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ARTICLE

Thermodynamic Optimization of Diesel Engines Equipped With Shemax Injection System Using Genetic Algorithm

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Abstract

Diesel engines equipped with spark plug injection system refer to those engines in which the mixture of natural gas and air enters the cylinder in the suction cycle and then a small amount of diesel fuel is sprayed into the cylinder during the power cycle as ignition fuel or spark plug injection to be One of the main features of spark plug injection engines is the drastic reduction of pollution caused by diesel burning. In this research, the investigation and analysis of these engines have been done from the point of view of energy and exergy. For this purpose, the thermodynamic relationships governing such an engine are coded by MATLAB software, and the program presents its performance parameters by receiving the engine characteristics. The main goal of this research is to find the optimal point of the engine performance in a state where both the efficiency of the first and second law is maximum and finding the main cause of exergy destruction. The findings of this research show that with the increase in the amount of gas injection in the engine, the efficiency of the first law of thermodynamics first decreases and then increases. Meanwhile, the efficiency of the second law of thermodynamics increases as the amount of gas injection increases in the engine. If λ the percentage of gas injected into the engine, the results of optimization with the help of genetic algorithm show that the optimal point of operation of such an engine in terms of the first and second law of thermodynamics will be in the condition that $\lambda \geq 0.75$ Also, the main cause of exergy destruction in engines equipped with spark plug injection system seems to be related to diesel fuel combustion.

Keywords: Exergy combustion, Energy analysis, Diesel engine, Dual fuel engine

1. Introduction

The history of today's internal combustion engines goes back to 1879 and 1892, when the spark ignition auto engine and the compression ignition diesel engine were introduced for the first time. With the progress of human societies and the need to use internal combustion engines in various industries, especially the transportation industry, extensive efforts were made to optimize the performance of such engines in order to reduce emissions and increase efficiency [1]. Internal combustion engines consist of four processes: suction, compression, explosion and discharge. In a compression ignition type internal combustion engine, only air enters the cylinder during the suction stage. After the compression stage, the diesel fuel is directly injected into the combustion cylinder, and as a result, the expansion stroke and then the discharge are performed [2].

The spark plug injection engine system is an almost new technology that has been used in the field of diesel engines. In these engines, some natural gas is injected into the air manifold, in which case a mixture of gas and air enters the cylinder during the suction stage. In the compression stage, a small amount of diesel fuel is sprayed into the compressed gas and air mixture as spark plug injection and combustion is carried out [3]. It is clear that in these engines, a smaller amount of diesel fuel is burned and instead the cleaner natural gas fuel participates in the combustion process. Fig. 1 shows the operation schematic of an engine equipped with

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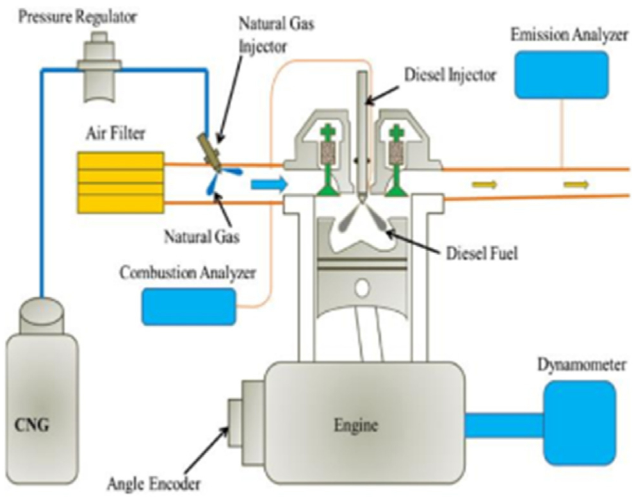

Fig. 1. Schematic diagram of a diesel engine equipped with spark plug injection system [5].

spark plug injection system. By placing the control sensors in the right places, the amount and pressure of diesel fuel injection is determined by the engine's computer controller [4]. Another advantage of this system is pure diesel engine operation if natural gas is not available. A lot of extensive research has been done on various performance parameters of engines equipped with spark plug injection system. But what is more important in the design of mechanical systems today is to check the efficiency of the second law of thermodynamics, or in other words, to analyze the exergy of a system.

