Original Research Article: full paper

(2024), «EUREKA: Physics and Engineering» Number 2

INVESTIGATE THE EFFECT OF EGR STRATERGIES ON THE HCCI ENGINE'S PERFORMANCE WITH N-HEPTANE FUEL INJECTION

Khuong Thi Ha
Faculty of Mechanical Engineering
University of Transport and Communications
3 Cau Giay str., Lang Thuong Ward, Hanoi, Vietnam, 100000

Tran Lan Anh
School of Languages and Tourism[1]

Nguyen Tuan Nghia✉
Automotive Practice and Inspection Center[1]
nghiant@haui.edu.vn

[1]Hanoi University of Industry
298 Cau Dien str., Bac Tu Liem District, Hanoi, Vietnam, 100000

✉**Corresponding author**

Abstract

In this study, the effect of EGR strategies on the HCCI engine's performance is studied. To achieve the research goal, a program was established to control the injection fuel and amount of exhaust gas recirculation in the HCCI engine. The set signal entered through the control computer is compared with the signal collected from the sensor's crankshaft and camshaft plates to give control signals to the injectors and EGR valves. The signals collected from the camshaft and crankshaft sensors are processed before being compared with the reference signal. A diesel engine that is 1 cylinder, 4-stroke, non-turbocharged, and air-cooled has an improved intake manifold to convert to combustion mode due to homogeneous charge compression ignition characteristics. The conversion engine is tested with 1200 rpm to 2800 rpm speed modes, a 400 rpm split point, 50 % load, 0 % to 30 % increased EGR rates, and a 5 % EGR split point. The fuel used in this research is n-heptane. The CEB-II gas analysis cabinet was used to investigate the exhaust gas components. The results of the research show that when the EGR rate is greater than 30 %, the engine operates stably according to the HCCI combustion characteristic at speeds lower than 2400 rpm. But when engine speeds are higher than 2400 rpm, the HCCI combustion characteristic is unstable; the torque and the indicator efficiency decrease rapidly. When the HCCI engine increases the EGR rate, the start of combustion is gradually later, the coefficient gradually decreases to close to the black smoke characteristic curve, the CO, HC, and CO_2 emissions increase. The NO_x emissions tend to increase when the EGR ratio increases from 0–25 %, but the NO_x decreases when the EGR rate is greater than 25 %.

Keywords: HCCI, EGR rate, engine performance, emission, NO_x, CO, HC, CO_2, HCCI engine, torque, n-heptane.

DOI: 10.21303/2461-4262.2024.003235

1. Introduction

The internal combustion engine plays an important role in the global economy. It serves the needs of commuting, transporting goods, etc. The problem of engine emissions and their great dependence on fossil fuels pose a challenge for researchers to find solutions to improve internal combustion engine progress. The main research directions that have been and are being implemented are reducing environmental pollution caused by engines, saving fuel, improving performance, and finding alternative fuel sources [1–4].

In 1979, research on the CAI (control auto ignition) fire model was carried out [5, 6]. After these first studies, there was a trend toward research and development of engines using homogeneous compression charge ignition (HCCI) models for automotive engines. Worldwide studies on HCCI engines are expected to overcome the disadvantages of traditional engines with the following advantages: low NO_x and PM emissions (HCCI engines reduce NO_x due to lower combustion temperatures compared to diesel engines). Original and burned with many combustion centers uniformly distributed in the combustion chamber space, so it burns cleanly, reducing PM [7–9], its thermal efficiency is equivalent to a direct injection engine [10], and it has the possibility of using a variety of alternative fuels [11–17].

In studies on HCCI engines, it is shown that the new combustion model has so many different characteristics than the combustion model of traditional gasoline and diesel engines [18]. On the original gasoline engine, the flame spread film originates from the spark plug, while on the engine operating with the HCCI principle, there is no spreading of the flame film in the cylinder; the combustion process takes place simultaneously in all positions. In the cylinder (in this case, the spark plug does not ignite; the spark plug serves for the transition between the normal mode and the HCCI mode). Similar to gasoline engines, but different from conventional diesel engines, with HCCI engines, the fuel and air mixture is preformed (in the intake manifold or in the cylinder). After that, the mixture compressed up to the auto-ignition temperature at the end of the compression stroke, similar to that of a diesel engine. In addition, it is possible to increase the mixture temperature at the end of the intake process by heating the intake air, heating it with a spark plug, or using leftover air in the cylinder. All of these methods can help the mixture reach the ignition temperature more quickly and become more homogeneous.

However, there are still many problems to be solved for HCCI engines, such as the inability to control combustion directly, high CO and HC emissions, and a concentrated working area at low loads. The problem of controlling the start of combustion on HCCI engines is not as simple as on gasoline and diesel engines; it is necessary to ensure the properties of the mixture so that the start of combustion is near the TDC. On gasoline engines, part of the fuel-air mixture gets stuck in the interstitial joints; when the piston goes down, this component will be ignited due to the high temperature (greater than 2500 K). However, on HCCI engines, the combustion temperature is very low (less than 1800 K), so this mixture is not decomposed, and CO and HC emissions are high. At low loads, the maximum temperature value is very small (only about 1200 K), not enough for CO to convert into CO_2, so the auto ignition process is more difficult. At high load, because the mixture is burned at the same time, the heat release rate is very fast, and the pressure increase rate is large, which adversely affects the engine. The working area of the HCCI engine is limited by two factors: no ignition and «knocking» at high speeds. The mixture is more difficult to auto-ignite because there is not enough time to react [19]. In comparison with previous studies about HCCI engines, this research applied the EGR strategy method and used N-heptane as fuel to increase the homogeneous air-fuel mixture and decrease NO_x emissions. Using N-heptane fuel is an effective method to reduce knocking happen. Due to the high latent heat of evaporation, the evaporation process of n-heptane in the intake pipe results in lowering of the intake air temperature, which can negatively affect the auto-ignition process of the fuel-air mixture in the combustion chamber.

