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Analyzing the S.I. Engine's Changeable Exhaust Valve Timing and Exhaust Gas Recirculation (EGR) System in Comparison

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ABSTRACT

The effectiveness of an exhaust gas recirculation (EGR) system and variable exhaust valve closing (EVC) timings is compared in this study. Diesel-RK software was used on the gasoline engine to conduct an array of simulation evaluations. The findings showed that the EGR system had a significant impact on braking power (BP). At 5000 rpm and 10, 20, and 30% EGR ratios, the BP reduced at rates of 11.8, 23.7, and 35.9%, respectively, in comparison to a Standard engine with 0% EGR setup. When compared to conventional engine configuration (EVOD 232° CA), the reduction rates for variable EVC timing settings are 6, 26, and 38%, respectively. A little difference is observed when comparing EGR ratios at 5000 rpm and 215, 207, and 199° crank angle (CA) exhaust valve opening duration (EVOD). At 6000 rpm and 215, 207, and 199° CA-EVOD, the temperature reduction rates are 3, 4.2, and 5.8% respectively when compared with standard duration. The maximum combustion temperature increased as the engine speed increased, and a significant reduction in temperature occurred as the EGR ratio increased. Nitrogen oxide (NOx) emissions peaked at 2000 rpm and steadily declined as engine speed increased. An EGR system significantly reduces NOx emissions, especially at high ratios, while variable EVOD setups result in a slight reduction; at 7000 rpm and 223.5, 207, 199, and 190° CA durations, the diminution rates are 5, 18.8, 29, and 45% in comparison to the standard engine (232° CA).

1. INTRODUCTION

Exhaust gas recirculation (EGR) is a system used in petrol and diesel engines to decrease the release of nitrogen oxide (NOx). A part of exhaust gas recirculates back to the cylinders after mixes with fresh air. This process reduces the oxygen content in the intake air stream and delivers inert gases to combustion to decrease peak combustion temperatures. EGR is an emissions reduction technique that recirculates a portion of an engine's exhaust gases back into the intake manifold. By reintroducing these inert gases into the combustion chamber, the overall oxygen concentration is reduced, leading to lower peak combustion temperatures. This temperature reduction is crucial, as high combustion temperatures facilitate the formation of nitrogen oxides (NOx), harmful pollutants contributing to air quality issues. By lowering the combustion temperature, EGR effectively diminishes NOx emissions. The EGR system typically comprises an EGR valve that regulates the flow of exhaust gases based on engine load and speed, ensuring optimal performance and emission control [1, 2].

Variable valve timing (VVT) is the method of changing the valves (intake and exhaust) timing which is used in internal combustion to increase the performance and exhaust emissions. There are several techniques in which this can be carried out, ranging from electro-hydraulic to camless systems and mechanical devices. Gradually stringent exhaust emissions rules are causing most engine manufacturers to usage VVT systems. VVT is a technology designed to optimize engine performance, fuel efficiency, and emissions by adjusting the timing of valve operations. In traditional engines, valve timing is fixed, leading to compromises between low-end torque and high-end power. VVT systems overcome this limitation by dynamically altering the timing of the intake and/or exhaust valves during engine operation [3].

The exhaust valve closing timing (EVC) has a highly considerable influence on how much of the exhaust gas is missing inside the cylinder at the beginning of the intake stroke. Thus advanced EVC can improve idle excellence, and reductions the engine's capability to scavenge the residue of exhaust gases particularly at high engine speeds [3].

2. LIERATURE REVIEW

The internal combustion engines (IC Engines) are the most current useful power generating instruments, which are widely used, such as diesel engines, engines gasoline, rocket propulsion, gas turbine. IC Engines are classified into two sets intermittent (recurrent) ignition engines and continuous ignition engines. The recurrent ignition engine is considered as a periodic ignition of the mixture (oxidizer and fuel) and is usually denoted as reciprocating engines such as gasoline engines and diesel engines [4]. The continuous ignition engine is described as a steady flow of the mixture into the engine, thus a constant ignition is preserved inside the engine such as jet engines. The most widespread intermittent ignition engine is the four-stroke spark-ignition SI engine (gasoline engine in North America, and petrol engines in Britain), due to its excellent performance as a prime mover in the transport. Gasoline engines depend on a spark plug to burn the mixture (air and fuel) inside the combustion chamber to rotate the crankshaft. The four strokes of the spark-ignition engine are shown in Figure 1. The stroke of the engine is a term used to describe the distance moves by the piston from the upper point in the cylinder to the lower point in the cylinder. This is double the distance from the crankshaft center to the center of the bearing fastening the connecting rod to the crankshaft. The location where the piston is at the upper position in the cylinder is termed as the top dead center (TDC), while the location where the piston is at the lower position in the cylinder is named as the bottom dead center (BDC) [4].

The four strokes include an intake stroke where the mixture (fuel and air) is drawn into the engine when the piston moves from upper position (TDC) to the lower position (BDC), while the intake valve is kept open. The second stroke is the compression, at which the mixture is compressed when the piston moves upwards from BTC to TDC (reduces mixture volume). The third stroke is the power stroke, at which the mixture is ignited. The expansion of the combustion gas causes to push the piston downward from TDC to BDC (the pressure of the hot exhaust gas rises dramatically and forces the piston downwards). The intake and exhaust valves are kept closed during the compression and power strokes. The fourth stroke is the exhaust stroke, at which the combustion gases are ejected out of the cylinder as a result of the piston moving again upwards from BDC to TDC, the exhaust valve is during the exhaust stroke is opened. It is expected that no energy is produced or used through the intake and exhaust strokes [5].