One of the most general researches about the second law of thermodynamics for internal combustion engines in the last twenty years is the current research [6] He has done a review study about the investigation of the second law of thermodynamics around internal combustion engines. He has collected and discussed the primary researches that have been conducted on the second law of thermodynamics on diesel and auto engines before 1950s. Canoglu et al. [7] have investigated exergy degradation in a power plant diesel engine. They have investigated the second law of thermodynamics in a 120 MW power plant powered by a diesel engine. The said power plant works with seven similar diesel engines that have turbochargers. The results of their research show that combustion has the largest contribution to irreversibility. The next factors are the intercooler and compressor of the power plant, respectively. Also, Abusoglu and Kanoglu [8], in a similar work, have discussed a diesel engine in the system of simultaneous production of work and heat, in terms of the analysis of the first and second laws of thermodynamics. In this article, a thermodynamic analysis has been done on the diesel engine system in the simultaneous production plant, located in the gas city of Antep, Turkey. They have studied the exergy analysis in each component as well as the exergenic efficiency of the power plant. Da Costa et al. [9] have studied the energy and exergy of dual fuel engines in a research. The results of this research show that the efficiency of the second law of thermodynamics is higher for the double-burnt mode than for the pure diesel mode. Momer Ozkan and colleagues have studied the effects of pre-injection on the energy analysis and exergy of a diesel engine experimentally. They showed that by applying pre-injection in the engine, the performance of the system is improved [10]. In another research, Niromand and Saeedi investigated the exergy of a marine diesel engine. The results of their research show that the main cause of destruction.

The exergy is the diesel engine itself (along with the turbocharger, intercooler and oil exchanger) [11].

In this research, an attempt is made to find the optimal point of the system performance in terms of different amounts of injected gas by examining the changes in the functional parameters of the engine, the efficiency of the first and second law of thermodynamics, by thermodynamic modeling of the dual fuel engine in the MATLAB software environment. The two efficiencies of the first and second law of thermodynamics will be discussed with the help of genetic algorithm and then the main cause of engine exergy destruction.

2. Exergy and efficiency of the second law of thermodynamics

In order to check the performance of the power generation system, which is dependent on the performance of its sub-systems, second law analysis can be more useful than energy analysis, because it gives a better view and insight into the processes carried out within the system. Second law analysis between high-quality energies, A distinction is made between work-oriented and low-level energies [12]. To analyze the second law, exergy (the ability to do work) is considered a key concept. In recent years, studies in the field of energy and exergy for cost analysis in the design of power generation systems and chemical industries have received much attention from researchers [13]. Exergy studies are very important from an economic point of view in the performance of engineering systems, because the results of these studies show that controlling the exergy destruction of a system will be economically beneficial. The main mechanisms of entropy production and exergy destruction can be friction, heat transfer, suffocation, Mixing pointed out.

3. Dynamic relations governing the problem

Knowledge of the geometric characteristics of the engine is the first step of analyzing the assumed problem. According to Fig. 2, equation (1) is extracted from this geometry, which provides the volume of the cylinder in terms of crank angle [14].

$$m_{air} = (m_{total})\frac{A/F}{(A/F)+1}$$
$$m_{fuel} = \frac{m_{total}}{(A/F)} \qquad (1)$$
$$m_{C_{12}H_{24}} = (1-\lambda)m_{fuel}$$
$$m_{CH_4} = \lambda m_{fuel}$$

