

Design and Modeling of a Novel Internal Combustion Engine with Direct Hydraulic Power Take-off

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ABSTRACT

This paper introduces a Hydraulic Linear Engine (HLE) concept and describes a model to simulate instantaneous engine behavior. The United States Environmental Protection Agency has developed an HLE prototype as an evolution of their previous six-cylinder, four-stroke, free-piston engine (FPE) hardware. The HLE design extracts work hydraulically, in a fashion identical to the initial FPE, and is intended for use in a series hydraulic hybrid vehicle. Unlike the FPE, however, the HLE utilizes a crank for improved timing control and increased robustness. Preliminary experimental results show significant speed fluctuations and cylinder imbalance that require careful controls design. This paper also introduces a model of the HLE that exhibits similar behavior, making it an indispensable tool for controls design. Further, the model's behavior is evaluated over a range of operating conditions currently unobtainable by the experimental setup.

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INTRODUCTION

A number of institutions are currently investigating the free-piston engine (FPE) for its potentially low cost and highly efficient power delivery [1,2,3,4,5,6,7,18,19,20]. Without a crank, an FPE has fewer rubbing parts than a conventional engine and significantly lower piston side loads [1,2]. These effects compound to reduce overall friction. Additionally, the stroke length is variable and enables compression ratio optimization for a variety of fuels or combustion strategies.

In a conventional engine, power is transmitted through the crank shaft to the load. Conversely, an FPE extracts energy directly from the linear motion of the pistons via one of two mechanisms: electric or hydraulic power take-off.

For electrical power generation, a linear alternator is attached to the piston assembly. Sandia National Laboratories is working to develop an electric FPE and have reported thermal efficiencies as high as 56% when evaluating the concept with a rapid compression and expansion machine [3].

Alternatively, Xu and Chang have developed a two-stroke concept with 32% electrical generation efficiency [4].

A hydraulic FPE uses the linear piston motion as a reciprocating pump. Energy is stored in a high-pressure hydraulic accumulator or used to actuate a hydraulic motor. Netherlands based company Innas BV has created a single-cylinder, two-stroke, hydraulic FPE capable of indicated efficiencies up to 50% [5]. Researchers at the University of Minnesota are investigating an FPE that uses both electric and hydraulic power-take off [6].



Although current research shows that the concept holds promise, FPE design has yet to evolve as a practical alternative to conventional technology. While the added flexibility afforded by variable stroke length has advantages, it is also the biggest hurdle preventing a viable engine. Robust control is necessary to maintain consistent and efficient operation and to ensure damaging mechanical failures or combustion instabilities do not occur. Too much fuel added to an FPE will drive the piston assembly beyond the desired location and potentially into the cylinder head. Conversely, underestimating fuel will lower compression ratios and efficiency. In the worst case scenario, the engine will stall entirely.

In light of the aforementioned difficulties, the United States Environment Protection Agency (EPA) has modified their previously reported FPE prototype [1]. This paper begins with a review of the original FPE concept to illustrate the numerous components in common with the new engine. Next, the key features of the updated design are described. Because of mechanical damage to the prototype, engine data is limited, necessitating a model to evaluate engine trends and to supplement controller design. The model is subsequently expressed, followed by an assessment of engine behavior.

ENGINE DESIGN

The EPA has reported two prototype FPEs produced as part of a hydraulic hybrid vehicle development program [1]. The larger, six-cylinder design was capable of four-stroke operation and achieved hydraulic conversion efficiencies near 40%. The novel engine presented in this paper shares numerous components with this original design, as such, a review of the predecessor six-cylinder FPE follows.

Six-Cylinder FPE

The original EPA FPE concept, shown in Figure 1, consists of three dual-piston assemblies rigidly coupled by a rack and pinion mechanism. The six-cylinder geometry and fully flexible, electro-hydraulic valve train enabled a four-stroke cycle. To accomplish balanced operation, cylinders 3_a and 3_b compress and ignite simultaneously and are considered a single cylinder, i.e. cylinder 3. Ideally, the combined displacement of cylinders 3_a and 3_b is equivalent to the displacement of one of the center cylinders (1 or 2). As a result, cylinders 3_a and 3_b have a smaller diameter than cylinders 1 and 2. Likewise, cylinders 4_a and 4_b are considered a single cylinder and also possess the smaller diameter.

