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Optimizing the Performance of the Engine Variable Valve Drive System by Particle Swarm Optimization Algorithm

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Abstract

To improve valve motion stability and reduce energy consumption in the engine valve train, this study utilized an enhanced Particle Swarm Optimization (PSO) algorithm to optimize the variable valve drive system. It constructed a schematic representation of the engine's variable valve mechanism, and developed a corresponding optimization problem to support the analysis. The objective function is defined based on power consumption, with appropriate constraints introduced. Key design variables for the variable valve system are identified. The PSO algorithm is adapted and improved to suit the specific optimization requirements. Detailed optimization steps using the PSO algorithm with a linearly varying inertia weight algorithm are outlined, leading to the identification of optimal parameter values for the design variables. To validate the optimization results, MATLAB is used to simulate engine power consumption, valve lift, valve velocity, and valve acceleration—comparing the performance before and after optimization. Simulation results indicate that, prior to optimization, the engine exhibits high power consumption, wide variations in valve lift, speed, and acceleration, and unstable valve seating. Post-optimization, there is a significant reduction in power consumption, along with more stable and constrained valve motion characteristics. These improvements confirm that the application of PSO to the variable valve drive system effectively reduces fuel consumption and enhances valve motion stability.

Keywords: Engine; Variable Valve Actuation; Valve Lift; Power; Particle Swarm Optimization.

1. Introduction

Energy is a basic industry for national development and an important guarantee for the sustainable, stable and healthy development of the economy of each country [1]. With the continuous improvement of living standards, global energy consumption has exhibited a consistent upward trend in recent years. How to reduce energy consumption is a common topic studied by experts and scholars around the world. The main area of energy consumption is the transportation field, and the proportion of automobile consumption is relatively large. Therefore, reducing the energy consumption of road vehicles is of great significance for saving energy.

Minimizing vehicular energy consumption primarily hinges on decreasing engine power losses, with the enhancement of engine thermal efficiency serving as a critical factor in achieving this objective. The main fuels used by engines are gasoline and diesel. In China, approximately 66% of petroleum fuels are consumed by internal combustion engines. The output efficiency of diesel engines is about 30% - 45%, and that of gasoline engines is even lower, about 20% - 30% [2]. Therefore, most of the heat energy of the internal combustion engine is wasted through heat dissipation, incomplete combustion, and exhaust. Studying advanced engines and improving engine thermal efficiency is one of the effective ways to solve the energy crisis.

In order to reduce the energy consumption of engines, researchers studied on engines from different directions. For example: References [3, 4] studied the energy-saving dispatching algorithm of automobiles, analyzed the principle of the energy-saving dispatching method of automobiles, deduced the energy consumption equation of automobile operation, i.e., the mathematical model describing the power consumption during engine operation, established the dispatching model of the automobile engine, and verified the energy consumption of the engine through experiments, thereby reducing the energy consumption during automobile operation. References [5, 6] developed power loss models for hybrid electric vehicles and conducted an in-depth analysis of energy consumption under varying operating conditions. They formulated the mathematical model of power consumption to characterize system energy losses and validated the vehicle's energy performance through simulation, demonstrating an effective approach to minimizing power losses in hybrid electric drive trains. References [7, 8] investigated the design of a variable-valve-lift mechanism, examine valve-timing characteristics across diverse operating conditions, and present a control-circuit schematic for the stepper

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motor. Employing BOOST software for thermodynamic optimization, these studies report notable improvements in both fuel economy and power output. Although the energy consumption of the engine valve train studied in previous research has been reduced, the valve landing speed is relatively large, resulting in a relatively large engine noise. To address this challenge, the present study developed a simplified schematic representation of the engine's variable valve actuation system, analyzed the working principle of the variable valve drive, established an optimization problem, defined the objective function through energy consumption, added constraint conditions, defined the optimization design variables, optimized the design parameters using the particle swarm algorithm, and simulated the engine energy consumption, valve lift, valve speed, and valve acceleration using Matlab software to verify the output effect after the engine optimization, providing a theoretical basis for in-depth research on engine energy-saving methods.