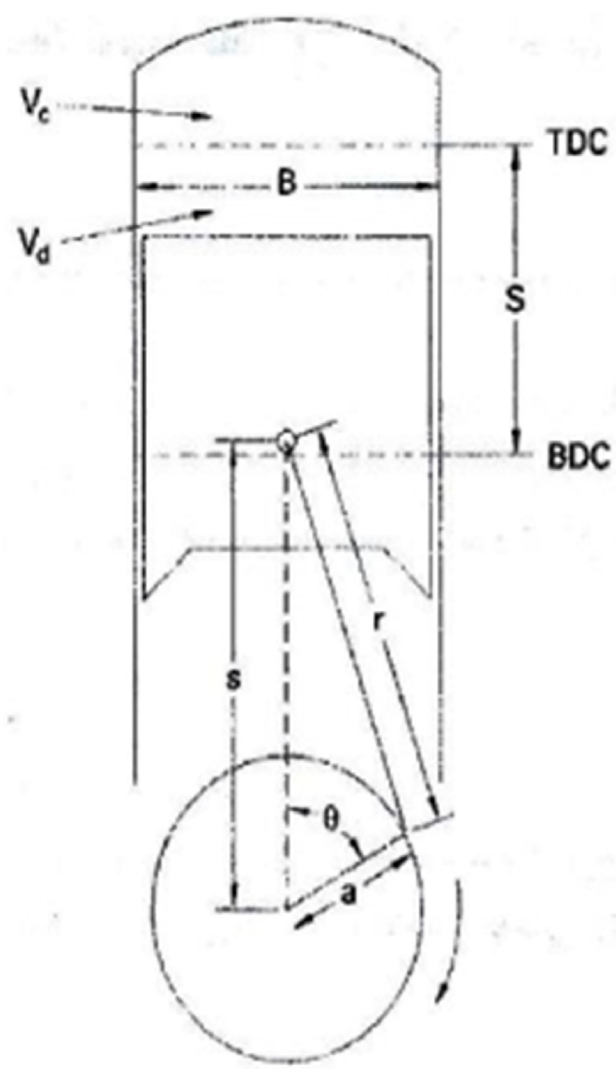

Fig. 2. Schematic of cylinder and piston geometry.

In the above relationship, V_c combustion chamber volume in cubic meters, V_d displacement volume in cubic meters, B piston diameter [m], θ crank angle [Rad], $S, (\theta = 0; @TDC)$ piston stroke [m], r piston handle length [m], r_c compression ratio [dimensionless], a crank radius [m], TDC is top dead center and BDC is bottom dead center [15].

Other geometric characteristics of the engine are obtained from the following relations

$$V_d = \frac{\pi B^2 S}{4} \qquad (2)$$

$$r_c = \frac{V_d}{V_c} + 1 \qquad (3)$$

$$C_r = \frac{V_4}{V_3} \qquad (4)$$

In equation (4), C is the separation ratio of the engine.

4. Thermodynamic modeling

Double cycle model is used for thermodynamic simulation of engines equipped with candle injection system, which is one of the best approximation models for diesel engines. The assumptions used in this research are [16]:

(1) The working fluid is air, which continuously circulates in a closed loop and behaves like a perfect gas.
(2) All processes that make up the cycle are internally reversible.
(3) The combustion process is replaced by an endothermic process from an external source.
(4) The exit process is replaced by an exothermic process that returns the working fluid to its initial state.
(5) Specific heats are considered constant at room temperature.

4.1. Combustion modeling

During the process of computer modeling in the process of combustion engines, various models have been offered. These combustion models can be divided into three categories according to the spatial dimension of the desired variables that are used in their formulation [17]:

1 Zero-dimensional models (thermodynamic single zone).

2 Pseudo-dimensional models (multi-zone thermodynamics).
3 Multidimensional models (CFD computational fluid dynamics).

In this research, zero-dimensional model is used. To get the amount of heat produced in the engine, the combustion equation must be written and solved. For this purpose, the combustion is assumed to be complete, in which case all the hydrocarbons burn completely and are not present in the combustion products (NO and CO). In the combustion equation, gas is approximated by methane (CH) and diesel or dodecane by (CH) [18].

$$\lambda CH_4 + (1-\lambda)C_{12}H_{24} + a_{th}(O_2 + 3.76N_2) \xrightarrow{\text{Complete Combustion}} xCO_2 + yH_2O + zN_2 \tag{5}$$

In which, the percentage of methane injected in the gas manifold and the amount of theoretical air for complete combustion are complete. Combustion products and the amount of theoretical air for reaction can be obtained through chemical balance. After balancing, the ratio of air to fuel (A/F) is calculated. Then, using the relations (6–9), the mass of air, diesel and methane that participated in the combustion equation is calculated [19].

