Simulation Study of Hydraulic Differential Drive Free-Piston Engine

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Abstract

The hydraulic free piston engine is a complex mechanical-electro-liquid system, in order to simplify the complex system of the single hydraulic free piston engine, a new method for the driving of hydraulic free piston engine is proposed. Hydraulic differential drive achieves the compression stroke automatically rather than special recovery system. The structure and principle of hydraulic differential drive free-piston engine are analyzed and the mathematical model is established based on the piston force analysis and the hydraulic system working principle. In addition, the control strategy of this novel hydraulic driving engine is also introduced. Finally, the transient results of dynamics are obtained through simulation. Then we compare our results to the ones from the hydraulic free piston engine made by the company Innas. The results show that: 1) the simplified engine can achieve the similar performance of the Innas BV concept. 2) The maximum frequency and the maximum power of the engine are increased. 3) The indicated work of the novel engine decreases and the BSFC increases by 2.3%.

Introduction

The potential advantages of the HFPE, such as higher thermal efficiency, lower frictional losses and better emission performance, make the research of HFPE flourishing [1, 2, 3, 4]. However, compared with the conventional internal combustion engine, the HFPE has its own disadvantages [2, 3]: The variation of TDC and BDC and the complex hydraulic system prevent it from being widespread in practical application. Therefore the appropriate hydraulic system for the stability of HFPE is still a concern in this field.

Throughout the history of HFPE, it has been presented as an economical and high efficiency power unit for a fork-lift truck by Innas Company [4]. From then on, the free piston engine has attracted many researchers in this field: Zero-dimensional modelling of free piston engines has been discussed [5, 6, 7, 8]. Some literatures [4, 9, 10, 11] have investigated free piston engines using multidimensional simulation models. In either case, the numerical methods could achieve the instantaneous dynamics of the engine piston with a high degree of precision. Actually, the 1D model of HFPE have been analyzed by many researchers for the control strategy, S. Tikkanen etc deem that the energy balance principle in the control of HFPE's compression ratio works and that the control system is stable, they also found that the load of the HPFE should be constant but can change within certain limits [12]. Mikalsen R etc also researched the free piston engine control [13]. What's more, the 1D solution makes a contribution to analyzing the dynamic characters of HFPE.

In this paper, a new method for the driving of hydraulic free piston engine is proposed. Hydraulic differential drive achieves the compression stroke automatically rather than special recovery system. The structure and principle of hydraulic differential drive free piston engine is analyzed and the mathematical model is established based on the piston force analysis and the hydraulic system working principle. We get the corresponding results according to the mathematical models of the Inna BV HFPE and the novel HFPE. In addition, the control strategy of this novel hydraulic driving engine is also introduced. Finally, the transient results of dynamics are obtained through simulation. Then we make a compare with the hydraulic free piston engine from Innas Company and get the results.

Novel Driving Concept of HFPE

The prototype of HFPE from Innas BV is shown in Figure 1 [5].The HFPE is a two-stroke diesel engine with uniflow scavenging and direct fuel injection. A hydraulic-electronic unit injector from Caterpillar is used for the injection system. The prototype of the HFPE consists of the hydraulic pump part and the combustion engine part, the pump chamber and the compression chamber. The check chamber restricts the rebound of the piston around BDC after the expansion stroke. The pump chamber delivers one part of the high pressure fuel in the expansion stroke. The other is delivered by the check chamber in the compression stroke. The compression chamber completes the compression stroke by the hydraulic power stored in the compression accumulator. The frequency control valve is used to control the working frequency of the HFPE. The specific working process can be obtained in literature [2]. However, the working process of the novel HFPE is different from the Innas one. The novel engine combines the pump chamber with the compression chamber. The working process can be shortly described as follows. During the compression stroke, the piston assembly is driven by the hydraulic energy from the secondary high pressure to compress the in-cylinder gas. After the frequency control valve opens and the piston assembly moves to TDC, the pump piston sucks the secondary high pressure oil automatically through the suction valve when the piston reaches the given position. When the piston assembly approaches TDC, fuel is injected into the combustion chamber and burns. During the expansion stroke, the combustion energy of the fuel is absorbed by the high pressure hydraulic system. High-pressure oil is output by the pump piston through the pressure valve. Finally the high pressure fluid passes the CPR load and the generated secondary pressure fluid reserves in the accumulator. The novel HFPE prototype, shown in Figure 2, is built to validate the feasibility of the technical scheme that discussed above.

