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THERMODYNAMIC PERFORMANCE OPTIMIZATION OF AN OVER-EXPANDED DIESEL CYCLE: A THEORETICAL STUDY

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ABSTRACT

Heat engines are most widely used machines for transportation and producing energy. Almost 80% of the energy in the world is obtained from fossil fuels. During the last decade, efficient use of energy became very important and many rules are introduced to achieve energy efficiency. Therefore the efficiency of heat engines such as internal combustion engines should be paid more attention. Recently, heat engine cycles such as diesel cycle have been combined with other cycles for increasing the efficiency of internal combustion engines. In this paper, a thermodynamic performance analysis and optimization of an over-expanded Diesel cycle is studied. These studies based on cycle parameters. Also, an optimization function which is based on power and compression ratio is defined which haven't been studied yet. Even more, a new parameter which is called over-expansion ratio is defined and included in the optimization function. The influence of over-expansion ratio to power output and thermal efficiency are inspected for different compression ratios, the maximum temperature ratios, and the cut-off ratios. One of the most important outputs of the study is that the theoretical maximum efficiency can be increased to 70%.

Keywords: Energy, Thermodynamics, Thermal efficiency, Over-expanded diesel cycle, Optimization

1. INTRODUCTION

In 1876 the heat engine was invented by Nikolaus Otto as a spark ignition (gasoline) engine. Almost two decades later the Diesel engine cycle was invented by Rudolph Diesel in 1893. It is also called compression engine due to injection of fuel into compressed air in a cylinder and combustion takes place in constant pressure. It is well known that the maximum thermal efficiency of contemporary heat engines is around 50% and a hybrid car engine can reach 60% [1, 2]. Besides, the combination of different cycles have been studied to increase the efficiency of heat engines such as Atkinson-Diesel cycle [3-7], Miller-Otto cycle[8], Otto-Atkinson cycle [9-12] etc.

Some studies have been performed on thermodynamic analysis of over-expanded Diesel cycle which is also called Miller-Diesel cycle. In the study of Wu [13] which is about basic design and optimization of thermodynamic cycles, the Miller-Diesel cycle is briefly explained. Schutting et al. [7] implemented both Atkinson and Miller cycles on a turbocharged Diesel engine of a passenger car. They investigated the NO_x emissions from these cycles. Dembinski and Lewis [14] focused on the problem of emissions (CO_2, NO_x, and smoke) and showed the emission reduction by applying Miller cycle on a Diesel engine. Martins et al. [15] carried out the thermodynamic analysis of an over-expanded cycle. They performed analyses and comparison on Otto cycle, Diesel cycle, Dual cycle, and Miller cycle especially at partial load conditions. This study is mainly focused on spark ignition engines (SI). As the result of this study, over-expanded SI engine reached the highest efficiency about 73% at light load or part load situations. Martins et al. [16] improved their study by inspecting the in- cylinder swirl analysis to different over-expanded cycles of hybrid vehicles. They created a computer model in GAMBIT and used this model for FLUENT simulations. They measured the swirl on a SI engine working under the Otto cycle and the Miller cycle, with early and late intake valve closure. They concluded that the Miller cycle with late intake valve closure was the one with better swirl conditions. Sher and Bar-Kohany [17] studied the

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performance of a commercial unthrottled SI engine installed with variable valve timing. They searched for optimal valve timing strategies for maximizing engine torque in terms of the exhaust opening, intake opening and intake closing timings. One of the important results of their study is that the maximum torque at any engine load, is shifted towards a lower engine speed by application of variable valve timing.

Gonca et al. [18] investigated the effect of late closing of intake valve by 10 and 20 crank degrees on exhaust emissions and efficiency. They resulted out that Miller cycled diesel engine is more efficient and clean whereas the produced power output is less than standard diesel cycle. The authors [19] similarly studied reduction of exhaust emissions by late closing of intake valve by 30 crank angle and steam injection to combustion chamber.

In this paper, thermodynamic performance analysis and optimization of timing of air inlet valve for an over-expanded Diesel cycle is studied. A novel optimization function based on power and compression ratio is defined. Even more, over-expansion ratio (ψ) is introduced and included in the optimization function. The relations between the over-expansion ratio and the power output, and the over-expansion ratio and the thermal efficiency are inspected for different compression ratio, the maximum temperature ratio, and the cut-off ratio. Furthermore, optimum range for over-expansion ratio, compression ratio, thermal efficiency and power output are illustrated.

