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The Effect of Variable Valve Timing on SIE Performance and Emissions

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Abstract— The present work deals with the overlap period variation effects; by changing the intake and exhausts valves open and close timings. The study focuses on the impact of this timing variation on engine performance and exhaust gases emission. The experimental work was conducted on the experimental single cylinder research engine SI engine type "Prodit". The valve timings variation carried out by changing the clearance distance between the rocker arm and valve stem. Three valve overlaps (104˚, 108˚, and 112˚) investigated theoretically and experimentally.

From the experimental results, the reduction in overlap period (overlap=104˚) with the compression ratio (CR=9) shows a better compromises between the overall performance and exhaust emissions. The results indicated increments in the volumetric efficiency (12.58%) and the brake thermal efficiency by (5.65%). The fuel consumption reduced by (3%) and exhaust emission by reduced about 4.45% for the HC and by 20.19% for CO.

Index Terms— Variable valve timing, overlap period, spark ignition engine, performance, emissions.

1 INTRODUCTION

The vast majorities of automobiles and trucks use engines operate either SI engine or on the CI engine. The valves in these engines have a mechanically actuated valve motion fixed with respect to the crankshaft position for all operating conditions. These valve motions such as (valve lift profile, valve event, and open duration) determined during the engine design stage by fixing the cam profile and its position. Valve timings designed, in general, for average operation and had no method of varying either timing or lift. This procedure gave adequate operation at medium speeds and loads but was not optimized for high or for low and idle speeds [1]-[2].

In the modern world, one of the largest concerns is the ever depleting supply of oil. The automotive industry especially impacted, in 2011, the world consumed 85 million barrels of oil a day [3]. The oil still a significant source of energy as well it is into the future. In spite of the world consumption of fossil fuels continue to grow to 118 million barrels per day by 2030 [4]. Besides, the released emission from internal combustion engines polluting the environment. The global demand for cars is soaring-one forecast has the number of worldwide cars increasing five-fold by 2050 to 2.9 billion [5]. The control of Greenhouse gas emission has begun to add the numerous constraints that vehicle manufacturers to satisfy. The engine fuel consumption reduction becomes a primary requirement for manufacturers and designers, as well as it must meet the current and future emission legislations naturally [6].

The Variable Valve Timing (VVT) is a remarkable technology to attain improvement of fuel economy, output power, and reducing emissions by optimization of intake and exhaust valves. Various mechanisms introduced into the markets from the second half of the 1980s and adopted for many automobile manufacturers' engines [7]-[8]. Since the late 1990's, designers of engines with (VVT) have used several methods to obtain this goal. Most of these earlier systems worked only on valves and had no lift variation. Most modern systems can now control timing and lift. One of the developed new systems of (VVT) has no camshaft and uses an electrically controlled actuator to adjust the motion of the valve with respect to the operating condition [1].

The addition of Variable Valve Timing (VVT) technology makes it possible to control the valve lift, duration, and valve timing. Various types of VVT have been proposed and designed to enhance the overall engine performance and to get full benefits from this technology. The products of some of these mechanisms have shown primary benefits in enhancing engine performance [9].

At present, the simplest, cheapest and most common mechanism used is Cam-Phasing (VVT). It varies the valve timing by shifting the phase angle of the camshaft about the crankshaft; it can only control the valve timing and cannot control the valve lift [10]. The intake and exhaust camshafts can vary simultaneously with notable improvement in fuel economy and emission reduction [11]. When every individual valve has its timing system, the largest degree of variability in the valve train is achieved. The valves enabled to open and close as and when required [10]. The actuating of each valve individually exhibits a better precise control than the valve timing, but the cost is often an obstacle [10].

Moriya [12], investigated the requirements for higher horsepower and torque as well as lower fuel consumption and emission. The research team developed the "Intelligent Variable Valve Timing" (VVT-I) and utilized to a 3-liter, in-line 6 engine. The (VVT-I) pulley installed at the front of the intake camshaft provides a timing difference between the camshaft and the crankshaft via a hydraulic actuator. This hydraulic actuator is activated by the oil control valve (denoted as OCV). This valve controls oil pressure supplied to the (VVT-I) pulley taking the commands from the electronic control unit (ECU). It continuously controls oil feeding and draining between the (VVT-I) pulley and (OCV) to achieve continual phasing.

