rod diameter | 63 | mm |
| | Stroke | 200 | mm |
| Check valve | Maximum pressure | 31.5 | MPa |
| High-pressure accumulator | Maximum pressure | 31.5 | MPa |
| | Nominal volume | 6.3 | L |
| | Pre-charge gas pressure | 4.5 | MPa |
| Relief valve | Maximum cracking pressure | 31.5 | MPa |
| Flow control valve | Maximum pressure | 31.5 | MPa |
| | Adjuconsistent flow-rate range: | 0~25 | L/min |
| Hydraulic motor | Maximum pressure | 35 | MPa |
| | Maximum flow-rate | 206 | L/min |
| | Maximum displacement | 54.8 | mL/r |
| | Minimum displacement | 16.4 | mL/r |
| Low-pressure tank | Volume | 175 | L |
| Magnetic powder brake | Torque range | 0~100 | Nm |
| | Maximum power | 10 | kW |
| Bidirectional force transducer | Force range | −100~100 | kN |
| Displacement transducer | Displacement range | 0~225 | mm |
| Flow-rate transducer | Flow-rate range | 3.3~20 | L/min |
| Pressure transducer | Pressure range | 0~25 | MPa |
| Torque-speed transducer | Torque range | 0~100 | Nm |
| | Speed range | 0~1000 | r/min |

## 4. Results and Discussion

This section begins with experiments to validate the hydraulic PTO model proposed in Section 3. The experiments were conducted in a hydraulic PTO test platform and the ECD movements were simplified as prescribed piston motions. The PTO system start up tests were carried out at the ambient temperature of 6 °C, 12 °C, and 18 °C. Temperature is thought to be almost immutable when the PTO system starts up within a short of time. Thus, we studied the influence of different start-up temperatures on the characteristics of the PTO system. The system pressure, flow, and output power are tested in the numerical model and experimental conditions.

### *4.1. Simulation and Experiments of Hydraulic PTO System at Different Temperatures*

The wave with a period of 8 s and an amplitude of 0.075 m was used to carry out the simulation and experiments. The pressure changes of the hydraulic PTO system within one minute of the startup process are shown in Figure 4. The three pictures (a), (b), and (c) respectively represent the corresponding system flow when the startup temperature is 6 °C, 12 °C, and 18 °C. It can be seen that the simulation results obtained by using the model in Section 3 generally agree with those obtained by experiments at different startup temperatures, °C, the system pressure dropped at the beginning. The analysis found that the accumulator was not working because of the low temperature at the beginning and the relatively high viscosity of the hydraulic oil. When the ambient temperature was 12 °C or 18 °C, there was little difference at the beginning. The system pressure in experimental conditions cannot reach the charging pressure because the compressibility of hydraulic oil was not considered in this model. However, the system pressure in the experimental condition was consistent with that in the simulation eventually.

