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THE STUDY ON THE CONVERSION OF SINGLE CYLINDER ENGINE INTO HOMOGENEOUS CHARGE COMPRESSION IGNITION (HCCI) ENGINE

NGHIÊN CỨU VỀ VIỆC CHUYỂN ĐỔI ĐỘNG CƠ MỘT XI LANH THÀNH ĐỘNG CƠ ĐỐT CHÁY NÉN NHIỆT ĐỒNG NHẤT (HCCI)

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ABSTRACT

Scientists are interested in a homogeneous charge compression ignition combustion model (HCCI) to replace a conventional combustion model on an internal combustion engine. This combustion model has the same performance as a direct-injection gasoline engine and the same type of ignition as a diesel engine. However, the NO_x emission component is significantly reduced, while the smoke level is almost zero. With this combustion model, it's perfectly capable of meeting emissions requirements without having to be equipped with an additional emission after-treatment system. In this study, the author converted a traditional one-cylinder diesel engine into a new engine based on HCCI mode. In order to be able to work in HCCI mode, it is necessary to modify the engine's parameters, such as reducing the compression ratio, equipping it with a port fuel supply system, or combining air heating and exhaust recirculation systems. The study results have shown that the converted engine can operate according to the principle of HCCI in low load conditions of up to 30%. The combustion characteristics and engine operation parameters before and after conversion to HCCI mode are tested and evaluated on the engine test bed. Experimental research to quantify and assess the engine's HCCI operating mode and construct a set of control parameters for setting and widening the working regime in HCCI mode.

Keywords: *HCCI; n-heptane/ethanol blend fuel; combustion characteristics; heat release rate; COV.*

TÓM TẮT

Nghiên cứu này tập trung vào ứng dụng mô hình đốt cháy nén nhiệt đồng nhất (HCCI) nhằm thay thế mô hình đốt cháy truyền thống trên động cơ đốt trong. HCCI cung cấp hiệu suất tương đương với động cơ xăng trực tiếp phun nhiên liệu và kỹ thuật kích hoạt giống với động cơ diesel. Điều đáng chú ý là khí thải NO_x giảm đáng kể và mức độ khói gần như không đáng kể. Nghiên cứu đã chuyển đổi thành công một động cơ diesel một xi lanh truyền thống thành một động cơ mới hoạt động dựa trên nguyên tắc HCCI. Để đạt được điều này, các thông số của động cơ đã được điều chỉnh, bao gồm việc giảm tỷ lệ nén, trang bị hệ thống cung cấp nhiên liệu qua cổng và kết hợp các hệ thống sưởi ấm không khí và tái tuần hoàn khí thải. Kết quả cho thấy động cơ đã được chuyển đổi có thể hoạt động theo nguyên tắc HCCI trong điều kiện tải nhẹ lên đến 30%. Thông số đốt cháy và hoạt động của động cơ trước và sau khi chuyển đổi sang chế độ HCCI đã được thử nghiệm và đánh giá trên bệ thử nghiệm động cơ. Nghiên cứu thực nghiệm nhằm định lượng và đánh giá chế độ hoạt động HCCI của động cơ và xây dựng một tập hợp các thông số điều khiển để thiết lập và mở rộng phạm vi hoạt động trong chế độ HCCI.

Từ khóa: *HCCI; nhiên liệu pha trộn n-heptane/ethanol; đặc điểm đốt cháy; tốc độ tỏa nhiệt; COV.*

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1. INTRODUCTION

Figure 1 compares fuel combustion according to different combustion principles. The combustion of the layered mixture of a diesel engine, with a flame surrounding the nozzle and the fuel spontaneously ignites at the beginning of the combustion. Meanwhile, gasoline engines use a spark to start the combustion process, and the flame spreads in a homogeneous mixture. HCCI engines combine the characteristics of both gasoline and diesel engines with a homogeneous fuel mixture that ignites spontaneously at the beginning of combustion. For HCCI motors, the combustion time is short and the pressure gain rate is significant due to the large heat release rate. In theory, HCCI engines do not have direct spark plugs or nozzles to aid combustion, and once the fuel mixture reaches a self-ignition temperature, combustion takes place in various areas. As such, there are some differences between the combustion process of HCCI engines when compared to diesel and gasoline engines as mentioned in some previous studies [1-3].

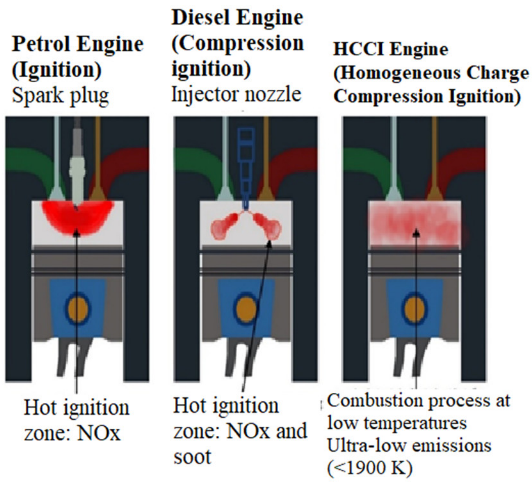


Figure 1. Comparison of HCCI motors with traditional motors

During an operation of the HCCI engine, fuel and air are mixed before combustion. The air mixture spontaneously ignites at some points in the combustion chamber due to the increased temperature of the compression stroke. This combustion mode uses a very lean mixture, resulting in reduced local combustion mass and temperature thereby reducing NO_x emissions. Furthermore, unlike traditional compression ignition internal combustion engines, HCCI combustion is well mixed in a homogeneous mixture, thus reducing PM formation.