In this study, let's present a method to convert a traditional diesel engine to operate under HCCI combustion through exhaust gas recirculation, with the desire to expand the HCCI combustion process for traditional engines without having to change the construction of the original engine. In order for the conversion engine to work, it is necessary to design a control system for the engine. It includes the system to control the amount of circulating gas, the amount of fuel injected in accordance with the modes of the engine, etc. The controller is prepared to be used for the test. The fuel used for the engine is volatile and is supplied on the intake pipe outside the combustion chamber. On the other hand, it is possible to investigate the influence of the recirculation gas ratio on the ability to set the HCCI working mode at each speed mode of the engine. Testing was conducted on diesel engines currently circulating in Vietnam.

2. Materials and methods

The test engine (Kubota BD178F (E)) in this study is a 1-cylinder diesel engine that is 1 cylinder, 4-stroke, non-turbocharged, and air-cooled with an improved intake manifold to convert to combustion mode due to homogeneous charge compression ignition characteristics (**Fig. 1**). The engine specification is as shown in **Table 1**. The fuel used in the experiment is n-heptane; the characteristics of the fuel are shown in **Table 2**. The conversion engine is tested with 1200 rpm to 2800 rpm speed modes, a 400 rpm split point, 50 % load, 0 % to 30 % increased EGR rates, and a 5 % EGR split point (**Table 3**).

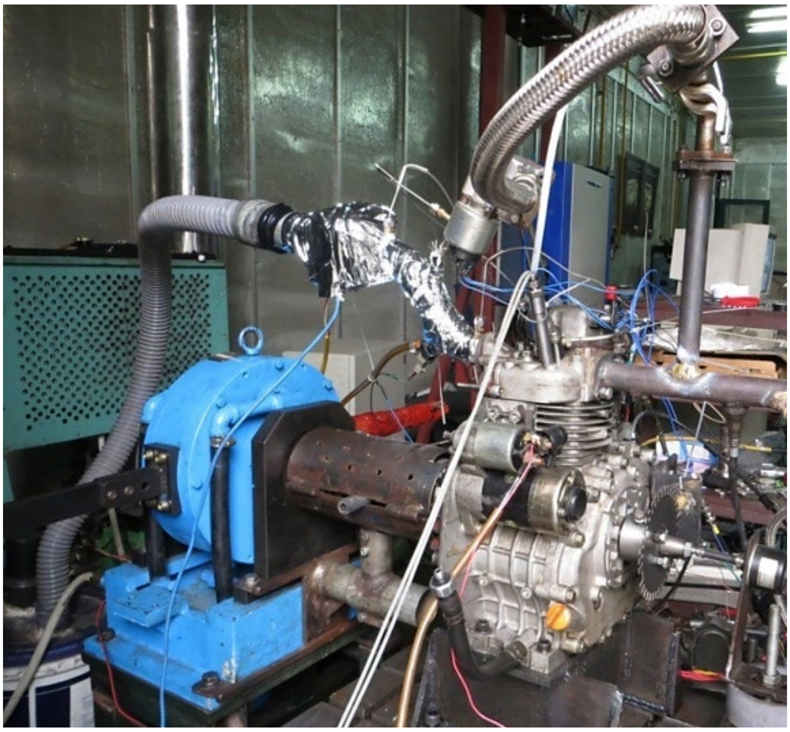

Fig. 1. The experimental system

Table 1
Specification of the test diesel engine

Parameter	Value
Displaced volume	273 cm^3
Type of engine	Direct Injection, naturally aspirated
Stroke	57 mm
Bore	78 mm
Compression ratio	20:1
Number of valves	2
Rated power/speed	4.4 kW/3600 rpm
Max. torque/speed	13 Nm/2000 rpm

Table 2
Specification of n-heptan

No.	Property	Unit	Test method	Value
1	Chemical formula	–	–	n-C_7H_{16}
2	Kinematic viscosity at 40 °C	mm^2/s	ASTM D445-00	0.567
3	Flash point	°C	ASTM D93-02	–4
4	Proportion at 15 °C	g/cm^3	ASTM D1298-05	0.68873
5	Steam pressure at 37.8 °C	PSIG	ASTM D323-99	1.8
6	Heat value	Kcal/kg	ASTM D240-02	11.125
7	Content n-heptan	%	GC/MS	97.5
8	Cetane number	–	–	56
9	A/F value	–	–	15.132

Table 3
Experimental modes

Rate EGR (%)	Speed engine (rpm)				
	1200	1600	2000	2400	2800
10	x	x	x	x	x
15	x	x	x	x	x
20	x	x	x	x	x
25	x	x	x	x	x
30	x	x	x	–	–

The schematic diagram of the control program built for the HCCI engine converted from a diesel engine is shown in **Fig. 2**. The set signal entered through the control computer is compared with the signal collected from the sensor's crankshaft and camshaft plates to give control signals to the injectors and EGR valves. The signals collected from the camshaft and crankshaft sensors are processed before being compared with the reference signal.

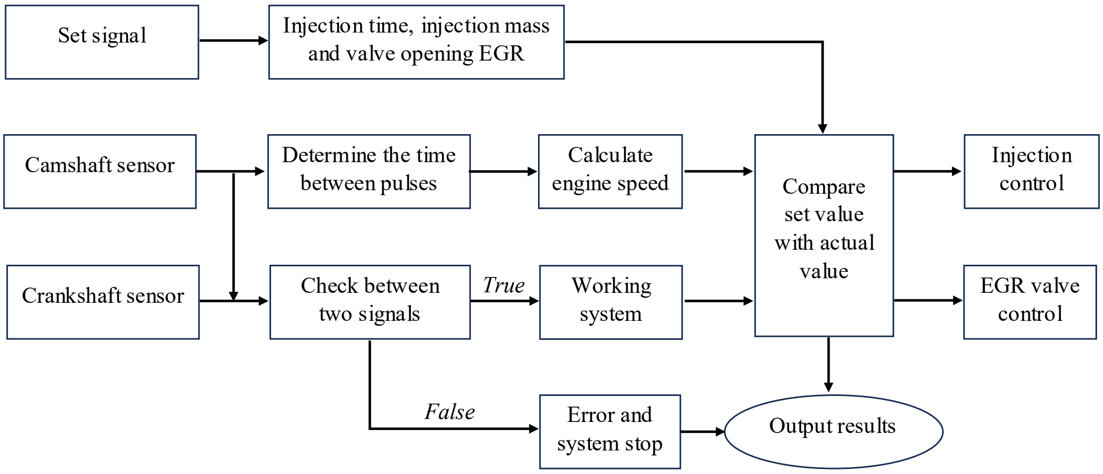

Fig. 2. Structure diagram of the program to control the fuel injection system and control the amount of exhaust gas recirculation on the HCCI engine

DW16 engine test bench: the test bench used during the experiment is an eddy current dynamometer-type engine test bench, denoted DW 16. Eddy current dynamometer test benches usually have a simple structure and a consistent delay. Small size, large torque measurement, high speed, good adaptability to motors, fast response speed, can work for a long time, easy maintenance and repair, besides the benefits The advantage of the DW 16 test bench is that it can determine the technical features of engines with small capacity or engines working in a narrow and small power range. The DW 16 test bench can test motors with a maximum torque of 70 Nm and a maximum power of 16 kW, and the speed of the bench varies from 0 to 13,000 rpm with an accuracy of 0.1 %.

