Experimental Investigation of Dynamic Characteristics and Stability of a Mechanically Started Free-Piston Engine

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This study presents an experimental investigation of the dynamic characteristics and operational stability of a two-stroke, two-cylinder free-piston engine (FPE) employing a purely mechanical starting mechanism, eliminating the need for the conventionally used linear electric motor. The engine was successfully initiated on the first ignition attempt through a forced oscillation procedure, producing a peak in-cylinder pressure of approximately 8 bar during starting phase. Under free-running, no-load conditions, pronounced cycle-to-cycle variations were observed, with peak in-cylinder pressures ranging from 6 to 15 bar and a non-stationary top dead center (TDC) position. Operational instability was primarily attributed to two coupled mechanisms: piston impacts with the stroke-limiting device and ignition timing deviations associated with piston velocity. Quantitative analysis revealed a coefficient of variation (COV) of peak pressure of 24.8%, substantially exceeding those of the peak pressure location and TDC position (both approximately 9.6%). These results provide one of the first experimental dataset describing combustion variability in mechanically started FPEs and highlight critical parameters required for stable operation.

Keywords: *free-piston engine, mechanical starting mechanism, operational stability, dynamic characteristics, combustion parameters.*

1. INTRODUCTION

The urgent need to reduce greenhouse gas emissions in the transportation sector has driven intensive research into alternative powertrain technologies. Among these, the free-piston engine (FPE) has emerged as a highly promising energy conversion platform for next-generation internal combustion engine applications [1-4]. By eliminating the crankshaft and connecting rod assembly used in conventional engines, the FPE offers a more direct power transmission path and the potential for enhanced thermal efficiency [5–8]. This architecture enables variable compression ratio operation and allows flexible adaptation to a wide range of fuel types. However, the absence of the crankshaft, which inherently constrains piston motion in conventional engines, fundamentally alters the piston dynamics of the FPE. In such systems, operational stability is not a mechanically guaranteed property but rather a dynamic equilibrium maintained through the balance among combustion, load, and feedback forces, which typically requires active control strategies and advanced dynamic modeling, similar to optimization techniques applied in conventional engine management [9–16]. Achieving consistent reciprocation under varying operating conditions therefore requires precise regulation of these forces, generally implemented through closed-loop con–trol of parameters such as fuel injection timing, injec–tion quantity, spark plug configurations,and load chara–cteristics [17,18]. This fundamental distinction creates opportunities for higher efficiency while simultaneously posing challenges for robust and reliable functioning.

Existing FPE control strategies primarily address two operational phases: starting and steady-state opera–tion. Regarding the starting phase, various control approaches for the linear actuator have been inves–tigated to generate the required driving force. Jia et al. [19,20] implemented a resonance-based starting strategy on a spark-ignition gasoline FPE prototype in which a linear generator operating as a linear motor provided an in-phase actuation force synchronized with piston motion. A closed-loop control system was employed to regulate this force precisely. Experimental results de–monstrated that this approach increased the peak in-cylinder pressure and compression ratio, allowing stable operation to be achieved within only a few cycles. Ignition occurred as early as the fourth cycle with a fixed actuation force of 125 N. The study also showed that larger starting forces produced higher piston velocities and cylinder pressures, thereby facilitating combustion initiation.

Feng et al. [21] focused on identifying the optimal compression ratio for a free-piston engine generator (FPEG) to ensure successful cold-start ignition. A nu–merical model developed in MATLAB/Simulink was validated using experimental data from a stably opera–ting prototype, and the deviation between simulated and measured in-cylinder pressures was maintained below

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5%. Simulation results indicated that the compression ratio was the dominant factor governing cold-start ignition performance, whereas operating frequency exerted a comparatively minor influence. Based on these findings, the study recommended optimal cold-start parameters that included an excess air ratio of 0.8–0.9, a compression ratio of 4–7, and an operating frequency of 10–16 Hz. Building on this work, Yuan et al. [22,23] conducted resonance-based starting phase experiments on a free-piston engine actuator (FPEA) prototype using an initial starting force of 570 N. During the starting process, piston motion exhibited self-excited oscillations with gradually increasing amplitude and frequency until a stable state was achieved after several cycles. The results confirmed that the resonance starting method allows the piston to reach a large stroke while maintaining stable peak in-cylinder pressure, compression ratio, and motion characteristics. For a diesel-fueled FPEA designed to achieve an auto-ignition pressure of 4.0 MPa, a linear alternator thrust of 600 N was sufficient to reach the target pressure within three cycles. However, when the starting force was reduced to 520 N or less, the required pressure could not be achieved. Additional simulations indicated that a starting force of 640 N rapidly stabilized the compression ratio at 22.7 while increasing the peak gas pressure to 7.2 MPa.

In the steady-state operational phase, research has predominantly focused on ensuring stability and continuous reciprocation. Zhang et al. [24] investigated a two-cylinder FPEG system and reported that the coefficient of variation (COV) of peak in-cylinder pressure was 22.78% without motor assistance. When thrust was applied, this value decreased to 10.12%, demonstrating the stabilizing effect of the linear motor. Under single-cylinder operation, the reported COV values were 0.84% for the TDC position, 3.33% for the compression ratio, 2.27% for the peak in-cylinder pressure, and 0.84% for the location of peak pressure. These values were substantially lower than the peak pressure COV of 10.11% observed during dual-cylinder operation, indicating that single-cylinder behavior provides significantly higher stability. Similarly, Liu et al. [25] examined the influence of operating parameters on the coefficient of variation of indicated mean effective pressure (COVIMEP). Simulation results showed that advancing the spark timing from −15° to −30° before top dead center reduced COVIMEP from 15.55% to 1.02%, indicating that ignition timing is a critical factor influencing combustion stability. Furthermore, COVIMEP remained below 10% when the fuel equivalence ratio ranged from 1.0 to 1.45. Consistent with the findings of Zhang et al. [21], COVIMEP in dual-cylinder functioning was 10.11%, which was substantially higher than the value of 0.83% observed in single-cylinder behavior. These results further confirm the superior stability associated with single-cylinder configurations.