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Whereas (ECU) computes optimal valve timing based on the engine operating condition and drives the (OCV). The entire system is capable of phasing the intake camshaft within 60° crank angles (CA). In this system, only the valve timing is varied without changing valve lift profile. Their results indicated that more than 10% improvements in low to mid-speed engine torque. Approximately 6% increase in fuel economy, in addition to, simultaneous emission reductions in NOx by 40% and HC by (10%).

Li [13] studied the influence of the intake valve closing (IVC) timing on the performance of the high-speed, spark-ignition engine. The investigators developed and utilized an electrically controlled Variable Valve Timing (VVT) system. The system based on the variable working position belt extender with successfully setting up in a four-cylinder, 1.6 liters, and Double Overhead Cam (DOHC) 5-valve. The SI engine supplied with a fixed belt extender that converted into a variable working position belt extender. Their investigation showed that an 11% increment in the torque achieved at low speed. More than 7% increment in the torque at high-speed. The fuel economy improvement made in the most of the speed range, especially at high speed. The CO emission reduced significantly, and the HC emission reduced only at high speed.

Schwarzenthal [14] developed another cam profile switching System. The system referred to as "Vario Cam Plus", which varies the timing as well as the lifting of intake valves. During the full load operation, both tappets are locked with each other by a hydraulically activated pin the "Fast" cam lobes drive the valve, supplying high lift and long duration opening. Otherwise, the valve actuates by the "slow" cam lobe via the inner tappet. The main study find is that the fuel saving at idle is up to 13% and an improvement of 45% in HC emission. In additional, the fuel economy is improved by 3.3% at partial load operation.

Hara [15] investigated the performance benefits of Variable Timing Control system (VTC) and Variable Valve Event and Lift (VEL) actuation system. These systems are an effective means of making engines more environmentally friendly. The team developed a design of a continuously variable valve actuated the device, capable of continuously varying the intake valve lift and event timing over the various engine operating conditions. The authors reported the performance benefits provided by (VEL) when deployed in an actual vehicle that include fuel saving of approximately 10%. A reduction of approximately 50% in the exhausted HC emission achieved at cold starting. An increase of approximately 10% in engine output rose.

Özdalyan [16] Designed and constructed an electro-mechanical valve (EMVA) system for the intake valve of a four strokes, single cylinder, overhead valve and spark ignition (SI) engine. The investigations test the effect on the engine with and without (EMVA) to observe the impacts of the (EMVA) on engine performance and emissions under full load at different speeds. They Improved power, engine torque and brake specific fuel consumption (BSFC) by 9.6%, 7.7%, and 29% respectively. A 66% decrease in CO emission. However, hydrocarbons HC and NOx emissions increased by 12% and 13% respectively.

2 EXPERIMENTAL SETUP

A single cylinder, variable compression ratio engine type (GR 306/000/037A) used in the experiments. The engine made by the Prodit Company, (Italy). The engine is 4 strokes; has popped overhead valve and connected to a hydraulic dynamometer. The engine adapted to run on an SI engine. The engine compression ratio varies from (4 to 17.5). The engine mounted on the main frame made of stainless steel sturdy that is designed to contain, support all the apparatuses, and to carry out all experimental tests. The system is mounted on a wooden base to increase safety and to reduce vibrations and noise. The wood works as a damper specifically made for this purpose. Table 1 presents the specifications of the engine while Fig. 1 shows a schematic of the tests used rig.