2. THE THEORY AND THE THERMODYNAMIC MODEL OF THE CYCLE

The theoretical cycle of an over-expanded Diesel cycle is illustrated in Figure 1. From the figure it is seen that the cycle is very similar to classical diesel cycle. However, its most important difference is existence of a constant pressure heat rejection process (process between 5-1) before compression. This difference result in increase of efficiency comparing to the classical diesel cycle. The thermal efficiency of an over-expanded Diesel cycle may reach 60% [5, 6].

In the cycle which is illustrated in Figure 1, the process between 1 and 2 is isentropic compression which is actually a negative work. Between 2 and 3, the heat input is realized by combustion process under constant pressure and an expansion takes place. Then, the exhaust gases continue to expand between 3 and 4 which is also an isentropic process. The heat rejection takes place under constant volume, between 4 and 5. And it continues between 5 and 1 while the volume starts to decrease under constant pressure.

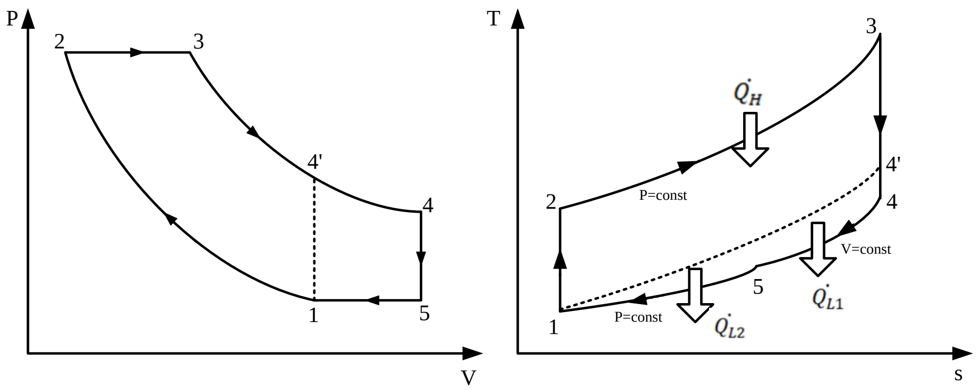

Figure 1. (a) P-V and (b) T-S diagrams of an over-expanded Diesel cycle

Although it is partially similar to the standard diesel cycle (1-2-3-4'-1), the most important difference is that there is a constant pressure heat rejection phase (process 5-1) which does not exist in standard diesel cycle. This phase starts at point 5 and ends at point 1 in Figure 1. In this figure we can see the difference from standard diesel cycle more clearly. In standard diesel cycle, the phase between 5 and 1 does not exist. Instead, the clean air is started to be charged and compressed from point 5 which is the starting point of a new cycle. This additional phase may result in an increase in thermal efficiency comparing to standard diesel cycle. In over-expanded Diesel cycle, the thermal efficiency may be increased up to 60% [5, 6].

The calculations of all of the processes of over-expanded Diesel heat engine are listed below. The equation for calculating the heat input of the cycle is:

$$\dot{Q}_H = C_p(T_3 - T_2) \tag{1}$$

where C_p is constant pressure heat capacity of working fluid which is calculated as $\dot{m} \cdot c_p$ in the unit of (kW/K). The heat output of the cycle is fulfilled in two steps: first under constant volume and then under constant pressure. The heat output equations during these two processes are expressed as:

$$\dot{Q}_{L1} = C_v(T_4 - T_5) \tag{2}$$

$$\dot{Q}_{L2} = C_p(T_5 - T_1) \tag{3}$$

$$\dot{Q}_L = \dot{Q}_{L1} + \dot{Q}_{L2} \tag{4}$$

where C_v is constant volume heat capacity of the working fluid which is calculated as $\dot{m} \cdot c_v$ in the unit of (kW/K). The power output of the cycle is calculated from the following equation considering the input and output heats.

$$\dot{W} = \dot{Q}_H - \dot{Q}_L \tag{5}$$

Substituting Eq.(1), Eq.(2) and Eq.(3) in Eq.(5), the power equation becomes:

$$\dot{W} = C_p(T_3 - T_2) - C_v(T_4 - T_5) - C_p(T_5 - T_1)\,. \tag{6}$$

Using the definition $\gamma = c_P / c_V$ which is adiabatic exponent, the power equation can be organized as follows.

$$\dot{W} = C_v T_1 \left[\gamma \left(\frac{T_3}{T_1} - \frac{T_2}{T_1} - \frac{T_5}{T_1} + 1 \right) - \left(\frac{T_4}{T_1} - \frac{T_5}{T_1} \right) \right] \tag{7}$$