Despite these advances, most existing studies rely on electromechanical actuation systems during the starting phase. Comparatively little attention has been given to purely mechanical starting mechanisms, although such systems could reduce structural complexity and cost while improving robustness in specific applications. In addition, experimental validation of FPE systems capable of maintaining stable free-running operation without active actuator assistance remains limited.

However, most previous experimental studies still rely on linear electric motors or linear generators to provide starting assistance and piston motion stabilization, resulting in increased system complexity and dependence on active control strategies. Experimental investigations of mechanically started FPE systems operating without actuator assistance remain very limited, particularly under free-running conditions.

To address these gaps, the present study investigates the starting and free-running performance of a two-stroke, two-cylinder, spark-ignition FPE prototype that replaces the linear actuator with a purely mechanical starting mechanism. The objective is to simplify the system architecture while maintaining the capability for stable and sustained functioning. Two experimental phases were conducted.

- **Starting phase:**A starting strategy specifically designed for the mechanical starter was developed to provide sufficient initial piston thrust for ignition without electronic actuation. Its feasibility was evaluated through repeated starting experiments.
- **Free-running phase:**After ignition was achieved, the engine operated without external thrust assistance. Key parameters influencing stability, including operating frequency, combustion pressure, and ignition timing, were systematically examined to evaluate their effects on piston motion regularity and combustion performance.

High-precision sensors and data acquisition systems were employed throughout the experiments to ensure accurate measurements of piston kinematics, in-cylinder pressure, and timing parameters. The primary objective of this study is to experimentally investigate the feasibility and operational stability of a mechanically started free-piston engine (FPE) operating without a linear electric motor or active actuator assistance. By eliminating the linear motor during the starting phase, the proposed architecture reduces system complexity, manufacturing cost, and control dependency. Experimental analyses of combustion variability through COV evaluation, piston dynamics, and ignition-timing effects under no-load operation provide fundamental insights and useful references for the future design, control, and development of efficient mechanically started FPE systems.

2. FPE MODEL SPECIFICATION

2.1 FPE Configuration

Figure 1 provides a schematic of the dual-cylinder FPE prototype developed for this investigation, and its primary technical parameters are detailed in Table 1. Unlike most experimental FPE systems, this prototype does not incorporate a linear electric motor. Instead, a purely mechanical starting mechanism was developed to provide the initial thrust required to initiate piston motion.

The configuration comprises two symmetrical combustion chambers, each functioning as an independent internal combustion unit, and a stroke-limiting mechanism to prevent piston–cylinder head collision. Fuel is delivered through a port fuel injection system, with the air-fuel mixture prepared in an auxiliary chamber that operates analogously to the crankcase of a traditional two-stroke engine. Sustained oscillatory motion is achieved through alternating combustion events in the two chambers, coordinated by a pre-programmed fuel injection and ignition control system. Due to the absence of a linear generator, the only moving compo–nents are the piston and the connecting shaft. Conse–quently, the prototype does not produce electrical power and operates under no-load conditions. This design allows the experimental setup to focus exclusively on investigating the fundamental dynamic and combustion characteristics of the FPE without the influence of electrical loading effects.The proposed FPE architecture is primarily intended for future FPEG applications, where the reciprocating motion of the piston can be directly coupled with a linear generator for compact electricity generation systems, such as portable power units and hybrid vehicle range extenders.

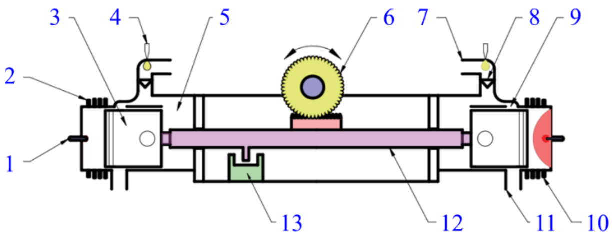

Figure 1.Schematic diagram of the FPE prototype

1 - Spark plug; 2 - Left cylinder; 3 - Piston; 4 - Fuel injector; 5 - Scavenging chamber; 6 - Mechanical starter; 7 - Air intake port; 8 - One-way valve; 9 - Scavenging port; 10 - Right cylinder; 11 - Exhaust port; 12 - Main connecting shaft; 13 - Piston displacement limiting mechanism.

Table 1. Specifications of the dual-cylinder FPE prototype

Parameter	Value
Number of cylinders	2
Engine type	Two-stroke
Bore	34 mm
Motoring stroke	22mm
Maximum stroke (limited)	30 mm
Reciprocating mass	0.5 kg
Compression ratio	9:1
Fuel	Gasoline

2.2 Mechanical starting system and control

The detailed structure of the mechanical starting mec–hanism is illustrated in Figure 2. Similar to FPE systems employing a linear electric machine during starting phase, this mechanism is designed to generate sufficient compression pressure within the combustion chamber to enable reliable ignition and combustion. The starter produces a controlled linear oscillation with a fixed amplitude of 22 mm, corresponding to a compression ratio of 3.0:1, thereby maintaining consistent comp–ression pressure throughout the initial starting cycles. This ensures repeatable ignition conditions without the need for active electronic control. The piston displa–cement profile during the starting process is presented in Figure 3, illustrating the oscillatory motion and the system's ability to consistently re-establish the con–ditions required for ignition across successive cycles.