The intake valve typically opens at 20° crank angle (CA) before the piston arriving the top dead center (bTDC) during the exhaust stroke, and closes at 35° CA after the piston arriving the bottom dead center (aBDC) during the movement of the piston upwards in the compression stroke, intake valve opening duration is about 235° CA. In intake process, the mixture (fuel and air) or fresh air are taken into the chamber very quickly, this is due to the vacuum pressure (less than atmospheric pressure) resulting from the downward movement of the piston in the cylinder, this displacement allows the mixture or the charged air to fill the space created by the piston movement [6]. If the intake valve closes early at BDC, the cylinder receives a lesser amount of the air-fuel mixture than its requirement. Therefore, the intake valve is kept open at the beginning of the compression stroke. At this moment, the mixture pressure come to be nearly equivalent to the atmospheric pressure.

Nitrogen oxides NOx are the common expression for a group of extremely reactive gases, all of which consist of different amount of nitrogen and oxygen including nitrogen dioxide (NO_2) and nitric oxide (NO), NOx affects health and environment such as Ground-level Ozone (Smog), Water Quality Deterioration, Global Warming, Particles, Toxic Chemicals, Acid Rain and Visibility Impairment. The higher temperature and excess air (Nitrogen and oxygen) are the main factors that lead to NOx formation [7].

EGR has been introduced to control NOx emissions from diesel engines effectively which lowers the oxygen concentration in the combustion chamber. Both biodiesel fuel and EGR are employed together to evaluate the engine performance and exhaust emission particularly NOx content. an experimental study was conducted on a Mitsubishi 4D68 four stroke, water cooled DI diesel engine fuelled with neat palm-biodiesel operating with diaphragm EGR Tests were performed under a steady state condition where conventional diesel fuel was used as a baseline fuel. According to the experimental results, diesel engine operating with palm-biodiesel and EGR reduced the brake power output, decreased the engine torque, increased fuel consumption, decreased NOx and absolute slight increment in other emissions include CO_2, CO, and particulate matter [8].

The exhaust valve opens at 35° CA-bBDC during the expansion stroke (power stroke), and it closes at 17° CA after the piston reaching the top dead center (aTDC) at the beginning of the suction stroke, therefore exhaust valve opening duration is about 232° CA. It is essential to open the exhaust valve earlier (before reaches the bottom dead center). This will permit the excess pressure of exhaust gas to escape before the piston reaches the bottom dead center. Figure 2 shows the valves timing diagram of the four-stroke petrol engine [9].

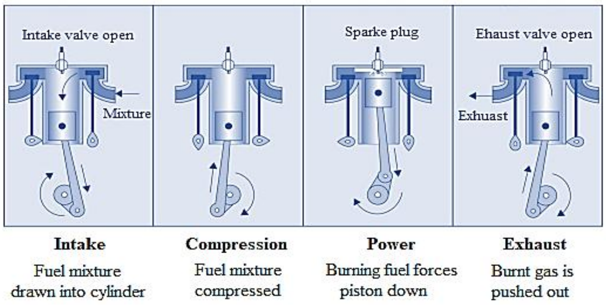

Figure 1. The four strokes of the gasoline engine [https://www.dreamstime.com/type-internal-combustion-engine-called-four-stroke-there-movements-strokes-piston-entire-firing-image158245426]

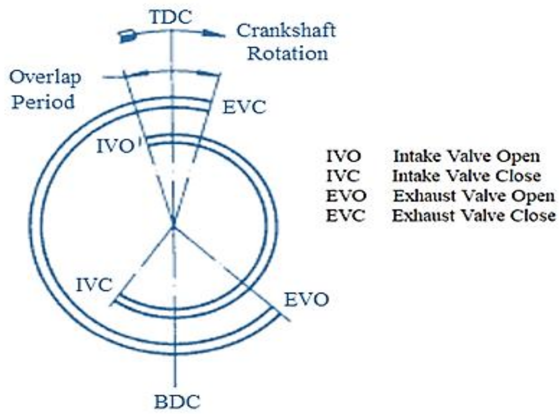

Figure 2. Intake and exhaust valves timing diagram of four stroke petrol engine [https://www.mecholic.com/2018/12/lead-lag-and-overlap-in-valve-timing.html]

EGR is a technique permits a specific amount of the exhaust gases to be recirculated back into the intake manifold to mix with the fresh air (see Figure 2). This method leads to a major decrease in the emissions of NOx, since it decreases the two main factors leading NO_X production excess oxygen and high combustion temperature (some of the oxygen needed for the combustion is exchanged by inactive gas) [10]. There are two kinds of EGR system, internal exhaust gas recirculation (iEGR), where the exhaust gases are reserved in the cylinder by overlapping the timing of exhaust and intake valves, the second type is the external exhaust gas recirculation (eEGR), in which the exhaust gas is recirculated back into the intake manifold by means of using external pipe and a EGR valve [11].

In iEGR a volume of exhaust gas is trapped inside the cylinder from the former engine cycle, this amount is subject to many factors for example the exhaust valve timing and engine speed. Adjusting the trapped gas fraction, usually be dependent on some techniques such as variable valve actuation, two stages cam lift, totally variable valve actuation, and camshaft phasing. External EGR is more effective than iEGR, since the exhaust gases may be cooled before returning back to the cylinders, the volume of recirculated exhaust gases is higher and the gas flow is better measured [12].