CEB-II gas analysis cabinet: the CEB-II exhaust gas analysis cabinet (Combustion Emission Bench) is a system that includes all modules that perform the analysis of exhaust gas components (analyzers) and other devices and equipment to ensure the correct working conditions of the system, such as the heating block (HSU), diagnostic block, control block, etc. In addition, the analysis cabinet is also equipped with an industrial computer with GEM110 control software. The connection of the control computer to the analyzers is done via digital signals. Depending on the analyzer, it can be connected to the computer via CAN, LON network, or RS232 serial cable. Analyzers installed in the cabinet are used to measure components in exhaust gases such as carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2), nitrogen oxides (NO and NO_x), and carbon hydrides (HC). They also measure the air residue coefficient with an accuracy of 0.1 %.

Smoke meter device: when a certain amount of exhaust gas passes through a standard filter paper membrane, PM will be retained, causing the filter paper to turn black. The determined blackness of the filter paper will reflect the smoke level of the exhaust gas. The Smoke Meter AVL 415 device has a measuring range from 0 to 9.99 FSN (filter smoke number) or from 0 to 3199 mg/m^3 with an accuracy of 0.1 %.

Pressure measuring sensor: QC33C cylinder pressure sensor, measuring range 0÷200 bar, sampling with resolution of 0.5 degrees crankshaft rotation angle, data acquisition device of pressure sensor with Indiwin software.

The n-heptane fuel injection process is based on the control system with the diagram shown in **Fig. 3**. The injection signal is connected to number 19 port of the microprocessor. To perform the fuel injection process, the microprocessor receives information about temperature, pressure, n-heptane fuel speed, throttle position, temperature, and intake air pressure through ports from 35 to 40. To maintain n-heptane fuel pressure before the injector, the fuel injection pressure is maintained at bar and is controlled via the microprocessor via port 22. In addition, to ensure for the safety of the control system, relays number 1 and 2 are used.

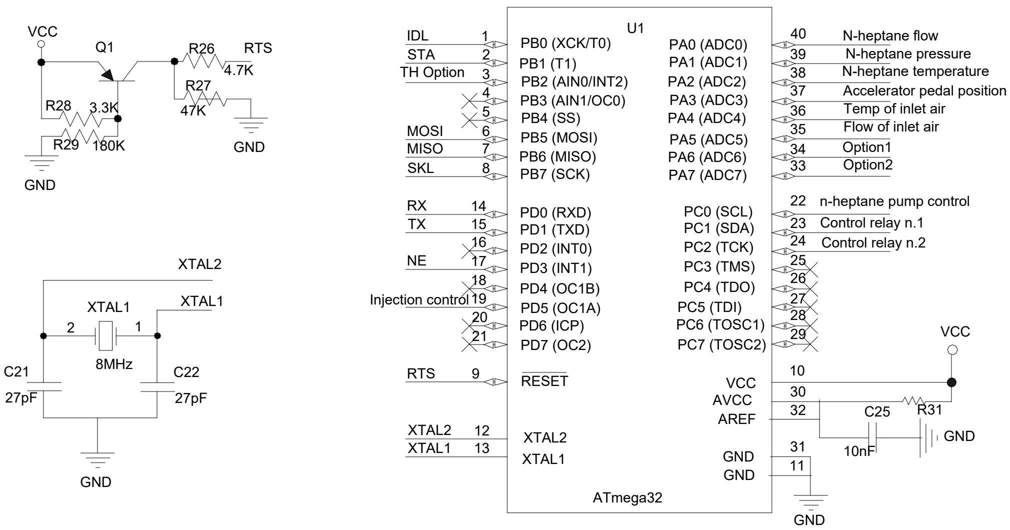

Fig. 3. N-heptane fuel injection control system for HCCI engine

In **Fig. 4**, the microprocessor and EGR valve opening results are displayed on the test engine. By using the display screen in the experiment, the opening of the EGR valve can be easily controlled and adjusted to suit the working mode of the engine.

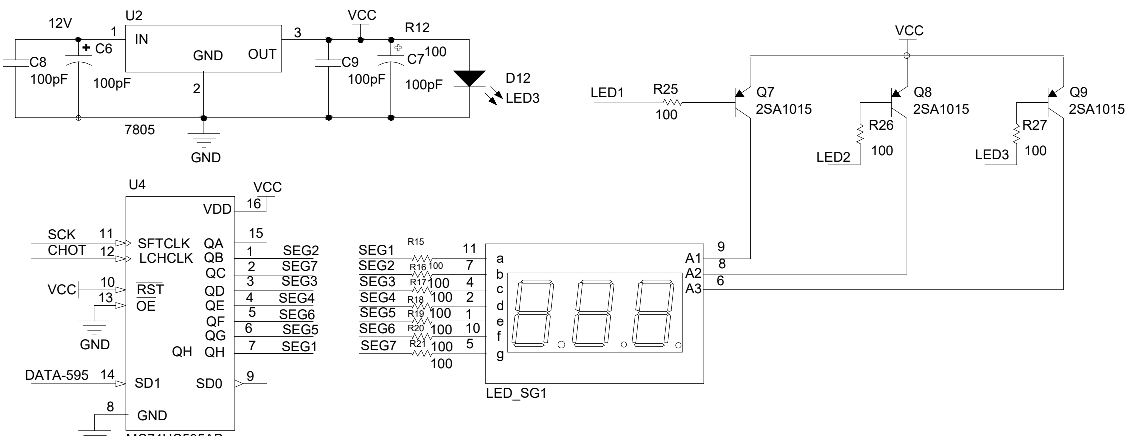

Fig. 4. Microprocessor and display of exhaust gas recirculation valve opening results