2. LITERATURE REVIEW

Variable Valve Drive System, Variable Valve Drive (VVD) systems have received considerable attention in recent years due to their potential to enhance engine efficiency, reduce emissions, and improve transient performance. Compared with conventional cam-driven mechanisms, hydraulic variable valve systems [9] offer significantly greater flexibility in shaping valve lift profiles, timing, and opening/closing rates, enabling precise control of cylinder charge, residual gas fraction, and combustion phasing under diverse operating conditions. Typical hydraulic VVD architectures include a high-pressure oil supply, hydraulic actuators, switching valves, and pressure stabilization components. The defining feature of these systems lies in their ability to actuate the valve motion directly through hydraulic force while relying on advanced valve-control strategies to achieve high responsiveness and wide adjustability.

A substantial body of research highlights the importance of hydraulic source design [10] in determining overall system performance. Dual-pump configurations—consisting of a primary pump coupled to an auxiliary or secondary pump—have been shown to improve the stability of upstream pressure dynamics while reducing the load on the main pump. Such arrangements help maintain adequate supply pressure for the actuator even under rapid valve events, thereby increasing efficiency and reducing parasitic losses. Furthermore, accumulators are frequently incorporated to mitigate pressure fluctuations and enhance hydraulic responsiveness during transient operation, resulting in more stable and accurate valve lift control. Studies also note that the valve return spring not only provides the mechanical restoring force required for closure but also contributes to energy recovery within the hydraulic circuit, reducing the net energy demand placed on the hydraulic source.

In terms of actuation control, hydraulic VVD systems commonly employ a high-pressure switching valve (HPSV) and a low-pressure switching valve (LPSV) to regulate the phases of valve opening, peak-lift holding, and closing. Prior research demonstrates that by adjusting the timing of these switching events, it is possible to finely tune the valve event schedule and, consequently, influence combustion phasing, volumetric efficiency, and internal exhaust gas recirculation. To further enhance control accuracy, researchers have proposed a range of strategies, including pressure-feedback-based switching control, model-predictive control (MPC), and integrated simulation frameworks coupling multi-body dynamics with hydraulic and gas-exchange models. These approaches aim to overcome the inherent nonlinearities of hydraulic systems, valve inertia, fluid dynamics delays, and switching-valve hysteresis.

Overall, recent research efforts have focused on hydraulic pressure regulation, energy optimization, and advanced switching-valve strategies as the core enablers of high-performance VVD systems. Through pump coupling, accumulator-based pressure stabilization, spring-assisted energy recovery, and precise HPSV/LPSV phase control, modern VVD architectures can significantly improve valve motion accuracy, reduce pump power consumption, and enhance combustion efficiency. As demands for lower energy consumption, higher controllability, and improved reliability continue to intensify, hydraulic variable valve systems are expected to remain a key technology pathway in the evolution of high-efficiency internal combustion engines.

The variable valve drive system examined in this study is depicted in Figure 1, and its operational mechanism is described as follows. The engine crankshaft powers a gear pump, which extracts oil from the fuel reservoir and supplies it to both the primary and secondary pumps. After being pressurized, the oil is directed into an accumulator that functions to stabilize the system's inlet pressure. A phase control mechanism governs the actuation timing of the High-Pressure Switching Valve (HPSV) and the Low-Pressure Switching Valve (LPSV), thereby enabling precise control over the valve's opening and closing phases. When the HPSV is activated, the engine valve begins its lifting motion. Once the HPSV closes, the valve reaches its peak lift and remains stationary until the LPSV opens, triggering the closing process. Final valve closure is completed once the LPSV is fully shut. The secondary pump, which is mechanically linked to the shaft of the primary pump, plays a role in modulating the upstream pressure of the primary pump-effectively influencing the downstream pressure in the hydraulic actuator. To ensure pressure stability during valve operation, the system leverages the energy stored in a return spring. This configuration effectively reduces the main pump's power consumption by lowering its operational demand.

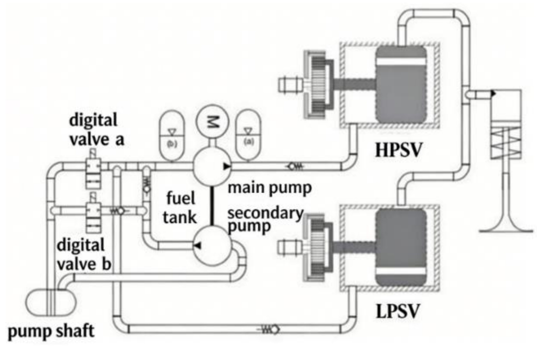

Figure 1. Variable valve drive system diagram

3. METHODOLOGY

3.1. Optimization Model

In this context, the present study establishes such a dynamic system model, as detailed in references [11, 12].