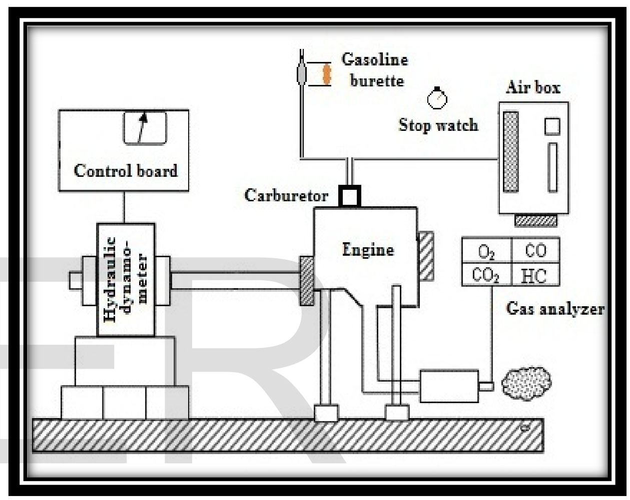

Fig. 1. Schematic of the experimental apparatus.

The hydraulic brake (or hydraulic dynamometer) used to measure the torque output from the engine by friction fluid. Water used as the friction fluid. To measure dynamometer load a hanging weight balance is used to overcome the tilt in the torque arm to keep it in the horizontal position, the following equation used to calculate the torque:

$$T_b = m*g*r \tag{1}$$

Where, m= torque weight balance reading (constant =5kg).

g= gravity acceleration ($10 \frac{m}{sec^2}$).

r= is the perpendicular distance between the weight centre and dynamometer axis (0.15 m).

Engine speed measured by an optical encoder locked to the crankshaft. When the engine is running a train of electric pulses is generated by this transducer that his frequency is proportional to the crankshaft speed. This signal conveyed through electric cables to the rpm display placed on the control board.

The compression ratio measured by the cylinder-head position transducer whose output signal is proportional to the applied extension of the transducer itself. GEFRAN Company manufactured this transducer; model PZ-12-A-025. Through this transducer, it is possible to know the position of the cylinder head and, therefore, to know the compression set. The output signal conveyed through electric cables to the compression ratio display located on the control board to visualize this value.

The exhaust gas analyzer type (mod 488 Italy) used to analyze the emissions from the exhaust. The analyzer detects the CO-CO2- HC- O2 contents. The exhaust gases drag from the engine exhaust pipe using a probe. The water moisture separated from the gases through the condensate filter and then these gases conveyed into the measuring cell. A transmitter generated a ray of infrared light sent to optical filters onto the measuring elements. The measuring cell gases absorb the ray of light at different wavelengths, according to their concentration.

The variable molecular compositions of H2 – N2 –O2 cases, prevent the absorption of the emitted ray. This process prevents the concentration measuring through the infrared system. The CO-CO2-HC gases due to their molecular composition absorb the infrared rays at specific wavelengths (absorption spectrum). The analyzer provided with a sort chemical sensor by which the oxygen rate (O2) measured [17].

TABLE 1
ENGINE SPECIFICATIONS

Technical Characteristics		Technical Characteristics	
Manufacturer	PRODIT S.A.S.	Cycle	Otto or Diesel, four strokes
Number of cylinders	1 vertical	Diameter	90mm
Stroke	85mm	Swept volume	541cm3
Compression ratio	4-17.5	Max. power	4 kW at 2800 rpm
Max. torque	28 Nm at 1600 rpm	No load speed range	500-3600 rpm (Otto cycle)
Engine cooling	Water cooled	Load speed range	1200-3600 rpm (Otto cycle)
Intake star	54^o before T.D.C	Intake end	22^o after B.D.C
Exhaust start	22^o before B.D.C	Exhaust end	54^o after T.D.C
Fixed spark advance	10^o before T.D.C (spark ignition)		

Brake Power B.p (kW):

$$\mathrm{B.P} = (2 * \pi * \mathrm{N} * T\_b)/(60 * 1000) \quad (2)$$

Where:
T_b: brake torque [N.m]
N: engine speed [rpm]
Actual Air Supply $\dot{m_a})_{act.}$ (kg /sec):

$$(m_a)^{\cdot}{}_{act} = (Q * \rho_{air})/3600 \quad (3)$$

Where:
Q = air flow rate [m^3/h]
ρ_{air}: Specific mass of the air [kg/m^3]

$$\rho\_air = P_{atm}/(R.T_{atm})$$

P_{atm} = Atmospheric Pressure [kPa]
R: Gas Constant
R= 0.287[kJ/ (kg.K)]
T_{atm} = Atmospheric Temprture [K]