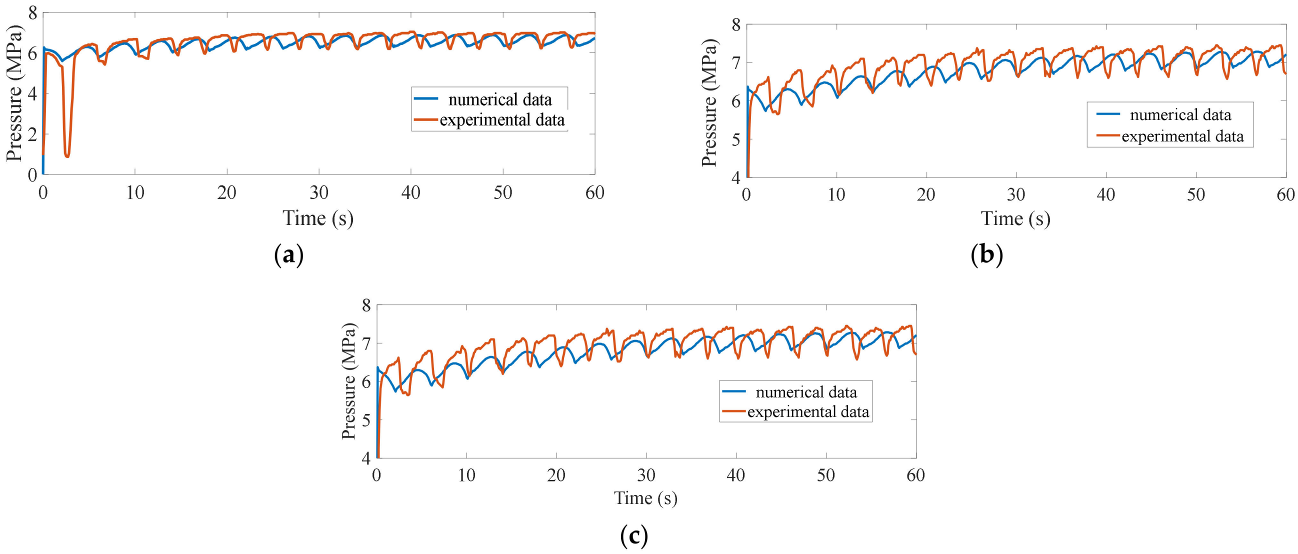


**Figure 4.** Comparison and experiment of hydraulic PTO system pressure between simulation at different temperatures. (**a**) ambient temperature is 6 °C. (**b**) ambient temperature is 12 °C. (**c**) ambient temperature is 18 °C.

The flow differences of the hydraulic PTO system at different temperatures in the start-up process are shown in Figure 5. The tested system flow is higher when the temperature is higher, close to the simulation results. Hence, within a certain temperature range, the higher the temperature the hydraulic oil is at the startup process, the greater the system flow rate will be.

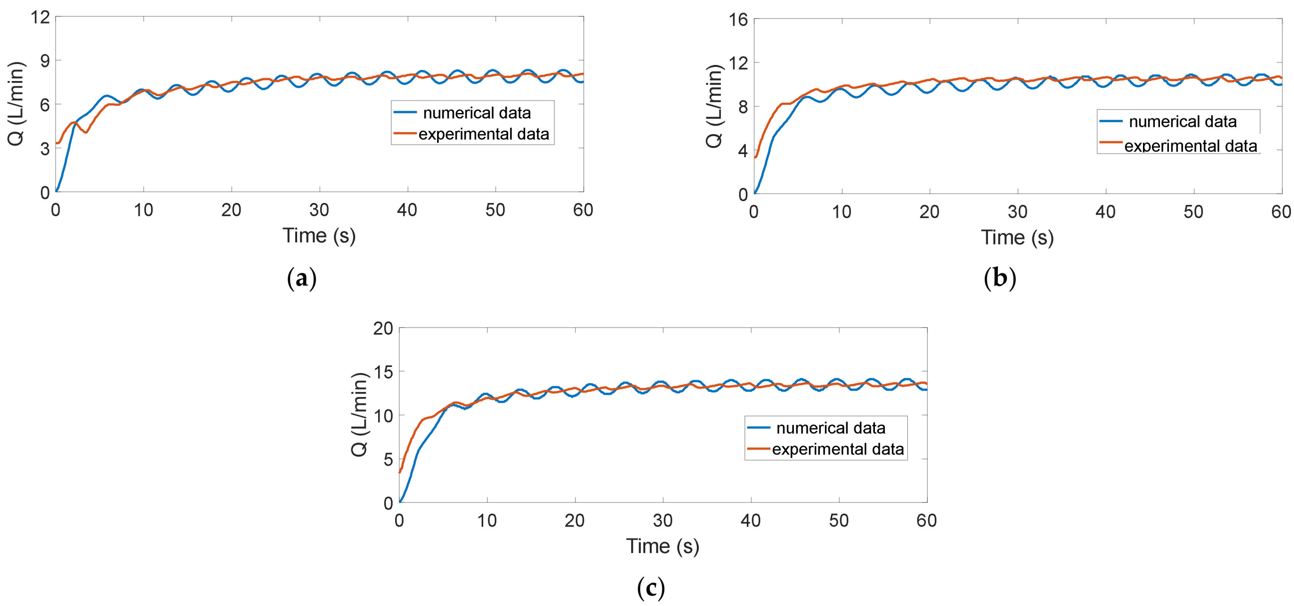


**Figure 5.** Comparison of hydraulic PTO system flow between simulation and experiment at different temperatures. (**a**) ambient temperature was 6 °C. (**b**) ambient temperature was 12 °C. (**c**) ambient temperature was 18 °C.