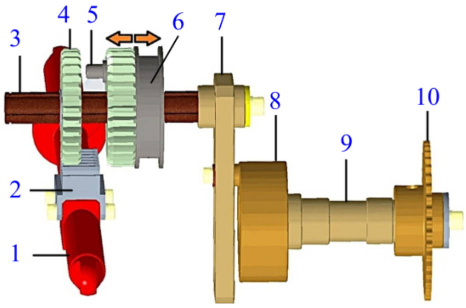

Figure 2. Mechanical starting mechanism

1 - Main shaft; 2 - Rack; 3 - Starting mechanism shaft, 4 - Driven gear; 5 - Fasteners; 6 - Magnetic clutch;7 - Handwheel; 8 - Eccentric wheel; 9 - Driven gear shaft; 10- Driven gear engaged with the starter gear of starter motor

The mechanical starting mechanism generates a driving force of approximately 400 N, supplied by an electric motor coupled to the starter gear. During start–ing phase, the positioning gear engages with the mag–netic clutch via a connecting pin, enabling motion tran–smission. The torque produced by the electric motor is delivered through a chain system to the starting sprocket mounted on the starting shaft, where it is converted into eccentric rotational motion by an eccentric wheel. This motion is subsequently transferred to the crank and the starting mechanism shaft. Upon completion of the starting process, the magnetic clutch disengages from the driven gear, allowing the main shaft to operate freely. Overall, the mechanism is engineered to transform the rotational motion of the electric motor into the linear reciprocating motion of the main shaft.

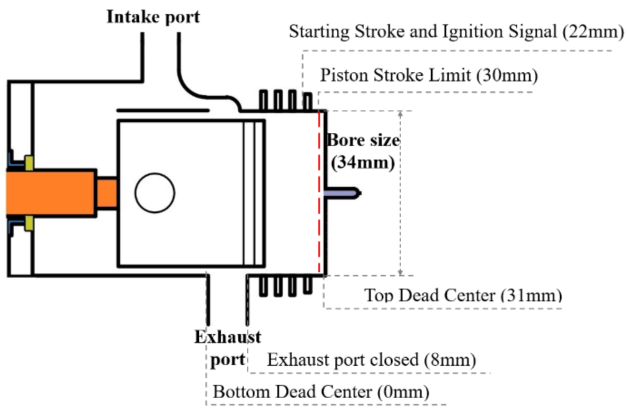

Figure 3. Schematic of the piston coordinate system

The FPE's control system diagram is presented in Figure 4. The control system performs three main func–tions: (1) engaging or disengaging the magnetic clutch to regulate the timing and duration of the piston's linear oscillation; (2) controlling the fuel injection process; and (3) controlling the ignition process. Two proximity sen–sors, one on the left (G1) and one on the right (G2), are mounted at each end of the cylinder to detect the direction of piston displacement relative to the left and right cylinders. In addition to position detection, these sensors provide input signals for the respective fuel injection and ignition control systems, enabling precise synchronization between piston motion and the combustion process.

The ignition process is initiated when either sensor G1 or G2 detects a signal, triggering a dwell time of 3.5 ms, after which the spark is generated, as illustrated in Figure 5.

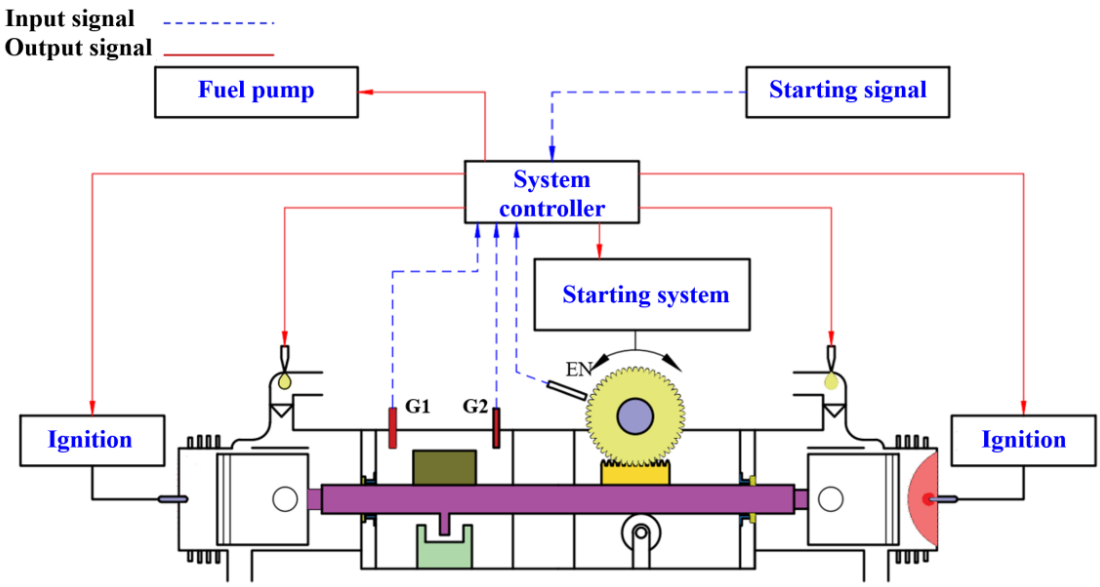

Figure 4. Schematic of the control system

Consequently, the ignition timing is not fixed but varies according to the piston's displacement speed, which can change in each operating cycle. For the fuel injection process, due to the two-stroke operating principle of the FPE, the compression stroke in one cylinder occurs simultaneously with the intake stroke in the opposite cylinder.

