The effect of Air to Fuel Equivalence Ratio on the Fuel Consumption of Slow Speed Diesel Engine (Sulzer RTA48)

Zuhaira M. Alhafi[1], and Abulkasim A. M. Asgyer[2]

1 *Department Mechanical and Industrial Engineering, Faculty of Engineering, University of Tripoli, Tripoli, Libya (zalhafi4@gmail.com)*

2 L*ibyan Academy for postgraduate studies. (l.A.P.S). Tripoli, Libya (abulkasim.asgyer @academy.edu.ly)*

ABSTRACT

In this paper, a RTA48 diesel engine have been tested under various equivalence ratios while monitoring its performance in terms of thermal efficiency, indicated and break power and specific fuel consumption. computer program and constricting mathematical models to simulate engine performance to obtain the results, which showed that by decreasing the equivalence ratio from closing to stoichiometric ratio to the very lean equivalence ratio, the thermal efficiency increase with 3% and the specific fuel consumption was decreased with 11 g/kW.hr , at very lean equivalence ratio and MCR (150 rpm) the break specific fuel consumption was 170. g/kWh and at closed to stoichiometric ratio was a round 181 g/kWh.

KEYWORDS: Diesel engine; thermal efficiency; fuel consumption; equivalence ratio.

INTRODUCTION

Diesel engines represent a unique feature in our contemporary civilization, which plays an important role in many industrial and agricultural projects, and the most common engines in generating power by land and sea, and they are frequently used in remote areas in particular to manage many machines for direct operation or to generate electric power. The world's largest thermal engines in operation are diesel engines, two-stroke 14-cylinder marine diesel engines; Produces a peak power of about 100 MW [1]

In previous studies in the mid-2010s, the main development goals of future diesel engines were described as improvements in exhaust emissions, reduced fuel consumption, and increased engine life, especially in the diesel engine for commercial vehicles and and sea transportation, which will remain the most important engine until the mid-2030s. The researchers hypothesize The complexity of the diesel engine will increase further in the future. Some researchers anticipate a future convergence of the operating principles of diesel engines because of the Otto engine development steps that have been made towards homogeneous charge pressure ignition [2]

The slow-speed diesel engine often used to drive the large propeller, is today the preferred method of propulsion for largest commercial ships and oil tankers due to its simplicity, reliability and low fuel consumption. Where great progress has been made in the design and construction of modern engines with slow speed, among these companies that have made a significant contribution to the production of this type of engine is SULZER company under the RTA series. The requirements for large thrust for merchant ships and their implications for engine design are presented, highlighting the issue of engine classification. The concept of engine designer companies and several engine builder licensees has been outlined, with a description of the practices in place for these arrangements. Recent advances in the design of large, slow-speed marine engines are summarized, while expanding briefly on each of the three overall goals, improving fuel consumption, reducing emissions, and increasing reliability.

Several researchers have developed numerical models of two-stroke marine diesel engines [3], in order to predict their operating information during propulsion [4] or to gain insight into the details of in-cylinder operations [5].

To improve the operating parameters of a two-stroke marine diesel engine and reduce emissions, several investigations have been carried out in order to implement alternative fuel combustion [6], using a bio-fuel blend for combustion [7] or using standard diesel fuel with some additives [8] of slow two stroke engine.

Vedran et. al (2017) [9] presents numerical analysis of efficiencies and non-measured operating parameters for the marine two-stroke slow speed turbocharged diesel engine 6S50MC MAN B&W with direct fuel injection. Numerical analysis was based on a measurement set performed at different engine loads. Calculated efficiencies were mechanical, indicated and effective efficiency.

In this paper, the effect of the equivalence ratio of fuel and air on the performance of a diesel engine represented in thermal efficiency and specific fuel exchange will be studied, through the development of a simulation program based on the thermodynamic laws of reciprocating internal combustion engines of a diesel cycle of a two-stroke diesel engine with speed SLOW MARINE SULZER RTA48

MATHEMATICAL MODEL OF COMPRESSION ENGINE CYCLE

For the calculations a theoretical computer program language has been developed, the model was built for working space of combustion chamber accordance with the real engine geometry of a two-Stroke Sulzer Diesel & Turbo engine.

In this paper the theoretical model is used to study the effect of the air to fuel equivalence ratio (ϕ) on the engine performance in terms of indicated and break power, specific fuel consumption.

The governor equations of this model are applied on an open system first law for the combustion chamber, typical calculations for a direct injection (DI) diesel engine use Equation (1) with the injected fuel explicitly included is [10,11]:

$$\frac{\dot{m}_{fi}}{\dot{m}_{ft}} = \frac{\omega}{\theta_d\ \Gamma(n)}\left(\frac{\theta-\theta_s}{\theta_d}\right)^{n-1} exp\left[\frac{-(\theta-\theta_s)}{\theta_d}\right] \tag{1}$$

Where:

θ is the Crank angle.