$$X'=f(X,t) \tag{1}$$

$$X=[x,x',P_1,P_2]^T \tag{2}$$

$$f(X,t)=\begin{bmatrix} x' \\ k\frac{P_2A_pK_sx-F_p-F_g-\mathrm{sign}(x')F_f}{m} \\ k\frac{P_1^{\frac{k+1}{k}}}{P_1^{0/k}V_{1,0}} \\ \frac{\beta}{P_2^{0/k}+A}\left(P_p-A_HC_d\mathrm{sign}(P_1-P_2)\sqrt{\frac{2[P_1-P_2]}{\rho}}\right) \\ \frac{\beta}{P_1P_2+A}\left(A_HC_d\mathrm{sign}(P_1-P_2)\sqrt{\frac{2[P_1-P_2]}{\rho}}-A_Hx'\right) \end{bmatrix} \tag{3}$$

$$A_H=\begin{cases} \frac{2A_{H,max}(t-t_1)}{t_2-t_1},\ t_1<t<\frac{t_1+t_2}{2} \\ A_{H,max}-2A_{H,max}[t-(t_1+t_2)/2]/(t_2-t_1),\ \frac{t_1+t_2}{2}<t<t_2 \end{cases} \tag{4}$$

where: P_1 is the air accumulator pressure; $P_{1,0}$ represents the initial pressure preset in the air accumulator prior to system pressurization; P_2 is the hydraulic cylinder pressure; x is the engine valve displacement; A_p is the piston area; K_s is the valve return spring stiffness; F_p represents the initial preload force applied to the engine valve return spring; F_g is the engine cylinder aerodynamic force; F_f is the Coulomb friction force; m is the engine valve moving mass; k is the exponential change coefficient; Q_p is the pump flow rate; A_H represents the effective flow area within the high-pressure region; C_d represents the flow coefficient of the rotary valve orifice; ρ is the density of the oil; β is the bulk modulus of the hydraulic fluid; V_1 is the accumulator volume; $V_{1,0}$ represents the initial gas volume charged into the accumulator prior to pressurization; V_2 is the hydraulic cylinder volume; $V_{2,0}$ is the pre-charge volume of the hydraulic cylinder; t is the time; t_1,t_2 is the time at different moments. X' refers to the first-order derivative of the state vector X with respect to time t.; X is the vector; x' is the engine valve speed.

The initial conditions of the above equations are as shown in Equation (5):

$$[x,x',P_1,P_2]^T=[0,0,P_{1,t_1},P_r]^T \tag{5}$$

where: $P_{1,t1}$ is the pressure of the air accumulator at the moment of $t=t_1$; P_r is the fuel tank pressure.

Under the assumption of a constant pump flow rate and negligible leakage, $P_{1,t1}$ can be determined as shown in Equation (6):

$$P_{1,t_1}=\frac{P_{1,0}V_{1,0}^k}{V_{1,t_1}^k}=\frac{P_{1,0}V_{1,0}^k}{\left(V_{1,t_0}-Q_p(t_1-t_0)\right)^k} \tag{6}$$

3.2. Optimization Design

3.2.1. Objective Function

The majority of power consumption within the system is attributed to the hydraulic pump, while the remaining portion is primarily distributed between the phase shifter and the rotary valve. Owing to the low friction between the rotary spool and its housing, the torque required for actuation is minimal, necessitating only limited rotational input to the spool shaft during valve operation. Furthermore, the phase shifter operates intermittently, and as such, its contribution to overall power consumption is considered negligible. The functioning of the system throughout the engine cycle is segmented into two distinct temporal phases. The first period starts when the HPSV is fully closed (t_0) and ends when the HPSV starts to open (t_1), and the remaining cycle (when the HPSV is open) is considered the second period $(t_1$ to $t_2)$. Accordingly, the objective function [13] is formulated as follows:

$$J=\frac{\int_{t_0}^{t_1}P_1Q_pdt}{T_c}+\frac{\int_{t_1}^{t_2}P_1Q_pdt}{T_c}=J_1+J_2 \tag{7}$$

where: T_c is the cycle period. J_1, J_2 are the powers in the first and second periods respectively.

