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Design of Gerotor Pump and Influence on Oil Supply System for Hybrid Transmission

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Abstract: Electric continuously variable transmission (E-CVT) is a vital part of the automobile in order to enhance the power coupling. The oil pump is an important power source component in the hybrid transmission system. Its efficiency exerts a significant impact on the efficiency of the oil supply system and even the hybrid transmission system. In this study, a gerotor pump is designed in line with the requirements of a certain type of hybrid electric vehicle. A Non-dominated Sorting Genetic Algorithm II (NSGA-II) genetic algorithm was employed to optimize the rotor tooth profile. The proportional-derivative (PD) control of the oil supply system was realized to lower the functional error of the oil supply system based on the AMESim simulation platform. In addition, the prototype test was performed to verify the rationality of the design.

Keywords: E-CVT; gerotor pump; oil supply system; hybrid transmission

Citation: Huang, M.; Shi, C.; Zhu, Y.; Zhang, J.; Zhang, F. Design of Gerotor Pump and Influence on Oil Supply System for Hybrid Transmission. *Energies* **2021**, *14*, 5649. <https://doi.org/10.3390/en14185649>

Academic Editor: Susan Krumdieck

Received: 27 July 2021

Accepted: 5 September 2021

Published: 8 September 2021

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1. Introduction

In recent years, although the continuous increase in the number of passenger cars has significantly promoted the development of the automotive industry, the corresponding energy shortages and environmental pollution problems have gradually entered the public's field of vision. To satisfy the current challenges and reduce oil consumption, researchers have adjusted their goals to the R&D and manufacturing of energy-saving vehicles or new energy vehicles [1–3]. Hybrid electric vehicles mainly refers to oil–electric hybrid electric vehicles, whose power sources contain a traditional internal combustion engine and electric motor. Its design goal is to make the engine always work at its highest efficiency through the power coupling input of the two power sources, aiming to realize the effects of energy saving and emission reduction. The hybrid transmission belongs to one of the core components of a hybrid vehicle. The hydraulic system of the hybrid transmission is designed to achieve the cooling and lubrication of the motor, lubrication of the gear shaft components in the box, and lubrication of the shifting components, as well as the shifting of various gears. The oil supply system is a critical part of the hydraulic system, and a certain flow of hydraulic oil is supplied to the entire tank through the oil supply system [4–7]. In the traditional continuously variable transmission (CVT), the power loss of the hydraulic oil pump and its oil supply accounts for approximately 40% of the power loss of the transmission, which also belongs to one of the important sources of the power loss of the entire CVT. As a result, the question of how to improve the efficiency of the oil pump and its oil supply dominates an essential role in increasing the power of the entire gearbox and achieving the energy-saving and emission-reducing effects of the entire vehicle [8,9].

Traditional automatic transmission oil pumps are mechanical pumps which are driven directly or indirectly by the engine. In pure electric mode, in addition to the necessary cooling and lubrication requirements, high-pressure oil is also required to control the shift for the design of a hybrid power system with shifting components. Therefore, the traditional single engine-related mechanical pump oil supply method cannot satisfy the demand for hybrid products [10,11]. This study designs the oil supply method of mechanical pump

auxiliary electric pumps. When the pump is directly driven by the engine, the oil pump is associated with the input shaft. The proposed method requires a longer oil inlet passage. It significantly increases the design difficulty of the tank. In the meanwhile, the long oil passage causes a certain degree of pressure to drop. In addition, with a direct connection to the input shaft, a higher speed produces a higher shear rate, resulting in increased friction and reduced mechanical efficiency. Regarding low-pressure lubricant circuits, high-speed and low-pressure vehicles are prone to poor lubrication. When the engine drives the oil pump indirectly, the system is installed flexibly, and the mechanical efficiency is higher compared with that of the direct drive solution. However, the mechanism of the intermediate shaft needs to be designed, and the number of components of the system become large, leading to an increase in the design cost and a decrease in the reliability of the mechanism. At present, with the development of electronic control technology and the miniaturization of motors, an increasing number of motors are used to replace mechanical or hydraulic structures in automobile research. Electrification can achieve greater maneuverability, which can save on cost to a certain extent. Based on the development of automobile electrification, and considering that the engine directly or indirectly drives the mechanical pump confronts numerous problems, this study only considers the design of electric pumps [12,13]. For the electric pump system, the hydraulic system contains a low-pressure cooling oil circuit and a high-pressure shift brake oil circuit. As shown in Figure 1, with this functional requirement, three integration modes of oil pump and motor are proposed, respectively: single-motor single-pump, single-motor dual-pump, and double-motor double-pump. The power transmission of the oil pump and motor is illustrated in Figure 2. By calculating the power loss of the three modes, the single-motor dual-pump integrated mode is determined [14].