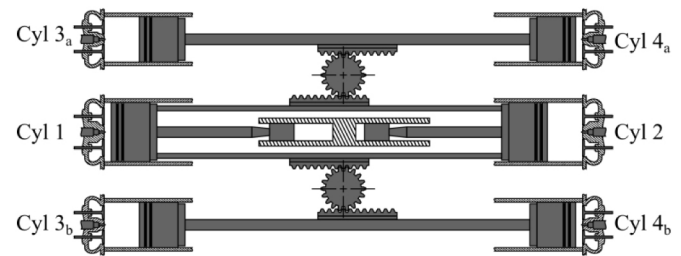


Figure 1. Six-cylinder FPE with hydraulic power take-off.

The center piston assembly acts as reciprocating pump for hydraulic power take-off. Highlighted in Figure 2, the engine draws hydraulic fluid from a Low Pressure (LP) reservoir or pumps hydraulic fluid into a High Pressure (HP) accumulator that stores energy in the form of compressed nitrogen gas, while check valves passively control flow. Using only check valves to control flow, the amount of work extracted from a set stroke length is fixed when assuming a steady HP accumulator pressure. Consequently, the power range of an FPE is limited because the operating frequency is fairly constant [2,7]. In order to adjust power output, the EPA designed the FPE with a circuit to bypass the LP check valves. During a specified fraction of each stroke, the bypass valves open to allow hydraulic fluid a path to low pressure and to prevent pumping to high pressure. While represented as a distinct path in Figure 2, the bypass is accomplished by physically forcing open the check valves using a set of actuating rods. The current design requires the bypass valves to open at the beginning of the stroke, before the hydraulic chamber is pressurized.

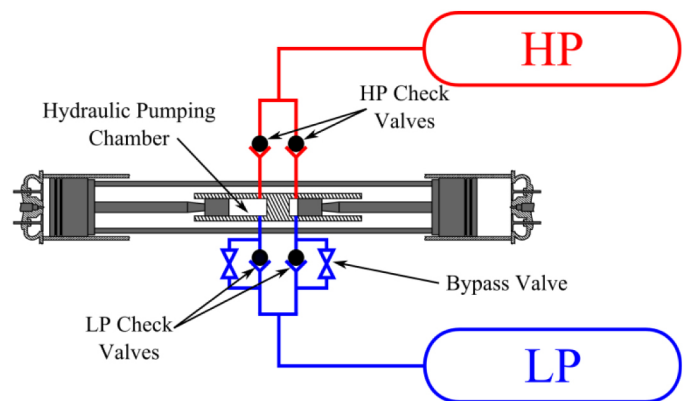


Figure 2. Hydraulic schematic of FPE including check valves to passively direct flow and bypass valves to control engine loading.

The intake and exhaust valves are a hydraulically-actuated, VVT system. The fully-flexible valve train allows for further customization and optimization of in-cylinder conditions, but creates a large hydraulic, accessory load. The FPE operates with compression ignition direct injection (CIDI) diesel combustion using a vertically-oriented,

hydraulically intensified fuel, HEUI-type injector. A magnetostrictive absolute position sensor monitors the linear piston location with 0.06 mm resolution and a hydraulic motoring circuit performs start-up.

Hydraulic Linear Engine

To address the robustness and control concerns associated with an FPE, the prototype has been modified. The new design is here referred to as a Hydraulic Linear Engine (HLE). As shown in Figure 3, the dual piston assembly associated with cylinders 3_b and 4_b of the six-cylinder FPE has been removed and replaced with a crank device mechanically balanced with existing hardware. Rather than delivering power to the load via the crank, as is the case for conventional engines, the HLE's crank constrains piston motion and actuates the valves. Power take-off remains hydraulic and the bypass valves continue to adjust load. The HLE utilizes CIDI diesel combustion but can be configured for port injection as well. The cylinder firing order is 1-2-4-3.