Then the non-dimensional power equation takes the following form:

$$\bar{\dot{W}} = \frac{\dot{W}}{C_v T_1} = \left[\gamma \left(\frac{T_3}{T_1} - \frac{T_2}{T_1} - \frac{T_5}{T_1} + 1 \right) - \left(\frac{T_4}{T_1} - \frac{T_5}{T_1} \right) \right]. \tag{8}$$

It is possible to write the temperature relations of the cycle using the second law of thermodynamics as:

$$\Delta S_{23} = \Delta S_{45} + \Delta S_{51} \tag{9}$$

$$c_p \ln \frac{T_3}{T_2} = c_v \ln \frac{T_4}{T_5} + c_p \ln \frac{T_5}{T_1} \tag{10}$$

Reorganizing the Eq.(10), we can express the temperature relations as:

$$T_4\left(T_2T_5\right)^{\gamma}=T_5\left(T_3T_1\right)^{\gamma} \tag{11}$$

After some more manipulation, the equation takes its final form as:

$$\frac{T_4}{T_1}=\frac{T_5}{T_1}\left(\frac{T_3T_1}{T_2T_5}\right)^{\gamma} \tag{12}$$

Besides, the temperature relations can be expressed in terms of cycle's parameters such as isentropic temperature ratio of compression, isentropic temperature ratio of expansion, maximum temperature ratio, cut off ratio, and over-expansion ratio as below. The cycle's isentropic temperature ratio of compression (ϕ_c) is:

$$\phi_c=\frac{T_2}{T_1}=\left(\frac{V_1}{V_2}\right)^{\gamma-1}=\left(r_c\right)^{\gamma-1} \tag{13}$$

where r_c is compression ratio which is formulized as:

$$r_c=\frac{V_1}{V_2}\,. \tag{14}$$

The isentropic temperature ratio of expansion (ϕ_e) is:

$$\phi_e=\frac{T_3}{T_4}=\left(\frac{V_4}{V_3}\right)^{\gamma-1}=\left(r_e\right)^{\gamma-1} \tag{15}$$

where r_e is expansion ratio which can be expressed as:

$$r_e=\frac{V_4}{V_3}\,. \tag{16}$$

The maximum temperature ratio of the cycle (α) is:

$$\alpha=\frac{T_{\max}}{T_{\min}}=\frac{T_3}{T_1} \tag{17}$$

Cut off ratio (β) is:

$$\beta=\frac{V_3}{V_2}=\frac{T_3}{T_2} \tag{18}$$

Over-expansion ratio (ψ) is:

$$\psi=\frac{V_5}{V_1}=\frac{T_5}{T_1}\,. \tag{19}$$

The relation between the maximum temperature ratio and the cut off ratio of the cycle is expressed as:

$$\alpha=\beta\phi_c\,. \tag{20}$$

Besides, it is possible to express Eq.(12) in terms of over-expansion ratio and cut off ratio as follows:

$$\frac{T_4}{T_1}=\frac{T_5}{T_1}\left(\frac{T_3T_1}{T_2T_5}\right)^{\gamma}=\psi\left(\beta/\psi\right)^{\gamma} \tag{21}$$

Substituting Eq.(15) and Eq.(17) in Eq.(21), we can obtain the maximum temperature ratio (α) as:

$$\alpha=\phi_e\left(\beta^{\gamma}/\psi^{\gamma-1}\right). \tag{22}$$

The relation between the isentropic temperature ratio of compression and isentropic temperature ratio of expansion can be expressed using Eq.(20) and Eq.(22) as below:

$$\phi_c=\phi_e\left(\beta/\psi\right)^{\gamma-1} \tag{23}$$

Assigning the equations above in main power equations (Eq.(7) and Eq.(8)), the power and non-dimensional power equations can be reorganized as:

$$\dot{W}=C_vT_1\left[\gamma\left(\alpha-\phi_c+1\right)-\psi\left(\left(\beta/\psi\right)^{\gamma}+\gamma-1\right)\right] \tag{24}$$

$$\bar{\dot{W}}=\left[\gamma\left(\alpha-\phi_c+1\right)-\psi\left(\left(\beta/\psi\right)^{\gamma}+\gamma-1\right)\right] \tag{25}$$