θ_s is the start of fuel injection.

θ_d is a measure of the injection duration.

$\Gamma(n)$is the Gamma Function = $\int_0^\infty t^{n-1}\, e^{-t} dt$ (Abramowitz, 1972).

ω is the Engine frequency, [1/sec].

n is the Shape parameter.

$1 \leq n \leq 2\ \ forDIchamber$

$3 \leq n \leq 5\ fordividedchamber$

The rate of change of "Burned fuel" in the cylinder is given by equation (2):

$$\frac{d_{m_f}}{d_\theta} = \frac{1}{\omega}\left(\dot{m}_{fi} - \frac{\dot{m}_{t\emptyset F_s}}{1+\emptyset F_s}\right) \tag{2}$$

The rate of change of "Burned air" in the cylinder is given by equation (3)

$$\frac{dm_a}{d\theta} = \frac{-\dot{m}_t/_\omega}{1+\emptyset F_s} \tag{3}$$

And the equivalence ratio at any time is given by equation (4):

$$\emptyset = \frac{m_f}{F_s m_a} \tag{4}$$

The mass in the cylinder at any time is:

$$m = m_a + m_f \tag{5}$$

The energy equation applied to the cylinder contents is given in equation (6)

$$\frac{dU}{d\theta} = -\frac{Q_l}{\omega} - P\frac{dV}{d\theta} - \frac{\dot{m}_l h_l}{\omega} + \frac{\dot{m}_{fi} h_f}{\omega} \tag{6}$$

Where

F_S is the stoichiometric fuel - air ratio by mass

U is the Internal energy.

$\dot{Q}_l$ is the Rate of heat loss.

P is the pressure in the cylinder.

V is the Volume.

h_l is the enthalpy of blow-by fluid.

h_f is the enthalpy of fuel.

By assuming that the system is a homogeneous.

$U = mu = mu(T, P, \emptyset)$

$V = mv = mv(T, P, \emptyset)$

$h = mh = mh(T, P, \emptyset)$

The functional relationships indicated are for equilibrium combustion products, a subroutine ECP is used.

The heat loss rate is given by equation (7);

$$\dot{Q}_l = H\left(\frac{\pi B^2}{2} + \frac{4V}{B}\right)(T - T_w) \qquad (7)$$

Where:

B is the cylinder Bore, [m].

H is heat transfer coefficient $[J/(m^2.K.sec)]$.

T is the combustion temperature [K].

In this case; assuming that the cylindrical combustion chamber with walls temperature T_w [K]. The volume is calculated as a function known from the engine geometry and engine speed.

The principle behind the operation of a diesel engine is the compression-ignition cycle. Downward movement of the piston causes air to be drawn or air forced charged into the engine cylinder where it is compressed on the upward stroke of the piston. Fuel is injected as the piston approaches the end of the compression stroke, and ignites spontaneously. The increase in pressure generated by the fuel burning provides the power of the engine. Table 1 shows the input data of the engine

Table 1 INPUT DATA:

1	Compression ratio	R	---	17
2	Bore,	B	m	0.48

3	Stroke,	S	m	1.4
4	Half stroke to rod ratio	EPS	---	0.25
5	Engine speed	RPM	---	150
6	Heat transfer coefficient	H	$J/m^2/K/s$	500
7	Blowby coefficient	C	1/s	0.03
8	Mean equivalence ratio	ϕ		0.6
9	Residual mass fraction	F		0.05
10	Initial pressure	P_i	MPa	0.162
11	Initial temperature	T_i	K	318
12	Wall temperature	T_W	K	420
13	Start of injection	θ_S	deg	-20
14	Injection duration parameter	θ_D	---	5.33
15	Injection parameter	EN	---	2
16	Fuel injection pressure	---	MPa	120
17	Crank advance angle	$\Delta\theta$	deg	2
18	Fuel		Diesel	

RESULTS AND DISCUSSIONS

The equivalence ratio of air to fuel is the key of all heat engine design, and has a significant effect on the engine performance and engine specific fuel consumption. In this project a sulzer RTA48 slow speed diesel engine has been selected to study the effect of the equivalence ratio on its performance, the equivalence ratio of 0.6 at very lean mixture to 0.95 closed to the stoichiometric ratio have been studied. Modern diesel engines are always using the turbo-charger system with inter-cooler to improve the engine volumetric efficiency. In this paper the pressure ratio of the turbocharger (Pr) is 1.6 and inter cooler air temperature of air charging to the cylinder is 45 ^oC. the slow diesel engine was selected with speed of 150 rpm. Its specifications are described in Table (1). Figure (1) shows the relationship between the crank angle and the pressure of gases inside the cylinder at different equivalence ratio (ϕ).

