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# PERFORMANCE ANALYSIS OF TWO-STROKE REVERSE UNIFLOW SPARK IGNITION ENGINE BY USING MATLAB

# İKİ ZAMANLI TERS-DOĞRU AKIŞLI BUJİ İLE ATEŞMELİ BİR MOTORUN MATLAB YARDIMI İLE PERFORMANS ANALİZİ

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## ÖZET

Günümüzde, petrol ve türevi kaynakların hızla azalıyor olması ve küresel ısınmanın da hızla artması nedeni ile içten yanmalı motorların ekonomik ve efektif kullanımı hayli önemli hale gelmiştir. Bu nedenle, içten yanmalı motorların performanslarının iyileştirilmesi konusunda yoğun çalışmalar yapılmaktadır. Motor performansının belirlenmesi iki aşamada yapılmaktadır. Birinci aşamada, henüz tasarı halindeki motorun performansı çevrim analizi aracılığı ile teorik matematiksel model kullanılarak belirlenmekte, motor parametrelerinden her birinin motor karakteristiklerine etkileri incelenmektedir. İkinci aşamada, teorik çalışma doğrultusunda deneysel çalışmalar yapılmaktadır. Bu çalışmada, iki zamanlı ters doğru akışlı, tek silindirli, buji ile ateşlemeli bir motorun performansının analizi için MATLAB programlama dilinin kullanılmasıyla bir bilgisayar programı geliştirilmiştir.

**Anahtar Kelimeler**: İki zamanlı motor, simülasyon, motor performansı, matematiksel model

## ABSTRACT

Today, due to the rapid decrease of resources such as petroleum and its derivatives and rapid increase global warming, the economic and effective use of internal combustion engines had become very significant. Therefore, intense studies about increasing the performance of the engines are performed. Determination of the engine performance has been done in two stages. At the first stage, the performance of the engine still in a draft form is determined through the cycle analysis in a theoretical way by using a mathematical model. At this stage each parameter of the engine is changed and its effect on engine characteristics is determined. At the second stage, experimental studies are done in accordance with theoretical work. In this study, a computer program was developed to simulate the performance of a two-stroke reverse uniflow spark ignition engine by using MATLAB.

**Keywords:** Two stroke engine, simulation, engine performance, mathematical modeling

## 1. INTRODUCTION

In parallel with the developments in the technology, the developments relevant to internal combustion engines are astonishing. But no essential change had been made in the main structures of engines. The characteristics of engine take an important place in the design of a new engine or in the improvement of a current engine. Determination of the characteristics of engine constitutes the basis of engine tests, and it does not only enable knowing the performance of engine as bare and static but also provides significant information regarding performance on vehicle under real service conditions. With this purpose, simulation programs were been used in order to examine of the thermodynamic behavior of engine and determine the performance under real service conditions. In product development, engineers simulate the underlying partial differential equation many times with commercial tools for different geometries (Junga, Paterac, Haasdonkd, Lohmannb, 2011: 17).

Simulation studies are beneficial for the analysis of various operations in manufacture methods (Babic, Miljkovic, Vukovic, Antic, 2012: 36). When digital modeling of facts realized during operation of an engine is made through computers and submitted for the use of manufacturers, the period of experimental operations which will take long was been shortened. The involvement of computers in the technology and improvements in the numerical calculations had provided to obtain more realistic results compared to calculation made in order to determine the characteristics of internal combustion engines. For example, the use of theoretical model is a good tool in meeting demands for a good fuel economy and for decreasing exhaust emissions. By the mathematical modeling of engines;

1. It was been able to determine the ability of engine to meet the expected behavior under different operation conditions (Sekmen, Erduranlı, Koca, 2007: 12).
2. Through the increasing speed of computers and shortening analysis periods, the cost can be calculated more effectively and through the applied digital methods, improvement was been obtained in simulation (Sekmen, Erduranlı, Koca, 2007: 12).
3. Change in performance is able to be searched through the analysis of mathematical model for each of the engine’s parameters and optimum engine characteristics are able to be determined (Sekmen, Erduranlı, Koca, 2007: 12).

Conventional 2-stroke engines are commonly used for two-wheeled and marine vehicles that have small engine displacement (Junpei, Yasuo, 2004). When two stroke engines are compared with four stroke engines having the same characteristics, they are lighter and have a higher output (Kashani, 2004: 28). But, in order to improve the new generation two stroke engines, there are many problems required to be addressed such as loss of fresh fill, irregular combustion at low load, decreasing exhaust emissions (Galindo, Climent, Pla, Jimenez, 2011: 31). The most urgent problems are increase of fuel consumption caused by short circuit along scavenging and very intense HC emission. It is not an exaggeration to say the all the performance of two stroke engine is determined by its scavenging behavior. Combustion simulation in two stroke engines becomes necessary not only for engine performance prediction but also for scavenge evaluation, since in-cylinder pressure and temperature are highly influenced by combustion process evolution (Abu-Nada, Al-Hinti, Akash, Al-Sarkhi, 2007: 31).

Small two-stroke engines, which are widely used in mobile machinery, will have to accomplish strict anti-pollution standards in imminent years. Active and passive solutions applied to two stroke engines design have arisen in the last decades. Conforming active solutions, exhaust emissions reduction technologies have been investigated (Duret, 1988, Nuti, 1997, Payri, 2011), and on the other hand, within passive solutions, current engine systems with different Technologies (carburettor, direct injection, catalyst) were been improved and evaluated in terms of emission control (Czerwinski, Compte, Reutimann, 2006: N3).

The purpose of this study is to determine the real cycle parameters of a two stroke, reverse uniflow, single cylinder, spark ignition engine under computer environment. In the preparation of simulation program, MATLAB programming language had been preferred as it’s a valid and widespread language in the field of engineering.

## 2. MATHEMATICAL MODEL

In this study, the real engine cycle and engine performance of a two-stroke reverse-uniflow single cylinder spark ignition engine had been calculated. For this, a computer program had been prepared by which the effects of parameters such as compression ratio, engine speed and excess air coefficient on the performance of engine are numerically examined. In calculation realized through the designed mathematical model, pressure and temperature values had been calculated iteratively with the Newton-Raphson method with 1 degree intervals for 360 ºCA where the cycle realizes.