By using the EGR system some of the exhaust gas recirculated back into the intake to mix with fresh air, this leads some of the fresh air necessary for the combustion process is replaced by inactive gas (exhaust gas), which leads to minimizing the amount of excess oxygen. Likewise, as the exhaust gases absorb some of the heat produced during combustion, the maximum combustion temperature is also decreased. Too much exhaust gas presented into the intake manifold to mix with fresh air, have an effect on the increase in the emissions of hydrocarbons (HC), particulate matter (PM), and carbon monoxide (CO), due to the incomplete combustion as a result of the shortage of fresh air [13]. If hot EGR is added to the fresh air, the exhaust gas caused to heat and increase the temperature of the charged air in the intake manifold, so thermal efficiency and combustion efficiency are NAimproved. Though, cooled EGR rises inlet charged density and hence increases the engine volumetric efficiency. All studies show that as the ratio of EGR increases (0-21%), the exhaust gas temperature unceasingly reductions, as mentioned before, the most significant reason for thermal NOx formation in the combustion chamber is extremely high temperature. EGR reduces both oxygen content and flame temperature in the engine cylinder [13].

It is possible to differentiate between EGR technology and variable exhaust valve closing timing, in the EGR system the exhaust gas is cooled by ambient temperature and also controlling (EGR valve) the quantity required to be added in the intake manifold according to the engine speed and load. A variable valve timing system has been introduced to improve engine performance, fuel consumption, or exhaust emissions. Previously, the engine's intake and exhaust valves opened for a certain quantity at an exact point in the four-stroke engine, and through a definite duration of time, it was that simple. Today, several IC Engines can change their valves open timing and durations (when they open and for how long). Modern vehicles can change valve duration, valve lift, and valve timing [14].

The delay in fuel injection time results in a decrease in NOx emissions and an increase in brake thermal efficiency. Potentially useful software for simulating retardation conditions is the Diesel-RK engine simulation, and the outcomes of the simulation are compared to experimental values [15].

Rajbir Singh ErHarmandeep Singh ErPrabhjit Singh Er to find the ideal compression ratio for biodiesel made from used cooking oil in various amounts. The process is greatly impacted by the compression ratio, which also offers an exceptional level of motor performance control. For traditional internal combustion engines, the compression ratio is fixed, leading to a compromise between the conflicting needs. A variable compression rate engine may be used to study the compression impact of various biodiesel mixes under various loads [16].

Al-obaidi and Jamshed [17] state that the present study examines numerically the combustion, performance, and emissions parameters of diesel engines powered by different grades of diesel. The Diesel-RK software version 4.3.0.189 is used to simulate the combustion process with a multi-zone model. The Iraqi diesel, EN 590, Heavy diesel, and light diesel are considered. Their energy content, sulfur, cetane number, and other additives are different, hence it's logical to observe different results. The condition of full load point is selected since the air-to-fuel ratio is minimum, hence a better comparison among the fuels is captured [17].

Elkelawy et al. [18] reviewed on, the effect of EGR on (Reactivity Controlled Compression Ignition) RCCI engine operations is reviewed and different techniques to employ EGR. Generally, RCCI engine operation needs EGR support, especially at high engine loads to limit the pressure increase rate, and around 50 percent of EGR may be essential based on fuel used and engine load. The RCCI combustion engine needs a substantially low EGR rate since the rate of burning is regulated by altering the mixture reactivity by employing two fuels having significantly varied reactivity. RCCI operation with cooled EGR yielded lower pressure rise rate, cyclic variation, and NOx emissions but greater (Total hydrocarbons) THC emissions than hot EGR operation. The higher EGR percentage has more benefits on extending maximum load and lowering soot and NOx emissions, whereas the combustion and indicated thermal combustion efficiencies decline to utilize higher EGR percentage. Thus, to accomplish the greater performance and efficient process of combustion, the collaborated regulation is important between EGR rate, premixed ratio, and direct-injection timing.

Ghassembaglou and Torkaman [19] investigated on cold EGR method with variable venturi and turbocharger has very significant effect, simultaneously on the reduction of NOX and grime simultaneously. EGR cooler is one of the most important parts in the cold EGR circuit. optimum design of cooler for working in different percentages of EGR, besides determination of optimum temperature of exhausted gases, efficiency growth, weight reduction, dimension and expenditures reduction, sediment reduction, and optimum performance by using gasoil which has significant amounts of brimstone, is investigated and optimized.

The turbocharged gasoline engine is illustrative of downscaled engines. EGR method merging with a turbocharged engine is the tendency of today's gasoline engine. Once EGR is used, the intake charge involves the fresh air and recirculated exhaust gas. EGR typically acts the fraction of the exhaust gas that mixed with the fresh air. The ratio of EGR is known as the ratio of recirculated exhaust gas in the entire intake mixture, Eqs. (1)-(3) [20].

$$m_{intake} = m_{air} + m_{fuel} + m_{EGR} \tag{1}$$

where,

m_{intake} =mass of mixture in intake manifold
m_{air} =mass of the fresh air
m_{fuel} =mass of fuel
m_{EGR} =mass of recirculated exhaust gas

$$EGR\ (\%) = \frac{m_{EGR}}{m_{Intake}} \times 100\% \tag{2}$$

$$EGR(\%) = \frac{m_{EGR}}{m_{air} + m_{fuel} + m_{EGR}} \times 100\% \tag{3}$$

The EGR ratio is known as the percentage of exhaust gases from the total gas mass induced into the engine. For sample, an EGR rate of 25% means that one-fourth of the gas which passes in the cylinders is really exhausted gas and 75% is fresh air [11, 21].