Figure (1) shows the pressure at the compression and expansion (power) strokes, at the beginning of the compression stroke the air charge under the pressure of the turbocharger at a pressure rate of 1.6, intial pressure (Pi) is 0.162 MPa, when the piston moves from the bottom dead centre (-180 Deg) towards the top dead centre (0 Deg), the pressure gradually to rise, when the piston reaches the angle of the crank shaft of 20 Deg before the top dead center (BTDC) at which diesel fuel is injected, and then the combustion occurs, causing a strong rise in pressure. The maximum pressure inside the cylinder occurrence just after top dead centre with very small angle less than 10 degs, at equivalence ratio of 0.95 the maximum pressure is about 26.7 MPa and about 21.3 MPa at ϕ =0.6. The pressure then decreases in the expansion stroke, at which the piston moves away from the top dead centre, towards the bottom dead centre.

The figure demonstrates the strong effect of the equivalence ratio on the pressure inside the cylinder. Maximum pressure occurs when the equivalence ratio approaching to the stiochiometric ratio and decreases when the mixture is very lean.

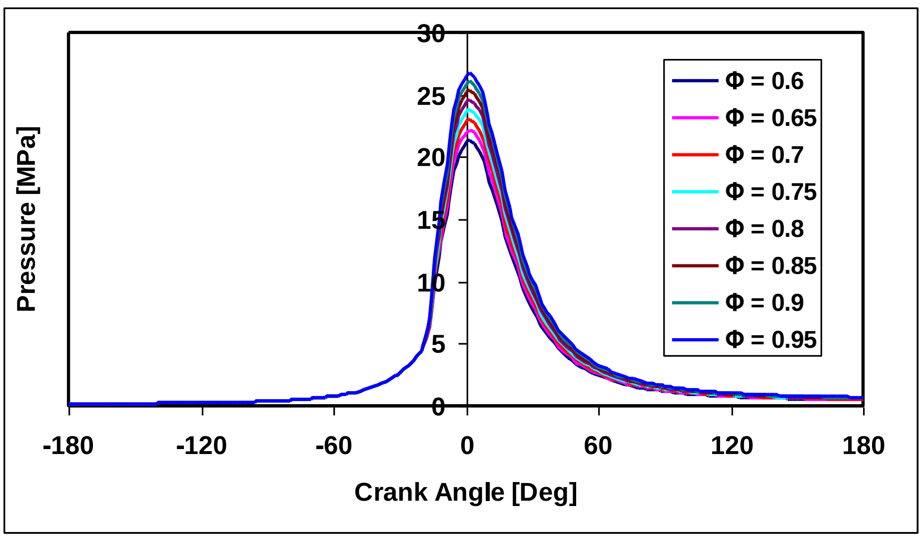

Fig. (4.1). Pressure of the gases inside the cylinder as a function of the crank angle during the compression and expansion strokes at different equivalence ratio Φ

Figure (2) shows the temperature of gases inside the cylinder and the crank angle at different equivalence ratio (ϕ), From the results; it is shown that the maximum temperatures inside the cylinder are influenced by the equivalence ratio, at the beginning of the compression stroke the temperature is equal the temperature of air charged by the inter-cooler turbocharger system (45 ^{o}C). The maximum temperature inside the cylinder was at high equivalence ratio closed to the stiochiometric ratio, and decreased at lean mixture. The maximum temperature was about 2950K at equivalence ratio of 0.95 and 2420 K at equivalence ratio of 0.6.

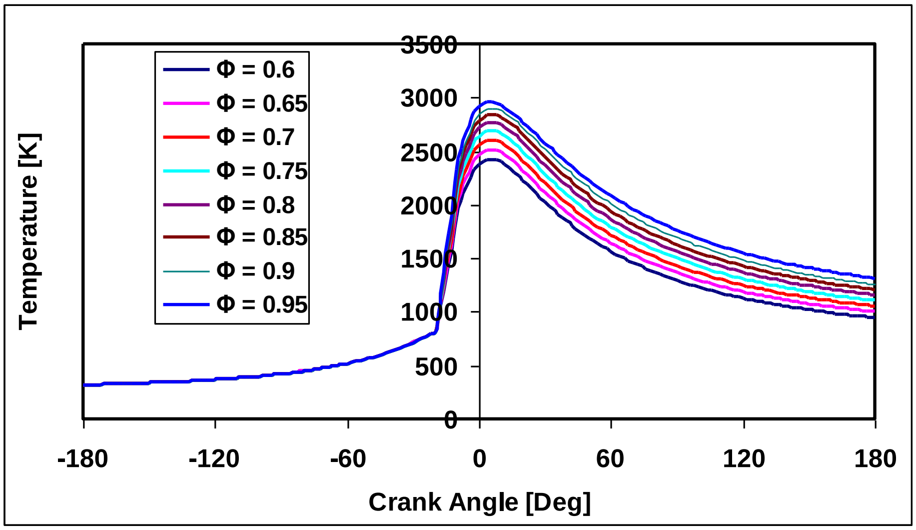

Fig. (2). Temperature of the gases inside the cylinder as a function of the crank angle during the compression and expansion strokes at different equivalence ratio Φ

Figure (3) shows the relationship between the gas mass inside the engine cylinder and the crank shaft angle of the engine under different equivalence ratios (ϕ), in this study the ratios of fuel and air equivalence ratio were selected from the very lean mixture when the equivalence ratio is 0.6 to equivalence ratio very close to the stoichiometric ratio of 0.95. In this type of engine, use a turbocharger to insert air into the cylinder, with turbocharger pressure ratio (Pr) of 1.6. Also use an intercooler to increase engine volumetric efficiency. The compressed air outlet the compressor of the turbocharger was cooled to 45°C, using seawater heat exchanger. In these operating conditions, the volumetric efficiency of the engine was 150%.