3. Results and discussions

In this section, the results are presented at 50 % load, speeds of 1200–2800 rpm, and different EGR rates to evaluate the maximum performance of the HCCI engine. The specific results are as follows: in **Fig. 5** it is shown that when the EGR rate increases gradually, which is the result of the pressure rise rate at low speeds (1200–1600 rpm), the engine tends to knock, and the test engine at this moment appears to vibrate. At speeds less than 2400 rpm, the test engine operates stably, but at 2800 rpm, it cannot be maintained well because the pressure rise rate gradually decreases and the pressure rise rate decreases so fast when increasing the EGR rate. If the EGR rate increases to 30 %, the test engine will not maintain the HCCI combustion process at 2400 rpm.

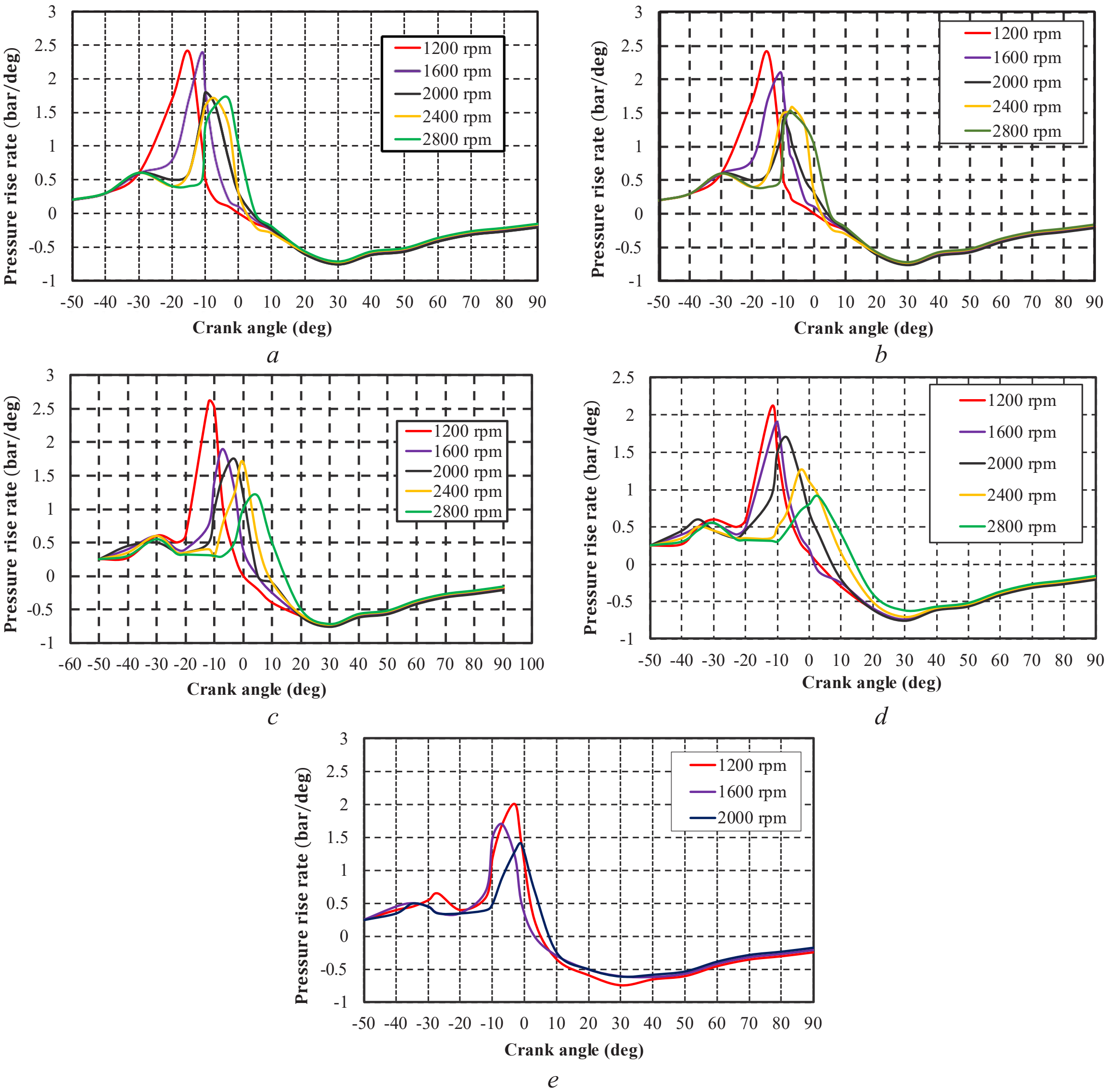

Fig. 5. The effect of exhaust gas recirculation rate on pressure rise rate with various exhaust gas recirculation rates: *a* – 10 % EGR rate; *b* – 15 % EGR rate; *c* – 20 % EGR rate; *d* – 25 % EGR rate; *e* – 30 % EGR rate

Fig. 6 shows that when the EGR rate increases from 10 % EGR to 25 % EGR, the results of the heat release rate of HCCI curves at small and medium engine speeds (1200–2000 rpm) follow the theory of the HCCI combustion characteristics. The rate of heat release of HCCI curves had two flames that formed a cold and hot flame, which correspond to the cold flame start of combustion and the hot flame start of combustion. But at higher engine speeds (2400–2800 rpm) and when the

EGR rate is greater than 30 %, diffuse combustion (similar to the combustion process in a traditional diesel engine) occurs at the end of the hot flame's burning phase, so it tends to be difficult to maintain the HCCI combustion characteristics. At the same time, when gradually increasing the rate of EGR, the maximum value of the rate of heat release gradually decreases, and the time of reaching the maximum value of the rate of heat release is also later, which adversely affects the technical parameters of the HCCI engine.