Fuel is injected during the intake stroke. Signals from sensors G1 and G2 are employed to accurately identify the intake and compression phases of each cylinder, thereby determining the precise fuel injection timing in synchronization with the piston's motion. To precisely control the piston's displacement position, an encoder (EN) is integrated into the driven gear of the drive mechanism, providing continuous feedback to the control system.

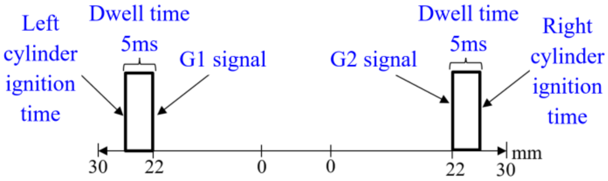

Figure 5. Ignition signals for the left and right cylinders

2.3 Piston Force and Stroke Limits

The total force acting on the system in a free-oscillation state can be expressed using Newton's Second Law.

Starting process:

$$F_{st} - F_{sl} - F_{sr} - F_f = m\frac{d^2x}{dt^2} \tag{1}$$

Operation process:

$$F_{cyl} - F_{cyr} - F_f = m\frac{d^2x}{dt^2} \tag{2}$$

where F_{cyl} and F_{cyr} are the gas forces generated by heat release in the left and right cylinders, respectively; P_{cyl} and P_{cyr} are the combustion pressures in the left and right cylinders; and A is the piston crown area. F_f denotes the friction force, m is the mass of the moving system (piston and connecting shaft), and d^2x/dt^2 is the linear acceleration of the piston. F_{sl}, F_{sr} represent the comp–ression forces on the left and right sides, respectively. F_{st} is the force from the starting mechanism during the starting phase.

$$F_{cyl} = P_{cyl} \times A \tag{3}$$

$$F_{cyr} = P_{cyr} \times A \tag{4}$$

For the piston to execute a power stroke, the thrust generated by combustion in one cylinder must exceed the total resisting forces acting on the system.

$$F_{cyl} > m\frac{d^2x}{dt^2} + F_f + F_{sr} \tag{5}$$

$$F_{cyr} > m\frac{d^2x}{dt^2} + F_f + F_{sl} \tag{6}$$

Unlike a conventional crankshaft-based piston engi–ne, where piston motion is constrained by the crankshaft mechanism, the piston in a FPE moves entirely without mechanical linkage constraints and is simultaneously influenced by all acting forces. Among these forces, the inertial force, which is proportional to the mass of the piston assembly and its acceleration, plays a crucial role but is inherently difficult to control.

A sharp increase in acceleration prior to combustion can cause the inertial force to become excessive, posing a risk of piston-to-cylinder head collision. Therefore, the design of the stroke-limiting mechanism is essential to dissipate or counteract excess inertial force, thereby preventing mechanical impact and ensuring operational stability. Owing to the complex dynamic behavior of the FPE, precise control over the piston's position and trajectory is a key requirement for maintaining stable and safe operation of the system.

2.4 Statistical Analysis of Cycle-to-Cycle Variability

The Coefficient of Variation (COV) is commonly used as an indicator to quantify the fluctuation level of key operating parameters of internal combustion engines, such as Top Dead Center (TDC) position, compression ratio, peak in-cylinder pressure, and the position at which peak pressure occurs. Moreover, COV serves as a critical metric reflecting the engine's continuous operational capability and stability. A lower COV value corresponds to reduced cycle-to-cycle variation, thereby indicating higher operational stability of the system.

$$COV = \frac{\sigma}{\bar{p}} \tag{7}$$

where σ is the standard deviation and p̄ is the mean va–lue of the measured parameter.

Specifically, the COV of the peak in-cylinder pressure (COV_{Pmax}) is calculated as:

$$COV_{P_{\max}} = \frac{\sigma_{P_{\max}}}{\bar{p}_{\max}} \tag{8}$$

where σ_{Pmax} denotes the standard deviation of the peak in-cylinder pressure, and $\bar{p}_{\max}$ represents its mean value of the peak pressure.

The standard deviation (σ_{Pmax}) is determined by:

$$\sigma_{P_{\max}} = \sqrt{\frac{\sum_{i=1}^{N}\left(p_{\max}^{(i)} - \bar{p}_{\max}\right)^2}{N}} \tag{9}$$

where $p_{\max}^{(i)}$ is the peak pressure of the *i-th* cycle, $\bar{p}_{\max}$ is the mean peak pressure over all cycles, and N is the total number of cycles. An analogous approach is used to calculate the COV of the TDC position.

3. EXPERIMENTAL SETUP AND TEST CONDITIO–NS

3.1 Experimental Prototype

The experimental layout of the FPE and its associated instrumentation are depicted in Figure 6. A data acquisition system, built on the LabVIEW platform, utilized an NI-USB 2 interface and a PicoScope 4425 high-speed data logger to guarantee high-accuracy, rapid data sampling. An AVL-ZF43 pressure transducer, coupled with a Kistler 5010 signal amplifier, was used to measure in-cylinder combustion pressure. Further–more, an optical encoder with a 0.4 mm per pulse resolution was mounted on the engine to obtain precise measurements of the piston's position and velocity.

3.2 Test Conditions and Procedure

The experiment was conducted in two sequential stages: the starting phase and the free-running phase. In the starting phase, the mechanical starter induced a linear oscillatory motion in the piston at a 10 Hz frequency and a fixed 22 mm amplitude. The purpose of this phase was to generate sufficiently high in-cylinder compression pressure to initiate combustion. Once the required compression pressure was reached, fuel was injected into the intake port. Similar to the performance of a conventional two-stroke engine, the air-fuel mixture was drawn into an auxiliary chamber during the intake stroke and retained by a one-way valve. The incorporation of this auxiliary chamber enabled naturally aspirated operation without the need for external air-compressor assistance. Upon achieving the target compression pressure, the starting mechanism disengaged from the connecting shaft, and the ignition process was imme–diately triggered, thereby concluding the starting phase.