During the initial phase, the pump's input power is primarily utilized to elevate the pressure within the accumulator. This relationship can be described by the following equation:

$$\frac{dP_1}{dt}=-k\frac{P_1^{\frac{k+1}{k}}}{P_{1,0}^{\nu k}V_{1,0}}\frac{dV_1}{dt} \tag{8}$$

Assuming zero exhaust valve opening and negligible pump internal leakage, all pump fluid flows into the accumulator, and the expression for the change in accumulator volume is

$$\frac{dV_1}{dt}=-Q_p \tag{9}$$

From Equations (7), (8) and (9), we can get

$$J_1=P_{1,0}V_{1,0}^k\left\{\left(1/\left(V_{1,t_0}-Q_p(t_1-t_0)\right)\right)^{k-1}-\left(1/V_{1,t_0}\right)^{k-1}\right\}/(k-1)T_c,\ k\neq1 \tag{10}$$

$$J_1=P_{1,0}V_{1,0}\ln\left(V_{1,t_0}/\left(V_{1,t_0}-Q_p(t_1-t_0)\right)\right)/T_c,\ k=1 \tag{11}$$

where: $V_{1,t0}$ represents the initial volume of the accumulator at the beginning of the first time interval (t_0), that is, the period from when the high-pressure pump safety valve (HPSV) is fully close to when the HPSV opens and starts.

During the second phase of the time interval (t_1-t_2), the nonlinear dynamic system represented by Equation (3) requires numerical methods for its solution., and then P_1 is obtained in the time period (t_1-t_2) of estimating J_2. Nevertheless, the approximate values of J_1 and J_2 are required for both calculations. Under steady operating conditions, the pressure in the accumulator stabilizes and provides a relatively constant energy supply to the valve actuator. At a specific moment within the engine cycle, the accumulator pressure is considered equivalent to that at the same relative time in the adjacent cycle (preceding or following), and the corresponding estimation for $V_{1,t0}$ is constrained by the following conditions:

$$\text{find } V_{1,t0} \text{ s.t. } P_{1,t_0}=P_{1,t_2} \tag{12}$$

For this purpose, the function $g(V_{1,t0})$ is defined as follows:

$$g(V_{1,t0})=\frac{P_{1,0}V_{1,0}^k}{V_{1,t_0}^k}-[0\quad 0\quad 1\quad 0]\times\left(\int_{t_1}^{t_2}f(X,t)dt+\mathrm{i.c.}\right) \tag{13}$$

where:

$$\mathrm{i.c.}=[0\quad 0\quad V_{1,t_0}\quad P_r] \tag{14}$$

Therefore, the problem of finding $V_{1,t0}$ may be reformulated as the following problem:

$$\text{find } V_{1,t_0}>0 \text{ s.t. } g(V_{1,t_0})\rightarrow0 \tag{15}$$

The Newton-Raphson algorithm is applied to obtain a numerical solution to the afore mentioned optimization problem. Accordingly, under steady-state operating conditions, the accumulator pressure during the valve opening phase can be determined as follows:

$$P_{1,t_0}=\frac{P_{1,0}V_{1,0}^k}{V_{1,t_0}^k} \tag{16}$$

3.2.2. Constraint Conditions

In order to reduce the power consumption of the variable valve drive system under study, the optimization procedure must satisfy a series of prescribed constraints. Accordingly, critical design parameters—such as the hydraulic cylinder diameter and the stiffness coefficient of the engine valve return spring—are subject to optimization. However, variations in spring preload can lead to incomplete valve closure at the end of the closing phase, particularly under high engine speed conditions. This incomplete closure results in a significant loss of engine power, which is undesirable and must be avoided. The dynamics of the valve closing phase can be characterized using the following set of differential equations:

$$\frac{d}{dt}\begin{bmatrix}x\\x'\\P_2\end{bmatrix}=\begin{bmatrix}x'\\(P_2A_p-K_sx-F_p-F_g-\mathrm{sign}(x')F_f)/m\\\frac{\beta}{V_{r0}+A_sx}\left(A_LC_d\sqrt{2(P_2-P_r)/\rho}-A_px\right)\end{bmatrix} \tag{17}$$

Consequently, the primary equality constraint associated with the optimization problem is defined as:

$$c_1:[1\quad 0\quad 0]\begin{pmatrix}\frac{\varphi_s+\varphi_c}{3N_{e,max}}\\\int H(t)dt+i.c.\end{pmatrix}=0 \tag{18}$$

where:

$$i.c.=\begin{bmatrix}l_d & 0 & \frac{F_p+K_sl_d}{A_p}\end{bmatrix} \tag{19}$$

where: $N_{e,max}$ is the maximum engine speed; φ_s,φ_c is the variable valve opening angle; l_d is the valve lift.