Figure 1. Three integration modes: (a) single-motor single-pump, (b) single-motor dual-pump, and (c) dual-motor dual-pump.

Figure 2. Schematic diagram of the total success rate of the oil pump.

The hydraulic system of mass-produced cars in the world includes numerous modes such as Prius, Malibu, Lexus, and Trumpchi. The integration modes and the number and types of pumps used in the hydraulic system vary from each other.

(1) Hydraulic system in Prius hybrid system

The Prius hybrid system is based on Toyota's hybrid technology "THS" system, which is currently a mature hybrid technology product [15]. The hydraulic system of the Prius gearbox only adopts one mechanical pump, which is driven by the engine shaft in order to achieve the cooling and lubrication. The structure is simple while fulfilling the functional requirements in this way [16–20].

(2) Hydraulic system in Malibu hybrid system

The Malibu hybrid system is based on General Motors' technology, which is dedicated to the research of double-row and three-row planetary gear mechanisms. In the oil supply system, a single electric pump provides pressure and flow for realizing the electrification of the oil supply system. The oil pump is a gerotor pump [21,22].

(3) Hydraulic system in Lexus hybrid system

Based on the original "THS", the GS450h hybrid transmission mounted on the Lexus LS600h is a "THS-II" longitudinal two-stage hybrid transmission. The hydraulic system consists of a mechanical pump and an electric pump [23].

(4) Hydraulic system in Trumpchi hybrid system

The GAC Mechatronic Coupling hybrid power system equipped with Trumpchi GA3S is a set of electromechanical coupling systems with a simple structure but superior performance. The hydraulic system is composed of a mechanical pump and an electric pump, and the mechanical pump assists the electric pump for cooling and lubricating [24].

In the present study, based on a certain type of hybrid transmission, the oil pump is designed in line with the needs of the hydraulic system, and the influence of the pump on the oil supply system is further analyzed. Moreover, an experiment platform was built, and the design and simulation results were verified.

2. Design of Gerotor Pump Oil Supply System for Hybrid Transmission

The oil pump refers to the power source of the hydraulic system. Designing the type and structure of the oil pump satisfying the requirements of the hybrid transmission is one of the important tasks for the design of the hybrid transmission [25]. The gerotor pump has the advantages of having few parts, a compact structure, small pulsation, and low noise. Moreover, it has been extensively applied in aerospace, aviation, and automobile fields. As shown in Figure 3, the base circle with radius r_1 and the radius circle with radius r_2 roll purely around the base circle. In addition, the connection point C on the circle is formed to construct a short-circle cycloid C1C2. The center of the circle and a certain radius R make a series of circles. The formed inner envelope is the tooth profile of the inner rotor of the gerotor pump, and the arc conjugate with the inner rotor is the outer rotor arc. In a gerotor pump, the rotation center of the outer rotor is the center, and the circle whose radius refers to the distance (L) between the rotation center of the outer rotor and the arc tooth profile center of the outer rotor is defined as the creation circle. The ratio of the generative circle radius L to the outer rotor pitch circle radius r_2 is determined as the generative coefficient k . The ratio of the outer rotor tooth profile arc radius R to the outer rotor pitch circle radius r_2 is described as the arc diameter coefficient h [26]. As

presented in Figure 3, the coordinate axis $S0:xO_1y$ is based on the base circle O_1 as the coordinate origin; the coordinate axis $S1:x_1O_1y_1$ with the inner rotor rotation center O_1 as the coordinate origin; and the coordinate axis $S2: x_2O_2y_2$ with the outer rotor rotation center O_2 as the coordinate origin. Among them, the fixed point C locates on the abscissa of the $S2$ coordinate system. In the coordinate system, where the outer rotor is the center of rotation, the outer rotor tooth profile arc equation can be written as:

$$(x - L \sin \varphi)^2 + (y - L \cos \varphi)^2 = R^2 \quad (1)$$

According to the coordinate transformation, the inner rotor tooth profile curve equation can be expressed as [27]:

$$\begin{cases} X = L \cos(\psi_2 - \psi_1) - R \cos(\psi_2 + \theta - \psi_1) - e \cos \psi_1 \\ Y = L \sin(\psi_2 - \psi_1) - R \sin(\psi_2 + \theta - \psi_1) + e \sin \psi_1 \end{cases} \quad (2)$$

where, ψ_1 and ψ_2 satisfy: $\psi_1 = r_2 \cdot \psi_2 / r_1$; θ denotes the angle between the straight line O_2C and the normal PC of the cycloid; θ satisfies: $r_2 / \sin \theta = L / \sin(\pi - \psi_2 - \theta)$.

Figure 3. Rotor tooth profile.

The gerotor pump refers to the positive displacement pump, which pumps oil by changing the volume of the closed chamber. The instantaneous flow rate is characterized by the volume change rate of the cavity, and the differential–integral method is adopted to obtain the instantaneous discharge flow rate:

$$q_i = \begin{cases} \frac{dV_i}{d\psi_2}, \frac{dV_i}{d\psi_2} \leq 0 \\ 0, \frac{dV_i}{d\psi_2} > 0 \end{cases}, \quad Q_{in} = \sum_{i=1}^Z q_i \quad (3)$$

Based on the mathematical model of the instantaneous flow of cycloidal rotor pump, the numerical calculation and analysis of the instantaneous flow can be realized by writing a script file with open-source software [28,29].

NSGA-II is one of the most popular multi-objective genetic algorithms, which lowers the complexity of non-inferior ranking genetic algorithms. It has the advantages of having a fast running speed and good solution set convergence. Moreover, it has also become the benchmark for the performance of other multi-objective optimization algorithms. Based on the NSGA-II genetic algorithm, the rotor tooth design is carried out. The design parameters of the tooth shape include the following four parameters, respectively: the number Z of inner and outer rotor teeth, the eccentricity e , the radius of the created circle L , and the radius R of the outer rotor arc [30–32]. Considering the range of the teeth of the commonly used hybrid transmission gerotor pump, the number of teeth of the outer rotor is selected as

four sets of data containing 5, 7, 9, and 11. The remaining design parameters are determined by the final design results. As a result, the design variables are determined as:

$$X = [x_1, x_2, x_3]^T = [L, e, R]^T \quad (4)$$

According to the main principles of compactness and pulsation that should be followed in the design of the hybrid transmission oil pump, the design goals of OF1 and OF2 are proposed in this study. The compactness is characterized by the smallest unit displacement volume, and the pulsatility is the deviation of the instantaneous flow rate from the average flow rate:

$$OF1: \min\left(\frac{\pi \cdot (L - R + 2e)^2 \cdot H}{\pi H [(L - R + e)^2 - (L - R - e)^2]}\right) \quad (5)$$

$$OF2: \min\left(\frac{Z}{\pi} \int_{\psi_2=0}^{\psi_2=\frac{\pi}{Z}} (Q - \bar{Q})^2 d\psi_2\right) \quad (6)$$

Constraints on design variables are proposed, aiming to simplify the complexity of the design process. The optimization results have practical significance. The tooth shape of the Equation (7) should not have the basic constraints of failure. In addition, basic boundary conditions include: generative coefficient k value range: 1.1 to 1.8, an arc diameter coefficient h value ranging from 0.2 to 0.8, and an eccentricity value ranging from: 1.5 to 3. The radius of the outer rotor tooth root circle is required to be greater than the radius of the creation circle. Moreover, all design variables should be positive real numbers.