For the solution of model, parameters such as compression ratio, excess air coefficient (EAC), engine speed, environment's pressure and temperature, engine design characteristics had been introduced at the beginning of the program developed by using MATLAB, and changes of pressure and temperature along the cycle had been determined with the numeric solutions of simple differential equations. According to different compression ratio (6:1–8:1-10:1) and excess air coefficient (0.80–0.88-0.96), engine performance at engine speeds in between $n$=1000–3500 rpm had been examined.

### *2.1 Intake process*

By the beginning of intake operation at an operating engine, it is assumed that there exists waste exhaust gases under $P_r$ and $T_r$ conditions ( $P_r$=(1.05-1.25).$P_0$, $T_r$=(900-1100)$K$ ) which had remained from the previous cycle and as much as the dead volume within the cylinder. In natural intake engines, it is considered that fresh fill is started to be taken in under $P_0$ and $T_0$ conditions.

Due to all losses arising along the intake operation, *a* pressure decrease of $\Delta P_a$ occurs according to $P_0$ pressure. This pressure decrease is determined with the equation of;

$$\Delta P_a = (\beta^2 + \zeta)\left(\frac{V_m n}{n_N}\right)\frac{\rho}{2}10^{-6} \tag{1}$$

Where;

$\beta^2 + \zeta$ : Entrapment and friction losses,

$V_m$ : The speed at the most narrow cross-section,

$\rho$ : Density of air,

$n$ : Engine speed (*rpm*),

$n_N$ : Nominal speed (*rpm*).

Thus the pressure by the end of intake becomes;

$$P_a = P_0 - \Delta P_a \tag{2}$$

And the temperature by the end of intake becomes;

$$T_a = \frac{T_0 + \Delta T + \gamma_r . T_r}{1 + \gamma_r} \tag{3}$$

Where;

$\Delta T$ : Pre-heating of fresh fill

$\gamma_r$ : Coefficient of waste gases

$T_r$ : Temperature of waste gases

### *2.2 Compression process*

If it is considered that the compression operation is realized with one $n_1$ polytrophic exponent, the pressure and temperature by the end of compression becomes;

$$P_2 = P_a \varepsilon^{n_1} \tag{4}$$

$$T_2 = T_a \varepsilon^{n_1 - 1} \tag{5}$$

Where *"$n_1$"* is found by the solution of the following equation obtained under the assumptions that the compressed mixture consists only of air and that the air's $\overline{C_V}$ specific heat changes linearly with temperature;

$$A + BT_1\left(\varepsilon^{n_1-1} + 1\right) = \frac{\overline{R}}{n_1 - 1} \tag{6}$$

Where along the compression operation, mixtures of fuel mostly consisting of air in gasoline engines and waste exhaust gases exist within the cylinder. Thus, in approximate calculations, the *A* and *B* coefficients will be taken as coefficients relevant to air.

### *2.3 Combustion process*

If the elemental constitution of fuel is being known as *C, H, $O_y$, S, W*, the required air amount $H_{min}$ to combust 1 kg fuel is [13];

$$H_{\min} = \frac{1}{0.208} . \left[ \frac{c'}{12} + \frac{h'}{4} + \frac{O_y'}{32} \right] \tag{7}$$

Where depending on the value of $\alpha$ excess air coefficient, *($M_1$)* fresh fill amount for 1 kg fuel is found with the formula of;

$$M_1 = \alpha . H\min + \frac{1}{M_y} \tag{8}$$

Where lower thermic value of fuel is found by the Mendeleyev formula;

$$H_u = 33.91c' + 125.6h' - 10.89(O_y' - S') - 2.51(9h' + W) \tag{9}$$

Where;

$H_u$ : Lower thermic value of fuel (*kJ/kg*)

$c'$ : Carbon amount emitted by the combustion of 1 kg fuel

$h'$ : Hydrogen amount emitted by the combustion of 1 kg fuel

$O_y'$ : Oxygen amount emitted by the combustion of 1 kg fuel

$S'$ : Sulphur amount emitted by the combustion of 1 kg fuel

$W$ : Water amount emitted by the combustion of 1 kg fuel

In gasoline engines, it can not be benefited from all the calorific value of fuel due to combustion with $\alpha \prec 1$ excess air coefficient. It can be benefited from calorific value as much as $H_u^{'} = H_u - \Delta H_u$. The heat loss due to deficient combustion can be taken as $\Delta H_u = 114.(1-\alpha).H\min$.

*2.3.1 Combustion equation*

$$\frac{\xi_z.H_u^{'}}{M_1.(1+\gamma_r)} + [A + B.T_2].T_2 = \mu.(A_g + B_g.T_3).T_3 \qquad (10)$$

Where;

$[A + B.T_2]$ : Mean specific heat of air

$(A_g + B_g.T_3)$ : Mean specific heat of combustion products

$\xi_z$ : Coefficient of benefiting from heat in combustion

$\mu$ : Molecular change coefficient

and the end of combustion pressure becomes;

$$P_3 = \lambda.P_2 \qquad (11)$$

***2.4 Expansion process***

If it is considered that the expansion operation is realized with one $n_2$ polytrophic exponent, the end of expansion characteristics become;

$$P_4 = \frac{P_3}{\varepsilon^{n2}} \qquad (12)$$

$$T_4 = \frac{T_3}{\varepsilon^{n_2 - 1}} \qquad (13)$$

And $n_2$ polytrophic exponent is found by the solution of the following equation;

$$A_g + B_g.T_3.(1 + \varepsilon^{1-n_2}) = \frac{\overline{R}}{n_2 - 1} \qquad (14)$$

***2.5 Exhaust process***

For exhaust temperature, the following formula is being provided;

$$T_r = \frac{T_4}{\sqrt[3]{\left(\frac{P_4}{P_r}\right)}} \qquad (15)$$

If the difference (or error ratio) in between this value and the $T_r$ value selected by the beginning of cycle calculations is larger than a specific value, the calculation shall be repeated until the difference decreases to required value by using the recently found temperature at the beginning of cycle calculation.