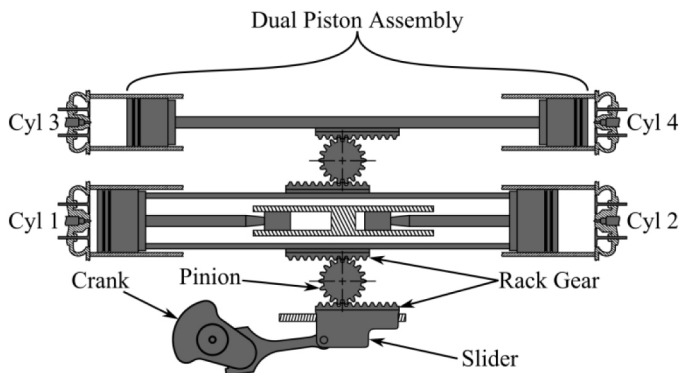


Figure 3. Four-cylinder hydraulic linear engine, utilizing a crank to constrain piston motion and actuate air valves.

While the crank eliminates some flexibility found in an FPE, many of the intended benefits are still present. The rack and pinion configuration reduces piston side loads and the HLE continues to have fewer moving parts than a conventional four-cylinder engine. Extracting power directly at the piston also reduces crank bearing loads.

The crank enables a number of traditional engine technologies. A starter motor turns the engine during start-up and a flywheel is present to ensure there is sufficient rotational kinetic energy available to carry the system beyond any given turn-around point. While the magnetostrictive sensor remains to monitor linear piston position, an absolute rotation encoder has been installed to record crank angle. The HLE crank also allows the engine to operate at higher speed, increasing the power output range.

Engine geometry requires a novel approach to cam-controlled valve timing. Shown in Figure 4, a cam depresses a plunger to pressurize an oil line. On the opposing end, the pressurized oil operates an actuator piston to open the valve. The process is analogous to a 'hydraulic pushrod'.

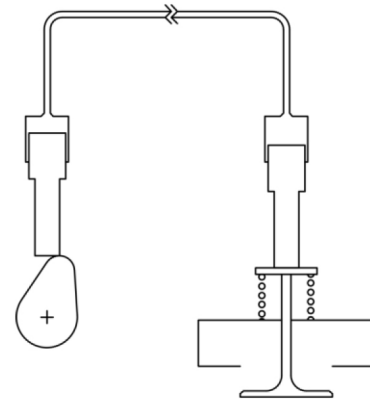


Figure 4. Cam-actuated, hydraulic air valves.

Because the HLE operates with a single crank and a small flywheel, the engine has low rotational inertial and exhibits minimal transients and fast start-up. At the same time, the engine responds to cylinder fueling discrepancies with significant speed fluctuations. Because cylinders 3 and 4 were designed for the original FPE, with a smaller diameter than cylinders 1 and 2, the HLE inherently has unbalanced cylinder displacements. It is therefore necessary to construct an HLE model to facilitate controls development with the purpose of correcting cylinder imbalance and to identify trends in engine performance in order to maximize useful engine testing on the limited lifespan of the prototype hardware. The described HLE configuration is a crucial development step more easily accomplished by converting and modifying existing FPE hardware than creating a completely new engine. An ideal HLE would likely have a balanced geometry designed specifically for its unique arrangement.

MODEL DESCRIPTION

The reported engine simulation was developed in MATLAB/Simulink and consists of three basic sub-models: dynamics, thermodynamics, and hydraulics. Figure 5 illustrates communication between individual components. The dynamics model uses pressure information to assess the torque acting on the crankshaft and calculates instantaneous engine speed. Gaseous pressure is provided by the thermodynamic subsystem which evaluates in-cylinder conditions and composition of each combustion chamber as a separate control volume. Meanwhile, the hydraulics model takes into account hydraulic compressibility and valve area to estimate pressure and flow.

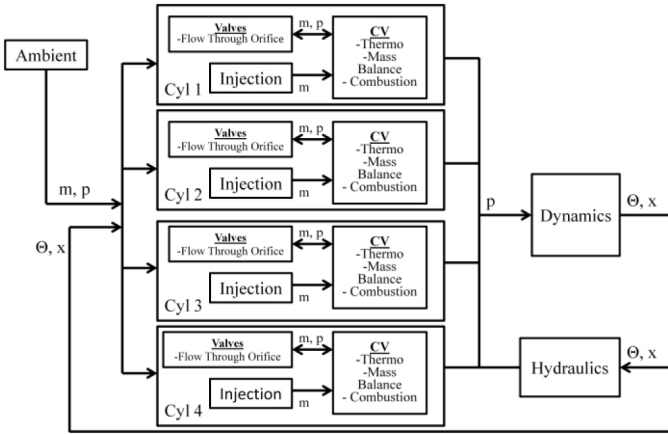


Figure 5. MATLAB / Simulink model structure.