$$R < \sqrt{\frac{27(Z-1)(L^2 - e^2 Z_2^2)}{(Z+1)^3}} \quad (7)$$

The optimization calculation program of NSGA-II algorithm is written via MATLAB. The initial population is designed to be 100. After several iteration calculations, it can be found that the population of 100 individuals remains basically stable after 300 iterations. Figure 4 displays the Pareto optimal solution when the number of teeth of the outer rotor is 5, 7, 9, and 11 teeth. Obviously, the pulsation performance is more superior with an increasing number of rotor teeth, while the compactness is more superior when the number of teeth decreases. Based on the selection of a hybrid vehicle with small pulsation and compact pumps, and the functional requirement that the pulsation performance weight is greater than the compact weight, a rotor pair with an outer rotor of nine teeth is selected in this study. The specific parameters include the number of teeth of the outer rotor (nine), the number of teeth of the inner rotor (eight), the eccentricity (e : 1.8 mm), the radius of the created circle (L : 19.5 mm), and the radius of the outer rotor arc (R : 3.8 mm). According to the designed parameters, a curve is drawn in Creo, as shown in Figure 5. Then, a complete gerotor pump drainage basin was formed through 3D modeling software [33,34]. At the same time, the pressure cloud diagram was obtained based on the CFD simulation software of professional motion machinery, as presented in Figure 6. The outlet pressure pulsation curve is illustrated in Figure 7. The analysis illustrates that the pulsation rate is 0.9%.

Figure 4. Pareto optimal solution.

Figure 5. Rotor tooth profile.

Figure 6. Pressure cloud diagram.

Figure 7. Pressure pulsation.

Based on the known rotor tooth shape, the oil supply pump that satisfies the flow requirements and working conditions can be calculated. The prototype presented in Figure 8 is made and tested for displacement, efficiency, starting performance and so on. Moreover, the rationality of the design is verified.

Figure 8. Prototype for testing.

3. Influence of Gerotor Pump Parameters on Oil Supply System

The traditional automatic transmission oil pump is generally a quantitative pump which is driven directly or indirectly by the engine. The flow output of the oil pump is associated with the engine speed. In order to satisfy the pressure and flow requirements, the displacement design of the oil pump is also determined by the minimum speed of the engine. During normal driving, the working range of the engine speed is often greater than the minimum speed. Therefore, the supply of hydraulic pump flow is often greater than the demand flow, as illustrated in Figure 9. The excess flow generates overflow loss, also causing the entire transmission to drive the decrease in efficiency. As a result, only improving the efficiency of the oil pump to control the output flow of the oil pump is not enough to decrease the power loss of the oil supply system. The self-adaptive performance of the oil pump refers to the control of the oil pump through timely output feedback under different working conditions, which is of great importance for the entire oil supply system.

Although the gerotor pump has good speed characteristics, high volumetric efficiency, and a simple structure, it still has a large pressure pulsation. Proportional Integral Derivative (PID) is an abbreviation for proportional integral differential control, which refers to a commonly used control algorithm. The P in PID control is a proportional link. The proportional control is actually an adjustable gain amplifier. The proportional control can

reduce the size of the error but cannot eliminate the steady-state error. The letter I indicates the integration link. The use of the integration link can eliminate the steady-state error, but increases the system overshoot and even cause the system to oscillate. D is a differential link, which is also known as an inertial link. The output through the inertial link has a time delay and cannot be changed in time with the input. In addition, it can reduce overshoot, eliminate system oscillation, and improve system stability. The construction of the proportional link module is employed to achieve error control [35]. The inertial link module is built to control the sudden change of error, and the output signal is delayed in time, thereby decreasing the system adjustment time. The oil pump speed control module shown in Figure 10 has been developed to realize the stability of oil supply. The control module includes a low-pressure flow feedback control, a high-pressure load pressure feedback control, and an oil pump torque feedback control. The maximum working pressure of the hydraulic system investigated in the current work is 17 bar, and the demand flow of the high-pressure oil circuit is 7 L/min. Besides, the demand flow of the low-pressure oil circuit is not less than 14 L/min. Considering the pump displacement, flow demand, and pump volumetric efficiency, the analysis demonstrates that, when the pump speed is about 1800 r/min, the maximum demand flow of the hydraulic system can be satisfied. The given pump speed of signal input is 1800 r/min, while the maximum speed is limited to 3500 r/min. The control model is equipped with the entire hydraulic system simulation model to simulate and control the pump speed in the entire drive cycle in order to achieve the oil supply target of the oil supply system [36].

Figure 9. Diagram of oil pump flow requires and supply.

Figure 10. Speed control module: low_con is the control of flow; pres_con is the control of pressure; torque_con is the control of torque and Pum_speed_con is the control of pump speed.