Fuel Consumption $\dot{m}_f$ (kg/sec):

$$(m_f)^{\cdot} = (V_f * \rho_f)/t_f \quad (4)$$

V_f: Volume of the Burette
ρ_f = Specific Mass of the Fuel [kg/L]
t_f = Time Necessary to Empty the Burette [sec]

Air fuel ratio (A/F):

$$A / (F = (\text{mass of actual air supply })/(\text{mass of fuel consumption})) = (m_a)^{\cdot}/(m_f)^{\cdot} \quad (5)$$

The theoretical air consumption $(\dot{m_a})_{theo.}$ (kg/sec):

$$(m_a)^{\cdot}{}_{theo.} = (Vs * K) * N/(2 * 60) * \rho_{air} \quad (6)$$

Vs = swept volume [m^3]
Vs = $\pi/4\ b^2\ s$
K = Number of engine cylinders = 1
N= Engine speed [rpm]
b= cylinder bore [m]
s= Stroke length

Volumetric efficiency ζ_v(%):

$$\zeta_v = (mass\ of\ actual\ air\ supply)/(mass\ of\ \text{theoretical air consumption}) * 100$$

$$\zeta_v = (m_a)^{\cdot}{}_{act}/(m_a)^{\cdot}{}_{theo.} * 100 \quad (7)$$

Break specific fuel consumption bsfc (kg/kW.hr):

$$\text{bsfc} = (mass\ of\ fuel\ consumption)/(brake\ power) * 3600$$

$$\text{bsfc} = (m_f)^{\cdot}/(B.P) * 3600 \quad (8)$$

Brake thermal efficiency ζ_{bth} (%):

$$\zeta_{bth} = (brake\ power)/(mass\ of\ fuel\ consumption * L.C.V) * 100$$

$$\zeta_{bth} = (B.P)/((m_f)^{\cdot} * L.C.V) * 100 \quad (9)$$

Where:

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L.C.V=Lower calorific value [kJ/kg].

2.1 Engine Preparation and Tests Procedure:

This research focused on the variation of intake and exhaust valve timing to investigate their effects on engine performance. Therefore, the clearance distance between the rocker arm and valve stem was changed. This clearance distance can be adjusted using a Feeler Gauges (a tool used to measure gap widths) a. This method is used to give two different valves timings in addition to the conventional timing, and then to make a comparison with the improvement over the conventional timing of the valves. Table (3.1) shows the three different cases.

TABLE 2
INTAKE AND EXHAUST VALVE TIMING DEGREES FOR EACH CASE

No. of Cases	*Case Type*	*Inlet valve*		*Exhaust Valve*		*Overlap period*
1	*Decreasing clearance distance*	52° *BTDC*	20° *ABDC*	20° *BBDC*	52° *ATDC*	104°
2	*Conventional clearance distance*	54° *BTDC*	22° *ABDC*	22° *BBDC*	54° *ATDC*	108°
3	*Increasing clearance distance*	56° *BTDC*	24° *ABDC*	24° *BBDC*	56° *ATDC*	112°

Each valve timing was tested in a constant load = 5 kg by using the hydraulic dynamometer. The speeds of 1400, 1800, 2000 and 2200 rpm examined under the same load. Two different compression ratios (7 and 9) for each speed were applied. These tests performed for one type of fuel Gasoline used for each case.

Firstly, the standard engine (conventional engine) was fully instrumented, and all sensors connected. The engine started and warmed up (to reach the study state) by running it at an average speed for nearly 15 min. The tests were performed for four chosen speeds ranging from 1400 to 2200 rpm with a constant load =5 kg, whereas the optimal running condition of the engine is between 1000 and 3000 rpm.

The required engine load was applied and using the hydraulic dynamometer and set the throttle control (for adjusting the amount air/ fuel mixture needed) to the usable value braking. To maintain the selected speed sometimes has to elapse before the engine reaches the required stable speed at the same time the gas analyzer prepared for measurement.