In the free-running phase, the air-fuel mixture stored in the auxiliary chamber was transferred into the com–bustion chamber, where compression and combustion occurred. The ignition timing was fixed in the same manner as during the starting stage, with sequential ignition in the two cylinders when the piston reaches the predetermined ignition position. A key operational consideration in this phase was the effect of the piston impact against the stroke limiter.

Such collisions generated a rebound force that altered the in-cylinder peak pressure, resulting in dif–ferences between the peak pressures before and after impact. Throughout both phases, key operational para–meters including in-cylinder pressure, piston velocity, and piston displacement were continuously recorded and analyzed to evaluate the performance characteristics of the FPE.

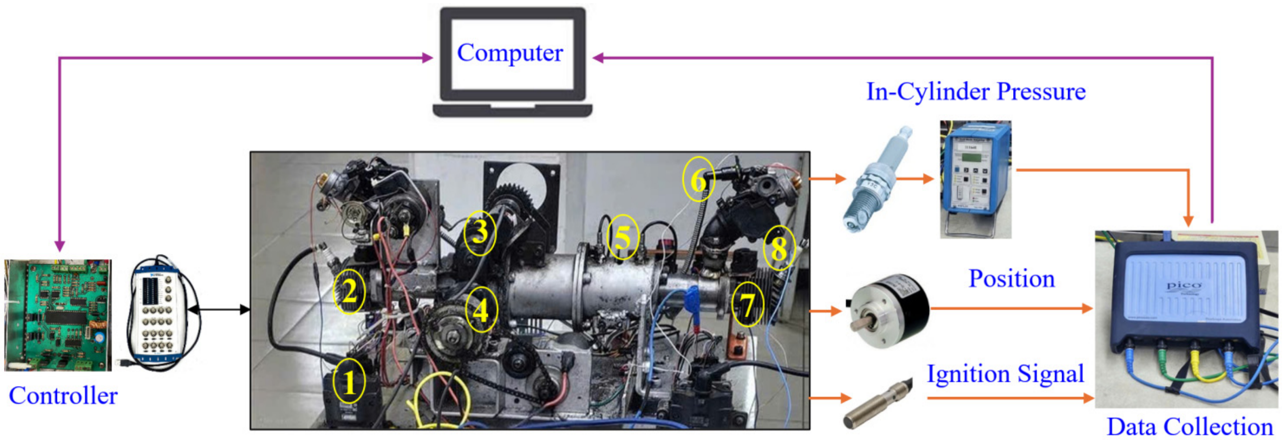

Figure 6. Experimental setup of the FPE test bench

1 - Ignition system; 2 - Left cylinder; 3 - Encoder; 4 - Mechanical starting mechanism; 5 - Proximity sensor; 6 - Fuel injector; 7 - Right Cylinder; 8 - Piezoelectric pressure sensor

4. RESULTS AND DISCUSSION

4.1 Mechanical Starting Process

At the onset of the starting process, an activation signal was transmitted to the controller to drive the electric motor, which was mechanically coupled to the main shaft connecting the two pistons. The piston oscillated with a constant amplitude and at a fixed frequency of 10 Hz. The in-cylinder compression pressure was initially maintained at approximately 4 bar. Once this pressure threshold was reached, the system initiated the ignition sequence to commence combustion within the chamber, as shown in Figure 7.

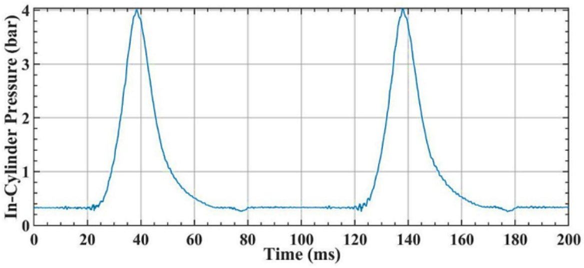

Figure 7. In-cylinder pressure profile during the starting phase

Figure 8 illustrates the transition in the engine's operating state. During the starting phase, lasting approximately the first 1.4 s, the piston was driven by the mechanical starting mechanism, undergoing forced oscillations with a stable amplitude of 22 mm and a consistent motion trajectory. Following the disenga–gement of the starting mechanism, the engine entered the free-running mode after a transition period of about 0.5 s. In this mode, the piston exhibited unstable oscillations with amplitudes increasing to as much as 30 mm, indicating the establishment of a self-sustaining combustion cycle and the characteristic unconstrained motion of the engine.

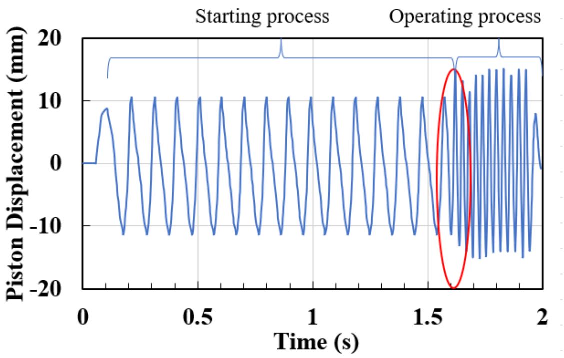

Figure 8. Piston displacement dynamics during the transition from the starting phase to free-running operation

The maximum piston velocity increased from approximately 0.7 m/s during the starting phase to 5 m/s upon transition to the free-running mode, as illustrated in Figure 9. This substantial increase resulted from the combustion force acting directly on the piston, replacing the mechanical driving mechanism and enabling the engine to achieve its characteristic operating frequency and velocity. This transition phase underscores the distinction between the forced motion during startingphase and the more complex dynamics that emerge when the engine operates autonomously under the influence of thermodynamic processes.