The second optimization constraint addresses the robustness of the system, with the objective of minimizing the sensitivity of engine valve operation to external perturbations. Fluctuations in peak in-cylinder pressure can cause substantial variations in the force exerted by combustion gases on the valve during its opening phase, resulting in significant cycle-to-cycle deviations. These variations can adversely affect the valve lift profile, potentially undermining the consistency and stability of valve motion

$$l_d\%=100[1\quad 0\quad 0\quad 0]\left(\int_{t_1}^{t_2}f(X,t)|_{P_c}dt-\int_{t_1}^{t_2}f(X,t)\Big|_{(1\pm0.5)P_c}dt\right)/\int_{t_1}^{t_2}f(X,t)\Big|_{P_c}dt \tag{20}$$

Accordingly, the second inequality constraint is formulated as follows:

$$c_2:(l_d-\varepsilon)\leq 0 \tag{21}$$

where: ε denotes the upper limit of allowable variation in the engine valve lift.

The final constraint is associated with the upper limit of the system's operating pressure. An increase in system pressure substantially raises the risk of hydraulic leakage. In addition, mechanical design constraints require that the pump's downstream pressure remain below its maximum allowable operating threshold to ensure system safety and reliability. This constraint, as defined in [14], is expressed as follows:

$$c_3:P_2(t)-P_{max}\leq 0,\ t_0\leq t\leq t_2 \tag{22}$$

3.2.3. Optimization Variables

Numerous design parameters influence the overall performance of the system; however, the impact of certain parameters can be considered negligible. For instance, enlarging the port dimensions of the rotary spool valve or the hydraulic cylinder can effectively reduce pressure losses, thereby lowering power consumption without breaching existing constraints. Nevertheless, such dimensional increases are inherently restricted by spatial limitations. Moreover, the enlargement of the rotary valve port's opening angle is restricted by the minimum valve opening duration dictated by engine operating requirements. In this study, the primary design parameters under consideration include the stiffness coefficient (K_S) of the engine valve return spring, the pre-tightening force (F_p), the hydraulic piston area (A_P), and the accumulator pre-charge volume $(V_{1,0})$.

3.2.4. Particle Swarm Optimization Algorithm

Particle Swarm Optimization (PSO) is an iterative, nature-inspired algorithm that seeks the global optimal solution by simulating the collective behavior of a swarm. Particle: an individual in the algorithm, representing a potential solution, with each particle having a position vector in the search space. Velocity: The direction and step size of the particle's movement. Assume the number of particles is n, the dimension is D, then the particle swarm set population is swarm = $\{h_1^k,h_2^k,\cdots,h_n^k\}$. Among them, k is the current iteration number. After k iterations, the position vector of the i th particle is $h_i^k=[h_{i1}^k,h_{i2}^k,\cdots,h_{iD}^k]$, and the velocity vector is denoted as $v_i^k=[v_{i1}^k,v_{i2}^k,\cdots,v_{iD}^k]$. Therefore, after k+1 iterations, the formulas for the position and velocity of the i th particle in the d -dimensional space are [15, 16]

$$\begin{cases}v_{id}^{k+1}=\omega v_{id}^k+c_1r_1(p_{id}^k-h_{id}^k)+c_2r_2\left(p_{gd}^k-h_{id}^k\right)\\h_{id}^{k+1}=h_{id}^k+v_{id}^{k+1}\end{cases} \tag{23}$$

In the formula: v_{id}^{k+1} is the velocity of particle i after k+1 iterations in the d -dimensional space; ω is the inertia weight coefficient; c_1,c_2 is the acceleration factor; r_1 , r_2 are random numbers; p_{id}^k is the optimal position of particle i after k iterations in the d -dimensional space; p_{gd}^k is the optimal position of the population particles after k iterations in the d -dimensional space; h_{id}^{k+1} is the position of particle i after k+1 iterations in the d -dimensional space.