From Figure (3) at the beginning of the compression stroke, the gases in the cylinder were air only, adding residual gases of exhaust gases in the previous cycle, which were assumed in this study by about mass fraction of F= 0.05. At the beginning of the compression when the piston was at the bottom dead centre (BDC), at crank angle of -180 degrees, the gas mass was 0.4758522 kg. With the rotation of the crank shaft of the engine during the compression stroke the mass of these gases, which are mostly air are decreased, as a result of blow-by effect between the piston and the cylinder wall, which was assumed here in this study by mass fraction of C = 0.03. The mass of air and residual gases decreased to 0.4736931 kg at the crank angle of 20 before the top dead centre (TDC) at which the fuel was started to injected. The mass has begun to increase as a result of fuel injection with combustion limits, here the amount of increase varies by varying air and fuel equivalence ratio. The rich mixture, the greater the amount of fuel injected. The maximum mass in the cycle is when the injection period is the highest values.

The maximum mass in the cycle is recorded when the injection period is completed. The maximum mass recorded is 0.5027271 kg at the crank angle of 24 degrees after the top dead centre (ATDC) and ϕ = 0.95. While the maximum gas mass was 0.4917964 kg when the when the equivalence ratio was 0.6. By the end of the power stroke, the total mass of gases was 0.5006243 kg at the ϕ = 0.95, which is entirely exhaust gases and at the very lean mixture (ϕ=0.6) the mass was 0.4897028 kg, also this decrease in mass from its maximum value as a result of the blow-by effect of the engine.

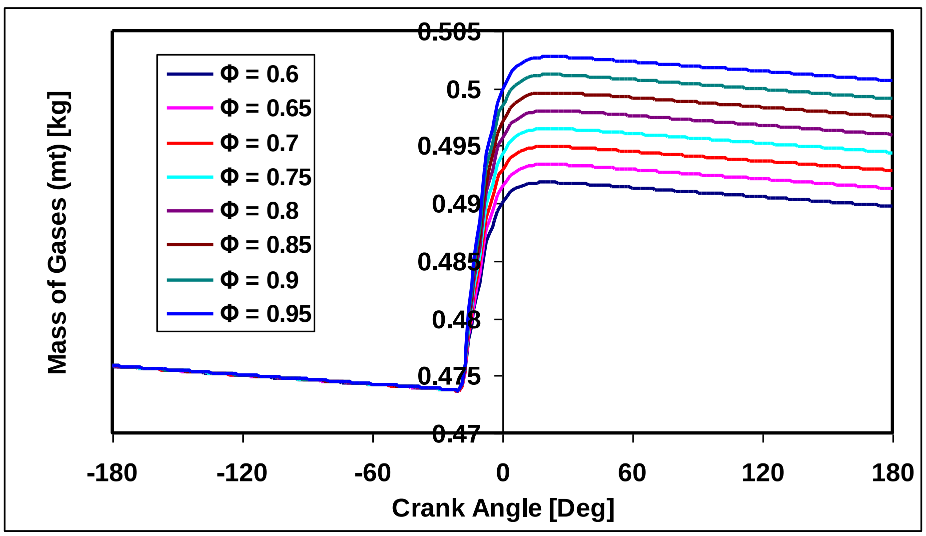

Fig. (3). Mass of the gases inside the cylinder as a function of the crank angle during the compression and expansion strokes at different equivalence ratio Φ

Figure (4) shows the relationship between the indicated mean effective pressure and the equivalence ratio. The engine's indicated power is known to be proportional to the indicated mean effective pressure, and the indicated mean effective pressure of the engine depends strongly on the equivalence ratio. At equivalence ratio closed to stoichiometric ratio induced higher indicated mean effective pressure, and the lower values of imep at lean mixture. Imep was 2.4588 MPa at ϕ = 0.95, while the imep was 1.6487 MPa at very lean mixture (ϕ= 0.6)