The data of experiment recorded, and the ambient temperature of the Lab measured. Beside that the data necessary for performing engine performance calculations recorded from the control board while the exhaust gas analyzer collects data of CO, CO2, O2, and HC. Each test repeated three to four times, to ensure the repeatability of experimental results. The average value of repeated tests adopted in the analysis. These procedures repeated to cover all cases; all chosen speeds range at the specified compression ratio and load.

3 RESULTS and DISCUSSIONS

Figs. 2 & 3 show the effect of VVT on the volumetric efficiency for variable engine speeds. At (overlap= 104˚) there is a decrease in volumetric efficiency with the engine speed increase. The same trend presents at both studied compression ratios (CR=7 and 9). This decrement main cause is the reduction of overlap period than the conventional period. As speed increases, a shorter time required to open and close the valves resulting in lower charge entered and the friction loss of air flow also increases with increasing speed.

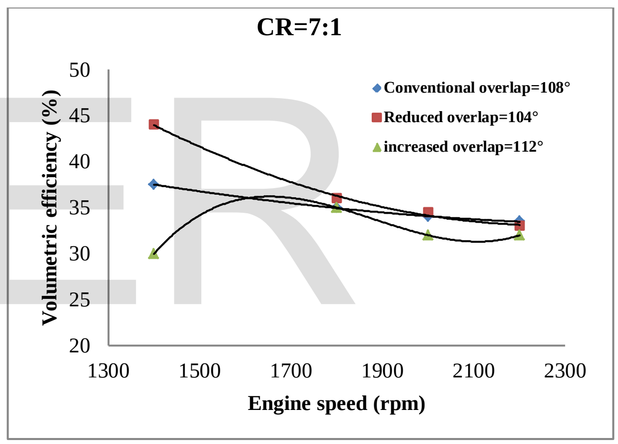

Fig. 2. The effect of VVT on volumetric efficiency for variable engine speeds at CR=7:1

Volumetric efficiency degraded with increased valve overlap=112° at CR=9:1 compared to the other overlaps. At CR=7:1 it was near conventional overlap efficiencies except for 1400 rpm. Increased valve overlap increases the time required for stilling exhaust gas inside the combustion chamber on the account of the new charge.

Figs. 4 & 5 reveal the variation of the brake specific fuel consumption (bsfc) at the studied compression ratios (CR=7 and 9) for the tested valve overlaps and variable speeds. The recent curves took the same trends of volumetric efficiency curves in the figures 2 & 3compared to the same engine speed. The volumetric efficiency decrement results in the reduction of the bsfc due to Internal Exhaust Gas Recirculation (IGER). IGER generated by employing Early Intake Valve Open (EIVO) and Early Exhaust Valve Close (EEVC) as reference [19] demonstrated. Another reason for the bsfc reduction is the pumping losses reduction that leads to decrease the power required to pump the charge into the cylinder during the intake stroke.

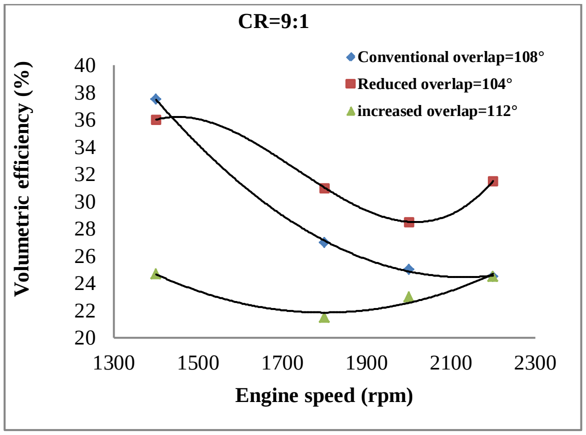

Fig. 3. The impact of VVT on volumetric efficiency for variable engine speeds at CR=9:1