Figure 11 presents the controlling speed of the oil pump. The speed is dynamically adjusted according to the output feedback. Simultaneously, it is limited to the speed range required by the pump. When parking, the signal can be received in time to reduce the speed. Figure 12 illustrates a comparison of the simulation results of the cooling oil flow without adding the speed module and the simulation results after adding the control module. Through the comparison, it is found that there are a wide range of conditions that do not meet the flow requirements before the control. After the addition of control module, the flow requirements can be basically met during the entire drive cycle. At the same time, the flow supply can be quickly reduced during parking in order to avoid waste. Figure 13 displays a simulation result of the main oil circuit pressure before and after control, as well as a comparison of the theoretical pressure. The simulation proves that the error between the main pressure and the theoretical pressure after control is smaller. Based on the analysis of the above simulation results, the PD control of the oil pump speed can effectively reduce the functional error of the oil supply system, especially under extreme conditions, such as low temperature and low pressure conditions. Additionally, the characteristics of the oil pump are greatly affected by the environment. Moreover, the dynamic control of oil pump speed is of great engineering significance.

Figure 11. Oil pump control speed.

Figure 12. Comparison of cooling and lubrication flow before and after control.

Figure 13. Comparison of main oil pressure before and after control.

An experiment platform was built according to the design of the gerotor pump. Displacement is regarded as the important data to check to determine whether the hy-

draulic pump is qualified. The evaluation standard refers to that when the temperature is $40\text{ }^{\circ}\text{C} \pm 3\text{ }^{\circ}\text{C}$ and the deviation between the theoretical flow and the design flow cannot exceed 5%. Table 1 presents the high-pressure pump displacement test data, and Table 2 indicates the low-pressure pump displacement test data.

Table 1. High-pressure pump displacement test data.

Test Conditions		Pressure: 0 Bar Theoretical Displacement: 4 mL/r		
Oil Pump Number	Rotating Speed (r/min)	Flow (L/min)	Calculated Displacement (mL/min)	Deviation (%)
1#	1000	3.85	3.85	3.75%
	1500	5.8	3.87	3.25%
2#	1000	3.84	3.84	4%
	1500	5.8	3.87	3.25%

Table 2. Low-pressure pump displacement test data.

Test Conditions		Pressure: 0 Bar Theoretical Displacement: 8 mL/r		
Oil Pump Number	Rotating Speed (r/min)	Flow (L/min)	Calculated Displacement (mL/min)	Deviation (%)
1#	1000	7.9	7.90	1.25%
	1500	11.8	7.87	1.63%
2#	1000	7.9	7.90	1.25%
	1500	11.8	7.87	1.63%

According to the test of the high- and low-pressure prototype pumps, respectively, the deviation between the calculated displacement value and the theoretical displacement value meets the requirements. As a result, the designed oil pump can satisfy the functional requirements of a hybrid transmission.

4. Conclusions

- (1) A gerotor pump was designed. The tooth profile was analyzed and designed in line with the requirements of the hydraulic system for hybrid transmission;
- (2) NSGA-II algorithm was employed to optimize the design of tooth profile parameters according to the mathematical model of instantaneous flow. The parameters of the gerotor pump can be determined;
- (3) Based on the AMESim platform, a PD control strategy was employed to develop the oil pump speed control model of the oil supply system. In addition, the influence of the designed gerotor pump on the performance of the oil supply system was simulated;
- (4) Prototypes were made, an oil pump test platform was built, and oil pump unit tests were conducted to confirm the rationality of the oil pump design and simulation reliability. Besides, the designed oil pump can meet the functional requirements of a hybrid transmission.

Author Contributions: Conceptualization, M.H. and F.Z.; methodology, M.H.; validation, C.S. and Y.Z.; resources, M.H. and F.Z.; writing—original draft, C.S. and J.Z.; writing—review and editing, F.Z. All authors have read and agreed to the published version of the manuscript.

Funding: This research was supported by A Priority Academic Program Development of Jiangsu Higher Education Institutions (PAPD); Engineering Research Center for Harmless Treatment and Resource Utilization of Aluminum Ash and Slag Solid Waste in Jiangsu Province; The Basic Scientific Research Project of Nantong City (grant number: JC2021200).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Acknowledgments: Sincere thanks go to anonymous reviewers for their valuable comments and great suggestions to improve this article. Thanks to Corun Hybrid Technology Co., Ltd. for providing the experimental conditions and simulation platform.

Conflicts of Interest: The authors declare no conflict of interest.

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