### *2.6 Engine performance characteristics*

The most significant disadvantage of classic two-stroke engines is the release of some amount of fresh fill from the exhaust port along the scavenging operation (short circuit), and the arise of high fuel consumption and high HC emissions as a result. In order to remove such faults of two-stroke engines, scavenging methods of uniflow or reverse uniflow type had been developed, and uniflow scavenging method was frequently been used in diesel engines of vessels. The design history of reverse uniflow scavenging method is not as old as uniflow scavenging method. It was been thought that a computer program which will reveal the performance characteristics of 2 stroke reverse uniflow, scavenging gasoline engines would provide more realistic terms for this type of scavenging method. In figure 1, the schematic picture of the 2 stroke, reverse uniflow engine was been seen.

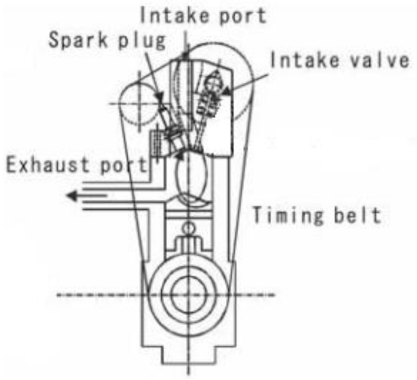


Figure 1. Schematic of reverse uniflow type engine

Engines will be operated in a wide range of rotation speeds. The least rotation speed is being limited by the required conditions for the stable operation of engine, in other words the composition and intake of air-fuel mixture in gasoline engines, and maximum rotation speed is being limited by efficiency of intake and exhaust operations, thermal stress of parts, increasing inertia forces, decrease in mechanical efficiency etc.

#### *2.6.1 Mean Indicated Pressure*

Mean indicated pressure can be used with the following formula by using the results of cycle calculation;

$$P_{mi} = \phi_i . P_{mi}^{'} \tag{16}$$

$$P_{mi}^{'} = \frac{P_a . \varepsilon^{n_1}}{\varepsilon - 1} . \left[ \frac{\lambda}{n_2 - 1} . \left( 1 - \frac{1}{\varepsilon^{n_2 - 1}} \right) - \frac{1}{n_1 - 1} . \left( 1 - \frac{1}{\varepsilon^{n_1 - 1}} \right) \right] \tag{17}$$

Where, “$\phi_i$” is the indicator diagram rounding coefficient.

#### *2.6.2 Mean Effective Pressure and Effective Power*

The work spent for mechanical losses and pumping work can be considered as mechanical losses mean pressure, and for $z \leq 8$ and $H / D \prec 1$ in gasoline engines;

$$P_{m,m} = 0.039 + 0.0132 . V_{p,m} \tag{18}$$

Where, $V_{p,m}$ specifies the mean piston speed. And for mean effective pressure;

$$P_{me} = P_{mi} - P_{mm} \tag{19}$$

And for the effective power of an engine with *z* cylinders,

$$N_e = \frac{P_{me}.z.V_h.n}{k.60} \tag{20}$$

Where;

*k* **:** for 2 stroke engines=1

*2.6.3 Effective Efficiency and Effective Specific Fuel Consumption*

By using the values available in operations until this point, the following equations can be written for effective efficiency and effective specific fuel consumption;

$$\eta_e = \frac{P_{me}.M_1.R.T_5}{H_u.P_5.\eta_v} \tag{21}$$

$$b_e = \frac{3600}{H_u.\eta_e} \tag{22}$$

*2.6.4 Effective Moment*

$$M_e = \frac{1000.N_e}{w} \tag{23}$$

Whereas;

$$\text{w=Angular velocity=}\frac{\pi.n}{30} \tag{24}$$

The aforementioned operations of the engine correspond to operate at full speed and nominal speed. The cycle calculations are required to be repeated at different speeds in order to determine the changes of characteristics of engine such as effective power, moment and effective efficiency along with rotation speed. For practical calculation, the values in other speeds may be found from values at nominal speed by using some empirical formulas. The characteristics at any "$n_x$" speed may be determined as follows (Kolchin, Demidov, 1984).

$$x = \frac{n_x}{n_N} \tag{25}$$

$$N_{e,x} = N_{e,N}.x.(1 + x - x^2) \tag{26}$$

$$M_{e,x} = \frac{N_{e,x}.1000}{w_x} \tag{27}$$

$$w_x = \frac{\pi.n_x}{30} \tag{28}$$

$$b_{e,x} = b_{e,N}.(1.2 - 1{,}2.x + x^2) \tag{29}$$

$$\eta_{e,x} = \frac{3600}{H_u . b_{e,x}} \tag{30}$$

## 3. RESULTS AND DISCUSSION

Some geometrical characteristics and operation parameters of the engine being simulated had been provided in Table 1.

Table 1. Engine specifications

| | |
|---|---|
| Engine type | Two-stroke , single cylinder, gasoline engine |
| Scavenging type | Reverse-uniflow scavenging |
| Bore x Stroke | 50 mm x 49.5 mm |
| Compression ratio | Changable ~ 6:1-8:1-10:1 |
| Engine speed (rpm) | 1000-3500 |
| Intake valve open - close | 40 deg.BBDC – 80 deg.ABDC |
| Exhaust port open - close | 70 deg.BBDC – 70 deg.ABDC |

In the simulation operation, the changes of compression ratio from 6 to 10 by 2 unit steps and of excess air coefficient from 0.80 to 0.96 with 0.8 unit steps and changes of engine speed from 1000 rpm to 3500 rpm with 500 rpm steps had been calculated. First the general characteristic curves of engine had been examined by keeping the engine speed stable at 3000 rpm, the compression ratio at 6:1 and excess air coefficient at 0.96, and then the effects of compression ratio and excess air coefficient on some characteristic curves of engine had been examined respectively.