$$m_{air} = (m_{total})\frac{A/F}{(A/F)+1} \tag{6}$$

$$m_{fuel} = \frac{m_{total}}{(A/F)} \tag{7}$$

$$m_{C_{12}H_{24}} = (1-\lambda)m_{fuel} \tag{8}$$

$$m_{CH_4} = \lambda m_{fuel} \tag{9}$$

The total thermal energy in kilojoules entered into the engine is obtained from equation (10)

$$Q_{in} = m_{C_{12}H_{24}}\left(Q_{HV_{C_{12}H_{24}}}\right) + m_{CH_4}\left(Q_{HV_{CH_4}}\right) \tag{10}$$

Where $Q_{HV_{C_{12}H_{24}}}$ and $Q_{HV_{CH_4}}$ respectively are the calorific value of diesel fuel and methane, which is the ratio of the heat transfer input to the engine at constant volume to the heat transfer input to the engine at constant pressure [20].

$$\psi = \frac{Q_{2-3}}{Q_{3-4}} \tag{11}$$

Although this parameter is received from the user by the software, ψ it cannot be assigned any value, because in the case where ψ 0 = double cycle, it becomes a normal diesel cycle, and if $Q_{3-4} \to 0$ this is the case, $\psi \to \infty$ spark ignition occurs. Therefore, a suitable value for this parameter should be found, which will be calculated later [21].

4.2. Justice ruling at different points of the cycle

Point 1) The initial working conditions of the engine are assumed as follows, which are:

P1 = 0.101 MPa

T1 = 60 C = 333 K

Process 1–2) is assumed to be a polytropic process with view n. The value of n is received by the software from the user. In the special case if $n = \frac{C_p}{C_v} = k$ the process will be isentropic. Where Cp and Cv are respectively the specific heat capacity of air at constant pressure and volume, the equation of process 1–2 is presented in relation (12):

$$P_{1-2}(\mathrm{V}) = \mathrm{P}_1\left(\frac{V_1}{V}\right)^n;\ V_c \le V \le (\mathrm{V_c} + \mathrm{V_d}) \tag{12}$$

Equation (12) presents the function of pressure changes in terms of cylinder volume in process 1–2.

Point 2) The temperature of point 2 is calculated using equation (13). Press this sharp point by placing V]V2 = Vc + Vd, it is obtained in relation (12)

$$T_2 = T_1(r_c)^{n-1} \tag{13}$$

Process 2–3) is a constant volume process in which heat transfer enters the engine. The amount of heat entered in this process depends on In the double cycle model, this process is drawn in a straight line parallel to the pressure axis in the diagram (volume pressure), whose equation is given in equation (15)

$$V = V_2 = V_c;\ P_2 \le P \le P_3 \tag{14}$$

Point 3) is derived from the first law of thermodynamics and according to the assumption of standard cold air, equation (15)

$$Q_{2-3} = m_{total}C_v(T_3 - T_2) \tag{15}$$

Due to the fact that only T3 is unknown in equation (15), the temperature of point 3 can be easily obtained by using equation (16) of the complete gas, and the pressure value at point 3 (3P) is calculated.

$$P_3 = \frac{m_{total}RT_3}{V_3} \tag{16}$$

In the last relation R = 0.287[kj/kgK] R = 0.287 is a general constant. Also, the volume of point 3 and 2 is the same and equal to the volume of the combustion chamber (Vc).

Process 3−4) In this process, heat is transferred to the system at constant pressure. This process is also drawn as a line parallel to the volume axis in the diagram (volume pressure), whose equation is:

$$P = P_3;\ V_3 \le V \le V_4 \quad (17)$$

Point 4) Temperature, pressure and volume of point 4 are obtained using relations (18), (19) and (20)

$$Q_{3-4} = m_{total} C_p (T_4 - T_3) \quad (18)$$

$$P_4 = P_3 \quad (19)$$

$$V_4 = \frac{m_{total} R T_4}{P_4} \quad (20)$$

Process 4−5) is polytropic expansion whose equation is calculated from equation (21)

$$P_{4-5}(V) = P_4 \left(\frac{V_4}{V}\right)^n \quad (21)$$

Equation (21) expresses the function of volume changes in relation to pressure in the expansion stage.