The thermal efficiency of the cycle is,

$$\eta_{th}=1-\frac{\dot{Q}_L}{\dot{Q}_H}=\frac{\dot{W}}{\dot{Q}_H} \tag{26}$$

Substituting the main power equation (Eq.(24) and heat input equation (Eq.(1)) in thermal efficiency equation (Eq.(26)), the obtained thermal efficiency equation is:

$$\eta_{th}=\frac{\gamma\left(\alpha-\phi_c+1\right)-\psi\left(\left(\beta/\psi\right)^{\gamma}+\gamma-1\right)}{\gamma\left(\alpha-\phi_c\right)} \tag{27}$$

Therefore, the non-dimensional power and thermal efficiency equations are expressed using the cycle parameters $\alpha,\beta,\gamma,\phi_c,\psi$. It is possible to express this as a function below:

$$\bar{\dot{W}}=f\left(\alpha,\beta,\gamma,\phi_c,\psi\right) \text{ and } \eta=f\left(\alpha,\beta,\gamma,\phi_c,\psi\right). \tag{28}$$

The objective function for the optimization of the cycle is described as:

$$F=\dot{W}/r_c \tag{29}$$

For this purpose the partial derivative of the objective function is calculated $\left(\partial\dot{W}/\partial r_c=0\right)$. According to the result of this derivation, the optimum isentropic compression ratio which maximizes the cycle power is obtained as:

$$\left(r_c\right)^* = \left(\alpha+1\right) - \frac{\psi}{\gamma}\left(\left(\beta/\psi\right)^{\gamma} + \gamma - 1\right) \tag{30}$$

Finally, assigning the optimum isentropic compression ratio $\left(r_c\right)^*$ in non-dimensional power equation (Eq.(25)) and thermal efficiency equation (Eq.(27)), the maximum power and maximum thermal efficiency of the cycle are obtained as:

$$\bar{\dot{W}}_{\max} = 2\psi\left(\left(\beta/\psi\right)^{\gamma} + \gamma - 1\right) \tag{31}$$

$$\eta_{th,\max} = \frac{2\psi\left(\left(\beta/\psi\right)^{\gamma} + \gamma - 1\right)}{\psi\left(\left(\beta/\psi\right)^{\gamma} + \gamma - 1\right) - \gamma}. \tag{32}$$

3. RESULTS AND DISCUSSION

In this section, the thermodynamic performance analysis of the cycle is carried out numerically using the theoretical performance equations given in previous section. The performance analysis of the cycle power and thermal efficiency is performed for various values of cycle parameters.

The effects of over-expansion ratio at various isentropic compression ratios to cycle's power and efficiency are shown in Figure 2. It can be clearly seen from the figure that as the over-expansion ratio increases, the cycle's power and thermal efficiency are increasing to an optimum value and then starts decreasing. Another point is that the power output and thermal efficiency of the cycle increases with the increase in compression ratio. The over-expansion ratio which maximizes the power and thermal efficiency is constant. For example, for various r_c values, the over-expansion ratio which maximizes the power output and thermal efficiency is ψ=3. Besides, the maximum value of thermal efficiency and power output are different depending upon the over-expansion ratio. In Figure 2.a we can observe that the maximum power output ($\dot{W}_{\max}$) for ψ =3.0 is 1227 kW for r_c=16 and 1398kW for r_c=20.

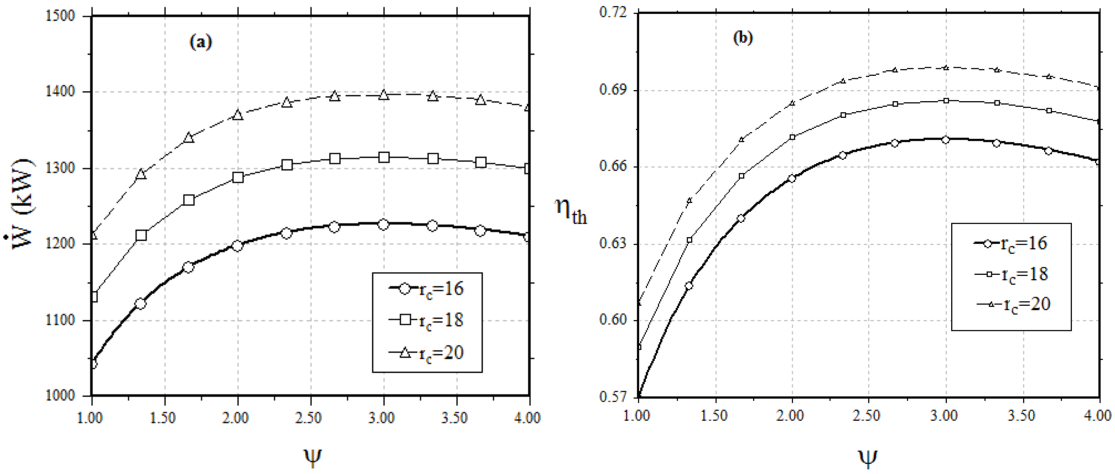

Figure 2. The effect of over-expansion ratio for different compression ratios (β=3) (a) to cycle's power output, (b) to cycle's thermal efficiency.