### *3.1 Cylinder pressure, temperature and engine characteristic curves*

The change of pressure within the cylinder had been examined according to cylinder volume and crank shaft angle. In figure 2, the change of cylinder pressure according to cylinder volume, and in figure 3, the change of cylinder pressure according to crankshaft angle is being seen.

In Figure 2 and Figure 3, the maximum cylinder pressure has a value of $P_{max}$=33.29 bars. The maximum cylinder pressure is the initiation of combustion towards the end of compression stroke and prior to reaching of piston the top dead center. In here, the purpose is realization of combustion after the top dead center. If the maximum pressure is applied when the crank is perpendicular, crank slots, piston pin or arm may be damaged. Moreover, the generation of knocking is prevented by this means.

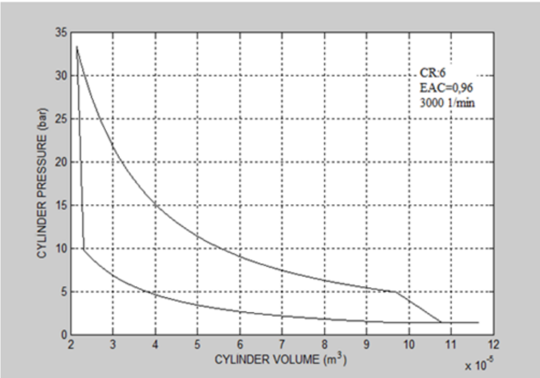


Figure 2. Cylinder pressure variations with cylinder volume

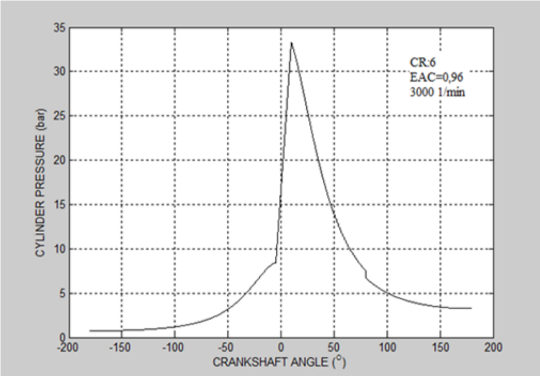


Figure 3. Cylinder pressure variations with crankshaft angle

The change of cylinder gas temperature according to crank shaft angle is being seen in Figure 4. It is possible to see the fast increase in temperature along with combustion. The end of compression temperature at cylinder is $T_2$=620.1 K, and maximum gas temperature is $T_3$=2386 K.

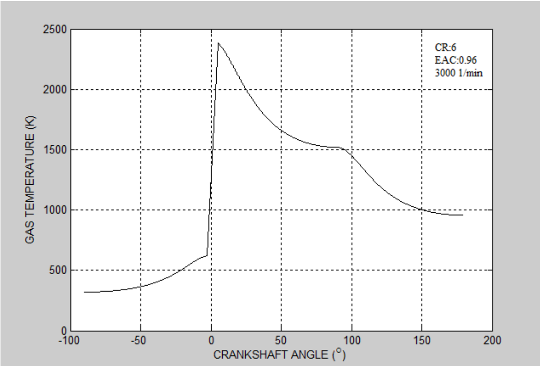


Figure 4. Gas temperature variations with crankshaft angle

Maximum mean effective pressure had been determined as $P_{memax}$=13.83 bars at n=1900 rpm, and maximum mean indicated pressure had been determined as $P_{mimax}$=14.6 bars at $n$=2000 rpm (Fig. 5). As the fill holding efficiency is low at low and high engine speeds, mean indicated and effective pressures are less than the values at optimum engine speed. Along with the increase of engine speed, the losses arising from friction also increases, and an increase is being observed in the difference in between mean indicated and effective pressures.

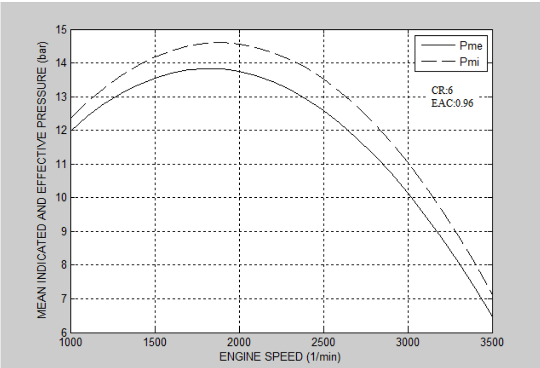


Figure 5. Mean effective and indicated pressure variations with respect to engine speed

As the engine moment is a function of mean pressure in the cylinder, it follows the indicated and effective moment curves with the same parallelism as mean indicated and effective pressure curves. Maximum effective moment had the value of $M_{emax}$=21.4 Nm at $n$=1900 rpm, and maximum indicated moment had the value of $M_{imax}$=22.58 Nm at $n$=2000 rpm (Fig. 6). The engine's moment decreases as the speed of engine increases. The reason of this is that as the rotation speed in a minute increases the intake time decreases. The amount of mixture entering the cylinders decreases and the volumetric efficiency of engine decreases. And at low speeds, more mixture may be transmitted to cylinder while the gas throttle is open.

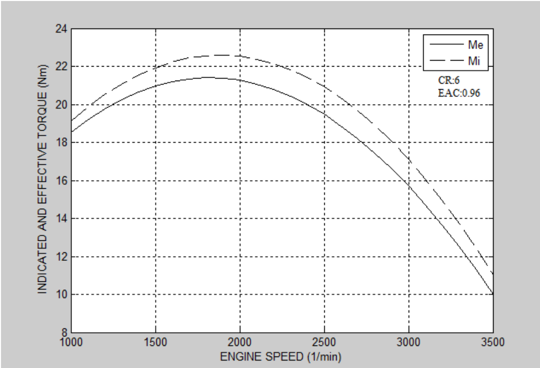


Figure 6. Mean indicated and effective torque variations with respect to engine speed

Maximum effective engine power has the value of $N_{emax}$=5.072 kW at $n$=2800 rpm, and maximum indicated engine power has the value of $N_{imax}$=5.483 kW at $n$=2800 rpm (Fig. 7). The power starts to decrease when the engine speed exceeds 2800 rpm. The volumetric efficiency decreases and frictions increase.