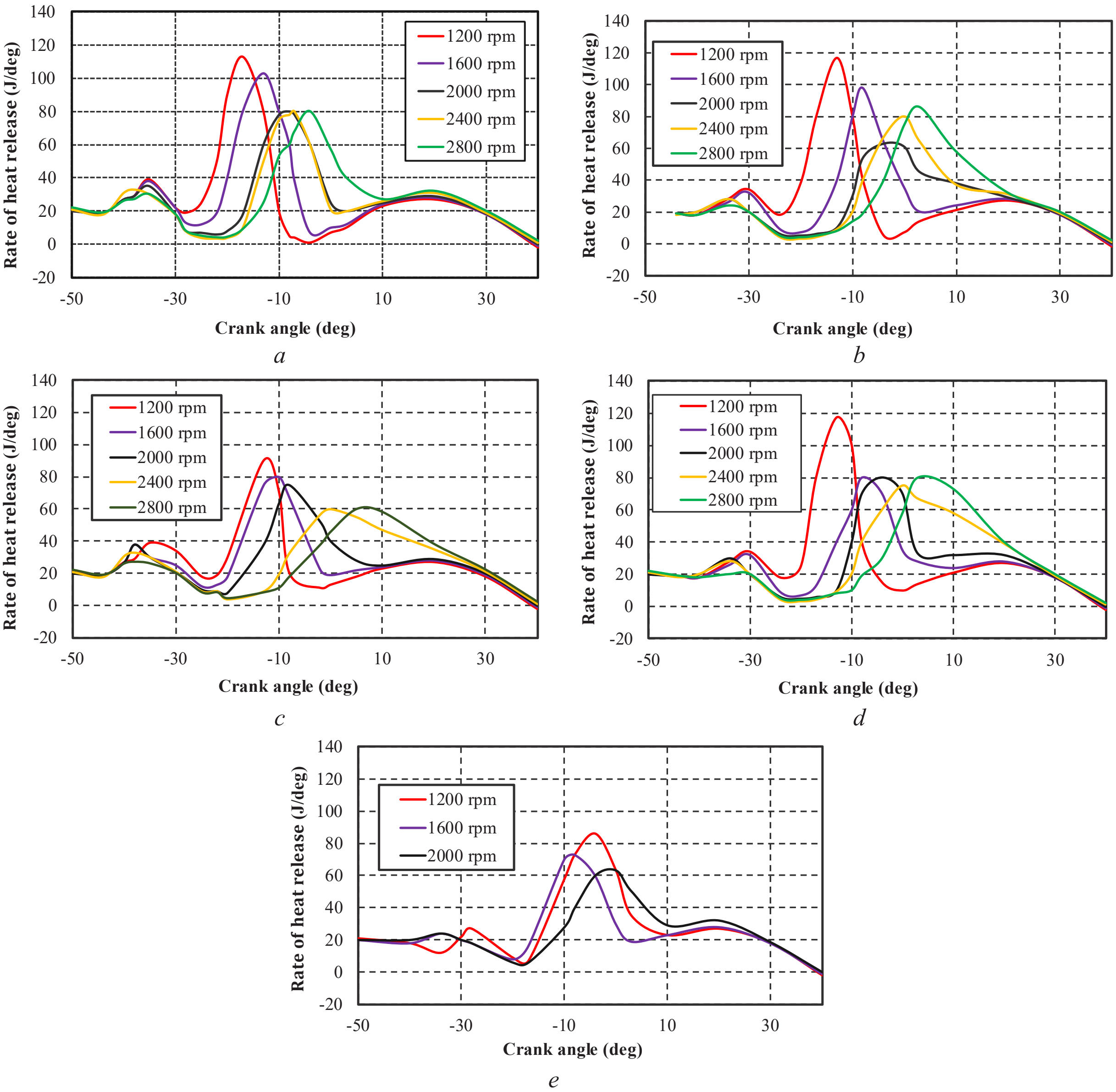

Fig. 6. The effect of the exhaust gas recirculation rate on the rate of heat release with various exhaust gas recirculation rates: *a* – 10 % EGR rate; *b* – 15 % EGR rate; *c* – 20 % EGR rate; *d* – 25 % EGR rate; *e* – 30 % EGR rate

Fig. 7 shows the result of the start of combustion of the HCCI engine at different EGR rates, and it is found that, without the addition of exhaust gas recirculation, the start of combustion of the cold flame (SOC1) and the hot flame (SOC2) takes place comes out early before the top dead center, when gradually increasing the EGR rate, the start of combustion is gradually later because the exhaust gas returns to mix with the new intake air to dilute the intake air, the combustion speed decreases, but at a speed of 2400÷2800 rpm and the recycling rate is greater than 30 %, the density of inert gas in the recirculation gas increases, the mixing time between fuel and air decreases, leading to the failure to maintain HCCI combustion characteristics. The start of combustion of the hot flame still occurs early before the top dead center, this is a factor affecting the technical parameters of the HCCI engine, at this time, the process of both combustion and compression will waste compression, leading to reduced engine power.

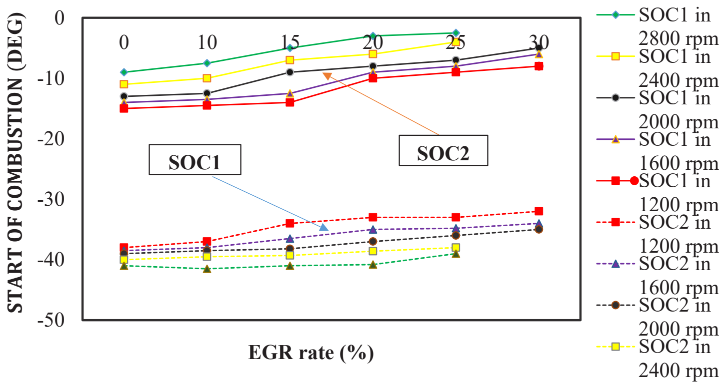

Fig. 7. The effect of exhaust gas recirculation rates on Start of combustion of cool flame and hot flame

Fig. 8 shows the result of the engine torque at the studied speeds when changing the EGR ratio. The torque values of the engine when conducting the experiment to convert a traditional diesel engine to operate according to HCCI combustion characteristics will be kept so that they are equivalent to the original engine, that is, 50 % of the engine loads calculated equivalent to 50 % of the effective torque value of the original diesel engine. The specific value is 6.2 Nm. In order to keep the useful torque value of the HCCI engine, the amount of fuel will be gradually adjusted in increasing speed mode. When compared to the original engine, when the EGR rate is less than 15 %, the HCCI engine maintains the same or better torque value due to the later start of combustion. The pressure rise rate and rate of heat release of the HCCI have a great value. In 20 % EGR, HCCI engines tend to reduce torque due to inert gas components in the cyclic gas making the combustion process worse.