In order to better search for the global optimal value, the inertia weight system is modified as follows:

$$\omega=\omega_0-\frac{c(\omega_0-\omega_1)}{T} \tag{24}$$

In the formula: ω_0 is the initial inertia weight coefficient; ω_1 is the final inertia weight coefficient; c is the current iteration number; T is the maximum iteration number.

The Particle Swarm Optimization algorithm proceeds through the following sequential steps:

- **Step 1**: Initialization of the Particle Population;
- **Step 2**: According to the actual problem, set the variation range of the particle swarm dimension;
- **Step 3**: Update the position and velocity vectors of each particle based on the formulation provided in Equation (23), and replace those exceeding the search range with the maximum or minimum dimension;
- **Step 4**: valuate the fitness of each particle based on the defined objective function, and continuously update the individual optimal p_{id}^k and the population optimal value p_{gd}^k of the particles;
- **Step 5**: If the optimization condition is met, stop the iteration, otherwise jump to Step 3 until the optimal solution is found.

The optimized values of the design variables obtained through the Particle Swarm Optimization algorithm are presented in Table 1.

Table 1. Simulation parameters

Parameter variable	Before optimization	After optimization
Spring stiffness K_S/(N·m)	45 000	31000
Preload F_p/N	0	0
Piston area A_P/mm^2	206	136
Pre-charge volume $V_{1,0}$/ bar	80	115

The preload is kept at 0 in both the initial and optimized designs because any increase in preload would introduce additional static force on the valve, which may hinder complete valve closure at high engine speeds and increase the risk of leakage. Sensitivity analysis (not shown in detail to keep the paper concise) indicated that varying the preload had only a marginal benefit on power reduction but could negatively affect valve-closure reliability. Therefore, the optimization mainly adjusts the spring stiffness, piston area, and accumulator pre-charge volume, while the preload is constrained to remain zero to ensure robust valve sealing.

4. RESULTS AND DISCUSSION

Simulation and Analysis：The design parameters of the engine valve have a notable impact on the overall performance of the engine. To more effectively evaluate the operational performance of the optimized valve mechanism, the engine's variable valve drive system—post-optimization—is simulated using MATLAB, and its results are compared with those of the pre-optimization configuration. Under various engine speed conditions, the corresponding power consumption is illustrated in Figure 2. The profiles of valve lift, valve velocity, and valve acceleration are presented in Figures 3, 4, and 5 respectively.

It can be seen from Figure 2 that: when the engine rotational speed is in the range of 0–5000 r/min, before optimization, as the engine rotational speed increases, the power consumption gradually increases, and the maximum power consumption is 238 W; after optimization, as the engine rotational speed increases, the power consumption gradually increases, and the maximum power consumption is 150 W. At the same time, the power consumption after optimization is always lower than that before optimization. It can be seen from Figure 3 that before optimization, the maximum valve lift is 7.3 mm, while after optimization, the maximum valve lift is 7.0 mm. It can be seen from Figure 4 that before optimization, the valve closing speed is 1.75 m/s, while after optimization, the valve closing speed is 1.05 m/s. It can be seen from Figure 5 that: before optimization, the peak acceleration generated by the valve is 1711.8 m/s^2; after optimization, the peak acceleration generated by the valve is 1105.2 m/s^2. Therefore, following optimization of the engine's variable valve drive system using the Particle Swarm Optimization algorithm, the valve seating velocity is reduced, resulting in lower noise levels and enhanced motion stability.