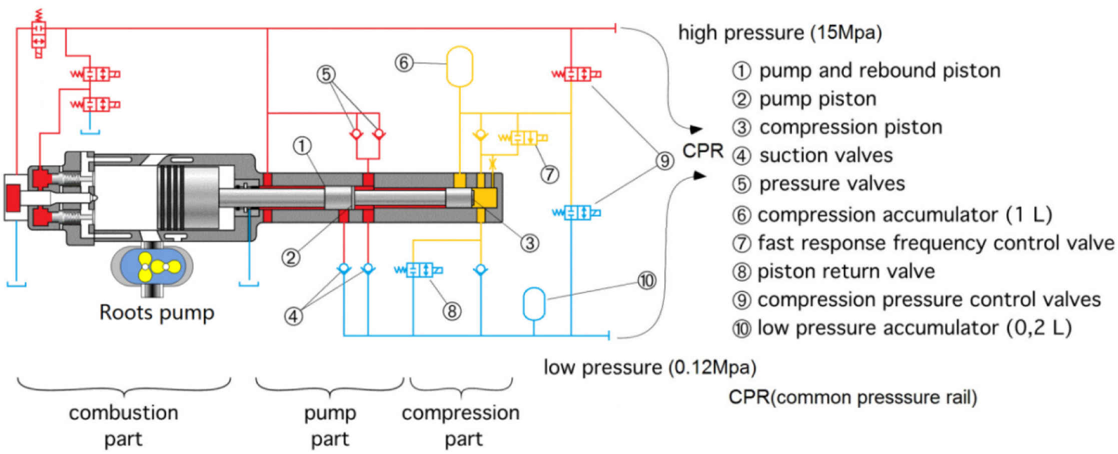

Figure 1. The hydraulic free piston engine concepts from Innas BV

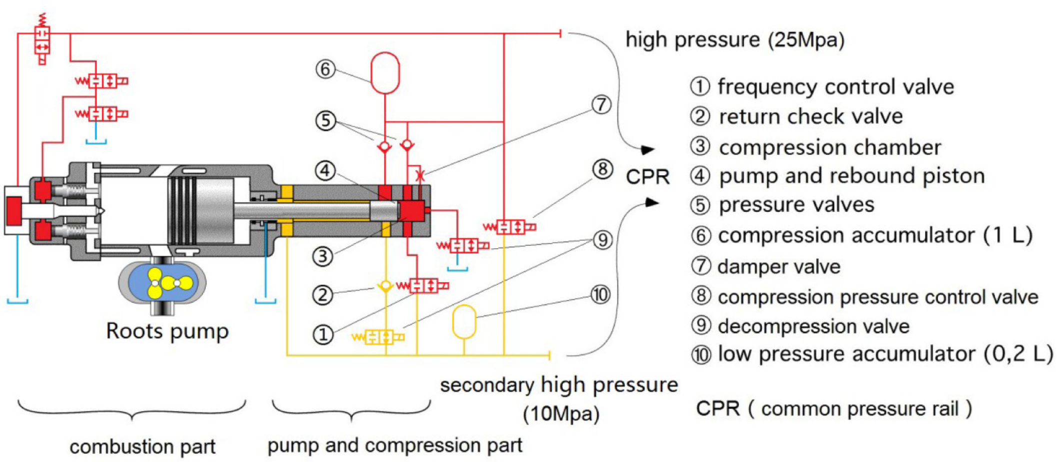

Figure 2. The hydraulic differential driving free piston engine concepts

Simulation Model of Engine

Piston Dynamics

The force analysis of HFPE is shown in Figure 3. A dynamic model of the piston motion can be derived from Newton's 2nd law. The model takes account of the different forces acting on the piston towards the piston centre and is given by the following equation:

$$p_3S_3 + p_2S_2 - p_1S_1 - \mathrm{sgn}(\dot{x})F_f = m_{\mathrm{Piston}} \frac{d^2x}{dt^2} \tag{1}$$

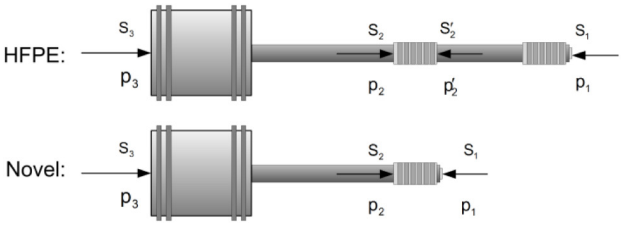

Figure 3. The force on the piston

Engine Model

The gas force in-cylinder may be more complex. However, we can get it indirectly. Based on the first law of thermodynamics:

$$\frac{\mathrm{d}U}{\mathrm{d}t} = \frac{\mathrm{d}Q}{\mathrm{d}t} - \frac{\mathrm{d}W}{\mathrm{d}t} + \sum \frac{\mathrm{d}m_{\mathrm{io}}}{\mathrm{d}t} h_{\mathrm{io}} \tag{2}$$

By the enthalpy definition, we have:

$$\frac{\mathrm{d}H}{\mathrm{d}t} = \frac{\mathrm{d}U}{\mathrm{d}t} + p\frac{\mathrm{d}V}{\mathrm{d}t} + V\frac{\mathrm{d}p}{\mathrm{d}t} \tag{3}$$