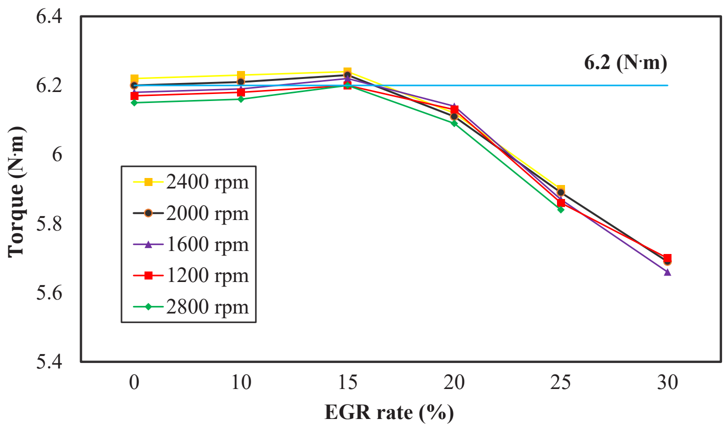

Fig. 8. The effect of exhaust gas recirculation rate on the engine torque

The results in **Fig. 9** show the value of the indicator efficiency of the HCCI engine when gradually increasing the EGR rate. The HCCI engine had a higher indicator efficiency than the traditional engine because of the following reasons: The first is that increasing the EGR rate has adjusted the start of combustion to be later, which helps the combustion process be better, reduces the work of the compression process, and increases the work of the work process, so the indicator performance has a better value. The second is that the engine operating according to the HCCI combustion characteristics will normally burn cleaner. However, when increasing the EGR rate by more than 25 %, the indicator efficiency decreases rapidly due to the inert gas composition in the circulating gas, which makes the combustion process worse and reduces the pressure rise rate and heat release value. A high proportion of inert gas makes the mixture thinner, so the combustion process takes place later and worse, leading to reduced efficiency.

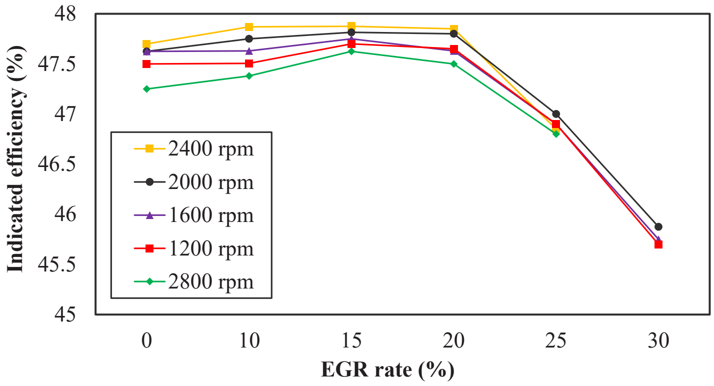

Fig. 9. The effect of exhaust gas recirculation rate on the engine indicated efficiency

The results in **Fig. 10**, **11** show the CO and HC emissions of the HCCI engine when increasing the EGR rate. Both of these emissions tend to increase gradually when increasing the EGR rate and the speed of the engine. The HC and CO emissions increased for the following reasons: the first is that when the EGR rate increases, the inert gas in the intake air ratio increases, and the reaction rate of fuel and air decreases. The second is that if engine speeds increase, the time spent mixing fuel and air will be reduced. At each EGR ratio value, when the speed increases, CO and HC decrease due to the influence of high vortex intensity, so it burns better.

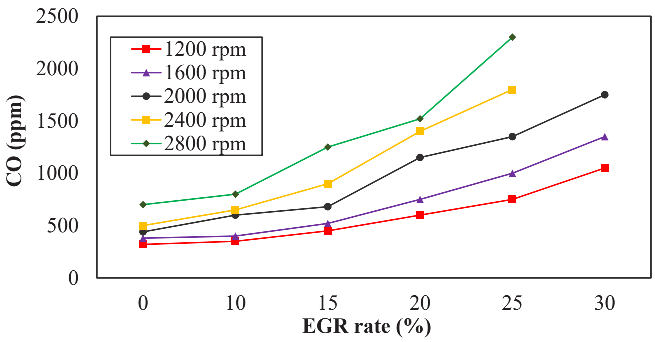

Fig. 10. The effect of exhaust gas recirculation rates on the indicated efficiency

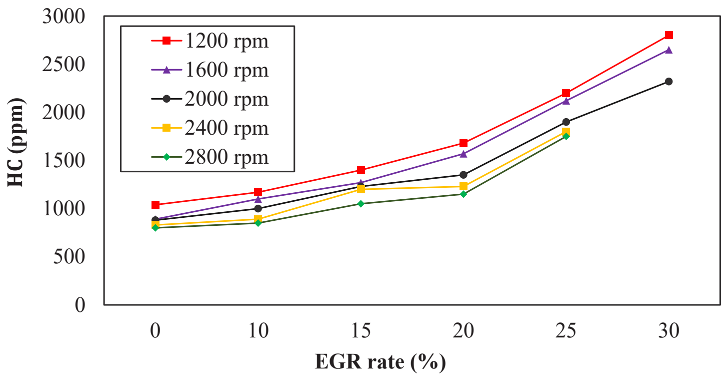

Fig. 11. The effect of exhaust gas recirculation rates on the indicated efficiency

Fig. 12 presents the NO_x emissions of the HCCI engine with increasing EGR rates. The NO_x emissions tend to increase when the EGR ratio increases from 0–25 %, but the NO_x decreases when the EGR rate is greater than 25 %. Because when the EGR rate is greater than 25 %, it will reduce the purity of the combustible gas mixture, which affects the quality of the fuel combustion process and leads to a decrease in the maximum combustion temperature and thermal efficiency of the engine (**Fig. 9**). The NO_x emission had a lower value when compared to the original engine due to the following reasons: the first, If the engine combustion is according to the combustion characteristics of HCCI, the combustion temperature will be lower than that of the original engine, so NO_x will be lowed. Secondly, when increasing the amount of circulating gas, it dilutes the intake air, leading to a decrease in the rate of nitrogen oxidation, thereby reducing NO_x emissions.