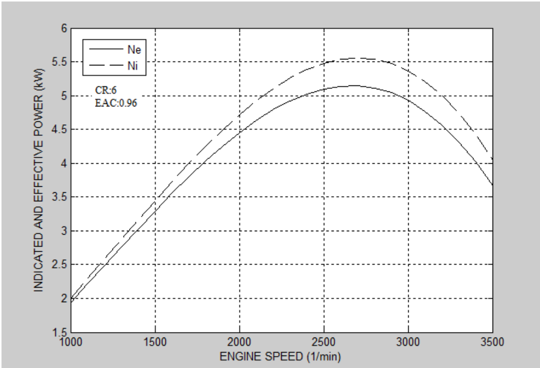


Figure 7. Mean indicated and effective power variations with respect to engine speed

The change of specific fuel consumption as per engine speed is shown in Figure 8. Along with the increase of engine speed, the heat losses in the cylinder decrease and thus the power which can be obtained from the fuel increases. Despite observing a slight decrease in specific fuel consumption along with the increase of engine speed, an increase in specific fuel consumption is being observed at high engine speeds. Minimum effective specific fuel consumption had the value of $b_{emax}$=0.3094 kg/kW-h at $n$=2000 rpm, and minimum indicated specific fuel consumption had the value of $b_{imax}$=0.2929 kg/kW-h at $n$=2000 rpm.

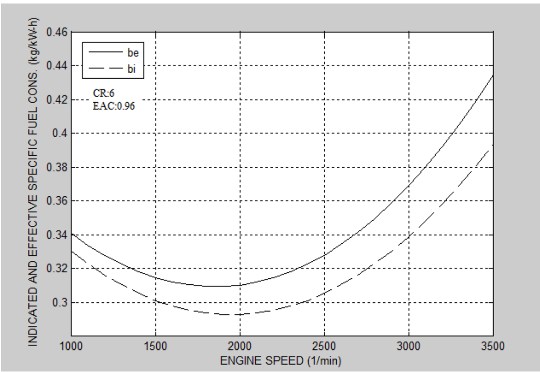


Figure 8. Mean indicated and effective specific fuel consumption variations with respect to engine speed

Maximum effective efficiency had the value of $\zeta_e$=0.3894 at $n$=1800 rpm, and maximum indicated efficiency had the value of $\zeta_i$=0.4097 at $n$=1900 rpm (Fig. 9).

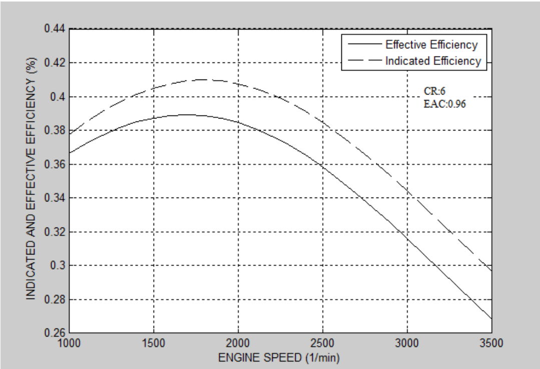


Figure 9. Mean indicated and effective efficiency variations with respect to engine speed

### 3.2 The effect of compression ratio

Cylinder pressure of compression ratio and its effect on gas temperature had been examined for $n$=3000 rpm, $EAC$=0.96 and for three different compression ratios ($\mathcal{E}$=6:1-8:1-10:1). And for engine performance curves, the change of engine speed from $n$=1000 rpm to $n$=3500 rpm had been addressed instead of fix engine speed. Ignition advance had been accepted the same for each compression ratio.

Along with the increase in compression ratio, increase is observed in cylinder pressure and gas temperature. Maximum cylinder pressure had the value of $P_{max}$=33.29 bars when the compression ratio was $\mathcal{E}$ =6:1, and it had the value of $P_{max}$=55.16 bars when the compression ratio increased to $\mathcal{E}$ =10:1 (Fig. 10-11). And maximum gas temperature had the value of $T_3$=2386 K when the compression ratio was $\mathcal{E}$ =6:1, and it had the value of $T_3$=2463 K when the compression ratio was $\mathcal{E}$ =10:1 (Fig. 12). As the net operation of cycle, end of compression pressures and temperatures increase along with the increase of compression ratio, the maximum pressures and temperatures of the cycle also increase.