The volume of EGR used in IC Engines is restricted by many factors, firstly is the requirement for supplying enough fresh air for the complete combustion, secondly, the large amount of EGR causes to decrease the engine efficiency. Additionally, on turbocharged engines, in high load points with the perfect turbocharger, the intake pressure may be greater than the exhaust gas pressure, it is difficult for EGR to mix with fresh air, due to no pressure difference to drive exhaust gas in the intake manifold. To defeat these difficulties, different EGR ratio is used. EGR ratio should correspond to the following operation conditions [11, 21]:

- Great EGR ratio is required throughout cruising and high engine speeds, while combustion temperatures are typically very high.
- Little EGR ratio is desirable for low speed and light load conditions.
- No EGR ratio should require throughout conditions when EGR adversely influences engine operation efficiency or vehicle drivability (engine warm up, ideal, wide open throttle, etc.).

2.1 Problems statements

Several researches are published concerning the influences of eEGR on diesel and gasoline engines performance and characteristics of exhaust emission. On other hand, a few researches executed to investigate the effect of iEGR on engines performance. In this study a comparison investigation will be carried out between the eEGR and iEGR in gasoline engine. by using Diesel –RK software. The targeted goal is to identify the optimum parameters that can improve engine performance and reduce the harmful NOx emissions.

2.2 Objective

The objectives of this research is to study and compare the influence of EGR and different EVOD on gasoline engine. The comparison is achieved on maximum combustion temperature, brake power, brake thermal efficiency, and emissions characteristics for gasoline engine. The engine performance and emissions will be measured and are studied here to highlight the improvements in the gasoline engine field.

3. METHODOLOGY AND MATERIALS

In the present work a professional thermodynamic engine simulation software Diesel-RK is applied to gasoline engine, Diesel-RK is created as a tool to simulate the internal combustion engines with variable engine parameters. The Diesel-RK software is a commercial IC Engines investigation software, extensively used as a manufacturing standard for motor vehicle and powertrain simulation software. The Diesel-RK software development has been at Bauman Moscow State Technical University (MSTU), at the IC Engines department, as a result, a certain implication was done on the mathematical models suitability and algorithms, and the potential has been constantly achieved in contact with IC Engines designers and computational scientists referring to the Internal Combustion Engines manufacturers' orders and needs [https://diesel-rk.bmstu.ru/]. Diesel-RK has been studied to suit multi types of engines of different uses, many computational processes and varieties were existing for the software to suit the demand of internal combustion engines manufacturers in world. Diesel-RK always uses advanced mathematical patterns of combustion process in IC Engines. In recent times, the Diesel-RK subtends the precision of fuel atomizing and injection characteristics, development of fuel sprays dynamics. Accounting the NO emissions is attained by the Zeldovich mechanism. The diesel-RK covers visualization of the fuel spray. This code allows the servant to investigate the animation picture of improvement the sprays of fuel, their interaction with combustion chamber. The code is helpful to design the shape of piston bowl and in producing a proper option of diameter, number of injector nozzles and directions for a specific fuel supply characteristic. The simulation software can be used for modeling the spark ignition petrol engines [17]. The Diesel-RK software model supports the library of different fuels including diesel, petrol and gas. In this software variable valve actuation can be arranged and adjusted independently for every operating mode, also NOx emission calculation is executed by two methods Zeldovich mechanism and Detail Kinetic Mechanism. User can select NO formation model from the list: Diesel-RK tool is easy to use, and allows trainees working without specific exercises [https://diesel-rk.bmstu.ru/].

3.1 Petrol engine specifications

The Nissan VQ20DE is a 1995 cubic centimeter, natural aspirated V6 60° 4-stroke gasoline engine from Nissan VQ-family, the engine code breakdown as follows **VQ-**Engine Family, 20**-**2.0 Liter Displacement, **D-**DOHC (Dual Overhead Camshafts), and **E**-Multi Point Fuel Injection [18]. The VQ20DE has the following specifications (see Table 1 and Table 2).

Table 1. Engine specifications [https://www.engine-specs.net/nissan/vq20de.html]

Engine Type	VQ20DE
Layout	Four stroke, V6, V-line engine design, and 6 cylinders-24 valves
Max. power	103kW at 6400rpm
Fuel type	Gasoline (petrol)
Displacement	2.0L (1995cm^3)
Intake valve duration:	Intake valve open at 15° CA-bTDC
225° CA	Intake valve close at 30° CA-aBDC
Exhaust valve duration:	Exhaust valve open at 35° CA-bBDC
232° CA	Exhaust valve close at 17° CA-aTDC
Cylinder bore×Stroke	76mm×73.3mm
Compression ratio	9.8

Table 2. Petrol fuel specification that used in simulation software

Chemical Composition of the Fuel	C_8H_{18}
Composition mass fraction	C=0.855 and H=0.145
Sulfur fraction in fuel	0.170%
Fuel Calorific value	44 MJ/kg
Heat capacity	255.68 J/K. mole
Octane ratings	91-premium
Fuel Density at 323 K	703 kg/m^3
Specific vaporization heat	230 kJ/kg
Molecular mass of fuel	114

3.2 Research approach

The investigates were achieved by using Diesel-RK software. The results of the simulations are shown in Appendix.