The variation of thermal efficiency due to the over-expansion ratio is shown in Figure 2.b. Here, the over-expansion ratio which maximizes the thermal efficiency is also ψ =3.0 for the different values of compression ratio. For example, the maximum thermal efficiency ($\eta_{th,max}$) for r_c=16 is 0.67 and for r_c=20 is 0.70 which means it is a super-efficient engine. In brief, while the compression ratio influences the power output and thermal efficiency, both of them reach optimum values for an identical over-expansion ratio.

The relation of over-expansion ratio with the cycle's power and thermal efficiency at different maximum temperature ratios (α) are shown in Figure 3. The power output reaches an optimum value and then slowly decreases. In Figure 3.a it is clearly seen that the maximum temperature ratio has a great positive effect on power output. For example, for ψ =1.89 which maximizes the power output for α=6.0, the maximum power output $\dot{W}_{\max}$ is 562.90 kW. For the same over-expansion ratio, the power output decreases to 347.90 kW for α=5.0 and 125.87 for α =4.0. By the increase in the maximum temperature ratios, the over-expansion ratio to reach the maximum power is slightly increasing. However, the maximum power values indicate a very large increase. When the maximum temperature ratio is α=4 the power reaches the maximum value of $\dot{W}_{\max}$=142.20 kW for the over-expansion ratio of ψ =1.28. However, increasing the maximum temperature ratio to α=5.0, the power reaches the maximum value of $\dot{W}_{\max}$=352.70 kW for the over-expansion ratio of ψ =1.47.

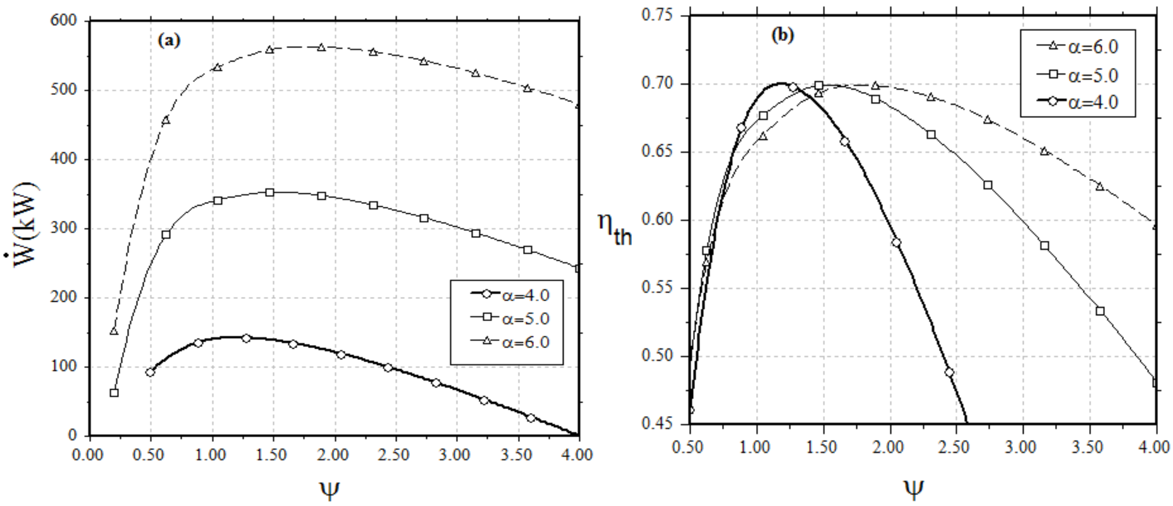

Figure 3. The effect of over-expansion ratio for different maximum temperature ratios (r_c=20) (a) to cycles power output, (b) to cycles thermal efficiency.