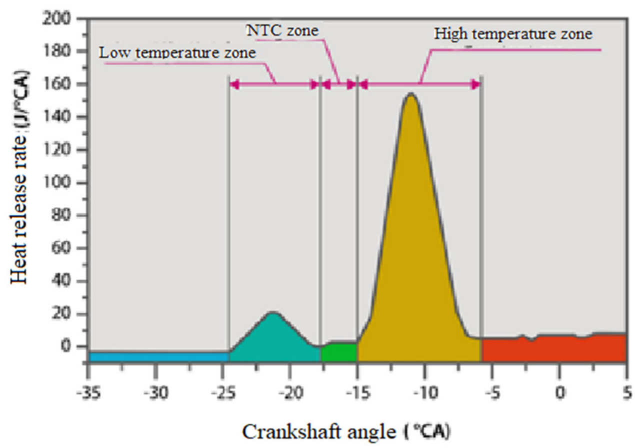


Figure 2. HCCI Exothermic Rate with n-Hueal Fuel

HCCI combustion consists of two stages: The initial heat release phase is called low-temperature heat release, where kinetic processes occur slowly and at temperatures below 850K, resulting in a heat release limit of 7% - 10% of the total heat released [4]. The remaining fuel's energy is released when the intake temperature exceeds 950K, a process known as high-temperature oxidation or high-temperature heat release. The negative temperature coefficient (NTC) region shown in Figure 2 is the zone that determines the fire periodicity because it determines the start of the primary heat release rate [5].

Scientists have carried out many studies related to HCCI combustion with different types of fuel. The compression ratio of the test engine changed from 20:1 to 14.8:1 by changing the thickness of the cowl cushion was also conducted in earlier studies [6]. Accordingly, the HCCI process is applied on single-cylinder diesel engines running on n-heptane fuel, although there is an engine knock at a compression ratio of 20:1, the operating range of HCCI combustion is limited to 2400rpm and 30% load. By reducing the compression ratio from 20:1 to 14.8:1, HCCI combustion range can be increased up to 3200rpm and 50% load. Gray and Ryan studied particulate emissions from HCCI combustion of diesel and hexadance/heptane mixtures. As a result, they showed that HCCI technology reduced contaminants by 27% compared to traditional compression-induced combustion diesel engines. Furthermore, they found a significant correlation between intake air temperature and PM emissions [7]. Agarwal AK and parter tested HCCI combustion of diesel engines with different EGR and air-fuel ratios. They found that increasing the air-fuel ratio (less) and EGR enhanced the organic content of particulate matter [8]. Kaiser EW, et al. measured PM from gasoline-powered HCCI engines with air/fuel ratios between 50 and 230 [9]. Price P et al. found that direct-injection HCCI gasoline engines had similar particulate gas emissions as traditional direct-injection engines. A significant accumulation (80 - 100nm) was observed. They found that enhancing exhaust gas recirculation reduced particulate matter concentrations in accumulation mode. A granulation mechanism with an average diameter of 10 - 20nm has also been observed [10].

In this study, the authors carried out a study of converting traditional diesel engines to operate in HCCI mode. On the basis of the evaluation analysis of previous studies, the necessary adjustments to the engine are made. The modified engine is tested on a power test stand to evaluate a number of criteria such as the ability to form HCCI combustion as well as the performance of the systems on the engine.

2. RESEARCH CONTENTS

2.1. Subjects of study

In this study, single-cylinder diesel engines were selected to perform conversions and equipped with systems to operate in HCCI mode with n-heptane. Engines are commonly used to drive small single-cylinder, non-turbocharged generators, used on agricultural machines, water pumps. This type of single-cylinder, air-cooled, uniform combustion chamber engine (Figure 3). The basic parameters of the engine are shown in Table 1.

Table 1. Basic parameters of the engine

Parameter	Unit	Value
Cylinder diameter, (D)	mm	78
Piston stroke, (S)	mm	57
Displacement volume, (V_h)	cm^3	273
Rated power, (N_{edm})	kW	4.4
Rated speed, (n_{dm})	vg/ph	3.600
Maximum torque, (M_{emax})	Nm	13
Speed at M_{emax},(n_M)	vg/ph	2000
Fuel consumption, (g_{emin})	g/kW.h	378
Speed at g_{emin}, (n_{ge})	vg/ph	2.400

Figure 3. Research subjects

2.2. Engine improved for HCCI mode

2.2.1. Adjust and decrease the compression ratio of the engine

The original motor is changed to reduce the compression ratio from 21 to 17 in order to convert to HCCI working mode. There are options for lowering the compression ratio of the engine, such as reducing the top of the piston or increasing the thickness of the cover pad. If the proper piston top cutting position can be calculated and chosen, it will raise the vortex in the fire stub, assist the mixture mix better, and improve the combustion process quality. However, cutting will modify the piston mass, which will affect the engine's dynamics. The possibility to raise the thickness of the lid cushion has numerous advantages, including minimal impact on other mechanical structures on the engine, less renovation work, lower cost. The lid cushion must provide tightness while preventing lubricant leaking and causing air leakage. In his investigation, increasing the thickness of the lid cushioning was the most effective strategy to minimize the compression ratio. The additional cushion thickness is solely determined by the piston stroke and compression ratio before and after restoration. The following formula will be used to determine the additional thickness of the mattress:

$$a = S.\frac{\varepsilon_o - \varepsilon}{(\varepsilon - 1)(\varepsilon_o - 1)} \quad (1)$$

In which ε and ε_o are compression ratios before and after renovation, S is the piston stroke, a is the pad thickness to be increased.