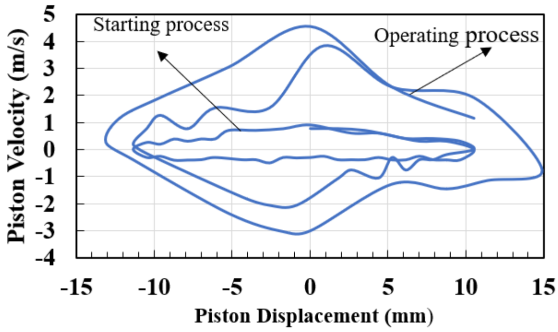

Figure 9. Piston velocity during the transition from starting phase to steady-state operation

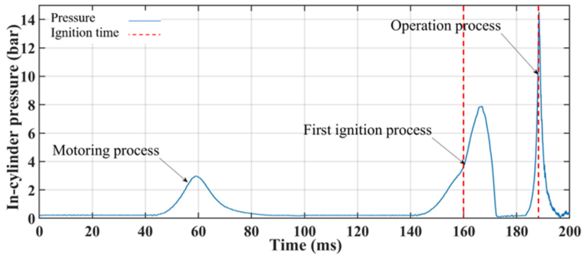

Figure 10. Pressure fluctuation characteristics during the transition from starting phase to steady-stateoperation

The variation in the timing of peak in-cylinder pressure between the initial starting phase and the stable operation of the FPE is shown in Figure 10. The pressure profile during the first ignition differs mar–kedly from that observed during stable operation, ref–lecting the distinct operating characteristics of these two phases.

During the first ignition, as the mechanical starter disengages from the main shaft, the piston continues its motion due to inertia, resulting in a gradual increase in compression pressure and force. When the compression force balances the inertial force, the piston momentarily halts, and the pressure generated by the post-ignition combustion causes the piston to reverse its direction. The gas force produced by the first combustion event only needs to overcome the inertia from the starting mechanism and frictional resistance to enable this re–versal. During this process, the peak pressure reaches approximately 8 bar and occurs with a notable delay relative to the ignition timing. In contrast, during stable operation, the TDC position is unconstrained and main–tained by the dynamic equilibrium of acting forces, inc–luding combustion-induced gas forces, inertial forces, and friction. In this phase, the in-cylinder pressure varies continuously and complexly with piston position and ignition timing throughout the heat release process.

4.2 Operation Process

As seen in Figure 11, the in-cylinder pressure across 50 consecutive cycles of the FPE shows substantial fluctuation. The peak pressure varied between a maxi–mum of 15 bar and a minimum of 6 bar, indicating a large amplitude of variation. This instability can largely be attributed to the absence of external loading in the FPE system. In the absence of electromagnetic drag from a linear generator, even a low combustion pressure of 6 bar is enough to reverse the piston's direction. The TDC position occurs at approximately 30 mm, which corresponds to a maximum compression ratio of 9:1.

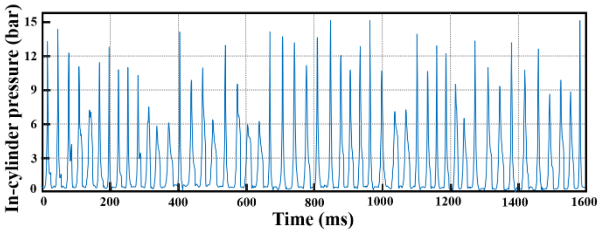

Figure 11. Continuous recording of 50 consecutive cycles during steady-state operation

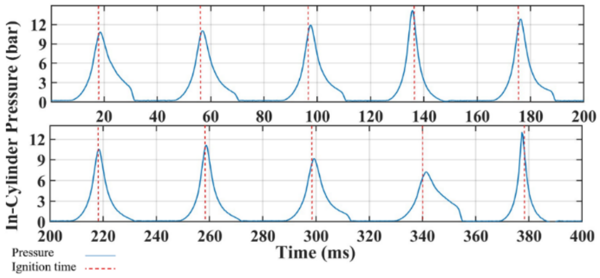

Figure 12. In-cylinder pressure over 10 consecutive cycles during steady-state operation

When the piston reaches its highest position, the stroke-limiting mechanism prevents direct collision with the cylinder head while simultaneously generating a reaction force that reverses the piston motion. At this point, the peak in-cylinder pressure is formed due to the maximum compression in the combustion chamber. Experimental results over 10 consecutive cycles show that the peak pressure may occur either before or after the ignition event. Due to a 3.5 ms dwell time, if the piston velocity is high, a delay in spark occurrence may cause ignition to take place after the peak pressure has already been reached, as illustrated in Figure 12.

4.3 Combustion Timing Effects

Figure 13(a) depicts the combustion process occurring before TDC, during which spark ignition initiates combustion, leading to the formation of peak in-cylinder pressure and causing the piston to reverse its direction of motion. Figure 13(b) presents the corresponding pressure-volume (p-V) diagram, in which the cycle is divided into three distinct regions: the gas-exchange region, the positive work region (where combustion pressure is higher than compression pressure), and the negative work region (where the compression pressure exceeds the combustion pressure). The negative work region arises from high compression forces, resulting in elevated in-cylinder pressures at the end of the compression stroke. While such high pressure can hinder piston motion, they simultaneously prevent direct contact between the piston and the cylinder head, thereby contributing to improved combustion efficiency. The piston reverses its motion when peak pressure is reached, specifically at the point where the gas force surpasses the inertial force.