In Figure 3.b the effect of over-expansion ratio to thermal efficiency for different maximum temperature ratios are shown. In the figure, for each maximum temperature ratio (a curve is assigned), the thermal efficiency increases to an optimum value rapidly while the over-expansion ratio s increases. After the optimum point, it decreases despite the increase in the over-expansion ratio. It is observed that the maximum thermal efficiency is reached at higher over-expansion ratio s for higher maximum temperature ratios. Another observation is the maximum value of the thermal efficiency which is not varying for different maximum temperature ratios. In other words, the maximum value of thermal efficiency is constant and it is $\eta_{th,\max} = 0.70$. This value is reached for maximum temperature ratio values of 4, 5 and 6 while the over-expansion ratio s are ψ =1.28, 1.47, and 1.89, respectively.

The relation of power output with over-expansion ratio for different cut-off ratio (β) is given in Figure 4.a. The power output slightly increases and reaches a maximum with the increase in over-expansion ratio until a particular value and then becomes stabilized at this value despite the increase in over-expansion ratio. Comparing this for different cut-off ratios, we observe that the power output increases with the increase in cut-off ratio. The mentioned observations above are valid for each different cut-off ratio. In other words, while the power output increases with an increase in cut-off ratio, it will reach a maximum value and becomes nearly constant after a particular over-expansion ratio. It means that the increase of over-expansion ratio after this value does not help increase in power output. The stabilized power output value is different for each cut-off ratio. For instance, for β=3.0 the power output is stabilized at $\dot{W}$ = 1396 kW for ψ =3.0. After this point there is not a significant change in power output which is around 10 kW. This situation is similar for other cut-off ratios.

In Figure 4.b the relation of thermal efficiency and over-expansion ratio is displayed for several β values. Here, it is observed that the thermal efficiency is decreasing by the increase in β values. This situation is valid for different β values until a particular over-expansion ratio, but after then the thermal efficiency characteristics are changing. For instance, for the point that over-expansion ratio is 3.42, the thermal efficiency which belongs to β=3.0 decreases and intersects with the thermal efficiency of β=4.0. However, this value is higher than the thermal efficiency which belongs to β=5.0. When ψ=3.58, the thermal efficiency which belongs to β=3.0 continues to decrease and becomes equal with the thermal efficiency value (which belongs to) β=5.0. At this point, the thermal efficiency of β=4.0 reaches a higher value which is the highest point for β=4.0. This situation can be expressed more clearly with equations as follows:

$$0 \leq \psi < 3.42 \qquad \eta_{\beta=3.0} > \eta_{\beta=4.0} > \eta_{\beta=5.0}$$
$$3.42 \leq \psi < 3.82 \qquad \eta_{\beta=4.0} > \eta_{\beta=3.0} > \eta_{\beta=5.0}$$
$$\psi \geq 3.82 \qquad \eta_{\beta=4.0} > \eta_{\beta=5.0} > \eta_{\beta=3.0}$$

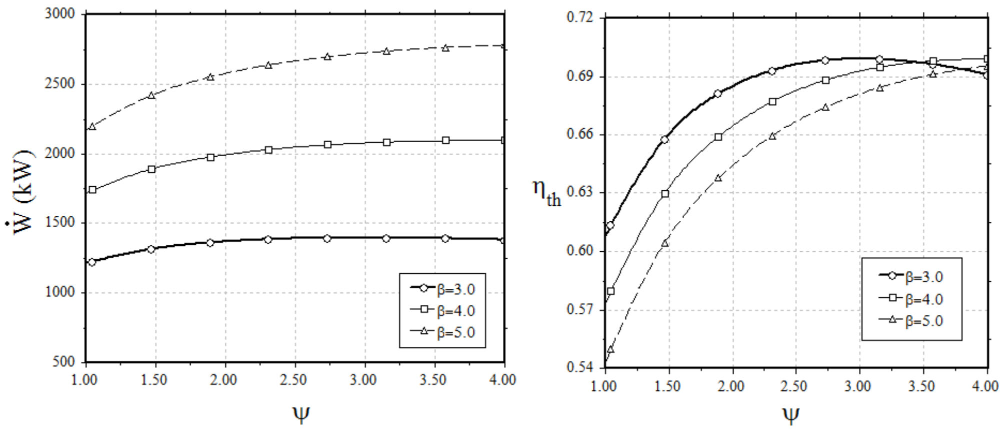

Figure 4. (a) The relation of power output and over-expansion ratio for different cut-off ratios, (b) the relation of over-expansion ratio and thermal efficiency for different β values.

The maximum thermal efficiency value is identical for all β values which are 0.70 and this efficiency is reached for different β values at different over-expansion ratio. For instance, the maximum thermal efficiency is reached at ψ=3.16 for β =3. We have to say that, the differences of thermal efficiency values corresponding to these over-expansion ratios are ignorable. That's why the β value which is easier to obtain can be preferred.