2.2.2. Design and manufacture HCCI engine parts and systems

Figure 4 depicts the overall diagram of systems planned and produced to allow the engine to function in HCCI mode, including the fuel supply system, intake air hot drying system, exhaust gas recirculation, and experimentation systems.

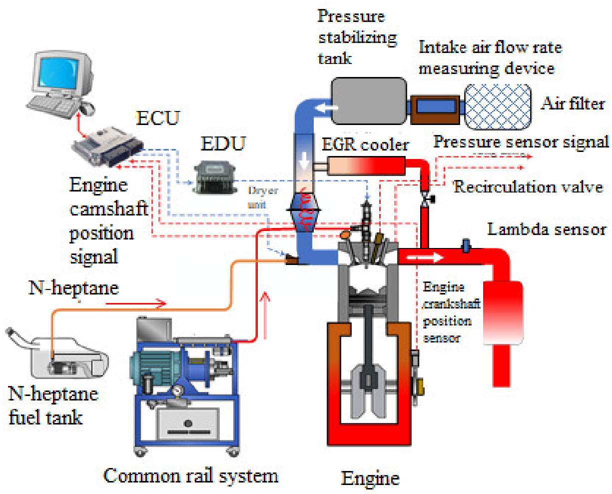


Figure 4. Overall diagram of engine systems

a) Intake tract indirect fuel injection system

Before loading into the engine, the fuel is indirectly sprayed on the intake manifold using a magnetic fill nozzle, as shown in Figure 4. The spray is directed towards the flow of intake air entering the engine via the nozzle, which is situated right in front of the intake valve. By utilizing the heat from the intake valve, fuel is injected and quickly dissipated before mixing with the intake air.

Because HCCI engines are difficult to start, a direct diesel fuel supply system, as shown in Figure 4, is employed to assist in starting and heating the engine. The engine's conventional fuel system was replaced with an electronically controlled common rail (CR) fuel system. To do this, the original high-pressure pump and mechanical nozzles are replaced with a collection of high-pressure pumping systems, injectors, pressure valves, sensors, and electronic control systems.

b) New intake air heating system combined with exhaust gas cycling

To ensure that the fuel mixes well with the air, the nozzle is positioned immediately in front of the intake pressure to generate an appropriate injection in the shape of the intake line before entering the engine. With the above configuration, the fuel will have the best chance of evaporating before entering the cylinder. A drying wire situated in front of the nozzle heats the intake air before it enters the engine. During the test, the temperature of the intake air can be controlled flexibly by altering the capacity of the drying wire. There are temperature sensors on the intake manifold body in front and behind the heater to determine the temperature of the air after it exits the heater. Furthermore, the exhaust gas recirculation (EGR) system is located in the middle of the intake manifold, where the cyclic gas will mix with the air before traveling through the heater, as illustrated in Figure 5.

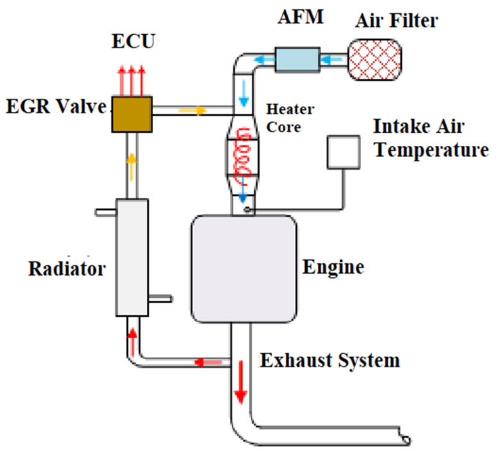


Figure 5. Intake tract design and exhaust gas recirculation system

Exhaust gases are partially extracted to the exhaust gas cooling system and then sent through the throttle in the case of exhaust gas recycling. Following that, both the new intake and the cyclic gas will be introduced into the engine cylinder. The throttle valve will alter the circulating air flow to fit each mode based on the engine's load mode. Return valves used in EGR systems are typically mechanical, vacuum-controlled, or electrically operated.

2.3. Fuel Test

During the test, regular commercial diesel and n-heptane were employed as fuel. When the engine is working in HCCI mode, n-heptane fuel is the primary fuel supplied to it. When switching to HCCI mode, diesel fuel is used like a typical engine and aids in the start-up procedure. Table 2 shows some of the basic features of the test fuel.

Table 2. Test of n-heptane fuel properties

No.	Property	Unit	Testing Method	Value
1	Chemical formula	-	-	n-C_7H_{16}
2	Kinematic viscosity at 40°C	mm^2/s	ASTM D445 - 00	0.567
3	Closed cup flash point temperature	°C	ASTM D93 - 02	-4
4	Density at 15°C	g/cm^3	ASTM D1298 - 05	0.68873
5	Vapor pressure at 37.8°C	PSIG	ASTM D323 - 99	1,8
6	Calorific value	kCal/kg	ASTM D240 - 02	11.125
7	n-heptane content	%	GC/MS	97.5
8	Cetane number	-	-	56
9	Air-fuel ratio	-	-	15.132

2.4. The implementation processes

Figure 6 depicts the procedure of carrying out the steps in this investigation.