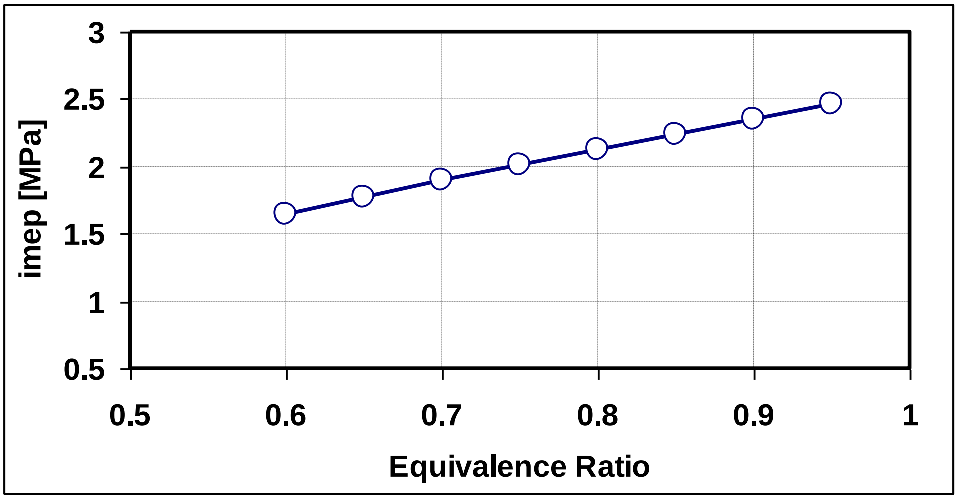

Fig. (4) Indicated mean effective pressure (imep) as a function of the equivalence ratio Φ

Figure 5 shows the relationship between the equivalence ratio and thermal efficiency. To metering any heat engine, thermal efficiency is an indicator of its performance. This is a ratio between engine power output to the heat given to the engine represented in the fuel energy given to the engine. From the figure, thermal efficiency is strongly influenced by the equivalence ratio. Thermal efficiency increased from 45.9% at equivalence ratio closed to the stoichiometric ratio (ϕ=0.95) to 48.8% at very lean mixture (ϕ=0.6).

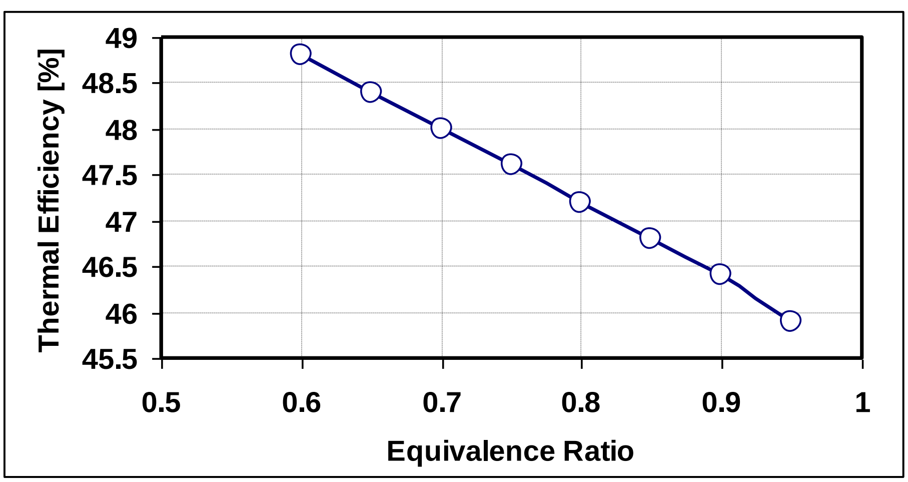

Fig. (5). Thermal efficiency as a function of the equivalence ratio Φ

Figure (6) demonstrates the relationship between maximum pressure of gases in the cylinder with air to fuel equivalence ratio. The maximum pressure of the gases inside the cylinder is strongly influenced by equivalence ratio. The results show that maximum pressure occurs when the equivalence ratio approaches the stoichiometric ratio, and decreases as the mixture is lean. The maximum record pressure was 28.7 MPa, when the equivalence ratio was 0.95, and it was 21.3 MPa when the mixture was lean (ϕ=0.6).

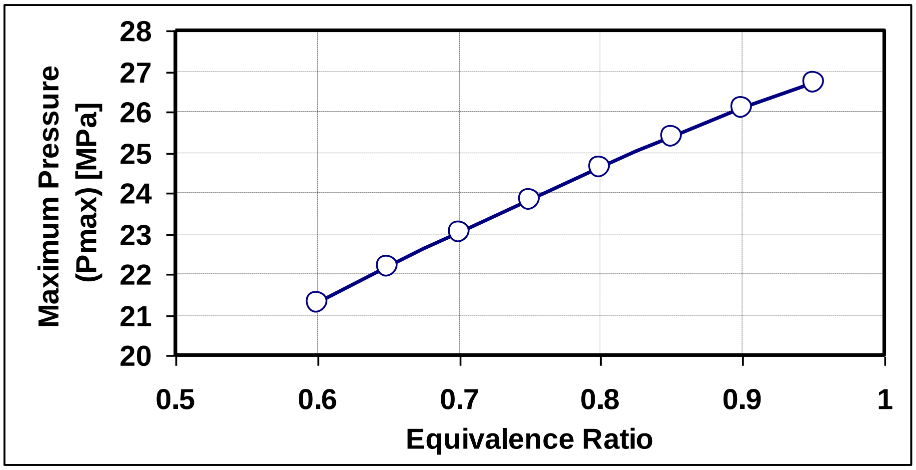

Fig. (6). Maximum pressure of the gases inside the cylinder (Pmax) as a function of the equivalence ratio Φ