In Figure 5, the relation of compression ratio versus power output and thermal efficiency are displayed for various over-expansion ratios. Either power outputs or thermal efficiencies perform a similar characteristic for different compression ratio and over-expansion ratio (Fig 6). As the compression ratio increases, the power output (Figure 5.a) and the thermal efficiency (Figure 5.b) is increasing by the increase in over-expansion ratio. For a practical compression ratio of r_c=20 and ψ=3.0 the obtained thermal efficiency is $\eta_{th,\max} = 0.70$ and power output is $\dot{W}$ = 1751 kW. One important point is that as the over-expansion ratio increases the rate of increase of thermal efficiency and power output is decreasing.

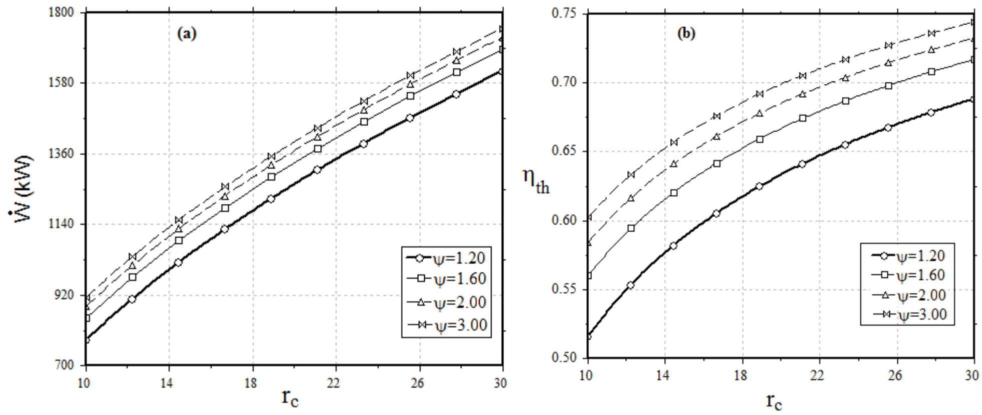

Figure 5. The relation of (a) compression ratio and power, (b) compression ratio and thermal efficiency for various over-expansion ratios.

The relation between the power output and thermal efficiency is given in Figure 6. From the figure it can be seen that as the over-expansion ratio increases, the thermal efficiency increases and the power output decreases. For instance, for the power output value $\dot{W}$ =1300 kW, the thermal efficiencies are η_{th}=0.64 and η_{th}=0.68 for ψ=1.2 and ψ=2.0, respectively. On the other side, for the thermal efficiency value of η_{th}=0.64, the power outputs are $\dot{W}$ =1300 kW and $\dot{W}$ =1126kW for ψ=1.2 and ψ=2.0, respectively.

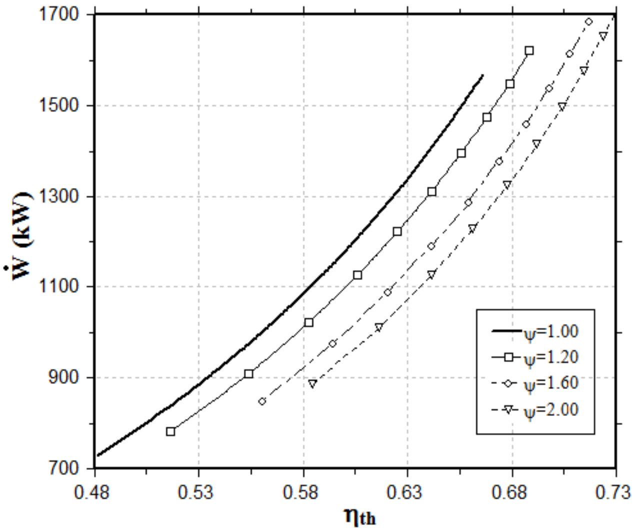

Figure 6. The relation of thermal efficiency and power output for various over-expansion ratios