The output power changes of the hydraulic PTO system within one minute of the startup process are shown in Figure 6. Picture (a) presents the comparison of system output power at 18 °C between the numerical model and the experimental conditions, and they are generally on the way up, though they varied at the beginning. Picture (b) demonstrates that although the output power of the PTO system increased in volatility, the average value only reached about 550 W when the initial temperature was 6 °C, which was much lower than the set system output power of 1200 W. In addition, the average value reached about 720 W and 1100 W when the initial temperature was 12 °C and 18 °C separately. Therefore, within the range of 6 °C to 18 °C, the higher the initial temperature, the greater the output power of the system, and it is closer to the designed output power.

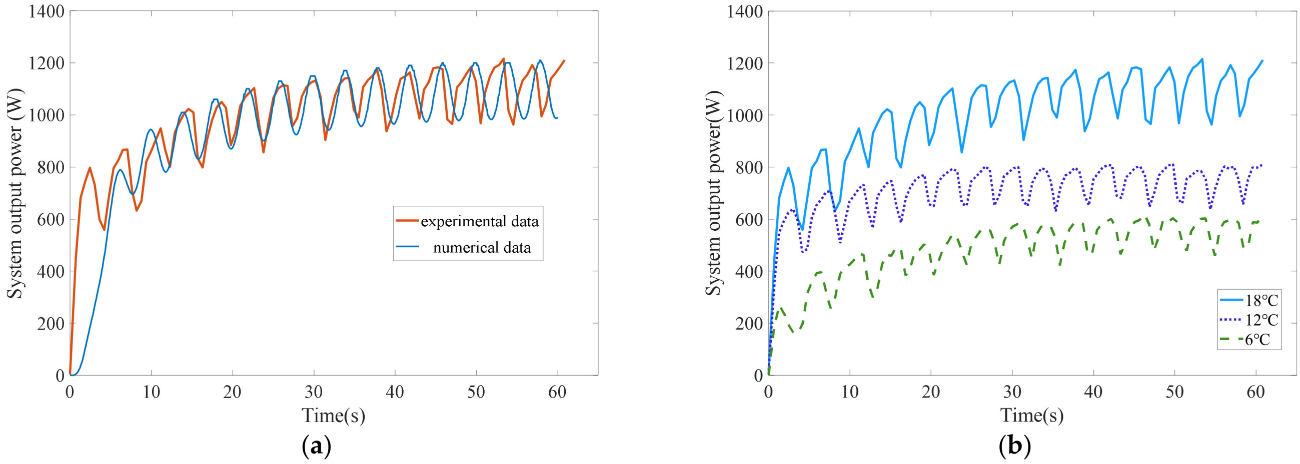


**Figure 6.** System output power at different start-up temperatures. (**a**) comparison of system output power at 18 °C between the numerical model and experimental conditions. (**b**) comparison of system output power at 6 °C, 12 °C, and 18 °C.

### *4.2. Simulation of Hydraulic PTO System at Different Temperatures*

The wave with a period of 6 s and an amplitude of 0.075 m was considered in this simulation model. The pressure and flow changes of the hydraulic PTO system in the startup process are shown in Figures 7 and 8. The system pressure had no evident diversity when the period of the input wave had changed from 8s to 6s. Furthermore, when the ambient temperature was higher, it was faster to achieve the balance, and the volatility also increased in the meantime. At the same time, the flow would have a clear distinction if the temperature increased.