The enthalpy can be also described as a sum of particular masses from the system multiplied by their specific enthalpies:

$$H = \sum_n m_n h_n \tag{4}$$

The HFPE model is limited to the specific gases then we can get the following equation by dealing with equation (2, 3, 4):

$$\frac{\mathrm{d}Q}{\mathrm{d}t} + V\frac{\mathrm{d}p}{\mathrm{d}t} + \sum \frac{\mathrm{d}m_{\mathrm{io}}}{\mathrm{d}t} h_{\mathrm{io}} = \sum_n m_n \frac{\mathrm{d}h_n}{\mathrm{d}t} + \sum_n \frac{\mathrm{d}m_n}{\mathrm{d}t} h_n \tag{5}$$

In order to get the rate of temperature change of the open system, the equation (5) can be changed as:

$$\frac{\mathrm{d}Q}{\mathrm{d}t} + V\frac{\mathrm{d}p}{\mathrm{d}t} + \sum \frac{\mathrm{d}m_{\mathrm{io}}}{\mathrm{d}t} h_{\mathrm{io}} = \frac{\mathrm{d}T}{\mathrm{d}t} \sum_n m_n \frac{\mathrm{d}h_n}{\mathrm{d}T} + \sum_n \frac{\mathrm{d}m_n}{\mathrm{d}t} h_n \tag{6}$$

From the ideal gas law:

$$pV = T\sum_n m_n r_n \tag{7}$$

Then the equation (7) can be expressed as:

$$p\frac{\mathrm{d}V}{\mathrm{d}t} + V\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{\mathrm{d}T}{\mathrm{d}t}\sum_n m_n r_n + T\sum_n \frac{\mathrm{d}m_n}{\mathrm{d}t} r_n \tag{8}$$

With equations (3) and (5), the rate of temperature change of the open system can be described as follows:

$$\begin{cases} \dfrac{\mathrm{d}T}{\mathrm{d}t} = \dfrac{\dfrac{\mathrm{d}Q}{\mathrm{d}t} + \sum \dfrac{\mathrm{d}m_{\mathrm{io}}}{\mathrm{d}t} h_{\mathrm{io}} - p\dfrac{\mathrm{d}V}{\mathrm{d}t} - \sum_n \dfrac{\mathrm{d}m_n}{\mathrm{d}t} h_n + T\sum_n \dfrac{\mathrm{d}m_n}{\mathrm{d}t} r_n}{\sum_n m_n C_{pn} - \sum_n m_n r_n} \\ C_{pn} = \dfrac{\mathrm{d}h_n}{\mathrm{d}T} \end{cases} \tag{9}$$

From the ideal gas law, we can get:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \rho\left(T\frac{\mathrm{d}r}{\mathrm{d}t} + r\frac{\mathrm{d}T}{\mathrm{d}t}\right) + rT\frac{\mathrm{d}\rho}{\mathrm{d}t} \tag{10}$$

Very little research exists on modeling of the heat release in the free-piston engines. Researchers usually deal with the combustion process by Wiebe function:

$$\frac{dX}{dt} = 6.9\frac{(\mathrm{m}+1)}{t_b}\left(\frac{t-t_s}{t_b}\right)^{\mathrm{m}}.\exp\left[-6.9\left(\frac{t-t_s}{t_b}\right)^{\mathrm{m}+1}\right] \tag{11}$$

$$\frac{dQ_{\mathrm{Hr}}}{dt} = \mathrm{m}_{\mathrm{fuel}}.H_u.\eta_u.\frac{dX}{dt} \tag{12}$$

The scavenging model used to estimate the flows inlet and outlet of the cylinder is reduced to describe the flows through a pneumatic orifice. The flow through an area restriction is governed by pressure difference.

For a compressible gas, we have:

$$\frac{\mathrm{d}m_{\mathrm{io}}}{\mathrm{d}t} = A_{\mathrm{or}}C_{\mathrm{or}}C_{\mathrm{io}}\frac{p_{\mathrm{ori}}}{\sqrt{T_{\mathrm{ori}}}} \tag{13}$$

The flow parameter C_{io} is a function of the pressure ratio as follows [14]:

$$C_{io} = \begin{cases} \sqrt{\dfrac{2\gamma_{or}}{r(\gamma_{or}-1)}}\sqrt{\varepsilon^{\frac{2}{\gamma_{or}}} - \varepsilon^{\frac{\gamma_{or}+1}{\gamma_{or}}}}, & \varepsilon > \left(\dfrac{2}{\gamma_{or}+1}\right)^{\frac{\gamma_{or}}{\gamma_{or}-1}} \\ \sqrt{\dfrac{\gamma_{or}}{r}\left(\dfrac{2}{\gamma_{or}+1}\right)^{\frac{\gamma_{or}+1}{\gamma_{or}-1}}}, \varepsilon = \dfrac{p_{oro}}{p_{ori}}, & \varepsilon \le \left(\dfrac{2}{\gamma_{or}+1}\right)^{\frac{\gamma_{or}}{\gamma_{or}-1}} \end{cases} \tag{14}$$

In equation (14), when the mass flow rate is sonic, the flow parameter C_{io} is constant.