4. CONCLUSION

One of the most important researches about energy is increasing the efficiency of internal combustion engines such as diesel engines due to the fact that 80% of the worlds' energy is produced in this way. That's why the heat engine studies try to obtain cycles closer to Carnot heat engine. Over-expanded Diesel cycle is a method to approximate the Diesel cycle to Carnot heat engine. The contemporary diesel engines efficiency can be increased to about 55% by cogeneration. In case of over-expanded Diesel cycle, the efficiency can theoretically reach 70% even without cogeneration. In Figure 7 the comparison of Diesel cycle and over-expanded Diesel cycle are illustrated which depend on over-expansion ratio. Power and thermal efficiency of a heat engine are reversely related parameters. The change of power and thermal efficiency by over-expansion ratio are individually inspected. One of the outcomes of this study is that the over-expansion ratios at which the maximum power and the maximum thermal efficiency obtained are calculated. This calculation can be interpreted as finding the optimum location of point one in over-expanded Diesel cycle (Figure 1) which is very important. It is possible to obtain theoretically higher efficiency values using unfeasible parameters. However, in this study the practically feasible parameter values are considered. For instance, the thermal efficiency of η_{th}=0.66 is reached at a practically feasible over-expansion ratio of ψ=3 which is shown in Figure 7. The previous studies [5, 6] reached the maximum thermal efficiency of $\eta_{th,max}$ =0.60. In this study, it is improved to $\eta_{th,max}$ =0.68 by optimization of cycle parameters.

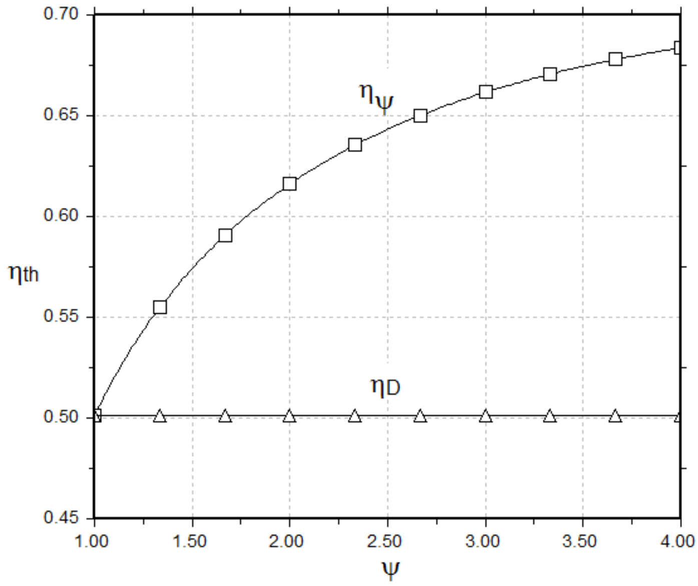

Figure 7. Comparison of Diesel cycle and over-expanded Diesel cycle depend on over-expansion ratio

Another important parameter which influences the over-expansion ratio is maximum temperature ratio. It is observed that increase in maximum temperature ratio results in an increase in over-expansion ratio. That's why it should be considered for designing feasible cycle parameters.

Another cycle parameter is cut-off ratio. The results of the study show that the cut-off ratio is more effective on power output of the cycle than thermal efficiency. While the power output increases with an increase in cut-off ratio, it will reach a maximum value and becomes constant after a particular over-expansion ratio. It means that the increase of over-expansion ratio after this value does not help increase in power output.

The mentioned cycle parameters are interrelated and they affect the over-expansion ratio. This induces the necessity of optimization of over-expansion ratio with respect to the other cycle parameters which is the topic of this paper.

The results of the study show that the over-expanded Diesel cycle is a promising method for increasing the efficiency of a Diesel engine. There are very few related studies in the literature and more research should be carried out. The experimental study of this paper is a future study of the authors. In that study, it is planned to analyze the over-expansion to a marine diesel engine with the engine specifications shown in Table 1.

Table 1. Specifications of the marine diesel engine.

Type	Sulzer 12RTA 84 C	-
Bore	84	cm
Stroke	240	cm
Cylinders	12	-
Engine speed	102	rpm (full load)
Power	42.85	MW (full load)
sfoc	168	g/kW.h (full load)

Abbreviations

P	pressure
V	volume
T	temperature
s	entropy
$\dot{Q}_H$	heat input
$\dot{Q}_L$	heat output
C_p	constant pressure heat capacity
C_v	constant volume heat capacity
$\dot{W}$	power output
$\overline{\dot{W}}$	non-dimensional power
γ	adiabatic exponent
ϕ	isentropic temperature ratio
ψ	over-expansion ratio
r_c	compression ratio
$(r_c)^*$	optimum isentropic compression ratio
r_e	expansion ratio
α	maximum temperature ratio
β	Cut off ratio
η_{th}	thermal efficiency
F	objective function
$\dot{W}_{\max}$	maximum power output
$\eta_{th,\max}$	maximum thermal efficiency

Subscript

c	compression
e	expansion

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