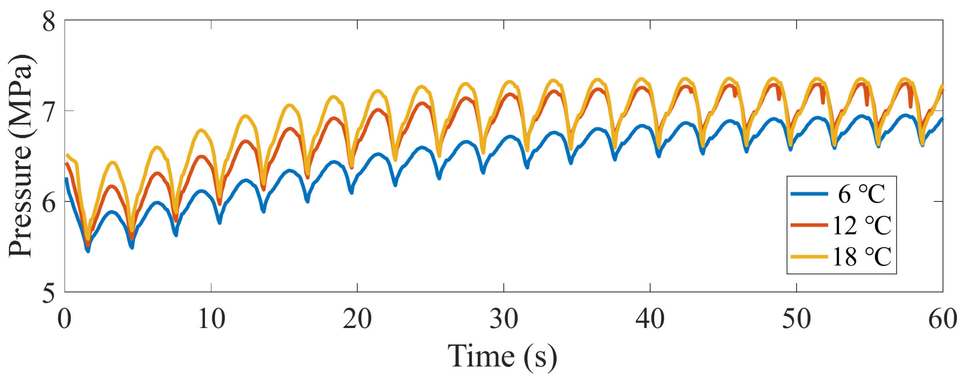


**Figure 7.** Comparison of the hydraulic PTO system pressure at different temperatures by simulation.

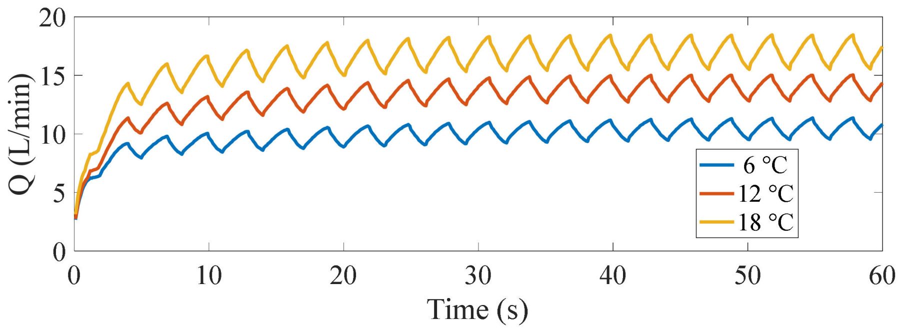


**Figure 8.** Comparison of hydraulic PTO system flow at different temperatures by simulation.

### *4.3. The Effect of Temperature Rise on the Consistency and Efficiency of Hydraulic PTO System*

Since the output power of the PTO system at low temperature was much lower than at slightly higher temperatures, a long-running test was carried out at the temperature of 6 °C, controlling the input force constant and using the sensors and meters to record the system state data at runtime. The test results are shown in Figures 9 and 10. It can be seen that when the device was started, the temperature gradually rose, while the hydraulic motor speed, system flow, and pressure rose rapidly. Then, the temperature continued to rise, and when the system pressure was consistent, the system flow, speed, and torque of the hydraulic motor continued to increase. The rise in temperature reduced the viscosity and the flow resistance of the hydraulic oil, while the torque loss of the hydraulic motor decreased, which would eventually increase the mechanical efficiency of the system. Then, 6 minutes later, the temperature of the hydraulic oil continued to rise and eventually reached about 13 °C; the pressure fluctuated slightly. The fluctuations in flow rate and hydraulic motor speed were more obvious. The continuous rise of temperature caused the viscosity of the hydraulic oil to be too low, leading to an increase in leakage, and finally, the volumetric efficiency of the hydraulic PTO system decreased.