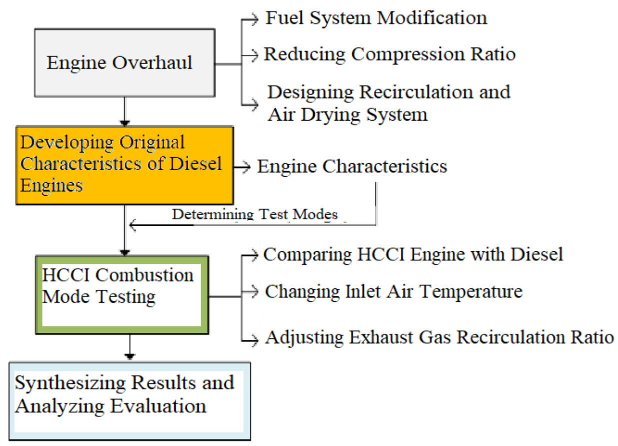


Figure 6. Process of conducting test studies

Step 1: Convert the engine selected for testing to use a common rail (CR) supply system. At the same time, design and manufacture other systems including indirect fuel supply systems on the intake tract; intake air heating systems; exhaust gas recirculation systems.

Step 2: Test the construction of the characteristics of pure diesel engines as a basis for determining further test modes.

Step 3: Conduct testing to establish HCCI combustion mode with n-heptane fuel and evaluate the technical features of the engine

Step 4: Control the exhaust gas cycle rate, new intake air temperature to control HCCI combustion.

2.5. Instrumentation for Testing

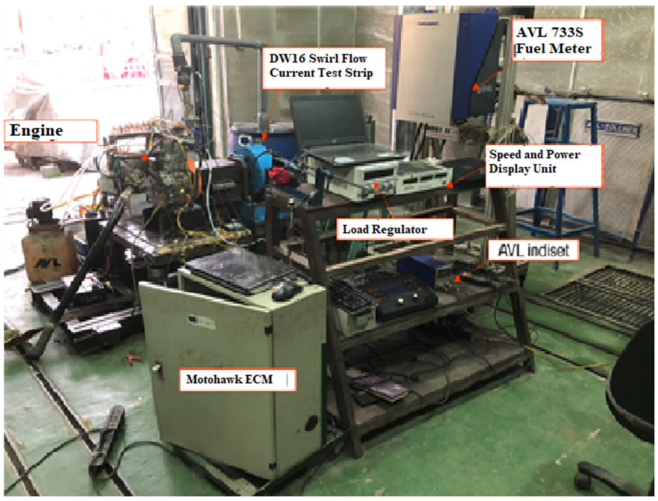


Figure 7. Test system layout diagram

The test was performed on a DW16 small brake test belt with a braking power range of up to 15kW and speeds ranging from 1000 volts per hour to 8000 volts per hour. In addition to the test brake, other auxiliary equipment such as a fuel consumption measurement device, an intake air flow measuring device, and a control ECU are used during the test. The pressure sensor positioned on the lid determines the cylinder pressure signal to determine the combustion process inside the cylinder. Other sensors, such as an intake air temperature sensor, post-drying gas, cyclic gas, and so on, are also installed on the engine to control the experimental process. Figure 7 depicts the installation of the engine and test equipment in the test room.

3. RESEARCH RESULTS AND DISCUSSION

3.1. N-heptane injection nozzle characteristics

In this study, the authors used liquid fuel injectors to supply n-heptane fuel on the intake manifold. To facilitate the process of controlling fuel injection and determining the amount of fuel consumed, the characteristic curve for injecting n-heptane into an open environment (pressure 1at) was built. The experimental system diagram to build the n-heptane injection nozzle characteristic curve is shown in Figure 8.