The engine model considers convection as the primary mode of heat transfer. The heat transfer by convection is modeled according to Woschni [15]:

$$\begin{cases} \dfrac{dQ_{Ht}}{dt} = \alpha_g \left(T - T_w\right) S_w \\ \alpha_g = 130 \dfrac{p^{0.8}}{T^{0.53} D^{0.2}} \left[C_1 \bar{\upsilon} + C_2 \dfrac{V T_{01}}{p_{01} V_{01}} \left(p - p_0\right) \right]^{0.8} \end{cases} \tag{15}$$

Hydraulic Pump Model

The pressure and the flow of the hydraulic chamber are calculated by the classical expression:

$$\begin{cases} \dfrac{dp_j}{dt} = \dfrac{B q_j}{V_j} \\ q_j = q_{j0} \pm S_j \dfrac{dx}{dt} \pm S_V \dfrac{dx_V}{dt} \end{cases} \tag{16}$$

The index j = 1, 2 represent the check chamber, the pump chamber (compression chamber), respectively.

The equation (16) chooses plus sign when calculating the pressure of the check chamber. The flow through the port on the check chamber is given by equation (17). The flow through the port on the FCV and the check valves also can be calculated by this form.

$$q_{10} = C_{ce} A_1 \sqrt{\dfrac{2\left|p_H - p_1\right|}{\rho_O}} \tag{17}$$

The check valve model is a spring loaded check valve model with the spool dynamics incorporated. The check valve coordinate was built with the origin at the spool complete closing position. When $x_V < x_{Vmax}$ the dynamic model of the spool is as follows:

$$m_V \dfrac{d^2 x_V}{dt^2} + c_{Vf} \dfrac{dx_V}{dt} \pm F_{Vfl} + F_{VK} + \left(p_{Vin} - p_{Vout}\right) S_V = 0 \tag{18}$$

When $x_V = x_{Vmax}$:

$$\dfrac{dx_V}{dt} = -C_{x_{Vmax}} \dfrac{dx_V}{dt} \tag{19}$$

The flow force without transient flow force incorporated and the effective flow area of the spool are approximately given by:

$$\begin{cases} F_{Vfl} = 2 C_V^2 A(x_V) \left|\Delta p_V\right| \cos\beta \\ A(x_V) = \pi x_V \sin\beta \left(d_V - x_V \sin\beta \cos\beta\right) \end{cases} \tag{20}$$

The flow rate of the check valve is given by:

$$q_V = C_V A(x_V) \sqrt{\dfrac{2\left|p_{Vin} - p_{Vout}\right|}{\rho_O}} + \dfrac{\pi}{4} D_V^2 \dot{x}_V \tag{21}$$

The spool movement effect on the flow is considered. The FCV is an electro-hydraulic servo valve. Its spool dynamic model is reduced to a second-order system as follows:

$$X_{FCV}(s) = \dfrac{\omega^2}{\omega^2 + 2\varsigma\omega s + s^2} \tag{22}$$

The flow is calculated by the equation (20) and the effective flow area is directly proportional to the valve spool displacement.

The hydraulic accumulator model based on the adiabatic process assumption is given by:

$$p_{A0} V_{A0}^{\gamma} = p_A \left(V_{A0} - V_A\right)^{\gamma} \tag{23}$$

The hydraulic port model describes variation of the cross sectional area of the chief port in the pump chamber (compression chamber). The cross sectional area is a function of the piston displacement.

Engine Control Strategy

The control of free-piston engines is a special challenge but it is attracted many researchers. Most researches above indicate that an accurate dead position control is significant for achieving a satisfactory engine performance and the load characteristics play an important role. The excellent controllability of such kind of free-piston engines has been demonstrated by Achten et al [5]. The HFPE control strategy is a standard feedback control method. The HFPE feedback control system contains different functions for its different operations. The novel HFPE control strategy is similar with the conventional one and our early efforts can be obtained in literature [16].