Point 5) The temperature at point 5 is obtained from equation (22). The pressure at this point is calculated by putting the maximum volume of the cylinder $V_{\max} = V_d + V_c$ in equation (22)

$$T_5 = T_4 \left(\frac{1}{r_c}\right)^{n-1} \quad (22)$$

Process 5-1) This constant volume process is also modeled by a line parallel to the pressure axis (equation (23)). From equation (24) the amount of heat removed from the system in process 5-1 is obtained:

$$V = V_4 = V_{\max};\ P_1 \le P \le P_4 \quad (23)$$

$$Q_{out} = m_{total} C_v (\mathrm{T}_5 - \mathrm{T}_1) \quad (24)$$

3-4) engine performance parameters

The amount of work done in process 1−2, 3−4, 4−5, net work, thermal efficiency and effective average pressure indicator are obtained from the following relationships [22]:

$$W_{1-2} = \frac{mR(T_2 - T_1)}{1-n} \quad (25)$$

$$W_{3-4} = P_3 (V_4 - V_3) \quad (26)$$

$$W_{4-5} = \frac{mR(T_5 - T_4)}{1-n} \quad (27)$$

$$W_{net} = |W_{4-5}| + |W_{3-4}| + |W_{1-2}| \quad (28)$$

$$\eta_{1st,law} = \frac{W_{net}}{Q_{in}} \quad (29)$$

$$\mathrm{Imep} = \frac{W_{net}}{V_d} \quad (30)$$

The amount of work of process 1−2 is negative because the engine loses energy by applying force to compress the gas and air mixture, but useful work is produced in two processes 3−4 and 5-4. And the difference between productive work and negative work shows the amount of net work. Thermal efficiency is also the ratio of net work to total energy input to the engine. The effective average pressure of the indicator is also the ratio of net work to displacement volume. To calculate the efficiency of the second law of thermodynamics, Berzostoski proposed the following relationship:

$$\eta_{2st,law} = \frac{W_{net}}{m_{fuel}(1.065 q_{LHV})} \quad (31)$$

where the low calorific value of the fuel and the adjacent number is the correction factor of the equation. Also, the total calorific value of the fuel participating in the combustion is obtained from the equation:

$$q_{LHV} = \lambda q_{LHV_{CH_4}} + (1-\lambda) q_{LHV_{C_{12}H_{24}}} \quad (32)$$

For a better modeling of a dual diesel process, Qabali et al. (20) have proposed the volume pressure parameter as follows:

$$\xi = \frac{P_3 - P_2}{V_4 - V_3} = \frac{P_3 - P_2}{V_3(C_r - 1)} \quad (33)$$

Equation (33) shows the ratio of pressure difference in constant volume (P3−P2) to volume difference in constant pressure (V4−V3). From the examination of the laboratory models, it can be concluded that in order to bring the double cycle model closer to the real process of the diesel engine, it should be about (30).

5. Results and discussion

Analytical examination of the results of changes in various parameters of a system is generally impossible.

Although this problem is feasible in a laboratory and has high accuracy, the high costs of testing and other limitations of this method have led to the growth of the use of numerical methods among researchers. MATLAB software, as one of the most powerful and at the same time the most flexible numerical programs, has a very high ability to solve various problems in the field of engineering. Therefore, this software has been used to model a diesel engine equipped with a spark plug injection system. The geometrical characteristics of the engine and the thermodynamic characteristics of natural gas and diesel are considered as inputs to the program, and the thermodynamic relationships for the double cycle compression ignition engine and combustion equations are coded. The most important engine performance parameters are considered as program outputs (Relations 28–31). But since the main focus of this research is on the efficiency of the first and second laws of thermodynamics, writing other codes that determine the parameters of ψ and ξ have also been done. Table 1 shows the geometric characteristics of the engine analyzed in this research (see Fig. 3).