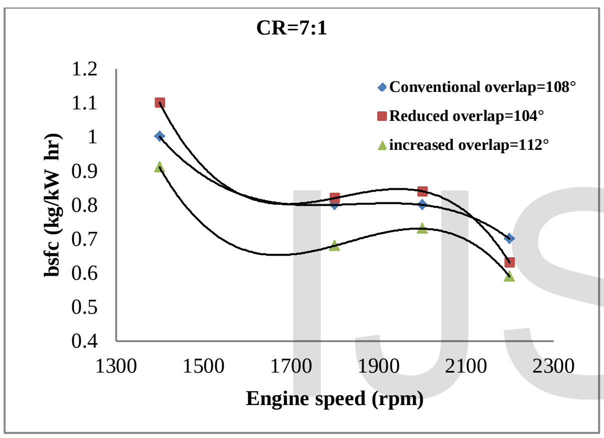

Fig. 4. The effect of VVT on bsfc for variable engine speeds at CR=7:1

By increasing valve overlap, the engine breathes more easily during the overlap period. In addition to the inlet valve remains open during a part of the compression stroke for a little longer, so a fraction of the charge is expelled back into the intake manifold. The pressure of the entrapped charge is little more than the atmospheric pressure. During the subsequent induction stroke, the entrapped charge gets readmitted at a pressure above that of the (air-fuel) mixture in the conventional engine. The suction pressure line partially deviates from the atmospheric line. Thus, the negative area is reduced, which result in reduced pumping losses to increasing in fuel economy at part load operation condition.

Figs. 6 & 7 depict the variation of the brake thermal efficiency of CR=7 & 9 respectively with variable engine speeds for the studied engine valves overlap. The brake thermal efficiency increase as the engine speed increase. Also, the curves indicate an increase in brake thermal efficiency with increase in compression ratio.

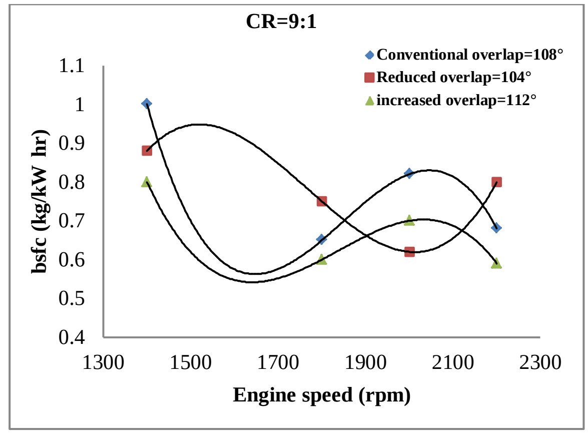

Fig. 5. The impact of VVT on bsfc for variable engine speeds at CR=9:1

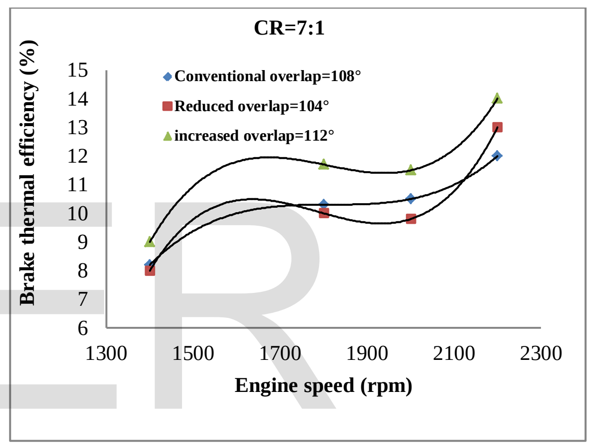

Fig. 6. The effect of VVT on brake thermal efficiency for variable engine speeds at CR=7:1.

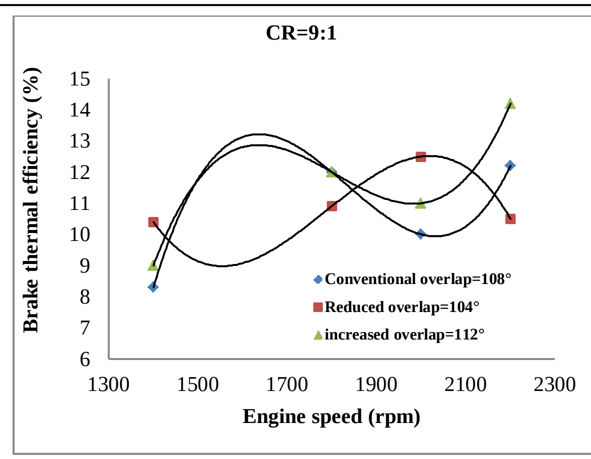

Fig. 7. The impact of VVT on brake thermal efficiency for variable engine speeds at CR=9:1.