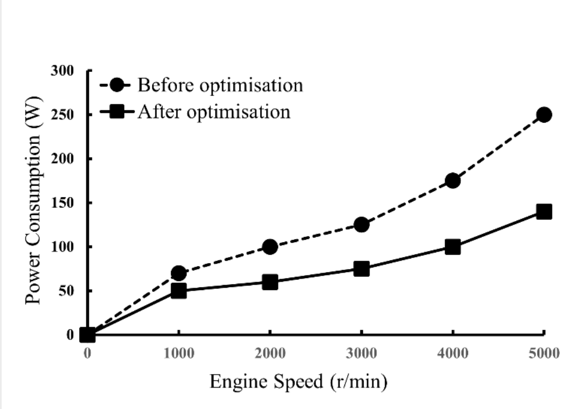

Figure 2. Engine power consumption

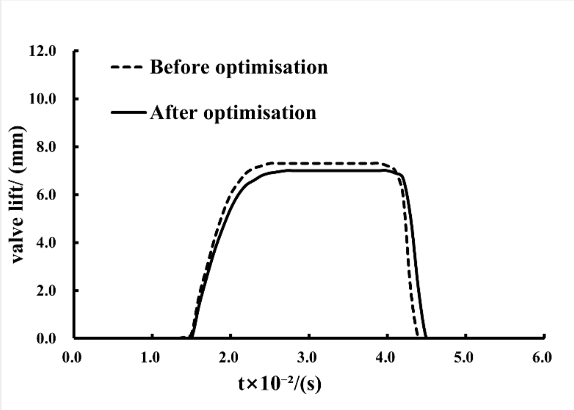

Figure 3. Valve lift

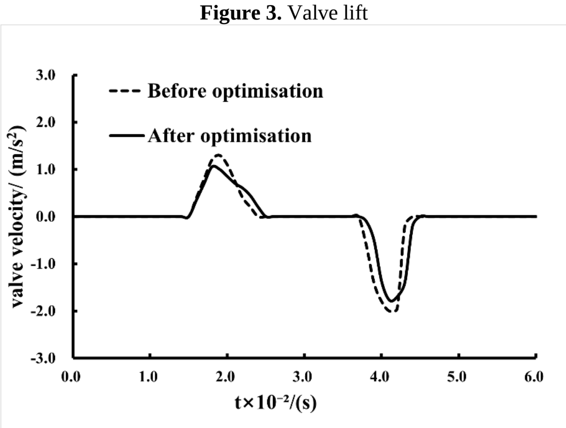

Figure 4. Valve speed

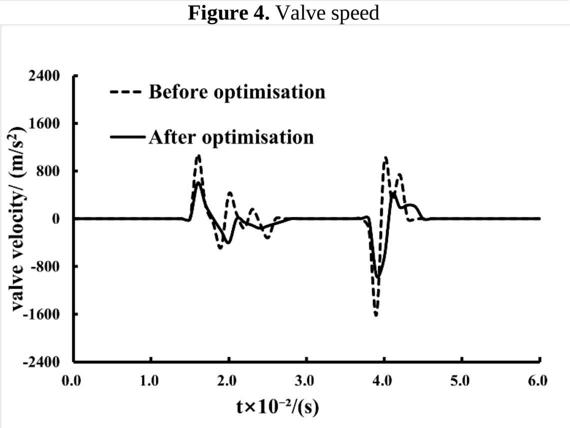

Figure 5. Valve acceleration

5. FURTHER DISCUSSION AND FUTURE PROSPECTS

In this study, an improved Particle Swarm Optimization (PSO) algorithm was employed to optimize key parameters in the engine's variable valve drive system. The simulation results verified that this method is capable of significantly reducing energy consumption and improving valve motion stability. While the current research has achieved promising results, several areas remain worthy of further investigation and development.

5.1. Engineering Insights from the Results

The post-optimization results indicate that the system exhibits reduced power consumption across a wide range of engine speeds, especially in high-speed conditions. This suggests the proposed optimization approach not only enhances fuel efficiency but also contributes to the design of more sustainable engine architectures. Furthermore, the reduction in valve lift, closing speed, and acceleration fluctuations improves the mechanical integrity of the valve train, reducing the likelihood of fatigue failure, valve floating, or sealing instability under dynamic loads.

Additionally, the design constraints integrated into the optimization—such as limiting hydraulic system pressure and maintaining valve closure integrity—ensure that the optimized parameters remain within the feasible boundaries of physical implementation. The successful regulation of the accumulator pre-charge volume and piston area serves as a reference for the practical deployment of fluid-mechanical systems where energy efficiency and mechanical reliability must be balanced.

5.2. Need for Multi-Objective Optimization

The current optimization approach targets a single-objective function—minimizing system power consumption. However, real-world engine applications require balancing multiple performance indicators, including noise reduction, emission control, cost minimization, and dynamic responsiveness. Future studies should incorporate multi-objective optimization methods using Pareto front analysis to identify trade-offs among these competing objectives and determine optimal compromises suited for practical implementation.