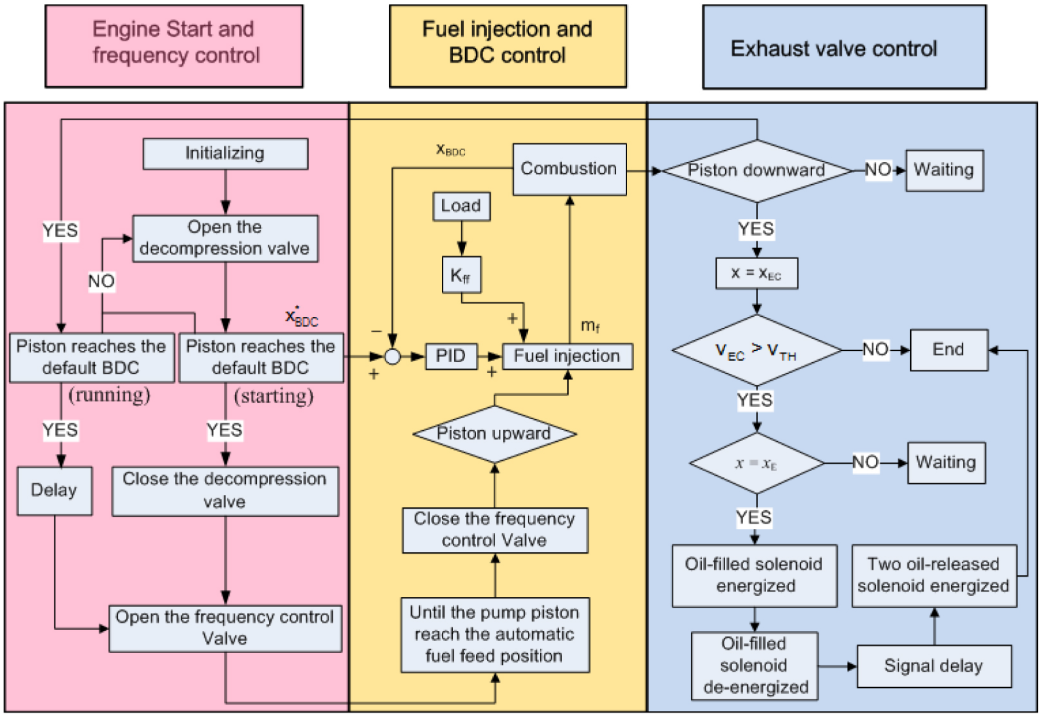

Figure 4. The control strategy of the novel HFPE

Notably, while the crankshaft engine started by starting motor, HFPE started by hydraulic energy. The pump pressure in the novel HFPE will decrease suddenly if the hydraulic leakage happens, therefore it needn't to adjust the compression pressure and cooperate with the FCV to avoid squirm of the piston for hydraulic leakages. It is very different from the Innas concept. To make HFPE starting successful, hydraulic leakage needn't be considered as the Innas concept one. As show in Figure 4, if the reset valves are in working condition, the piston will move to the TDC before engine starting. Therefore the reset valves must be open before the engine starting.

The starting control strategy is also shown in Figure 4, as the leakage of pump chamber is avoided, the piston won't creep to TDC without starting control strategy. However, in the conventional HFPE, when the chief-port is opened and FCV still not be opened, the piston will move to TDC uncontrollable for the leakage. The engine starting will be failed for lack of compression energy. So the novel driving model improves the controllability performance of HFPE.

HFPE's working frequency is controlled by the FCV. Innas BV presents a Pulse Phase Modulation (PPM) control method which has been applied to the control the HFPE working frequency. The piston position feedback control strategy with a PID controller based on the variation of BDC and the load is shown in Figure 4. The piston velocity during the expansion stroke is an important characteristic of normal engine operation. If the operation is abnormal, the piston velocity is much lower. So the abnormal engine operation can be judged by the variation between the real piston velocity v_{EC} and the standard piston velocity v_{TH} at the piston position x_{EC} before the aim piston position x_E of valve open. If the variation is larger than the error Δv, ECU will not send the valve open signal. The standard piston velocity v_{TH} can be obtained from experiment, and the whole control strategy of exhaust valves control is shown in Figure 4 as well.

On the other hand, the HFPE is a very fast moving system, so the control system must be sensitive and rapid. All control valves are high speed valves and the sensors are sensitive. The fast and correct response of the control system determines not only the performance but also the service life of the HFPE. An inaccurate response may damage the mechanical structure of the engine.

Comparison and Results

In order to demonstrate the performance of the dynamic simulation, the specific technical parameters should be given, and they are shown in Table 1.

In order to analyze the characteristics of the novel engine, we fetch some curves of both engines in the condition that the FCV is full open. The specific technical parameters in Table 1 show the changes from HFPE to the novel engine. Figure 5 shows the piston displacement and velocity at the similar operation condition. It seems that both of them have the similar dynamical features. In fact, the compression pressure of them isn't much different and the fuel injection quality is the same. What's more, the hydraulic load is alike. All these reasons lead to the similar results. The piston velocity of the novel engine shows the same trend when it is compared to the conventional HFPE. Distinctively, the lower compression pressure (10 MPa) makes less contribution on accelerating the piston which slows down the piston early. On the other hand, the novel engine gets rid of specialized compression pump, which leads to the hydraulic resistance decrease. All these reasons result in the increase of piston velocity in expansion stroke and the specific characteristics are shown in Figure 6.