In Fig. 4 the changes ψ compared to ξ are shown. This variation for each constant follows the approximately linear pattern shown. Therefore, to satisfy the ratio, $\xi = 30\left[\frac{GPa}{m^3}\right]$ the value ψ should be around 0.15. Therefore, $\psi = 0.15$ and ξ it is placed in the optimal range. In Table 2, the changes ψ of the engine for different values λ of As it is known, it is obtained $\psi = 0$ according to $\lambda = 0$ the model of the ideal cycle of diesel, in which all the heat transfer is entered into the system at a constant pressure.

Table 1. Geometric specifications of the engine [22].

Row	Attribute	Value
1	piston diameter	112 m
2	compression ratio	1:8
3	crank radius	125 m
4	piston strokes	115 m
5	The length of the connecting rod	250 m

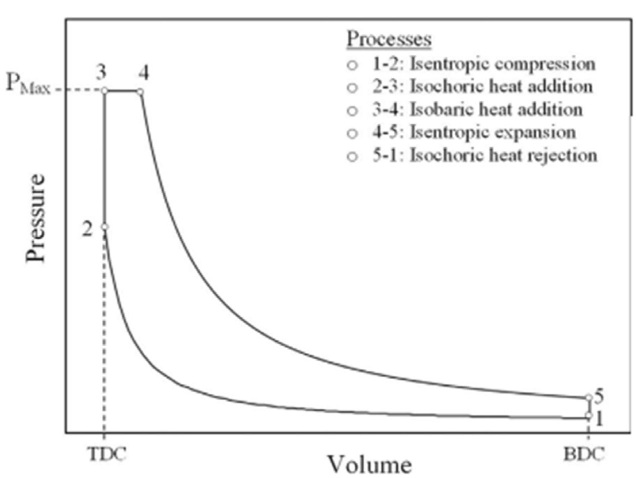

Fig. 3. Schematic diagram of volume pressure of a dual diesel engine.

In Tables 3 and 4, the performance parameters of the diesel engine also improve in the indicator. Of course, it should be noted that the isentropic and real conditions are shown for the optimal point $\psi = 0.15$. According to the data in these tables, the increase λ causes an increase in the pressure and temperature of the maximum cycle, and in the effective average pressure, increasing the maximum cycle pressure is not desirable in general, because it causes the phenomenon of quiche (4) to prevent this problem, it is necessary for the engines to Be equipped with candle injection system with low compression ratio. In the setting of Table 3, it was assumed that all processes are isentropic ($n = \frac{C_p}{C_v} = k$ i.e.), this assumption causes the efficiency of the second law of thermodynamics of such an engine to be practically one hundred percent. Therefore, it is clear that such an assumption moves the thermodynamic model of the assumed engine away from the real conditions. In order to achieve more realistic conditions, expansion and compression processes (n = 1.5) have been considered in the polytropic view program codes, and then the performance parameters of the engine and its efficiencies have been extracted and recorded in Table 1. In this case, the efficiency of the second law of thermodynamics has been calculated for different values. Fig. 5 shows the changes in the net work output of the engine in relation to the amount of gas injection. Carefully in this figure, it can be seen that from $\lambda = 0.4$, there is no change in the net work of the engine (see Fig. 6). As it is known, the output net work is not related to the increase of injected gas, so that the net work only increases with an increase of λ about 0.2 kJ and then remains constant. When it increases. The amount of thermal energy produced in the cycle also increases λ and because the net work of the system does not change much, according to equation (29), the efficiency of the first law of thermodynamics decreases. Fig. 7 shows the changes in the efficiency of the first and second laws of thermodynamics with the increase in the percentage of natural gas injection. By increasing the amount of gas injected into the engine, the efficiency of the second law of thermodynamics increases in an upward and almost linear trend. This problem is evident that if the assumed diesel engine works with 100% gas, it has a higher efficiency of the second law. Meanwhile, from the point of view of the first law of thermodynamics, the efficiency of the system is maximum when the