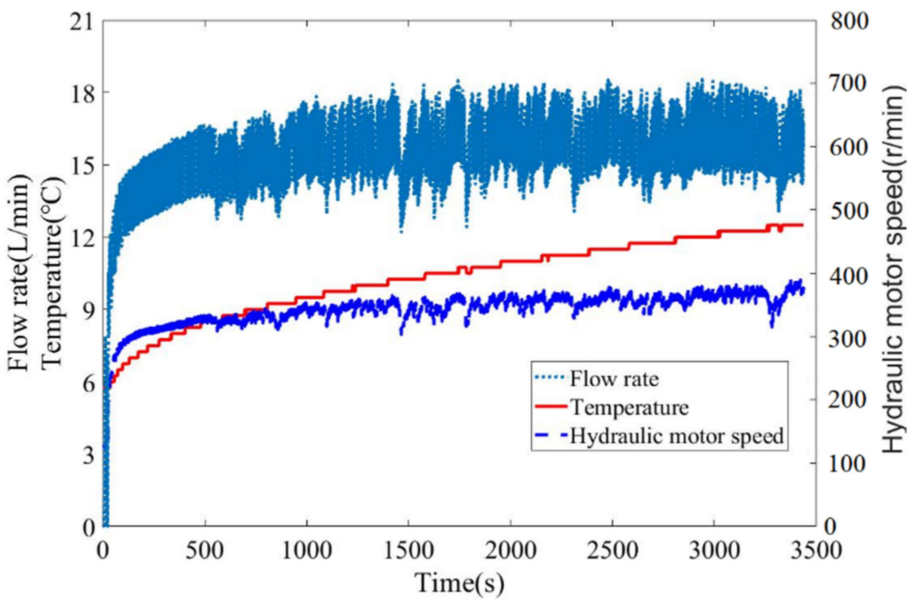


**Figure 9.** Changes of temperature, flow, and motor speed in the start-up to steady progress of power take-off system.

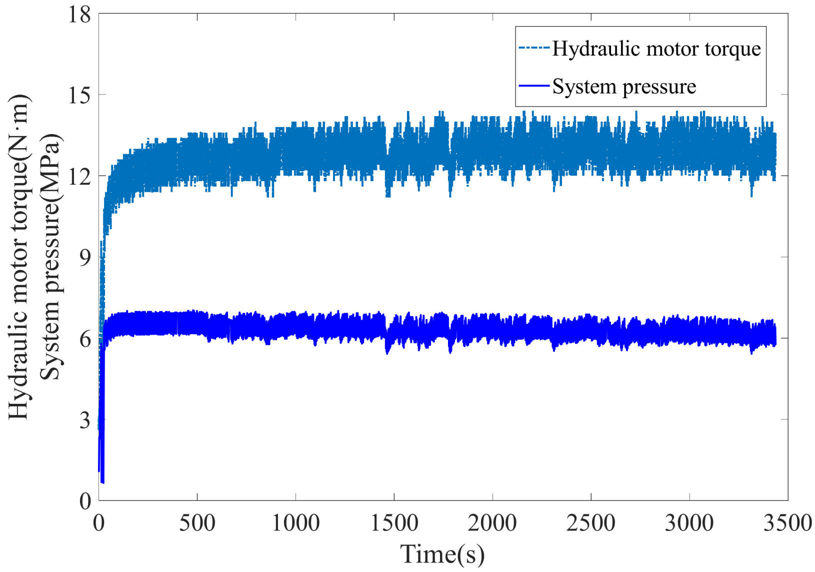


**Figure 10.** Changes in hydraulic motor torque and system pressure during long-term operation of the PTO system.

The changes in the mechanical efficiency and volumetric efficiency of the hydraulic motor with the operation of the PTO system are shown in Figure 11. The increase in the temperature of the hydraulic oil caused the mechanical efficiency to rise from about 80 percent to 97 percent, while the volumetric efficiency had a downward trend decreasing from around 93 percent to 85 percent. The total efficiency of the hydraulic motor shows a tendency that it increases firstly and then decreases, and the overall efficiency is maintained at the range of 70 percent to 88 percent.