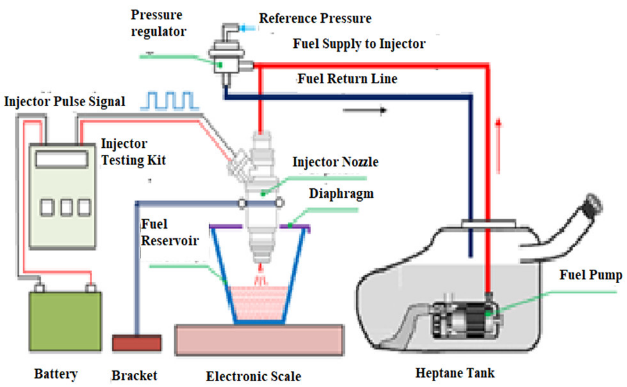


Figure 8. Experimental diagram to build n-heptane nozzle characteristics

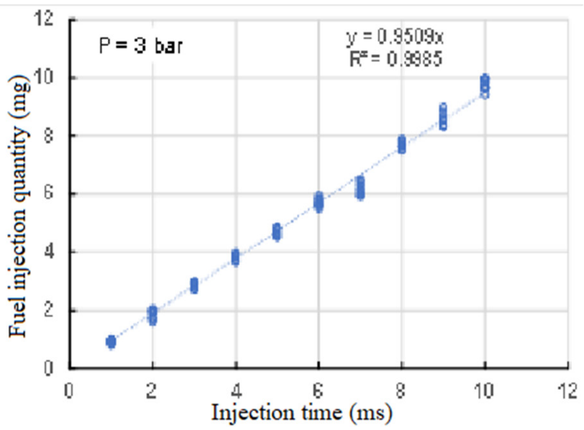


Figure 9. Results of n-heptane nozzle characteristic curve at 3 bar pressure

Conduct a test to measure the spray characteristics of an ethanol nozzle in an open environment (pressure 1 at) at an injection pressure of 3bar. Change the control pulse width to 2, 4, 6,…, 12ms respectively. In each of these modes, perform 5 times, each time delivering 300 pulses (300 sprays). After each injection, weigh the amount of fuel in the container to determine the amount of fuel in one injection. The results of the nozzle characteristics are shown in Figure 9, where the horizontal axis represents the injector opening time and the vertical axis is the amount of fuel in one injection.

3.2. Building characteristics of pure diesel engines

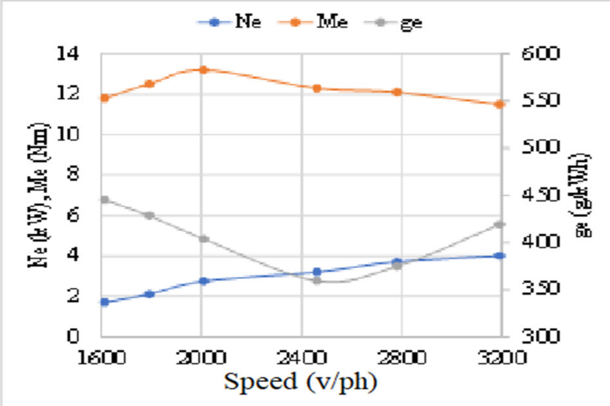


Figure 10. External characteristics of pure diesel engines

The external characteristics of pure diesel engines are determined by the limit of the air-fuel ratio. In each speed mode, the amount of fuel injected is adjusted to reach the lambda limit of around 1.2. Then, determine the maximum torque and power of the motor. The test results of building the external characteristic curve of the pure diesel engine are shown in Figure 10.

3.3. Evaluate the possibility of HCCI combustion formation

The results of the heat release rate tests for both the conventional diesel engine and the HCCI engine using only n-heptane fuel at a fixed speed of 2000rpm and varying loads of 10%, 20%, and 30% are depicted in Figure 11. The heat release rate plot of the HCCI engine is significantly different compared to the diesel engine, as the combustion process is divided into two phases: (1) the cold flame formation phase and (2) the hot flame phase. During the cold flame formation phase, the temperature inside the cylinder is low, and heat exchange occurs between the ignition point and the fuel-air mixture in the vicinity. The hot flame phase is the primary combustion phase of the HCCI engine, at this time which the mixture has become self-ignitable. The heat release rate increases rapidly within a very short time during this phase, and the majority of the heat released in the HCCI engine's combustion process originates from this stage.

For the HCCI engine, since the homogeneous mixture is formed outside the cylinder, the combustion process is more efficient. Additionally, n-heptane fuel has a higher self-ignition capability compared to diesel, leading to an earlier ignition of the combustion process. As the load increases, combustion occurs even earlier before the top dead center (TDC), resulting in the phenomenon of increased knocking. This is the reason for limiting the operating range of the HCCI engine. The results in Figure 11 also indicate that the start of combustion for both the cold and hot flames occurs relatively early before TDC, which increases the engine's compression work. Therefore, a suitable solution is needed to control the timing of ignition to ensure that the combustion process occurs at the correct timing.