Dynamics

Similar to a single-cylinder engine, the HLE has low rotational inertia and is subject to large speed fluctuations over the duration of a stroke. Filipi and Assanis [8] and Zweiri et al. [9,10] constructed models to capture the instantaneous dynamics of a single-cylinder engine. The model presented here uses the same principles.

A torque balance describes the rotation of the rigid crank,

$$J\ddot{\Theta} = \tau - \tau_f, \quad (1)$$

where J is the total rotating inertia, $\ddot{\Theta}$ is the angular acceleration, and τ_f is the frictional torque. Torque, τ , is a geometric projection of the forces acting on the piston, F , and is a function of crank angle, Θ , such that

$$\tau = FrG(\Theta), \quad (2)$$

where r is the crank radius and $G(\Theta)$ is the function

$$G(\Theta) = \left(\frac{\sin(\Theta + \beta)}{\cos \beta} \right). \quad (3)$$

The angle between the connection rod and axis of piston motion is defined as β while F is the sum pressure forces, F_p , and inertial forces, F_{in} .

Because the movement of the slider, the pistons, and the pinions are linearly coupled they are lumped together in a single inertial force balance as follows:

$$F_{in} = m_{eq}a, \quad (4)$$

where

$$m_{eq} = m_{12} + m_{34} + m_{slidr} + 2 \frac{J_{pin}}{r_{pin}^2}. \quad (5)$$

Mass and inertia are m and J respectively with subscripts denoting the specific assembly. Acceleration, a , is obtained

geometrically based on angular acceleration, velocity, and position,

$$a = \frac{d^2x}{dt^2} = r\ddot{\Theta}^2 \left[\cos \Theta + \frac{r(\cos 2\Theta + r^2 \sin^4 \Theta)}{(L^2 - r^2 \sin^2 \Theta)^{\frac{3}{2}}} \right] + r\ddot{\Theta} \sin \Theta \left[1 + \frac{r^2 \cos \Theta}{2\sqrt{L^2 - r^2 \sin^2 \Theta}} \right]. \quad (6)$$

Connection rod length is denoted as L . The pressure force resulting from hydraulic fluid and cylinder gases is a summation of the pressures, p , multiplied by the respective piston area, A , or

$$F_p = \sum A_i p_i = A_{12}(p_1 - p_2) + A_{34}(p_4 - p_3) + A_{hc}(p_{hc2} - p_{hc1}). \quad (7)$$

Friction

The friction model is divided into individual friction contributions from the piston rings, piston skirt, slider, bearings, and air valves. Piston ring friction, piston skirt friction, slider friction, and air valve friction are adapted from correlations reported by Zweiri et al. [10]. Bearing friction follows from the work of Rezek and Henein [11] and Cameron [12].

Piston Ring Friction

In conventional engines, friction generated from the piston ring acting against the cylinder wall has three main contributions: inertial forces, static tension forces, and gas pressure forces. Because the HLE piston assemblies interact through a rack and pinion mechanism, the crank does not impart side loads on the pistons. The inertial component of Zweiri's friction model [10] is therefore eliminated. It is also assumed that rack and pinion interaction, side loads introduced by the rack and pinion, and gravity are negligible. While decreased side loading enables the use of low tension piston rings that reduce friction, this paper assumes standard piston ring properties.

The elastic tension of a compressed piston ring exerts a normal force against the cylinder wall. The frictional torque on the crank shaft resulting from static ring tension, τ_{fr1} , is

$$\tau_{fr1} = \eta r |G(\Theta)| \sum_{k=1}^{n_r} \left[\frac{E_{Y,k} L_g}{7.07 d_r \left(\frac{d_r}{L_{r,i}} - 1 \right)^3} \right] \pi d_r L_{r,i}, \quad (8)$$

where E_Y is the modulus of elasticity of the piston ring, L_g is the gap closure of the piston ring, d_r is the piston ring diameter, L_r is the piston ring thickness, n_r is the number of piston rings, and η is the friction coefficient [10].

The torque contribution of gas pressure to ring friction, τ_{fr2} , is described as