Table 1. Main parameters of the HFPE and the novel engine

	HFPE	Novel		HFPE	Novel
parameter	Value	Value	parameter	Value	Value
A_1	201	201	p_2'	25	——
B	650	650	p_{A0}	10	10
C_{FCV}	0.6	0.6	p_H	15	25
C_V	0.65	0.65	S, S_3	7616	7616
C_{cc}	0.65	0.65	S_1	440	440
C_{or}	0.72	0.72	S_2	102	102
C_{xVmax}	0.45	0.45	S_2'	204	——
c_{Vf}	3.5/40	3.5/4.0	V_{A0}	4.0	4.0
d_V	20/25	20/25	x_{Vmax}	7.0	7.0
m	1.5	1.5	β	0.785	0.785
m_{Piston}	4.2	4.2	γ	1.4	1.4
m_{fuel}	24	24	η_u	0.9	0.9
H_u	42500	42500	ρ_O	850	850
p_1	15	25	ζ	0.3	0.3
p_2	25	10	ω	100	100

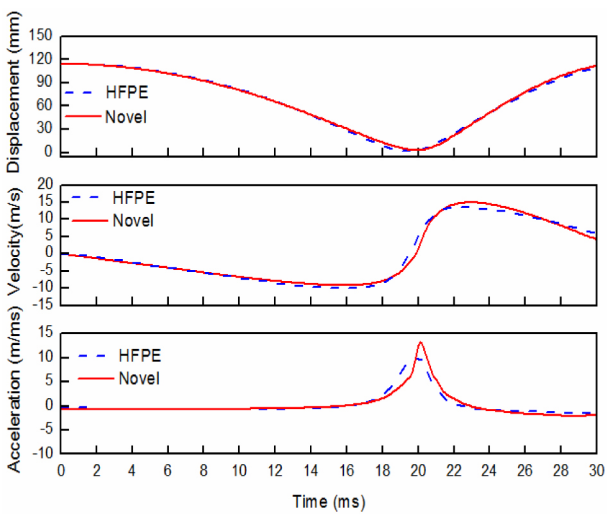

Figure 5. Piston displacements, velocities vs. time

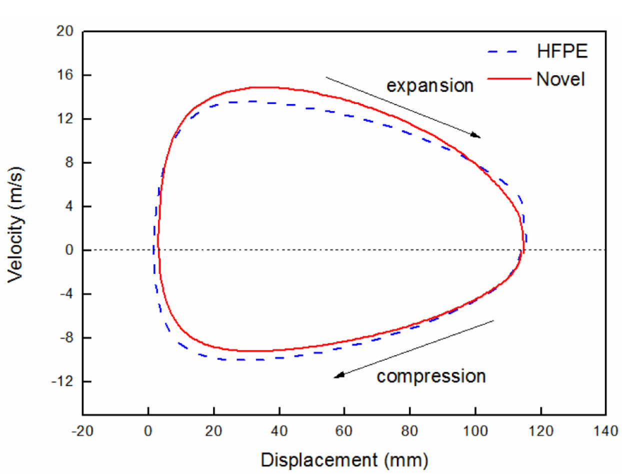

Figure 6. Piston velocities vs. displacement

Of course, there must be some difference in these two kinds of engines. When the FCV is full open, the piston frequency is obviously different for the novel engine has higher compression pressure. Therefore the novel engine has the higher frequency. Because the differential pressure of the load and the secondary high pressure is the same as the conventional HFPE, and the same sectional area of pump chamber, the power of the novel engine is enhanced based on equation (24).

$$P_e = \Delta p . S_1 . L_{stroke} . f \tag{24}$$

Actually, in Figure 7, it witnesses the preceding inference. The frequency of the novel engine is 32 Hz, and it increases 6.7% compared with the 30 Hz of conventional HFPE [17].