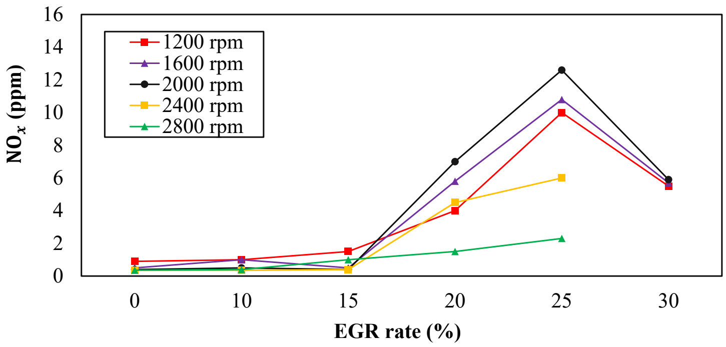

Fig. 12. The effect of exhaust gas recirculation rates on the indicated efficiency

The results in **Fig. 13** show that CO_2 emissions from HCCI engines tend to increase gradually when increasing the EGR rate and engine speed. The reason for the increased CO_2 emissions of the HCCI was the increased amount of fuel required to maintain the same torque as the original engine.

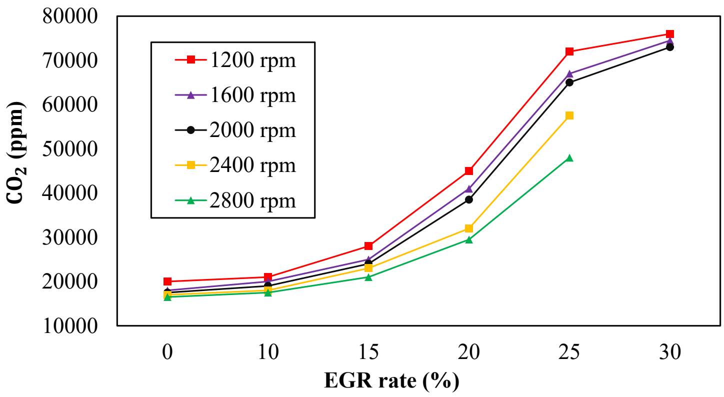

Fig. 13. The effect of exhaust gas recirculation rates on CO_2 emission

Fig. 14 is the result of the ratio between the actual amount of air divided by the theoretical amount of air required to completely burn 1 kg of fuel (λ) when changing the EGR rate. The results show that when increasing the EGR rate, the amount of circulating air takes the place of the intake air on the intake manifold, so the actual amount of air into the cylinder of the engine is reduced compared to the theoretical air volume, so λ gradually decreases at all test speed modes. With the EGR rate from 25 % to 30 %, the λ has a low value; it closes to 1, which means it is close to the black smoke characteristic curve, so the engine does not operate stably at the higher EGR rates. The technical parameters of the HCCI engine are reduced, the indicated pressure decreases, and the working of the HCCI engine is no longer reliable.

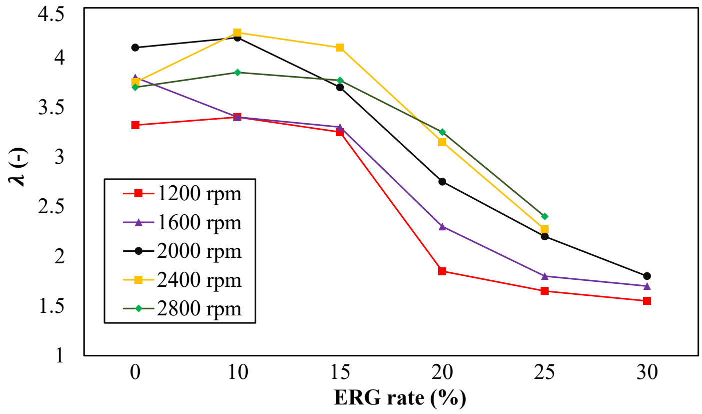

Fig. 14. The effect of exhaust gas recirculation rates on the λ

This research was successful in presenting the effect of EGR strategies on the HCCI engine's performance with n-heptane fuel injection. A diesel engine that is 1 cylinder, 4-stroke was converted to combustion mode due to homogeneous charge compression ignition characteristics, which helped reduce NO_x emissions when compared to the original engine. The limitation of this study is that the engine performance has not been investigated in all part load conditions. The engine speed was limited until 2800 rpm, so the engine operation condition in which the auto-ignition happened was not clearly and fully presented. In this research, the effect of EGR on the PM and auto-ignition is a limitation, so in the next research, this limitation will be elucidated.

4. Conclusions

Experimental results on the HCCI engine when controlling the change of the exhaust gas recirculation ratio in terms of combustion characteristics, start of combustion, torque, indicator efficiency, and emissions at 50 % load at engine speed from 1200 rpm to 2800 rpm were as follows:

– When EGR rate is more than 30 %, the engine operates stably according to the HCCI combustion characteristic at speeds lower 2400 rpm. When engine speeds higher than 2400 rpm the HCCI combustion characteristic is unstable, the torque and the indicator efficiency decrease rapidly.

– The increase in EGR rate leads to a gradual delay in the start of combustion, which helps to improve the technical features of the HCCI engine.

– When the HCCI engine increases the EGR rate, the coefficient λ gradually decreases to close to the black smoke characteristic curve, so it is not possible to increase the exhaust gas recirculation rate any more. The increase in EGR rate causes CO, HC, and CO_2 emissions to increase. The NO_x emissions tend to increase when the EGR ratio increases from 0–25 %, but when the EGR rate is greater than 25 %, NO_x decreases. The NO_x emission of the HCCI engine is very small compared to that of the original engine at all engine speeds and EGR rates.

Conflict of interest

The authors declare that they have no conflict of interest in relation to this research, whether financial, personal, authorship or otherwise, that could affect the research and its results presented in this paper.