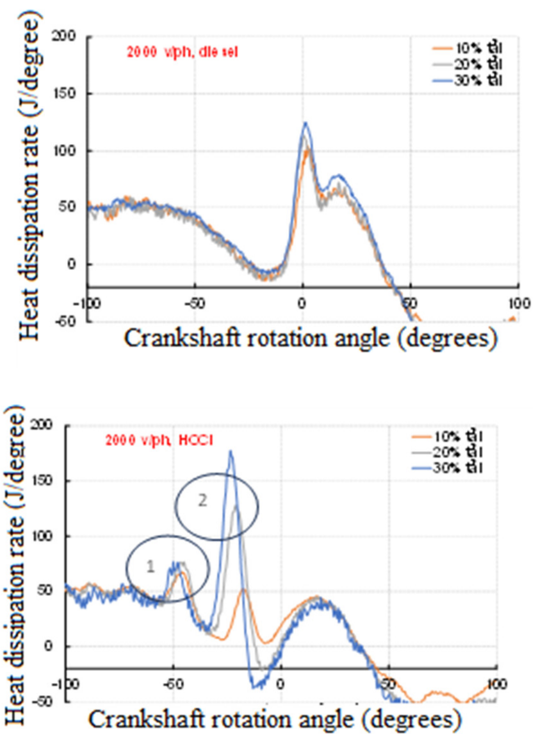


Figure 11. Course of the engine's heat release rate in original and HCCI modes

3.4. Compare the stability of the HCCI combustion process with that of the compression combustion process

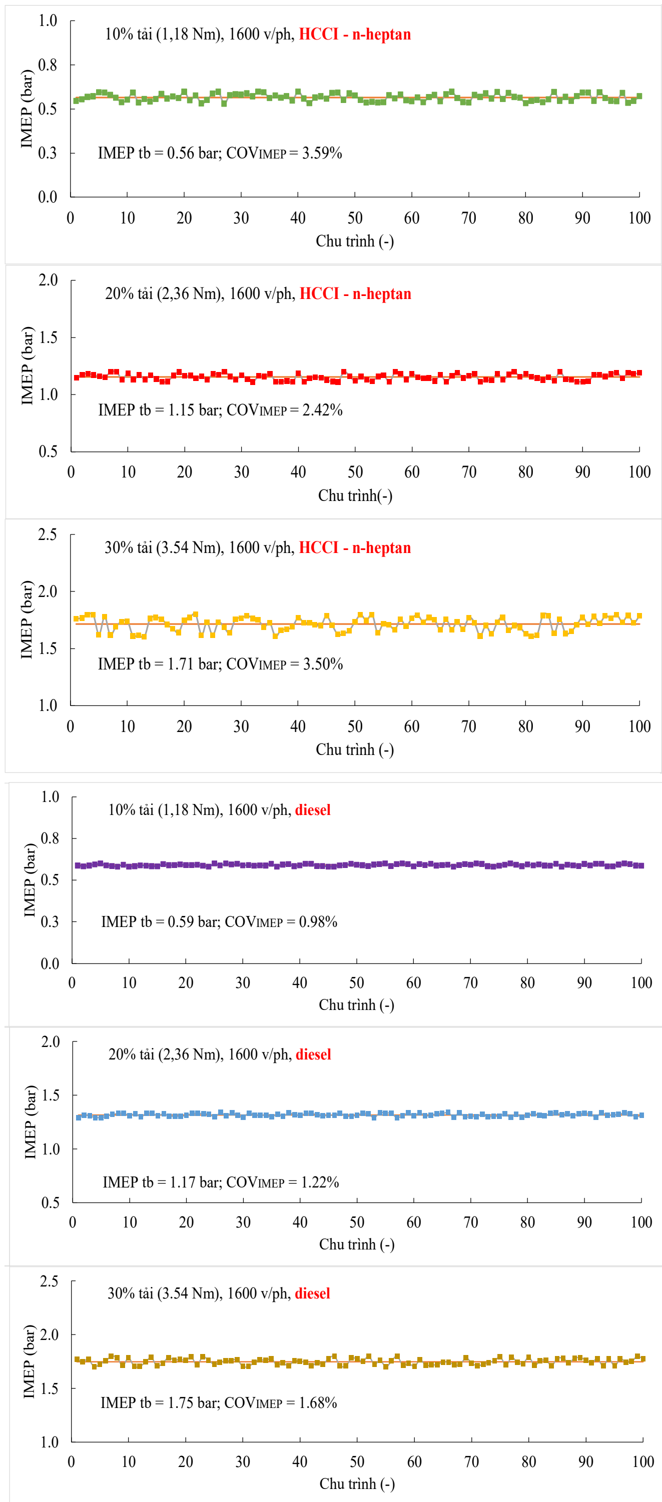
In this study, the authors investigated the performance of the HCCI engine using pure n-heptane fuel at speeds of 1600, 2000, and 2400rpm. The test results determine the average indicator pressure fluctuation coefficient COVIMEP shown in Figure 12. The COVIMEP coefficient represents the level of fluctuation in the engine's operation at different operating modes. The course in cylinder pressure with respect to the crankshaft angle was used to determine the IMEP values for each cycle according to Equation 2. The average IMEP value over 100 consecutive cycles and COVIMEP were determined according to Equations 3 and 4.

$$IMEP = \frac{\oint pdV}{V_h} \tag{2}$$

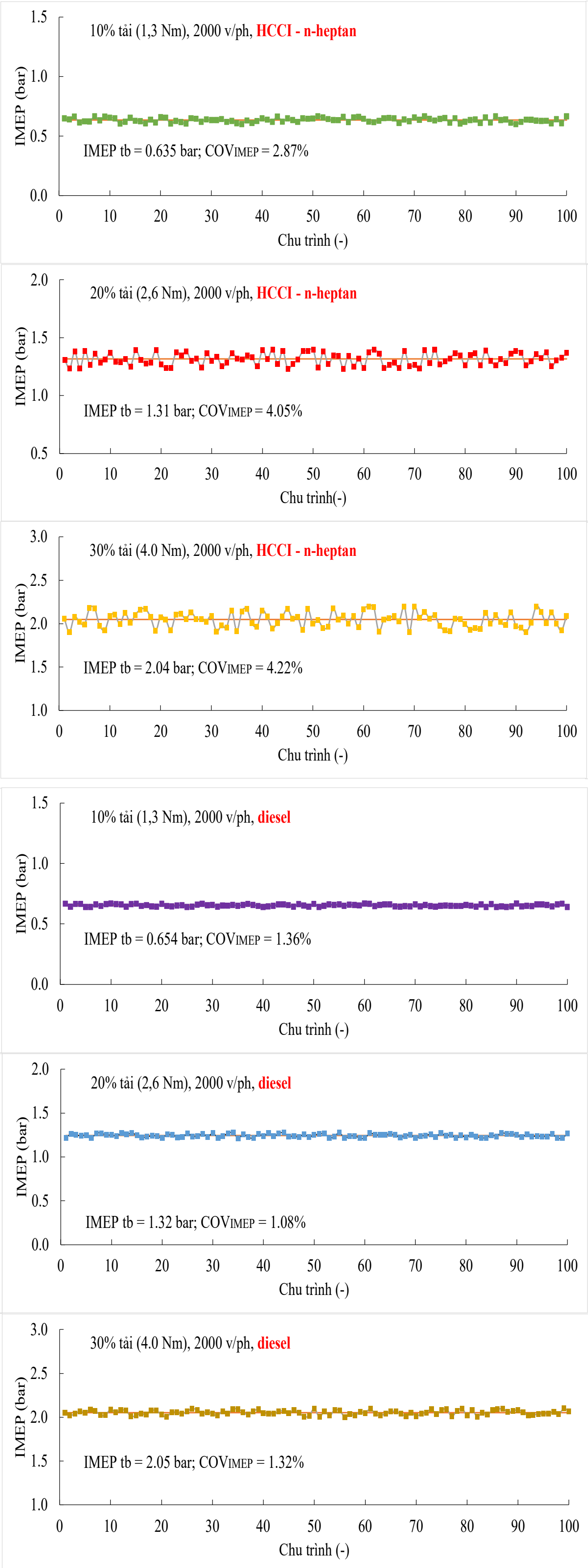
$$\overline{IMEP} = \frac{\sum_{i=1}^{n} IMEP_i}{n} \tag{3}$$