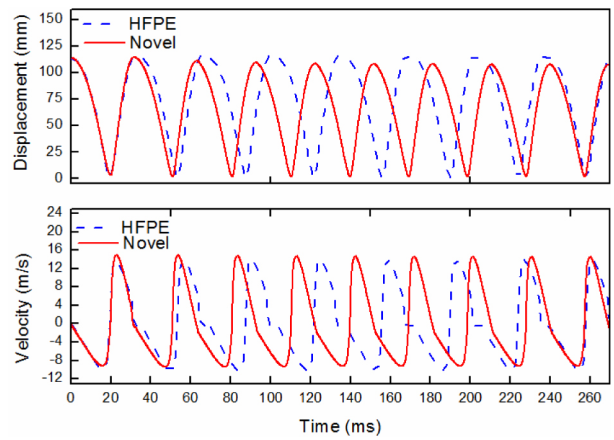

Figure 7. Piston displacements, velocity vs. time in multiple cycles

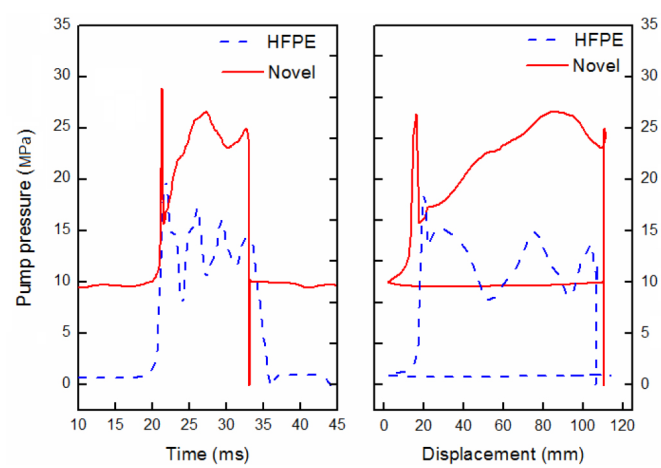

Figure 8. Pressures of pump chambers vs. time and displacement

Figure 8 illustrates that the hydraulic pressures of pump chamber. In the left figure, the pump chamber pressure fluctuates around 10 MPa and 0.12 MPa respectively in the compression stroke. However, the pump chamber pressure of the conventional HFPE doesn't reach 25 MPa for the pump cavity loss and lower initial pressure, the actual mean value of pump chamber pressure is less than 15 MPa. On the contrary, higher initial pump chamber pressure and less pump cavity loss lead to the increase of the maximum pump pressure, which results in the same difference of the initial and the final the pump pressure. On the other hand, the squirm of piston toward TDC leads to the pump pressure decreases suddenly and the pump pressure falls to 0.1 MPa. What's more, the pressure difference of the pump chamber and rebound chamber reduces to a lower level. These characters are benefit for the frequency control strategy as we consider less about the unstability caused by the squirm of piston toward TDC in the conventional HFPE [2].

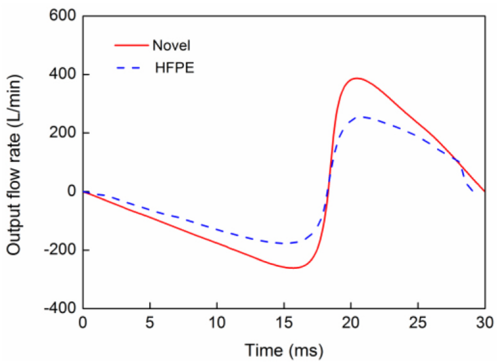

Figure 9. The output flow rate of HFPE and the novel engine vs. time

However, pump mean pressure is higher compared to the HFPE, power output is only increased slightly because power output is based on the difference between maximum and minimum pressure. Therefore the output flow rate is shown in Figure 9. The negative value means that the pump chamber sucks oil from the secondary pressure rail automatically. As the flow rate is decided by differential pressure, the novel engine increases this case. Hence, the output flow rate is higher than the conventional HFPE during the compression stroke and the expansion stroke. Actually, it side validates that the output power of this novel engine is improved.

Additionally, although the equivalent fuel injection quality is used in the simulation, BSFC may be different for the compression pressure is not the same. BSFC can be calculated by equation (25) and it indicates that BSFC of HFPE is decided by the indicated work. The higher compression pressure of conventional HFPE may lead to the TDC moves to the cylinder head. The compression ratio will increase a little and the maximum value of pressure in-cylinder enlarges. In Figure 10, the indicated work of the novel engine decreases and the BSFC increases by 2.3% based on equation (25) when it is compared with the conventional HFPE. Even through the output power increases by 6.7% and BSFC increases 2.3%, the economic improvement is achieved with the simplification of the hydraulic system. Less hydraulic components and the similar performance implement our original intention. However, the 1D simulation of in-cylinder process may be not accuracy and the further experiment work should be carried out.

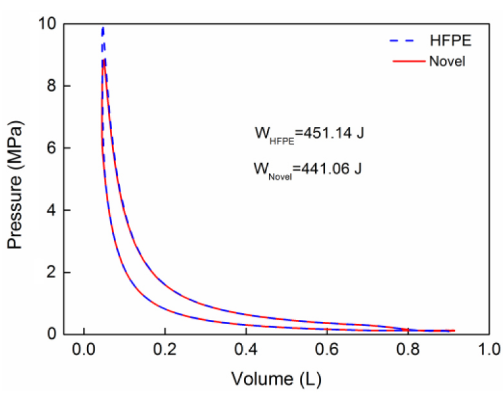

Figure 10. In-cylinder pressures of HFPE and novel engine vs. time

$$b_i = \frac{B_i}{P_i} \times 10^3 = \frac{m_{fuel} \times f \times 3600}{W_i \times f} = \frac{3600 m_{fuel}}{W_i} \tag{25}$$