The simulation working conditions on the petrol engine are as follow:

1) At variable speeds (1000, 2000, 3000, 4000, 5000, 6000. 7000, and 8000 rpm).
2) At different EGR ratio (0.00, 0.10, 0.20, 0.30, and 0.40).
3) At different exhaust valve closing timing (see Table 3).

The iEGR was planned to be achieved by applying variable exhaust valve timing to trap a specific amount of exhaust gas within the cylinder to be mixed with the fresh air during intake stroke. Designation of the valve closing timing is referred to crankshaft angle before top dead center during exhaust stroke. The setups of changing the exhaust valve timing, were chosen to be before the designing timing, the purpose of changing the timing to reserve a variable quantities of exhaust gas within the cylinder according to the angle of the crankshaft to study its effect on the engine and carryout the comparisons with external EGR. The inlet valve open timing is held constant at 35° CA-bBDC during expansion stroke.

Table 3. Variable exhaust valve timing and durations

Exhaust Valve Closing Timing (EVCT)	Exhaust Valve Opening Duration (EVOD)
17° CA-after TDC (Standard Engine)	232° CA
8.5° CA-after TDC	223.5° CA
0° CA-TDC	215° CA
8° CA-before TDC	207° CA
16° CA-before TDC	199° CA
25° CA-before TDC	190° CA

The impractical concept (software) is established on the experiments, in engineering fields, the validation of the theory is founded on the actual experiments. There are several prospects to describe the incongruity of the experimental results with the simulation results, some as a result of experimental mistakes and environments and others as a result of the numerical model. The model was a dependable instrument for forecasting emission characteristics and engine performance. Firstly, the Diesel-RK software effectiveness could be examined by compares the engine design output power (103 kW at 6400 rpm), with the average power produced by the software at the same engine speed which is found approximately 98.2 kW. The approximate effectiveness of the software can be calculated by finding the ratio between these values which is equal 95.3%, while the maximum brake power gained by the software is found at 7000 rpm and is equal 105.5 kW.

Engine specifications and fuel characteristics were applied to the Diesel-RK software, all data required to conduct this study are collected in Appendix. The study field includes the engine performance in term of brake power, the characteristics of exhaust emissions was considered (NOx) and finally maximum combustion temperature.

4. RESULTS AND DISCUSSIONS

4.1 Brake power

Figure 3 and Appendix show the both engine conditions which are under study, EGR system (solid curves) and variable exhaust valve closing timings (dashed curves).

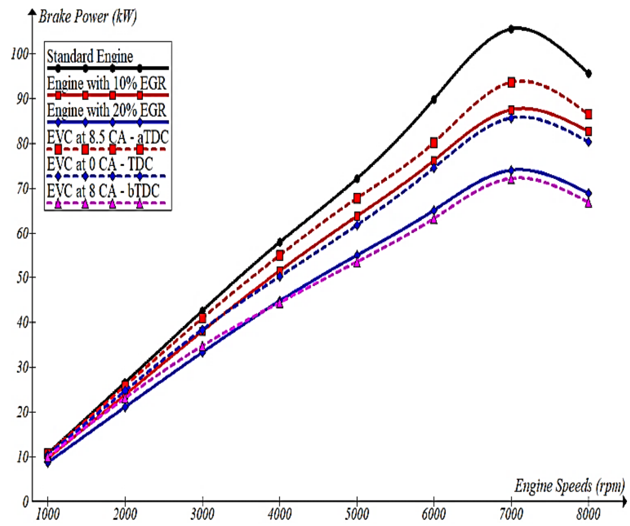

Figure 3. The engine output power versus engine speed

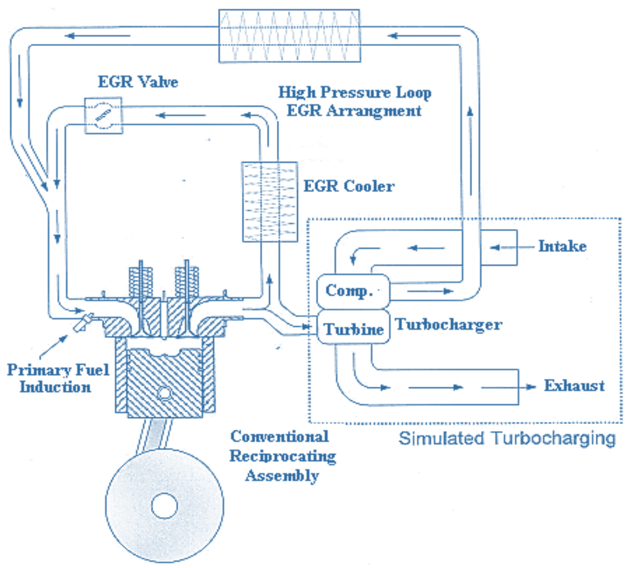

Figure 4. Schematic EGR system [https://www.researchgate.net/publication/283944151_A_Short_Review_of_Treatment_Methods_of_Marine_Diesel_Engine_Exhaust_Gases/figures?lo=1&utm_source=google&utm_medium=organic]

Table 4. The setups which are close to each other regarding brake power (kW)