$$COV_{IMEP} = \frac{\sqrt{\frac{\sum_{i=1}^{n}(IMEP_i - \overline{IMEP})^2}{n-1}}}{\overline{IMEP}} \tag{4}$$

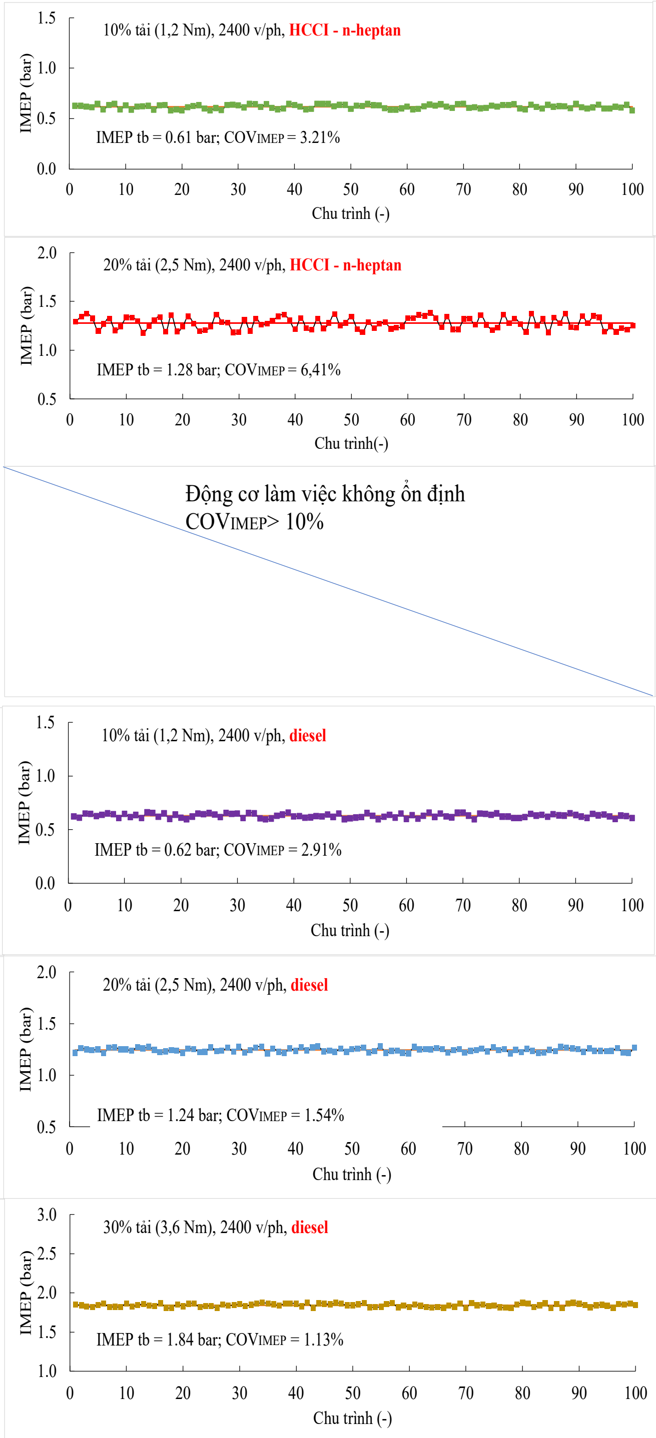
In which: p is the cylinder pressure according to the crankshaft rotation angle (bar), dV is the variation of the engine's working volume (liters), V_h is the engine's working capacity (liters); $IMEP_i$ is the average indicated pressure in a working cycle (bar); $\overline{IMEP}$ is the average indicated pressure over n consecutive cycles (bar); COV_{IMEP} is the coefficient of variation of average indicated pressure (%); n is the number of power cycles measured and stored.



a) Test results at constant speed of 1600rpm, load 10% - 20% - 30%



b) Test results at constant speed of 2000rpm, load 10% - 20% - 30%



c) Test results at constant speed of 2400rpm, load 10% - 20% - 30%

Figure 12. Comparison of COV_{IMEP} of diesel and HCCI-n-heptane engines

The test results in Figure 12 also indicate that, as the load and speed increase, the COVIMEP coefficient tends to increase for the HCCI-n-heptane engine. Especially at a 30% load, a speed of 2400rpm, corresponding to a torque of 3.6Nm, the HCCI-n-heptane engine cannot maintain its working condition, and there is significant engine knocking. At lower loads and speeds of 10% - 20% load, 1600 - 2000rpm, the HCCI engine can still operate, but the

COVIMEP coefficient fluctuates considerably compared to the diesel engine at most test points. This indicates that the HCCI combustion process is not well controlled, resulting in significant oscillations between cycles.