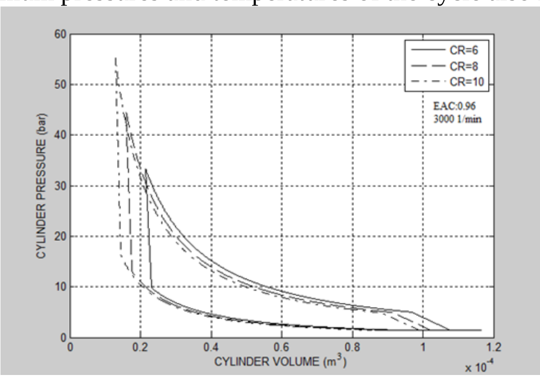


Figure 10. The effect of compression ratio on cylinder pressure

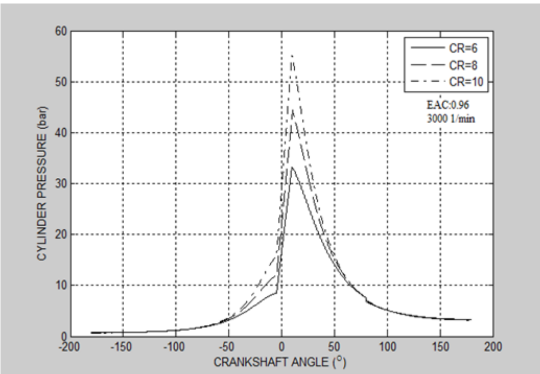


Figure 11. The effect of compression ratio on cylinder pressure

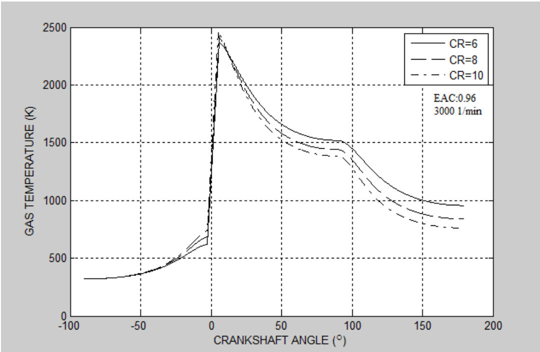


Figure 12. The effect of compression ratio on gas temperature

An increase in mean effective pressure is observed along with the increase of maximum cylinder pressure. Maximum mean effective pressure had the value of $P_{memax}$=13.83 bars at $n$=1900 rpm for $\mathcal{E}$=6:1, and compression ratio had the value of $P_{memax}$=18.84 bars at $n$=1900 rpm for $\mathcal{E}$=10:1 (Fig. 13). As the end of compression and end of combustion pressures and temperature increase along with the increase of compression ratio, the mean effective pressure also increases.

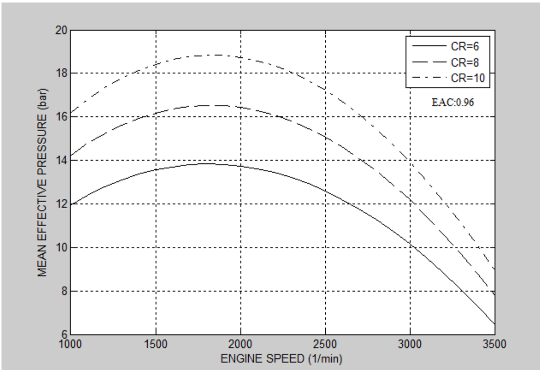


Figure 13. The effect of compression ratio on mean effective pressure

The change of effective moment is also parallel to the change of mean effective pressure. Maximum effective moment compression ratio had the value of $M_{emax}$=21.4 Nm at $n$=1900 rpm for $\mathcal{E}$=6:1, and compression ratio had the value of $M_{emax}$=29.14 Nm at $n$=1900 rpm for $\mathcal{E}$=10:1 (Fig. 14).

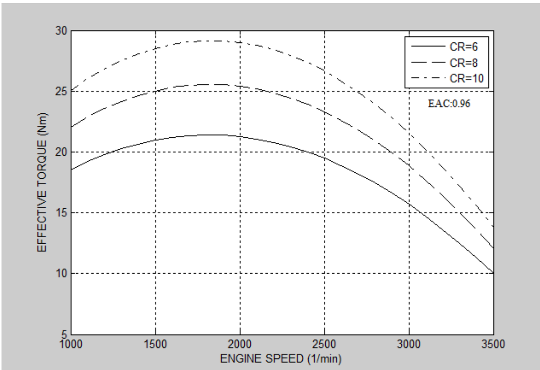


Figure 14. The effect of compression ratio on effective torque

An increase in effective power had been observed along with the increase of compression ratio. Maximum effective power had the value of $N_{emax}$=5.072 kW at $n$=2700 rpm for compression ratio $\mathcal{E}$=6:1, and compression ratio had the value of $N_{emax}$=7.05 kW at $n$=2700 rpm for $\mathcal{E}$=10:1 (Fig. 15). As the end of compression and end of combustion pressures and temperatures increase along with the increase of compression ratio, the effective power also increases.

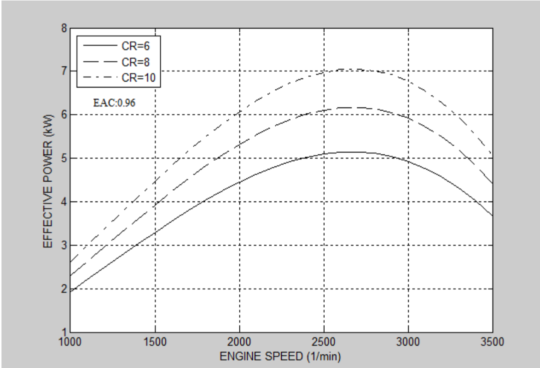


Figure 15. The effect of compression ratio on effective power

The increase observed in power along with the increase of compression ratio is sensed as a decrease in the effective specific fuel consumption. Effective specific fuel consumption compression ratio had the value of $b_{emax}$=0.3094 kg/kW-h at $n$=1900 rpm for $\mathcal{E}$=6:1, and compression ratio had the value of $b_{emax}$=0.2103 kg/kW-h at $n$=1900 rpm for $\mathcal{E}$=10:1 (Fig. 16). This condition arises from the fact that the possibility of volumetric efficiency and knocking operation is high at low engine speeds, and that the possibility of knocking disappears due to high turbulence, short heat transfer period and low volumetric efficiency at high speeds.

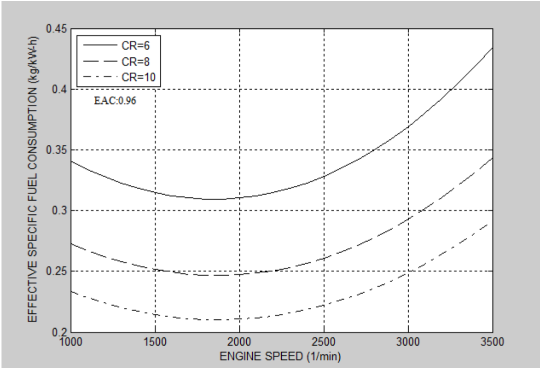


Figure 16. The effect of compression ratio on effective specific fuel consumption

Effective efficiency had the value of $\zeta_e$=0.3894 at $n$=1700 rpm for $\mathcal{E}$=6:1 along with the increase of compression ratio, and compression ratio had the value of $\zeta_i$=0.5722 at $n$=1700 rpm for $\mathcal{E}$=10:1 (Fig. 17).