N (rpm)	0° CA – TDC Setup is Close to 10% EGR		8° CA – bTDC Setup is Close to 20% EGR		16° CA - bTDC Setup is Close to 30% EGR	
	10% EGR	0° CA - TDC	20% EGR	8° CA - bTDC	30% EGR	16° CA - bTDC
1000	9.7	10.4	8.7	9.9	7.7	9.3
2000	23.9	24.7	21.2	23.1	18.4	20.8
3000	38.0	38.4	33.4	34.8	28.6	30.1
4000	51.5	50.3	44.7	44.5	38.2	38.6
5000	63.7	61.8	55.0	53.5	46.3	44.3
6000	76.0	74.5	65.0	63.2	54.1	52.5
7000	87.5	85.5	74.0	72.0	61.0	59.0
8000	82.7	80.4	68.8	66.9	55.3	53.0

Across all setups, the brake power increased as the engine speed augment until reached its extreme value at 7000 rpm, and then it began to decrease versus engine speed. At setup 20% EGR ratio and 2000, 5000, 7000, and 8000 rpm, the brake powers are equal to 21.2, 55, 74 (maximum power), and 67.8 kW respectively, this due to the drop of the cylinder's volumetric efficiency at high engine speeds, there is no enough time for the engine to intake the required charge for complete combustion. In EGR ratio arrangements, the EGR system affect inversely the brake power, therefore if the EGR ratio decreased the brake power decreased. At 5000 rpm and EGR ratios 0, 10, 20, and 30%, the brake powers are equal to 72.2, 63.7, 55.1, and 46.3 kW respectively, the decreasing rate when compared to the engine without EGR (0%) are 11.8, 23.7, and 35.9% respectively. This phenomenon is existing in all engine speeds, and also the peak output power of all setups is found repeatedly at 7000 rpm (see Figure 4 and Appendix). The brake power reductions rate is caused as a result of replacing a definite quantity of fresh air with exhaust gases (inert gases) which in turn decrease the quantity of the fresh air and hence the combustion efficiency. This is one of the drawbacks of the EGR technique since it is usually accompanied by a decrease in the output power, mostly at high EGR ratios. Under all simulated setups (EGR ratios and variable exhaust valve closing timings) there is no effect on brake power at 1000 rpm; all the values are close to the same.

Advanced exhaust valve closing time is used to keeping a part of exhaust gas in the cylinder, may play an important role in reducing NO_X emissions. Figure 4 and Appendix show some engine arrangements with different closing timing and their influences on engine brake power. The arrangements are prepared as follow at 17° (standard closing timing), 8.5° CA-aTDC, 0° CA-TDC, 8°, 16° and 25° CA - bTDC, thus the exhaust valve opening durations are 232°, 223.5°, 215°, 207°, 199°, and 190° CA respectively. If the exhaust valve opening duration is decreased the brake power is decreased, similar to the EGR system, those processes are indicated by dashed curves (see Figure 4 and Appendix). At engine speed 6000 rpm and opening durations 232°, 215°, 199°, and 190° CA, and the brake powers are equal to 89.8, 71.5, 50.5, and 39.4 kW respectively. The lessening rates are 20.5, 43.8, 56.2% respectively when compared to the design duration 232°. The lowest amount of power that generated by the engine is occurred at the minimum exhaust valve opening duration at 190° CA, due to the great amount of exhaust gas that trapped within the cylinder, which leads to a major reduction in the quantity of fresh air.

Some different setups are observed to be close with each other regarding the brake power (see Figure 4). The EVC at 0° CA - TDC system is close to 10% EGR system, EVC at 8° CA - bTDC system is close to 20% EGR system, and EVC at 16° CA - bTDC arrangement is close to 30% EGR system as shown in Table 4.

The setup of 0% EGR ratio and at EVC designing timing showed a straight relationship between engine brake power and charging quantity and quality, the lengthier exhaust valve opening duration increased the charge purity and hence the brake power, particularly at higher engine speeds. The valves (intake and exhaust) overlap, have a major effect on the quality of the in cylinder contents at the beginning of the intake stroke. To increase the brake power at the high engine speeds (greater than 3000 rpm), it is essential to minimize the exhaust gas to be held in the cylinder to lowest the amount of exhaust gas, this permits for maximum quantity of fresh air and fuel to pass in during the intake stroke, this requires EVC timing to be at, or closely after top dead center.

4.2 Maximum in-cylinder temperature

Figure 5 and Appendix show the maximum in-cylinder temperatures of different arrangements, EGR system displayed by solid curves, while the setups of different exhaust gas valve closing timings presented by dashed curves.

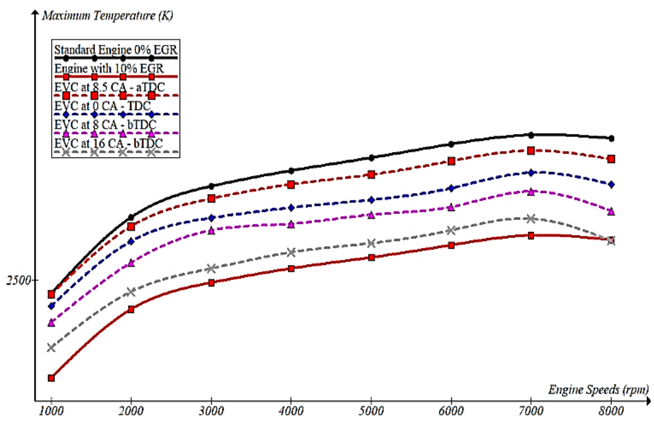

Figure 5. The in-cylinder maximum temperature versus engine speed

The higher temperature is attained by the setup of 0% EGR ratio and at EVC design timing, due to the absent of the exhaust gas in which just fresh air is drawn in the cylinder (see Figure 3 and Appendix). In all arrangements the in-cylinder temperature augmented as the engine speed rises until reached maximum value at 7000 rpm (maximum power), followed (after 7000 rpm) by a slight drop. At 10% EGR ratio setup and speeds 2000, 4000, 7000, and 8000 rpm, the temperatures readings were found 2445, 2520, 2582 and 2574 K respectively, the reduction rate at 8000 rpm compared to the reading at the maximum (7000 rpm) is about 0.30%. The reason is due to the reduction in combustion efficiency as a result of decreasing in the volumetric efficiency which always accompanies with the higher engine speeds.