Figure 14 illustrates the case in which combustion process occurs after TDC. In this scenario, the maxi–mum compression pressure is attained precisely when the piston reaches TDC. At this instant, the stroke-limi–ting mechanism generates a reaction force that reverses the piston's motion. After the piston begins its down–ward stroke, spark ignition is initiated, and combustion commences. Because this combustion takes place at an increased cylinder volume, after the piston has moved away from TDC, the resulting peak combustion pressure is generally lower than that achieved when combustion occurs at or before TDC. The gas force generated during this process continues to drive the piston downward throughout the expansion stroke.The experimental re–sults indicate that ignition timing control is one of the most critical factors affecting combustion stability in mechanically started FPE systems.

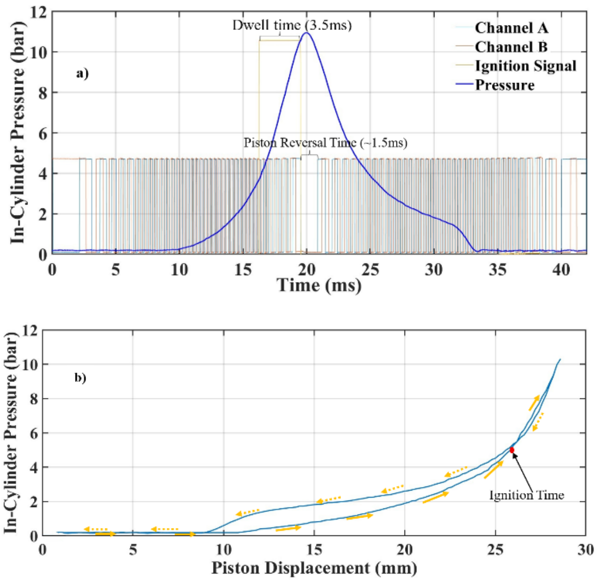

Figure 13. (a) Ignition occurring prior to the peak in-cylinder pressure; (b) In-cylinder pressure as a function of piston displacement

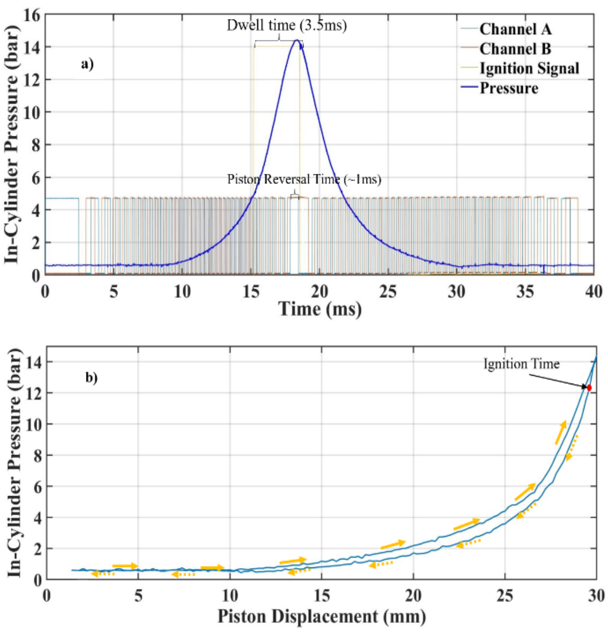

Figure 14. (a) Ignition occurring after the peak in-cylinder pressure; (b) In-cylinder pressure as a function of piston displacement

4.4 Cycle-to-Cycle Variability

As demonstrated in Figure 15, the in-cylinder com–bustion pressure of the FPE exhibits significant varia–bility over 50 consecutive cycles. The peak pressure fluctuates between 15 bar and 6 bar, reflecting the sys–tem's operational instability.

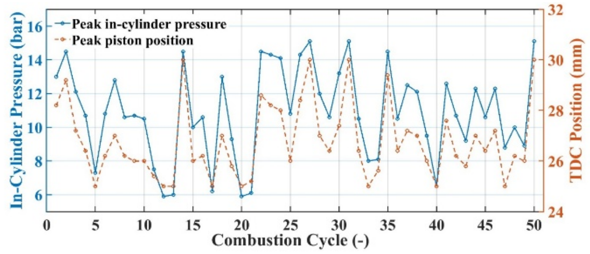

Figure 15. Fluctuations in peak in-cylinder pressure and TDC position during steady-state operation

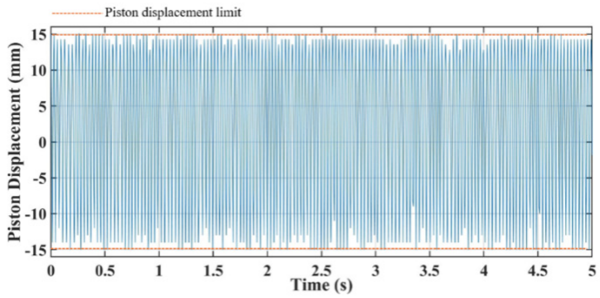

Figure 16. Piston displacement dynamics during 5 s of continuous steady-state operation

However, even at the lowest recorded peak pressure, the generated gas force was adequate to propel the pis–ton back to the ignition position of the opposing cy–linder. Using the collected data, the COV for key para–meters was calculated. The results show that fluctua–tions in the TDC position are relatively minor, largely because the piston frequently interacts with the stroke limiter. The COV of the peak in-cylinder pressure was found to be 24.2%, whereas the COV of the piston position was significantly lower at only about 9.6%.