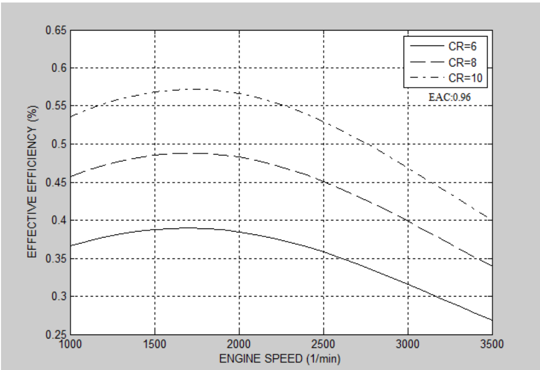


Figure 17. The effect of compression ratio on effective efficiency

### *3.3 The effect of excess air coefficient*

The change of cylinder pressure and gas temperature had been examined for $n$=3000 rpm, compression ratio $\mathcal{E}$=10:1 and for three different excess air coefficients (*EAC*=0.80-0.88-0.96).

Maximum cylinder pressure had been $P_{max}$=55.26 bars for *EAC*=0.80 value of the excess air coefficient, and maximum cylinder pressure had been $P_{max}$=55.16 bars for *EAC*=0.96 value (Fig. 18-19). Due to energy decrease caused by the decrease of fuel amount taken in due to increase of excess air coefficient and thus depletion of mixture, the maximum pressure and temperature values within cylinder decrease.

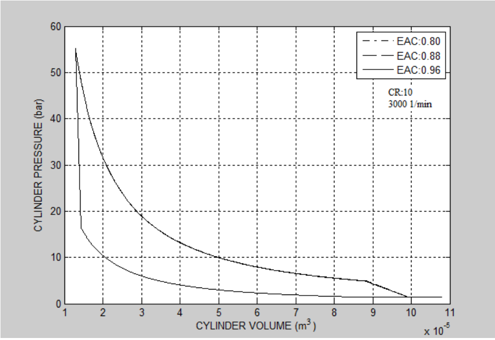


Figure 18. The effect of EAC on cylinder pressure

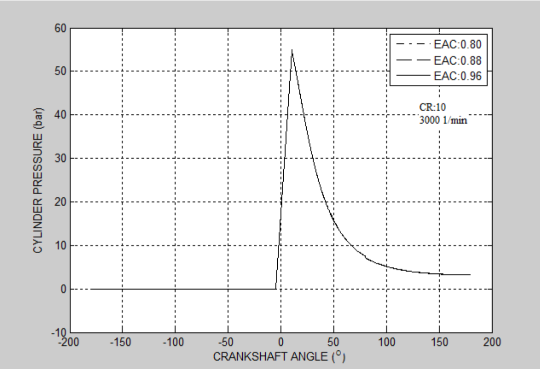


Figure 19. The effect of EAC on cylinder pressure

Along with the increase of excess air coefficient, the change in gas temperature is clearer. Maximum gas temperature has the value of $T_3$=2524 K for excess air coefficient *EAC*=0.80, and maximum gas temperature has the value of $T_3$=2463 K for excess air coefficient *EAC*=0.96 (Fig. 20).

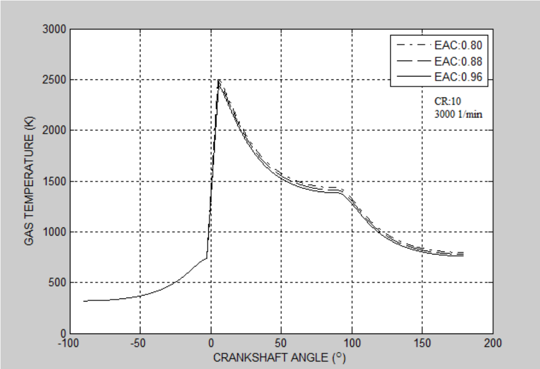


Figure 20. The effect of EAC on gas temperature

The maximum value of mean effective pressure had been calculated as $P_{memax}$=18.96 bars at *n*=1800 rpm for *EAC*=0.80 excess air coefficient, and it had been calculated as $P_{memax}$=18.84 bars at *n*=1800 rpm for *EAC*=0.96 excess air coefficient (Fig. 21).

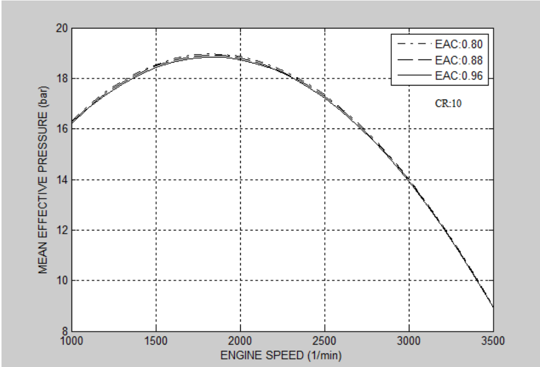


Figure 21. The effect of EAC on mean effective pressure

The maximum effective moment had the value of $M_{emax}$=29.33 Nm at $n$=1800 rpm for $EAC$=0.80 excess air coefficient, and it had the value of $M_{emax}$=29.14 Nm at $n$=1800 rpm for $EAC$=0.96 excess air coefficient (Fig. 22).

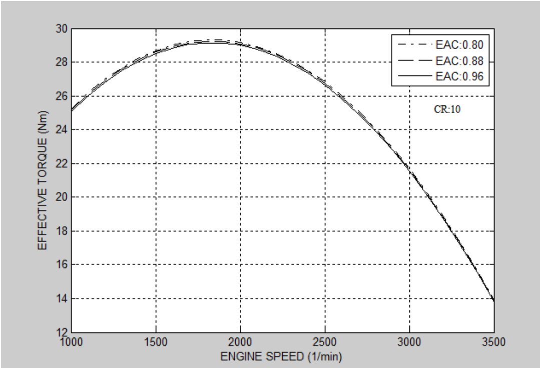


Figure 22. The effect of EAC on effective torque

The maximum effective engine power had the value of $N_{emax}$=7.096 kW at $n$=2700 rpm for $EAC$=0.80 excess air coefficient, and it had the value of $N_{emax}$=7.05 kW at $n$=2700 rpm for $EAC$=0.96 excess air coefficient (Fig. 23).