The variable exhaust valve closing timing setups attained the highest in-cylinder temperatures compared to the EGR systems (see Figure 3 and Appendix) at all engine speeds. Even all the variable EVC timing setups except EVC at 25° CA-bTDC, attained temperatures greater than 10% EGR ratio setup. At 6000 rpm and EVC at 8.5° CA-aTDC, EVC at 0° CA-TDC, EVC at 8° CA-bTDC, and EVC at 16° CA – bTDC setups, the maximum temperatures are equal to 2669.9, 2635.1, and 2591.3 K respectively, which are all greater than the temperature of 10% EGR ratio setup (2564.1 K). Since the combustion temperature inside the cylinder is directly linked to the power output, but there is conflict appears in this issue in some setups, when compared the brake power results to their maximum temperatures. The EVC at 8.5° CA-aTDC setup ranks fourth position (pink dashed curve) when comparing maximum in-cylinder temperatures (see Figure 4 and Appendix), while this setup ranks sixth position (pink dashed curve) when comparing brake powers (see Figure 3 and Appendix), it should be ranked fourth position. Same infliction appears at EVC at 16° CA-aTDC and EVC at 25° CA -aTDC arrangements. This conflict is due to that the exhaust valve close very early resulting in a great amount of extremely hot exhaust gases that's are trapped in the cylinder. This condition leads to pull a minor amount of mixture (air and fuel) during the intake stroke. In additional, the hot exhaust gas expands in the cylinder when the piston moves downwards avoiding the mixture entering the cylinder smoothly and easily.

4.3 Nitrogen oxides NO_X

NOx pollutant emissions are extremely dependent on the air/fuel ratio. Any method used to decrease the in-cylinder peak temperature and the amount of fresh air will lessen the formation of nitrogen oxides. Now the rich mixture (excess amount of fuel) generates less amount of NO_X compared to the lean mixture (excess amount of air). The EGR is introduced to recirculate 10% to 30% of the exhaust gases back into the intake manifold.

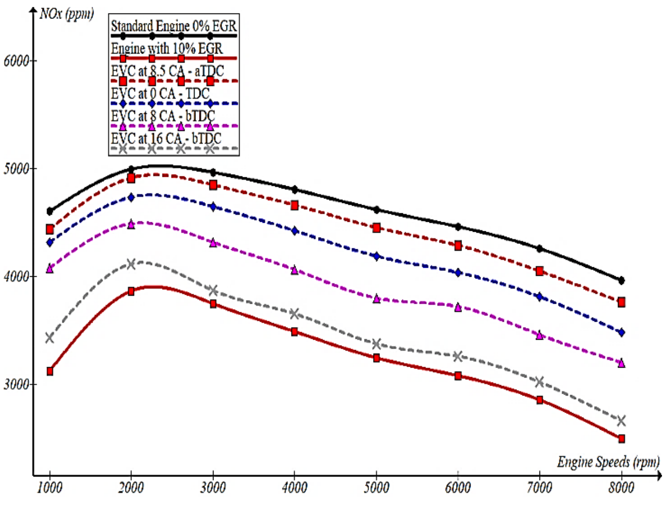

Figure 6. Nitrogen oxides versus engine speeds

Figure 6 and Appendix illustrate the nitrogen oxides emissions of different simulation systems, EGR system showed by solid curves, while the exhaust valve closing timing setups shown by dashed curves. The uppermost emissions happened with the standard engine (0% EGR and EVC at 17° CA-aTDC) at all engine speeds when compared with other systems (see Figure 5 and Appendix), due to the high in-cylinder temperature and lean mixture (excess air). Figure 5 and Appendix show that at all systems the NO_X formats at high rate mostly at low engine speeds (less than 2000 rpm), greater than 2000 rpm the formation starts to decrease. At system 10% EGR ratio and engine speeds 1000, 2000, 3000, 6000, and 8000 rpm, the emissions of NO_X are equal 2845, 3864, 3746, 3077, and 2493 ppm respectively. The reduction is due to there is no enough time for NO_X formation through high engine speeds. A significant reduction of NO_X emissions which achieved by high EGR rates setups (20 and 30%), when compared with other setups (see Figure 5 and Appendix). The NO_X emissions at 5000 rpm and standard engine, EVC at 8.5° CA-aTDC, 10% EGR ratio, EVC at 16° CA-bTDC, 20%, 30% EGR are equal to 4618, 4451, 3244, 1520, and 922 ppm respectively. The reduction percentages when compared with standard engine are 3.6, 29.8, 67, and 80% respectively, this is due to the presence of exhaust gases during combustion (decreasing combustion temperature because of the addition of exhaust gas to the inlet mixture which tend to reduce the combustion temperature). The different EVC timing systems are seen less efficient in reducing NO_X emissions compared with the EGR ratios (see Figure 5 and Appendix), shown as dashed curves. One of the reasons is the exhaust gases temperature. In EGR system, the exhaust gas is cooled before mixing with the mixture. Whereas in variable EVC timing systems the exhaust gases of high temperature are mixed with the air-fuel mixture inside the cylinder.