The percentage increasing of brake thermal efficiency at CR=9 than CR=7 by (4.034 % for Conv. overlap=108˚), (7.229 % for

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Reduce. overlap104˚) and (7.492 % for Increase. overlap112˚). The tests results clearly indicate the effect of compression ratio on the brake thermal efficiency. CR increases the pressure and temperature inside the combustion chamber to better combustion by improving charge homogenization prior to ignition.

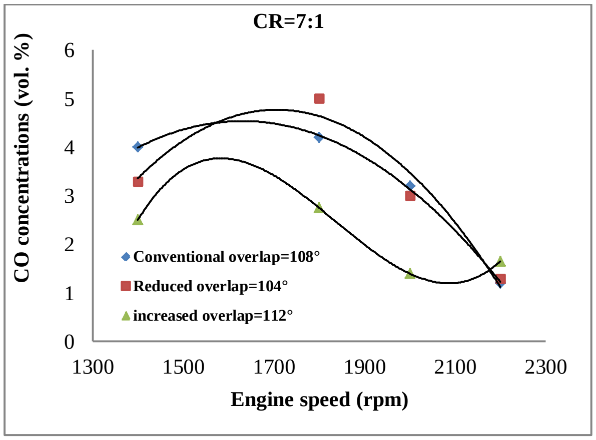

Fig. 8. The effect of VVT on CO for variable engine speeds at CR=7:1.

Figs. 8 & 9 reveal the carbon monoxide concentrations variations with the engine speed at the studied compression ratios and valve overlaps. There is a decrease in the emission of CO due to the volumetric efficiency increasing with reduced overlap than the conventional ones. The carbon monoxide arises mainly due to incomplete combustion, so, it is a measure of the combustion efficiency. There are increments in CO concentrations of the reduced overlap timing at higher engine speeds. The combustion process is relatively quick at high engine speeds; the combustion completion within the power stroke becomes harder. Therefore, the combustion might continue during the exhaust stroke and exhaust pipe.

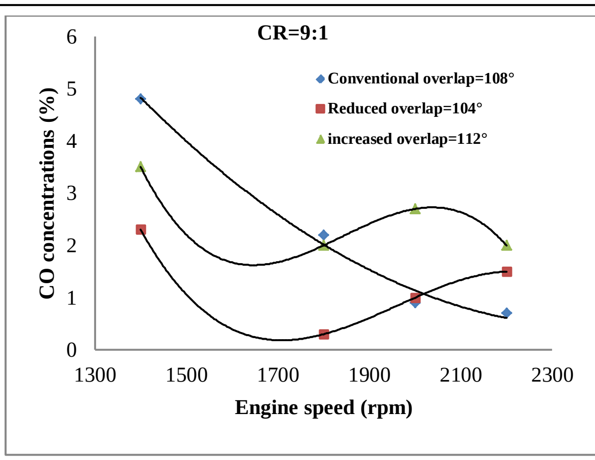

Fig. 9. The impact of VVT on CO for variable engine speeds at CR=9:1.

At (overlap=112˚) the CO concentrations decreased due to the combustion improvement. Also, the (IEGR) effects to decrease the CO emissions, with the method of Late Exhaust Valve Close (LEVC). A certain amount of exhaust gas is allowed to re-enter the engine from the exhaust port by downward piston motion during the beginning of intake stroke. By increasing valve overlap with Late Intake Valve Close (LIVC), the inlet valve remains open for a little longer during part of the compression stroke. So, a part of the charge is expelled back into the intake manifold as explained by references [20]-[21].