Financing

The study was performed without financial support.

Data availability

Manuscript has no associated data.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

References

[1] Nghia, N. T., Khoa, N. X., Cho, W., Lim, O. (2021). A Study the Effect of Biodiesel Blends and the Injection Timing on Performance and Emissions of Common Rail Diesel Engines. Energies, 15 (1), 242. https://doi.org/10.3390/en15010242

[2] Khoa, N. X., Kang, Y., Lim, O. (2019). The effects of combustion duration on residual gas, effective release energy and engine power of motorcycle engine at full load. Energy Procedia, 158, 1835–1841. https://doi.org/10.1016/j.egypro.2019.01.429

[3] Khoa, N. X., Lim, O. T. (2020). The effects of bore-stroke ratio on effective release energy, residual gas, peak pressure rise and combustion duration of a V-twin engine. Journal of Mechanical Science and Technology, 34 (6), 2657–2666. https://doi.org/10.1007/s12206-020-0539-z

[4] Khoa, N. X., Lim, O. (2021). A Study to Investigate the Effect of Valve Mechanisms on Exhaust Residual Gas and Effective Release Energy of a Motorcycle Engine. Energies, 14 (17), 5564. https://doi.org/10.3390/en14175564

[5] Onishi, S., Jo, S. H., Shoda, K., Jo, P. D., Kato, S. (1979). Active Thermo-Atmosphere Combustion (ATAC) – A New Combustion Process for Internal Combustion Engines. SAE Technical Paper Series. https://doi.org/10.4271/790501

[6] Noguchi, M., Tanaka, Y., Tanaka, T., Takeuchi, Y. (1979). A Study on Gasoline Engine Combustion by Observation of Intermediate Reactive Products during Combustion. SAE Technical Paper Series. https://doi.org/10.4271/790840

[7] Wang, Z., Shuai, S., Wang, J., Tian, G. (2006). A computational study of direct injection gasoline HCCI engine with secondary injection. Fuel, 85 (12-13), 1831–1841. https://doi.org/10.1016/j.fuel.2006.02.013

[8] Cinar, C., Can, Ö., Sahin, F., Yucesu, H. S. (2010). Effects of premixed diethyl ether (DEE) on combustion and exhaust emissions in a HCCI-DI diesel engine. Applied Thermal Engineering, 30 (4), 360–365. https://doi.org/10.1016/j.applthermaleng.2009.09.016

[9] Curran, H. J., Fischer, S. L., Dryer, F. L. (2000). The reaction kinetics of dimethyl ether. II: Low-temperature oxidation in flow reactors. International Journal of Chemical Kinetics, 32 (12), 741–759. https://doi.org/10.1002/1097-4601(2000)32:12<741::aid-kin2>3.0.co;2-9

[10] Maurya, R. K., Agarwal, A. K. (2014). Experimental investigations of performance, combustion and emission characteristics of ethanol and methanol fueled HCCI engine. Fuel Processing Technology, 126, 30–48. https://doi.org/10.1016/j.fuproc.2014.03.031

[11] Theinnoi, K., Suksompong, P., Temwutthikun, W. (2017). Engine Performance of Dual Fuel Operation with In-cylinder Injected Diesel Fuels and In-Port Injected DME. Energy Procedia, 142, 461–467. https://doi.org/10.1016/j.egypro.2017.12.072

[12] Khoa, N. X., Lim, O. (2022). Influence of Combustion Duration on the Performance and Emission Characteristics of a Spark-Ignition Engine Fueled with Pure Methanol and Ethanol. ACS Omega, 7 (17), 14505–14515. https://doi.org/10.1021/acsomega.1c05759

[13] Gan, S., Ng, H. K., Pang, K. M. (2011). Homogeneous Charge Compression Ignition (HCCI) combustion: Implementation and effects on pollutants in direct injection diesel engines. Applied Energy, 88 (3), 559–567. https://doi.org/10.1016/j.apenergy.2010.09.005

[14] Rabeti, M., Ranjbar, A. A., Jahanian, O., Safieddin Ardebili, S. M., Solmaz, H. (2021). Investigation of important semi-empirical heat transfer models for a natural gas-fueled HCCI engine. Energy Reports, 7, 8652–8666. https://doi.org/10.1016/j.egyr.2021.11.011

[15] Zapata-Mina, J., Safieddin Ardebili, S. M., Restrepo, A., Solmaz, H., Calam, A., Can, Ö. (2022). Exergy analysis in a HCCI engine operated with diethyl ether-fusel oil blends. Case Studies in Thermal Engineering, 32, 101899. https://doi.org/10.1016/j.csite.2022.101899

[16] Fernie, O., Megaritis, T., Ganippa, L., Tingas, E.-Al. (2022). Numerical Analysis of Zero-Carbon Hcci Engine Fuelled with Steam Diluted H_2/H_2O_2 Blends. SSRN Electronic Journal. https://doi.org/10.2139/ssrn.4100930

[17] Khoa, N. X., Nghia, N. T., Quan, V. H., Ngoc, N. A. (2023). The Effects of EGR and Oxygen Content on the GCI Engine Performance Under Two-Injection Modes and Fueled Biodiesel Blends. Arabian Journal for Science and Engineering. https://doi.org/10.1007/s13369-023-08477-2

[18] Shudo, T., Shima, Y., Fujii, T. (2009). Production of dimethyl ether and hydrogen by methanol reforming for an HCCI engine system with waste heat recovery – Continuous control of fuel ignitability and utilization of exhaust gas heat. International Journal of Hydrogen Energy, 34 (18), 7638–7647. https://doi.org/10.1016/j.ijhydene.2009.06.077

[19] Khoa, N. X., Lim, O. (2022). A Review of the External and Internal Residual Exhaust Gas in the Internal Combustion Engine. Energies, 15 (3), 1208. https://doi.org/10.3390/en15031208

Received date 15.01.2024
Accepted date 19.03.2024
Published date 29.03.2024

How to cite: *Khuong Thi, H., Tran Lan, A., Nguyen Tuan, N. (2024). Investigate the effect of EGR stratergies on the HCCI engine's performance with n-heptane fuel injection. EUREKA: Physics and Engineering, 2, 94–105. https://doi.org/10.21303/2461-4262.2024.003235*