Figure (7) shows the relationship between maximum temperature of gases in the cylinder with air to fuel equivalence ratio. The maximum temperature inside the cylinder with the equivalence ratio is the same as the maximum pressure behavior within the cylinder described earlier. The maximum temperature of the gases inside the cylinder is strongly influenced by equivalence ratio. The results show that maximum temperature occurs when the equivalence ratio approaches the stoichiometric ratio, and decreases as the mixture is lean. The maximum record temperature was 2955K, when the equivalence ratio was 0.95, and it was 2423K when the mixture was lean (ϕ=0.6).

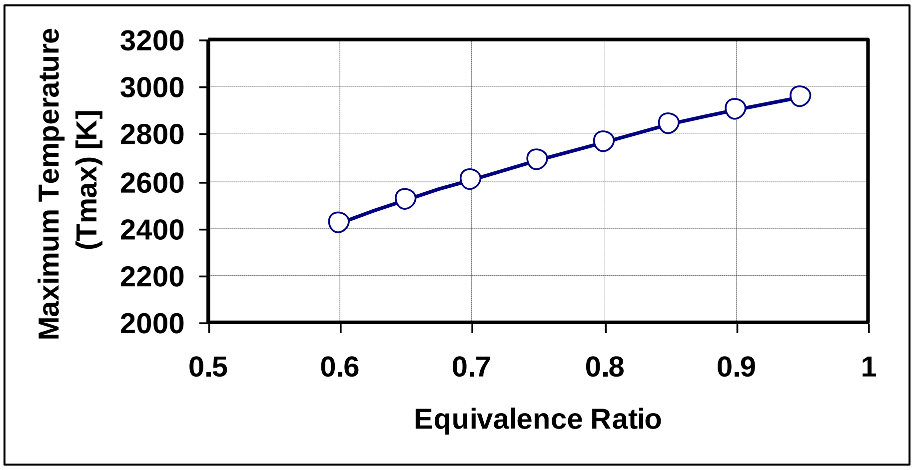

Fig. (7). Maximum temperature of the gases inside the cylinder (Tmax) as a function of the equivalence ratio Φ

It has also been noted in Figure (8) that the indicated power (Pi) of the engine depends directly on the indicated mean effective pressure (imep). Figure (8) demonstrates the relationship of indicated power with the equivalence ratio. From the figure, it is clear that the higher amount fuel in the mixture and equivalence ratio approaches closer to the stoichiometric ratio leads greater indicated power. When the engine is operated at very lean, the indicated power will reduced In this engine, the indicated power at equivalence ratio of 0.95 was 1557.8 kW/cyl., while the indicated power was 1044.6 kW/cyl. when operating with a very lean mixture (ϕ= 0.6). Also, Figure (8) demonstrates the engine's break power (Pb) and its relationship to equivalence ratio. By knowing the mechanical efficiency of the engine η_m, the break power of the engine can be calculated: by using Eq. (8):

$$P_b = P_i \eta_m \tag{8}$$

Where:

P_i - is the indicated power [kW/cyl.]

η_m - is the mechanical efficiency.

The mechanical efficiency of this engine selected (slow speed Sulzer RTA480) is 95%. the break power at equivalence ratio of 0.95 was 1480 kW/cyl., while the preak power was 992.6 kW/cyl. when operating with a very lean mixture (ϕ= 0.6).

When the amount of fuel increases it causes increased pressure inside the cylinder resulting in greater power.

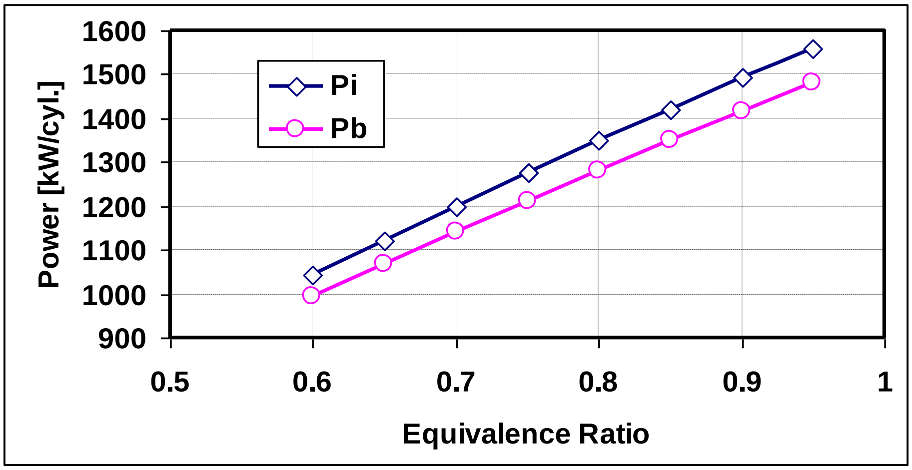

Fig. (8). Power (P) as a function of the equivalence ratio Φ

The fuel combustion is assumed to be complete, and the products will be CO_2, H_2O and N_2. The classic combustion equation for heavy diesel fuel, where a is the excess air coefficient is given by:

$$C_{14.6}H_{24.8} + 20.8\left(O_2 + \frac{79}{21}N_2\right) ==> 12.4H_2O + 14.6\,CO_2 + \ 20.8\left(\frac{79}{21}N_2\right)$$

Fuel	Closed Formula	Mole Weight	Heating Value		Stoichiometric
			HHV $[^{kJ}/_{kg}]$	LHV $[^{kJ}/_{kg}]$	$\left(\frac{A}{F}\right)_s$
Light Diesel	$C_{12.3}H_{22.2}$	170	44800	42500	14.5
Heavy Diesel	$C_{14.6}H_{24.8}$	200	43800	42400	14.5

$$\phi = \frac{\left(\frac{A}{F}\right)_s}{\left(\frac{A}{F}\right)_a} \qquad (9)$$

$$\left(\frac{A}{F}\right)_a = \frac{\left(\frac{A}{F}\right)_s}{\phi} \qquad (10)$$

Where :

$\left(\frac{A}{F}\right)_s$ is the stoichiometric air to fuel ratio

$\left(\frac{A}{F}\right)_a$ is the actual air to fuel ratio

Figure (4.9) plots actual air to fuel ratio $\left(\frac{A}{F}\right)_a$ as a function of equivalence ratio.

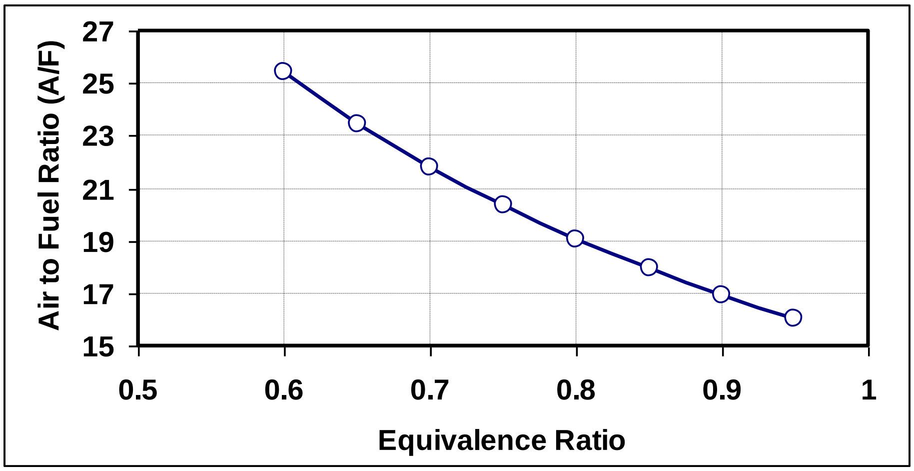

Fig. (9). Air to fuel ratio (A/F) as a function of the equivalence ratio Φ

Figure (9) shows the effect of the equivalence ratio on the indicated and break specific fuel consumption (ISFC and BSFC). One of the very important fundamentals that needs to be taken into account in the metering of the engines is the specific fuel consumption, due to its interest in reducing the cost of fuel consumption and obtaining the highest possible power of the engine, as well as reducing the specific fuel consumption causes a decrease in the pollutants produced by the engine from combustion process. The results, in Figure (9), show that the indicated and break specific fuel consumption is strongly influenced by the equivalence ratio of air to fuel. ISFC and BSFC decreased by more than 6% per cylinder when the equivalence ratio is reduced from 0.95 to very lean mixture with $\phi = 0.6$. ISFC decreased from 171.4 g/kW.h at $\phi = .95$ to 161.3 at $\phi = 0.6$, and BSFC decreased from 180.4 g/kW.h at $\phi = .95$ to 169.6 at $\phi = 0.6$,