3.5. Effect of intake air temperature and EGR ratio on HCCI combustion process

The authors conducted an assessment of the influence of new intake air temperature and exhaust gas recirculation rate on the engine's combustion process. The results of studying the pressure course in the cylinder at speed mode of 2000rpm, 30% load are shown in Figures 13 and 14.

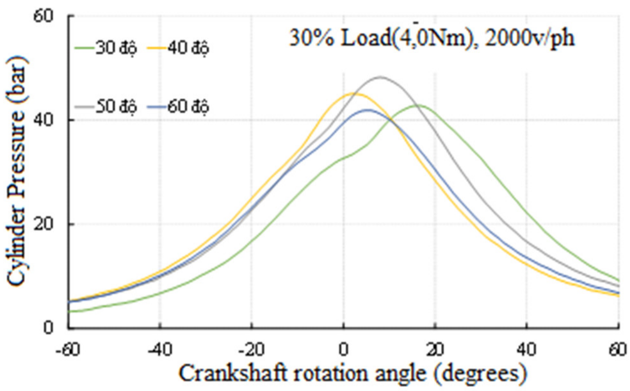


Figure 13. Pressure course in the cylinder when changing temperature of new intake mixture.

The intake air temperature is a parameter that affects the formation of a homogeneous mixture in the HCCI engine. To assess the impact of this parameter, the research team conducted intake air heating using a heater wire to control the temperature of the incoming air at levels of 30°C, 40°C, 50°C, and 60°C with a fluctuation range of +/-2°C. The electrical power supplied to the heater wire was controlled to maintain the temperature of the new intake mixture right at the intake port stable. The experimental process shown that when the temperature of the new intake mixture was varied, there was a noticeable change in the engine's combustion process, as evidenced by the pressure curve inside the cylinder shifting to the left, as shown in Figure 13. This implies that the flame formation process in the cylinder occurred earlier.

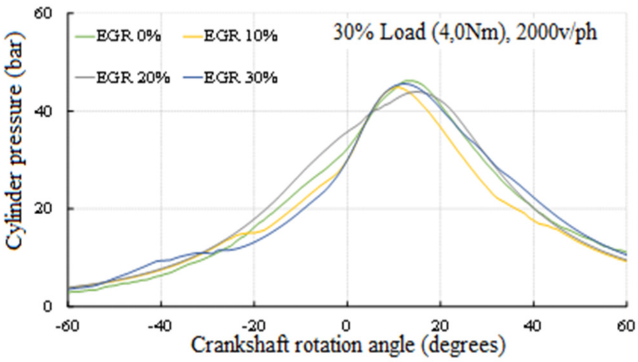


Figure 14. Course of cylinder pressure when changing the exhaust gas recirculation ratio

Exhaust gas recirculation (EGR) is controlled by adjusting the resistance on the engine's intake mainfoild and gradually opening the EGR valve to recirculate a portion of the exhaust gases back into the intake mainfoild. The recirculated air is passed through a cooler to control its temperature, ensuring that the temperature of the new intake air mixture at various test conditions, with and without recirculation, remains stable at 30°C (with a permissible deviation of +/-2°C). The results depicting the pressure course inside the engine cylinder with different EGR ratios of 10%, 20%, and 30% are shown in Figure 14. The results indicate that as the EGR ratio increases, the pressure inside the cylinder tends to decrease. This is because the recirculated air reduces the air-fuel mixture density, thereby reducing the combustion rate. The timing of reaching maximum pressure also gradually shifts towards top dead center (TDC), indicating a delayed start of combustion compared to before.

4. CONCLUSION

In the research aimed at transforming a conventional diesel engine into an HCCI engine, several results were obtained:

- The engine's compression ratio was adjusted down to 17 by changing the cylinder head gasket thickness to 0.8mm.

- The intake mainfoild of an indirect fuel supply system was designed to enable the engine to operate in HCCI mode. Simultaneously, the electronic diesel fuel injection system was also used when the engine operated in traditional diesel mode, supporting the starting process and warm-up phase when transitioning to HCCI mode. Other systems such as exhaust gas recirculation and air heating systems were designed and manufactured to assist in controlling the HCCI engine's combustion process.

- With the high self-ignition propensity fuel injection solution into the intake mainfoild, using n-heptane in the intake, the HCCI engine could operate stably at low-speed and low-load conditions. The stable operating speed range with n-heptane fuel was approximately from 1600rpm to 2000rpm, and the torque ranged from 1.2 to 3.4Nm. At higher load and speed regions, engine vibrations were significant, and COVIMEP exceeded the 10% threshold.

- The influence of intake air temperature and exhaust gas recirculation ratio on the HCCI engine's combustion process was also tested. Increasing the intake air temperature tended to advance the combustion process. Conversely, increasing the exhaust gas recirculation ratio tended to slow down the combustion process.

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