5.3. Integration with Real-Time Adaptive Control

The PSO algorithm utilized in this study operates offline, relying on predefined system models and static boundary conditions. In actual engine operations, working conditions—such as engine load, speed, and temperature—fluctuate frequently and unpredictably. Therefore, future work could integrate PSO with adaptive control frameworks, such as fuzzy logic controllers or reinforcement learning agents, to allow dynamic, real-time adjustment of valve parameters. This would enable optimal valve actuation across the entire operational envelope.

5.4. Potential for Hybrid Intelligent Algorithms

While the improved PSO algorithm demonstrated satisfactory convergence and global search performance, the risk of premature convergence and entrapment in local optima remains. To further improve optimization robustness, future research could explore hybrid intelligent algorithms, such as PSO combined with Genetic Algorithms (GA), Ant Colony Optimization (ACO), or Artificial Neural Networks (ANN). These hybrid approaches may enhance both global search efficiency and predictive adaptability under complex and non-stationary conditions.

5.5. Experimental Validation and Model Calibration

The present study relied on simulation using MATLAB to verify the effectiveness of the optimization strategy. However, the model's accuracy is highly dependent on the validity of initial parameters and boundary assumptions. For greater reliability and real-world applicability, future work should incorporate experimental validation through engine bench tests or high-speed data acquisition of valve lift and in-cylinder pressure. Such empirical data will support model refinement and ensure that the optimization approach aligns with practical mechanical behaviors and performance constraints.

5.6. Environmental and Economic Assessments

Beyond technical performance, environmental impact and cost-effectiveness must also be evaluated. Future studies may employ Life Cycle Assessment (LCA) frameworks to quantify the effects of optimization strategies on fuel savings, carbon emissions, and manufacturing costs across the engine's lifespan. Integrating economic and environmental metrics with performance-based optimization can guide manufacturers in developing greener, more cost-efficient engine solutions aligned with global sustainability objectives.

5.7. Summary and Outlook

In summary, the application of Particle Swarm Optimization in optimizing the variable valve drive system has demonstrated its potential for improving energy efficiency and mechanical stability. The algorithm's adaptability and convergence speed make it a valuable tool in engineering system design. Looking ahead, the integration of PSO with multi-objective frameworks, real-time control systems, and experimental feedback mechanisms will further elevate its utility. Such advancements will contribute to the development of intelligent, energy-efficient, and environmentally sustainable internal combustion engine technologies.

6. CONCLUSION

Aiming at the problems of large energy consumption and unstable movement during the engine operation, a variable valve drive system is designed, the design parameters are optimized using the Particle Swarm Optimization algorithm, and the engine's operational performance is assessed through simulation. The primary conclusions drawn from this study are as follows:

1. Integrated dynamic model for a dual-pump hydraulic VVD system:

We formulate an integrated nonlinear dynamic model that simultaneously considers the accumulator pressure dynamics, dual-pump coupling, valve motion, and hydraulic cylinder behavior under realistic constraints (pressure limits, valve closure reliability, and robustness to in-cylinder pressure fluctuations).

2. Power-oriented objective function with robustness constraints:

Unlike many existing PSO-based VVA studies that focus mainly on valve timing profiles or combustion performance, our work defines a power-consumption-oriented objective function explicitly targeting the reduction of hydraulic pump power, while incorporating constraints on valve lift variation, maximum system pressure, and cycle-to-cycle robustness.

3. Improved PSO strategy tailored to the VVD problem:

We employ an improved PSO algorithm with a variable inertia-weight strategy to enhance global search performance and convergence speed for this particular hydraulic actuation problem.

4. Comprehensive quantitative evaluation of valve motion quality:

The study not only reports power reduction but also systematically evaluates valve lift, velocity, and acceleration (including their peaks) before and after optimization, linking these dynamic characteristics to noise reduction and mechanical stability.

These clarified contributions distinguish the present work from previous PSO applications to valve actuation and show how our study advances the state of knowledge in energy-efficient hydraulic VVD systems.

- Before optimization, the engine working power consumption is large, resulting in relatively serious waste of resources. After optimization, the engine working power consumption is small, effectively saving resources.
- Optimizing the variable valve design parameters using the Particle Swarm Optimization algorithm can effectively reduce the peak valve lift values, valve speed, and valve acceleration, thus improving the stability of valve movement, with a significant optimization effect.
- Using Matlab software to simulate the optimized parameters of the engine variable valve can simulate the actual working conditions of the engine, can test the working effect of the engine, improve the design efficiency, and avoid waste of resources caused by unreasonable design.

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