Summary/Conclusions

From the compare of the Innas BV free piston engine and the novel one, we can get the following conclusions: The control strategy of this novel hydraulic driving engine is similar with the traditional one. However, hydraulic differential drive achieves the compression stroke automatically rather than special recovery system. The new method for the driving of hydraulic free piston engine can simplify the complex system of the single hydraulic free piston engine. The transient results of dynamics obtained through the mathematical model show that: the simplified engine can achieve the similar dynamic performance, such as displacement, velocity and accelerate compared to the Innas concept. The maximum frequency and the maximum power of the novel engine are increased by 6.7% for its rapid corresponding and simplified structure. However, BSFC also increases by 2.3% and the verification test should be carried out further.

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Definitions/Abbreviations

Definitions

A_1 - cross-sectional area of the check chamber port

A_{FCV} - the constant amplitude

A_V - check valve effective flow area

A_{or} - effective area of the mass flow

B - oil bulk modulus, MPa

B_i - fuel consumption per hour, g/h

b_i - Brake Specific Fuel Consumption, g/(kW.h)

C_{FCV} - flow coefficient of the frequency control valve

C_V - flow coefficient of the check valve

C_{cc} - flow coefficient of the check chamber port

C_{io} - mass flow parameter

C_{or} - flow coefficient

C_p - specific heat for constant pressure, J/(kg.K)

C_{xVmax} - spool speed coefficient

c_{Vf} - damping coefficient of the valve spool

d_V - check valve spool diameter, mm

F_{Vf1} - flow force applied to the check valve spool, N

F_{VK} - spring force of the check valve spring, N/mm

F_f - friction force applied to the piston, N

f - frequency of piston, Hz

m - combustion quality index

m - in-cylinder gas mass, kg

m_V - spool mass without spring mass incorporated, kg

m_{Piston} - piston moving mass, kg

m_{fuel} - fuel mass available in the cylinder, mg

m_{io} - gas mass entering and leaving the cylinder, kg

H - specific enthalpy of the in-cylinder gas, J/kg

H_u - the low heat value of the diesel, kJ/kg

n - group number of gases

h_{io} - specific enthalpy of the gas entering and leaving the cylinder, J/kg

j - 1, 2, and 3, the three hydraulic chambers

p, p_3 - in-cylinder gas pressure, MPa

p_1 - pressure of the compression chamber, MPa

p_2 - pressure of the rebound chamber, MPa

p'_2 - pressure of the pump chamber, MPa

p_A - pressure of the compression accumulator, MPa

p_{A0} - gas precharged pressure, MPa

P_i - Indicated power, kW

p_H - pressure of the high-pressure rail, MPa

p_{Vin} - pressure at the inlet port, MPa

p_{Vout} - pressure at the outlet port, MPa

P_j - pressures of the three hydraulic chambers, MPa

p_{ori} - upstream pressure, MPa

p_{oro} - downstream pressure, MPa

Q - heat transferred to the gas, J

Q_{Hr} - heat released by fuel combustion, J

Q_{Ht} - thermal exchange heat, J

q_i - volumetric flow rate, L/min

q_{FCV} - flow through the frequency control valve, L/min

t - time, s

t_b - duration of the combustion

t_s - the starting time of combustion

U - internal energy

V - in-cylinder gas volume, L

V_A - oil volume, L

V_{A0} - nominal capacity, L

V_j - volumes of the three hydraulic chambers, L

v_{EC} - the real piston velocity, m/s

v_{TH} - the standard piston velocity, m/s

W - work done by the in-cylinder gas, J

W_i - Indicated work, J

X - the burned mass fraction of fuel

X_{FCV} - transfer function for spool motion of FCV

x - piston displacement, mm

x_E - the aim piston position, mm

x_{EC} - the real piston position, mm

x_V - hydraulic valve spool displacement, mm

x_{Vmax} - maximum spool displacement, mm

Greeks

α_g - heat transfer coefficient, W/(m^2.K)

β - efflux angle (constant value), rad

γ - adiabatic index of the accumulator gas

γ_{or} - specific heat ratio

η_u - combustion efficiency

ρ - in-cylinder gas density, kg/m^3

ρ_o - hydraulic oil density, kg/m^3

ζ - damping ratio

ω - natural frequency, Hz

Abbreviations

CPR - Common Pressure Rail

BDC - Bottom Dead Center

BSFC - Brake Specific Fuel Consumption, g/(kW.h)

ECU - Electronic Control Unit

FCV - Fast Control Valve

HFPE - Hydraulic Free Piston Engine

PID - proportional-integral and derivative

PPM - Pulse Phase Modulation

TDC - Top Dead Center

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