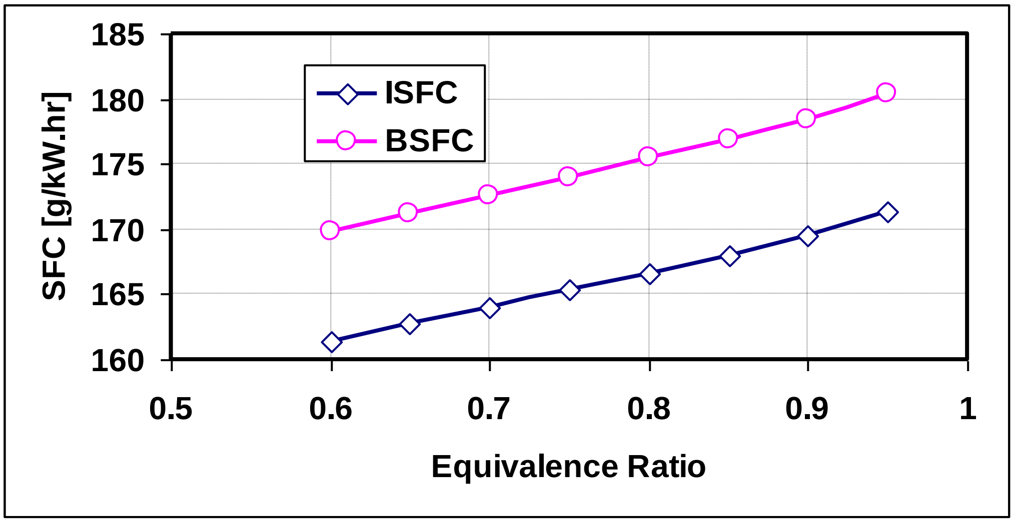

Fig. (10). Specific fuel consumption (SFC) as a function of the equivalence ratio Φ

Figure (8) show the relationship of the engine torque as a function of the equivalence ratio. From the figure, it is clear that the higher torque calculated at equivalence ratio closed to stoichiometric ratio. The engine torque can be calculated using eq. (11)

Torque (T) [N.m]= Break Power (Pb) [kW] * 9.549/ speed [rpm] (11)

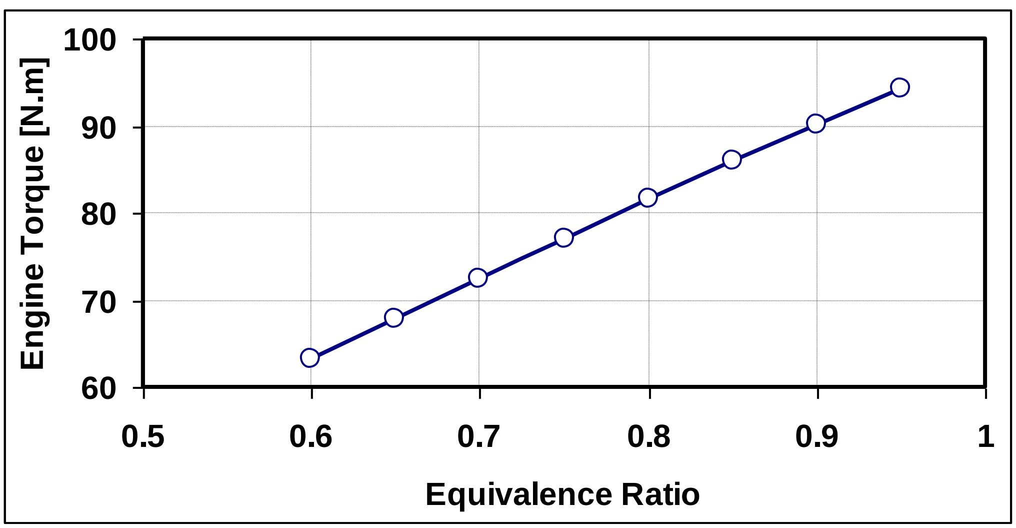

Fig. (11). Engine Torque as a function of the equivalence ratio Φ

CONCLUSIONS

In this paper the purpose of studying a the equivalence ratio effect on the engine performance at lean mixture is to increase the themal efficency and reduced the specific fuel consumption of slow speed diesel engine (Sulzer _ RTA 48). From results of testing the effect of the equivalence ratio on the engine performance show that ,when the engine opartes under the conditions of the stoichiometric mixture, the indicated mean effective pressure (imep) of the engine increases, causing an increase of the engine's indicated and break power and also an increase of engine torque, but on the other hand when operating the engine at stoichiometric ratio or closed to stoichiometric mixture ratio, results in an increase in the specific fuel consumption, and decrease of the thermal efficiency of the engine, and also when operating in these conditions, dangerous pollutants emissions are increased because the rise in temperatures of combustion gases, such as a nitrogen oxide (NOx). The engine must therefore be operates at very lean mixture as possible.

Results show that the thermal effciency and indicated and break specific fuel consumption are strongly influenced by the use of the lean equvalence ratio of air and fuel. Themal efficency increased from 45.9% at equivalenc ratio 0.95 which is closed to stoichiometric ratio to about 49% at lean lean mixture with equivalence ratio of 0.6. The ISFC and BSFC are also decreased by more than 6% per cylinder. Reducing of the specific fuel consumption causes a decrease in the pollutants produced by the engine from combustion process.

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