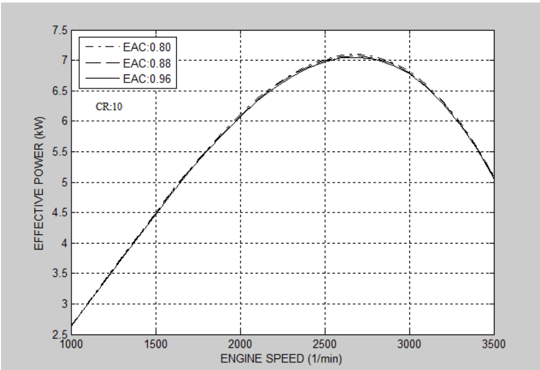


Figure 23. The effect of EAC on effective power

The minimum effective specific fuel consumption had the value of $b_{emax}$=0.2566 kg/Kw-h at $n$=1900 rpm for $EAC$=0.80 excess air coefficient, and it had the value of $b_{emax}$=0.2103 kg/kW-h at $n$=1900 rpm for $EAC$=0.96 excess air coefficient (Fig. 24).

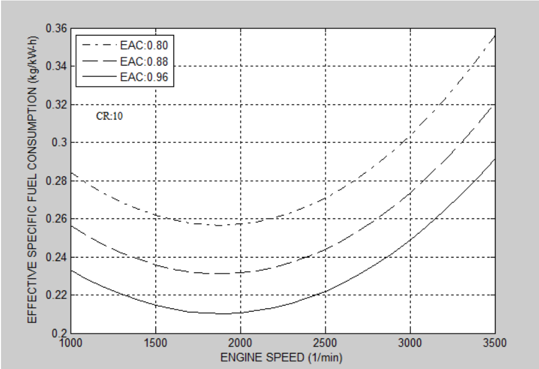


Figure 24. The effect of EAC on effective specific fuel consumption

As the excess air coefficient increases, an increase in effective efficiency is observed. The maximum effective efficiency had the value of $\zeta_e$=0.5604 at $n$=1700 rpm for $EAC$=0.80 excess air coefficient, and it had the value of $\zeta_e$=0.5722 at $n$=1700 rpm for $EAC$=0.96 excess air coefficient (Fig. 25). As the mixture depletes and as the fill approaches to ideal gas characteristics as the excess air coefficient increases and as it is benefited from the fuel within the cylinder at maximum degree, an increase in effective efficiency is being observed.

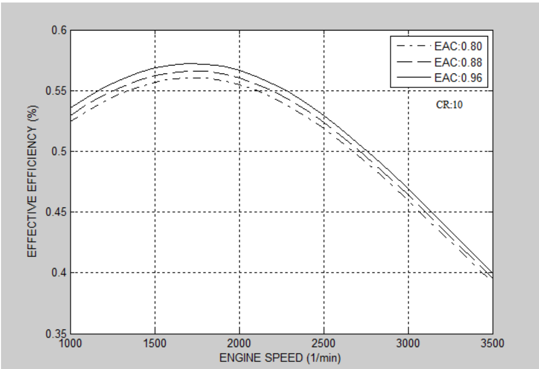


Figure 25. The effect of EAC on effective efficiency

## 4. CONCLUSIONS

In this study, a simulation program had been developed by using MATLAB programming language for the examination of the cycle of a two-stroke, reverse uniflow, single cylinder, and spark ignition engine. Each of the cycle operations had been calculated for CA changes of 1° and for various parameters, and as the result of calculations, graphics were drawn for cylinder pressure and temperature, mean indicated and effective pressure, indicated and effective moment, indicated and effective power, indicated and effective specific fuel consumption and indicated and effective efficiency values. By these graphics, the changes of cylinder pressure and gas temperature, and effects of compression ratio and excess air coefficient on the engine performance had been examined.

As the compression ratio increases, maximum cylinder pressures and temperatures increase. By the increase of maximum cylinder pressure, increases are observed in mean indicated and effective pressure, indicated and effective moment, indicated and effective power and indicated and effective efficiency. And the indicated and effective specific fuel consumption values decrease and become more economic.

Along with the increase of excess air coefficient, decreases had been observed in maximum cylinder pressures and temperatures, mean indicated and effective pressure values, engine moment, engine power and specific fuel consumption. And an increase in thermal efficiency was observed along with the decrease of specific fuel consumption.

**NOMENCLATURE**

| | | | |
|---|---|---|---|
| b | Specific fuel consumption (kg/kWh) | $\mu$ | Molecular change coefficient |
| CA | Crankshaft angle (deg) | | |
| | | ***Greek symbols*** | |
| H | Stroke (m) | $\lambda$ | Pressure increase ratio |
| $H_u$ | Lower calorific value of fuel (kJ/kg) | $\gamma_r$ | coefficient of waste exhaust gases |
| M | Torque (Nm) | $\phi$ | Indicator diagram rounding coefficient |
| n | Engine speed (rpm) | $\eta$ | Efficiency (%) |
| P | Pressure (bar) | $\rho$ | Density ($kgm^{-3}$) |
| N | Engine power (kW) | $\beta$ | Coefficient of entrapment losses |
| Q | Heat (kJ) | $\zeta$ | Coefficient of friction losses |
| R | Universal gas constant (kJ/kg K) | $\xi$ | Coefficient of benefiting from heat in combustion |
| T | Temperature (K) | $\varepsilon$ | Compression ratio |
| z | Number of cylinders | ***Subscripts*** | |
| w | Angular velocity (rad/sn) | e | effective |
| $\varphi$ | Recharging coefficient | i | indicated |
| $\alpha$ | Excess air coefficient | m | Mean |

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