Figure 16 presents the engine operating for appro–ximately 5 seconds. The maximum piston stroke reaches 30 mm, constrained by the stroke-limiting mechanism, while the primary displacement range fluctuates between 25 mm to 29 mm. Controlling the maximum stroke of the piston is a complex task that requires the simultaneous optimization of multiple parameters, including fuel quantity, injector position, and ignition timing, as well as, in some cases, tolerating minor impacts to maintain stable and durable engine operation.

5. CONCLUSION

This study experimentally investigated the dynamic characteristics and operational stability of a two-stroke, two-cylinder spark-ignition free-piston engine (FPE) initiated by a purely mechanical starting mechanism. The results demonstrate that stable engine operation can be achieved without the use of a linear electric motor. The mechanical starting system successfully drove the piston to oscillate at approximately 10 Hz with an amplitude of 22 mm, enabling successful ignition on the first attempt and generating a peak in-cylinder pressure of about 8 bar.

Under no-load, free-running conditions, piston motion was governed by the dynamic interaction among combustion, inertial, and load forces, resulting in significant cycle-to-cycle variations. During operation, the peak in-cylinder pressure varied between 6 and 15 bar, while the maximum piston velocity increased from 0.7 m/s to 5.0 m/s. The experimental observations further revealed that the combined effects of the stroke-limiting rebound mechanism and the velocity-dependent ignition timing, caused by the 3.5 ms system dwell time, constitute the primary sources of operational instability. These factors lead to deviations in the formation of peak pressure relative to the intended ignition timing in certain cycles.

Quantitative analysis of combustion variability showed that the coefficient of variation (COV) of peak in-cylinder pressure reached 24.8%, which is significantly higher than the COV values of the peak pressure location and TDC position (both approximately 9.6%). This result suggests that although the piston stroke is partially constrained by the mechanical mechanism, the combustion process remains highly susceptible to cycle-to-cycle fluctuations. The present work should be regarded as a foundational experimental investigation of mechanically started FPE operation. The proposed architecture provides a simplified alternative to complex linear actuator-assisted systems. The experimental results demonstrate that reducing mechanical losses alone is insufficient to ensure improved engine efficiency unless stable combustion phasing and controlled piston dynamics are simultaneously maintained.Significant cycle-to-cycle combustion variations were observed under no-load operation, while instability mechanisms such as piston rebound effects and velocity-dependent ignition timing deviations were identified as key factors affecting operational stability. These findings provide important insights for future studies focused on adaptive ignition control, combustion stabilization, and stable loaded operation before the practical efficiency advantages of mechanically started FPE systems can be fully realized.

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NOMENCLATURE

Roman Symbols

A	Piston crown area, m^2
COV	Coefficient of variation
COV_{Pmax}	Coefficient of variation of peak pressure
COV_{IMEP}	Coefficient of variation of indicated mean effective pressure
F	Force, N
F_{cyl}	Gas force in the left cylinder, N
F_{cyr}	Gas force in the right cylinder
F_f	Friction force acting on the moving

	components, N
FPE	Free-piston engine
FPEG	Free-piston engine generator
F_{sl}	Compression force on the left side, N
F_{sr}	Compression force on the right side, N
F_{st}	Force generated by the starting mechanism, N
IMEP	Indicated mean effective pressure, Pa
m	Mass of the moving system, kg
N	Number of cycles
p	In-cylinder pressure, Pa
p_{max}	Peak in-cylinder pressure, Pa
$p_{\max}^{(i)}$	Peak pressure of the i^{th} cycle, Pa
$\overline{p}$	Mean value of a parameter
$\overline{p}_{\max}$	Mean peak in-cylinder pressure, Pa
TDC	Top dead center
v	Piston velocity, m s^{-1}
x	Piston displacement, m
V	Cylinder volume, m^3

Greek Symbols

σ	Standard deviation
σ_{Pmax}	Standard deviation of peak pressure, Pa
Φ	Equivalence ratio
θ	Ignition timing (crank-angle equivalent), °
ω	Angular frequency, rad s^{-1}

Subscripts

cyl	Left cylinder
cyr	Right cylinder
f	Friction
i	Cycle index
max	Maximum value
sl	Left-side compression
sr	Right-side compression
st	Starting mechanism

ЕКСПЕРИМЕНТАЛНО ИСПИТИВАЊЕ ДИНАМИЧКИХ КАРАКТЕРИСТИКА И СТАБИЛНОСТИ МЕХАНИЧКИ ПОКРЕТАНОГ МОТОРА СА СЛОБОДНИМ КЛИПОМ

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Ова студија представља експериментално испити–вање динамичких карактеристика и оперативне стабилности двотактног, двоцилин–дричног мотора са слободним клипом (МСП) који користи иск–ључиво механички механизам покретања, елими–нишући потребу за конвенционално коришћеним линеарним електромотором. Мотор је успешно пок–ренут при првом покушају паљења путем поступка принудне осцилације, производећи вршни притисак у цилиндру од приближно 8 бара током фазе покретања. У условима слободног рада, без опте–рећења, примећене су изражене варијације од цик–луса до циклуса, са вршним притисцима у цилиндру у распону од 6 до 15 бара и нестационарним положајем горње мртве тачке (ГМТ). Оперативна нестабилност је првенствено приписана двама повезаним механизмима: ударима клипа са уређајем за ограничавање хода и одступањима времена па–љења повезаним са брзином клипа. Квантитативна анализа је открила коефицијент варијације (КОВ) вршног притиска од 24,8%, што знатно премашује коефицијент варијације места вршног притиска и положаја ГМТ (оба приближно 9,6%). Ови резултати пружају један од првих експерименталних скупова података који описују варијабилност сагоревања у механички покретаним ФПЕ и истичу критичне параметре потребне за стабилан рад.