Figs. 10 & 11 show the variation in HC concentrations with the engine speed at compression ratio (CR=7 and 9) for the studied valves overlap. The HC emissions decrease with the increase of engine speed for the studied compression ratios. The retained gases in the cylinder tend to be exhausted at the end of the exhaust stroke. This portion of residual gases tends to contain the majority of the gases from crevice volume in the cylinder that are the typical source of most of the unburned hydrocarbons.

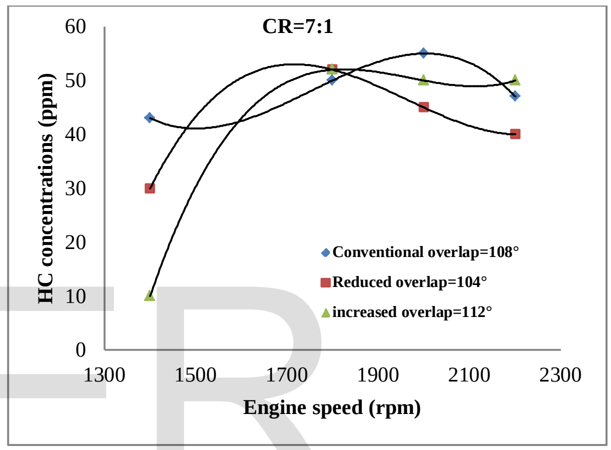

Fig. 10. The effect of VVT on HC for variable engine speeds at CR=7:1

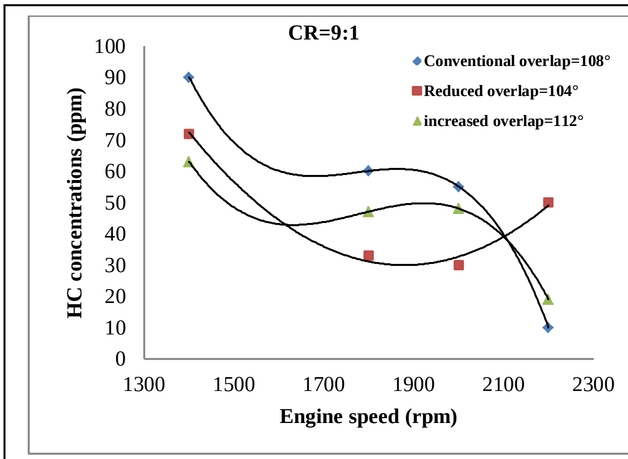

Fig. 11. The impact of VVT on HC for variable engine speeds at CR=9:1.

The figures depict the Hydrocarbon emissions for valve overlap (overlap=112˚). The curves indicate that the increase in the engine speed improved the Hydrocarbon emissions for early intake openings. Some amount of unburned and burned gas flows backwards into the intake port from the combustion chamber during the valve overlap. This reverse flow through the combustion process again with fresh (air-fuel) mixture, leads to the re-oxidation of the unburned gases. At higher engine speed, it is obvious that the (HC) emissions had similar tendencies to (CO) emissions for the same reasons. The high HC concentrations existed at engine partially misfiring.

4 Conclusions

The recent experimental investigation paper deals with the effect of the overlap period variation by changing the valve timings. The timing variations include the Intake Valve Open (IVO), Intake Valve Close (IVC), Exhausts Valve Open (EVO), and the Exhaust Valve Close (EVC) timings. The effect of this timing variation on engine performance and exhaust gases emission studied. The following conclusions extracted from the recent experimental investigations:

Effect of valve overlap variation at full load operation is not effective as in part load operation. The volumetric efficiency increased with (overlap=104˚) and decreased with (overlap=112˚) compared with the conventional overlap case. The resulted bsfc and brake thermal efficiency with changing the valve overlap were approximate. The compression ratio increment leads to increase Brake Thermal Efficiency at the same operating conditions. The increase in CR increases both temperature and pressure inside the combustion chamber that results in higher fuel evaporation rate and preferable mixing preparations before ignition. The reduction in overlap period (overlap=104˚) at compression ratio=9 shows the better compromises between the overall performance which means higher Volumetric Efficiency. Also, it improved the brake specific fuel consumptions, increased brake thermal efficiency, and reduced exhaust emissions.

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