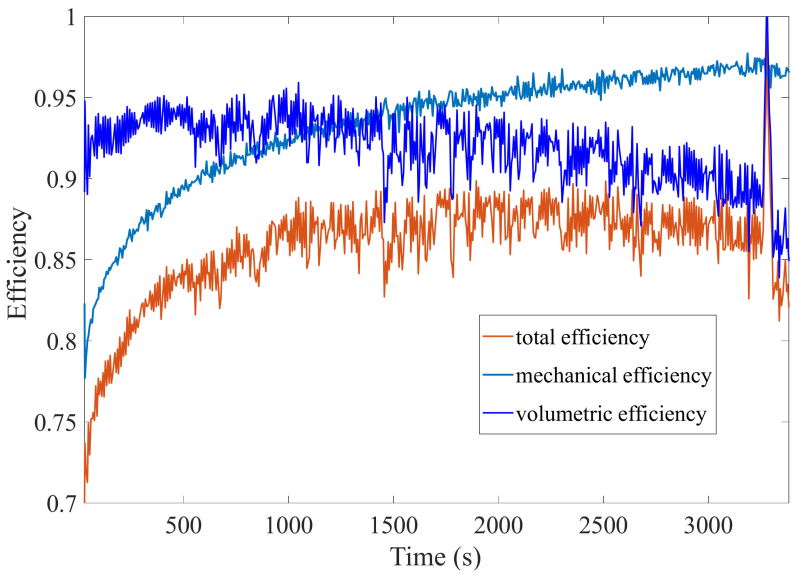


**Figure 11.** Mechanical, volumetric, and total efficiency of the hydraulic motor.

With the operation of the experimental device, the temperature of the hydraulic oil increased, the viscosity of the hydraulic oil decreased, and meanwhile, the hydraulic motor speed and system flow increased. Therefore, the influence of temperature on the system characteristics mainly altered the flow by affecting the hydraulic oil viscosity, thus acting on the hydraulic motor efficiency. This eventually had an impact on the system output power.

## 5. Conclusions

This paper focused on and studied the impact of start-up at variable temperatures and the rise of temperature on the characteristics of the hydraulic PTO system. The mathematical model of the temperature—hydraulic oil viscosity—hydraulic motor efficiency of the hydraulic PTO system was added into the simulation model of the hydraulic PTO system. Experiments were carried out to prove the correction of the simulation model, and different waves were emulated. The conclusions are as followed:

1. The system flow will increase owing to the rise of system temperature before it achieves the maximum, while system pressure can always reach a constant value;
2. The output power of the hydraulic PTO system is much lower than the designed one when it starts at low temperature, and it needs a reasonably long time for the system to heat up;
3. The results indicate that the thermal hydraulic model is reasonable and has high accuracy compared with to the real situation. For the cases of different wave excitation, the pressure of the hydraulic system is maintained at a relatively stable value, which ensures the consistent operation of the system at different input of energies;
4. The influence of temperature on the system characteristics is mainly altering the flow by affecting the hydraulic oil viscosity, thus acting on the hydraulic motor efficiency, eventually impacting the system output power.

In future work, we will study the operating characteristics of PTO in a larger range of temperatures and wave states and introduce a more accurate model to describe other components' performances to improve the efficiency of the hydraulic PTO system in different sea states.

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## Appendix A

**Table A1.** Parameters of numerical model.

| Components | Title | Unit | Value |
|---|---|---|---|
| Single-acting single-rod hydraulic cylinder | Piston diameter | mm | 80 |
| | Rod diameter | mm | 63 |
| High-pressure accumulator | Gas pre-charge pressure | bar | 60 |
| | Accumulator volume | L | 6.3 |
| | Polytropic index | □/ | 1.4 |
| Flow control valve | Set flow | L/min | 20 |
| | Minimum operating pressure difference | bar | 2 |
| | Flow-rate pressure gradient | L/min/bar | 0.05 |
| | Valve hysteresis | bar | 0 |
| Hydraulic motor | Displacement | cc/rev | 17.5 |

**Table A1.** *Cont.*

| Components | Title | Unit | Value |
|---|---|---|---|
| Generator | Number of pole pairs | □/ | 5 |
| | Reference temperature | degC | 18 |
| | Stator winding resistance at reference temperature | Ohm | 0.05 |
| | Corrective coefficient on stator winding resistance | 1/K | 0.2 |
| | Stator cyclic inductance on Park's d axis | H | 0.0035 |
| | Stator cyclic inductance on Park's q axis | H | 0.0018 |
| | Permanent magnet flux linkage at reference temperature | Wb | 0.4 |
| Load | Resistance | Ohm | 11 |
| Oil | Density | kg/m$^3$ | 850 |
| | Bulk modulus | bar | 17,000 |
| | Absolute viscosity | cP | 51 |
| Design operating parameters | Pressure | MPa | 7 |
| | Flow-rate | L/min | 14.7 |
| | Amplitude of piston motion | m | 0.075 |
| | Frequency of piston motion | Hz | 1/8 |
| | Rotational speed | r/min | 600 |
| | Output power | W | 1200 |

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