5. CONCLUSIONS

Founded on simulation software results, different EVOD and EGR system were associated, the analysis of the data showed the EGR system and different EVOD are found to decrease the engine power output considerably depending on the EGR percentage and exhaust valve opening duration. The results showed that the extreme combustion temperature declined meaningfully in the presence of the EGR system, and as the EGR rate is augmented the combustion temperatures are decreased, while in the different EVOD the combustion temperatures decreased at a lower rate compared to the EGR system. The NO_X emission reduced when the EGR systems are introduced to the intake system. The reduction of NO_X emissions be contingent mostly on the amount of EGR, and as the EGR ratio is increased the NO_X emissions are decreased. While a minor reduction of NO_X emissions is achieved by different EVOD when compared to the EGR system.

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NOMENCLATURE

aBDC	after bottom dead center
aTDC	after top dead center
bBDC	before bottom dead center
bTDC	before top dead center
BDC	bottom dead center
BP	brake power (kW)
CA	crank angle (° degree)
CI engine	compression ignition engine
CO	carbon monoxide (ppm)
EGR	exhaust gas recirculation
eEGR	external exhaust gas recirculation
EVC	exhaust valve close
EVOD	exhaust valve opening duration
HC	hydrocarbon
IC Engine	internal combustion engine
iEGR	internal exhaust gas recirculation
MSTU	Moscow State Technical University

NO	nitric oxide (ppm)	rpm	revolution per minute
NO_2	nitrogen dioxide (ppm)	SI engine	spark ignition engine
NO_X	nitrogen oxides (ppm)	TDC	top dead center
ppm	parts per million	VVT	variable valve timing
PM	particular matters		

APPENDIX

Engine with EGR System				
Engine Condition	**Engine Speed (rpm)**	**Brake Power (kW)**	**Maximum Temp. (K)**	**NO_X Emissions (ppm)**
Standard Engine Without EGR and EVC Timing at 17° CA-aTDC EVO Duration (232° CA)	1000	10.7	2474	4443
	2000	26.6	2615	4995
	3000	42.6	2673	4965
	4000	57.9	2702	4805
	5000	72.2	2726	4618
	6000	89.8	2751	4461
	7000	105.5	2768	4303
	8000	95.6	2762	3965
10% EGR	1000	9.7	2318	2845
	2000	23.9	2445	3864
	3000	38.0	2495	3746
	4000	51.5	2521	3493
	5000	63.7	2541	3244
	6000	76.0	2564	3077
	7000	87.5	2582	2854
	8000	82.7	2574	2845
20% EGR	1000	8.7	2159	1298
	2000	21.2	2268	1888
	3000	33.4	2310	1853
	4000	44.8	2333	1701
	5000	55.0	2354	1520
	6000	65.0	2373	1380
	7000	74.0	2388	1200
	8000	68.8	2379	991
30% EGR	1000	7.7	1998	452
	2000	18.4	2088	1214
	3000	28.7	2123	1230
	4000	38.2	2144	1051
	5000	46.3	2159	922
	6000	54.1	2178	783
	7000	61.0	2194	606
	8000	55.3	2185	440
40% EGR	1000	6.6	1832	130
	2000	15.6	1907	204
	3000	24.1	1937	219
	4000	31.5	1953	221
	5000	37.6	1965	218
	6000	43.2	1985	239
	7000	47.8	1998	248
	8000	41.5	1988	191
Variable Exhaust Valve Closing Timing				
EVC Timing at 8.5° CA-aTDC EVO Duration (223.5° CA)	1000	10.7	2473	4438
	2000	25.9	2598	4917
	3000	41.0	2650	4849
	4000	55.0	2677	4664
	5000	67.8	2695	4451
	6000	80.1	2711	4286
	7000	93.7	2739	4159
	8000	86.6	2724	3763
EVC Timing at 0º CA-TDC EVO Duration (215° CA)	1000	10.4	2451	4318
	2000	24.7	2571	4733
	3000	38.4	2614	4647
	4000	50.3	2633	4429

	5000	61.8	2648	4191
	6000	74.5	2669	4034
	7000	85.5	2699	3981
	8000	80.4	2676	3485
EVC Timing at 8° CA-bTDC EVO Duration (207° CA)	1000	9.9	2422	4079
	2000	23.1	2532	4491
	3000	34.8	2591	4317
	4000	44.5	2604	4069
	5000	53.5	2621	3801
	6000	63.1	2635	3717
	7000	72.0	2664	3734
	8000	66.9	2628	3199
EVC Timing at 16° CA-bTDC EVO Duration (199° CA)	1000	9.3	2375	3433
	2000	20.7	2478	4115
	3000	30.1	2521	3869
	4000	38.6	2551	3657
	5000	44.3	2567	3372
	6000	52.5	2591	3256
	7000	59.0	2614	3021
	8000	53.0	2572	2661
EVC Timing at 25° CA-bTDC EVO Duration (190° CA)	1000	8.4	2322	2731
	2000	18.3	2418	3242
	3000	26.1	2473	3149
	4000	32.4	2488	2998
	5000	39.8	2500	2744
	6000	46.3	2530	2590
	7000	51.9	2554	2340
	8000	42.4	2523	1992