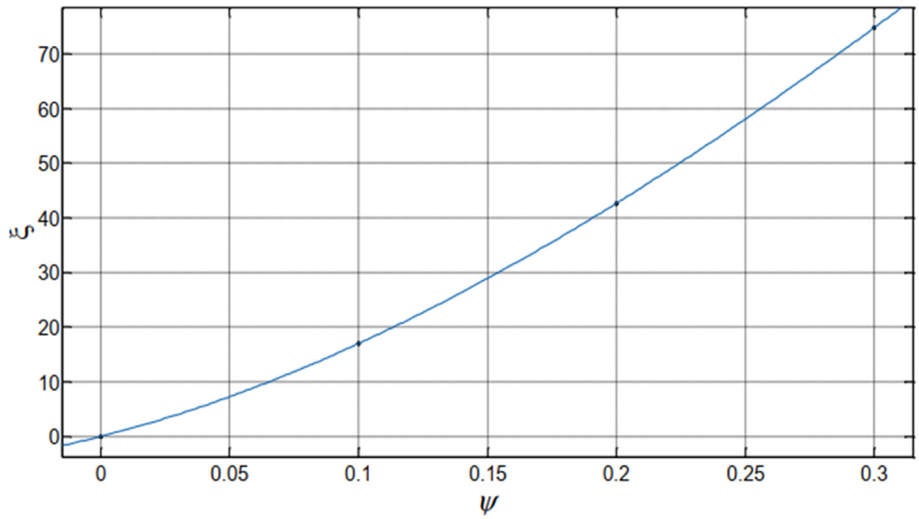

Fig. 4. How to change ψ relative to ξ.

Table 2. Engine changes ψ for different values of λ.

ψ	λ	$\xi = \left[\frac{GPa}{m^3}\right]$
0	0	0
0.1	0.9	17.1
	0.7	17.2
	0.5	17.1
0.2	0.9	42.5
	0.7	42.8
	0.5	42.6
0.3	0.5	74.2
	0.7	75.0
	0.9	75.6
1	1	379.2

engine is working as a pure diesel, an increase in the system causes a decrease in the efficiency of the first law. Although with an increase of λ more than 08, the efficiency goes up again, but the maximum value of the efficiency $\lambda = 0$ of the first law occurs. But from the point of view of the second law of thermodynamics, increasing the gas injected into the system always improves the efficiency of the second law, so in order to find a point where the efficiency of the system is in terms of the first and second laws, using the results presented in Table 4, Two univariate objective functions are defined as relation (34).

Now, with the help of genetic algorithm, the optimal point of the desired function is found. Table 5 shows that the software provides fit values per design variable with 200 repetitions $\lambda = 0.751$. Therefore, it can be said that $\lambda < 0.75$ the efficiency of the first and second laws of thermodynamics are somehow in conflict with each other. That is, as the efficiency of the first law of thermodynamics increases, it decreases and the efficiency of the second law of thermodynamics increases.

Therefore, in order for the system to be at an optimal point in terms of the efficiency of the first and second laws.

In order for thermodynamics to work, a balance must be established between them. This is $\lambda \geq 0.75$ then possible. Because it is in this value that both efficiencies show an upward behavior.

As mentioned, one of the mechanisms that destroys exergy is combustion. In general, hydrocarbons lose a large amount of their ability to perform their work in an oxidation reaction [21] Meanwhile, the type of fuel is also effective in destroying exergy, which the septic chemical investigation of hydrocarbon combustion process and their exergy analysis Separately required. In this research, it was seen that by increasing the amount of injected gas (methane) and naturally reducing the diesel fuel in

$$\begin{cases} \eta_{1st,law}(\lambda) = 0.09848\lambda^2 + 1.93\lambda + 59.96;\ 0 \leq \lambda \leq 1 \\ \eta_{2st,law}(\lambda) = 1.195\lambda^4 - 1.346\lambda^3 + 0.583\lambda^2 - 0.622\lambda + 59.96;\ 0 \leq \lambda \leq 1 \end{cases} \quad (34)$$

Table 3. Performance parameters of the motor for $\psi = 0.15$ in isentropic conditions.

ψ	λ	P_{max} (Mpa)	T_{max} (K)	W_{ret} (kJ)	Imep (MPa)	$\eta_{1st,kne}$ (%)	$\eta_{2st,kne}$ (%)	$\xi\left(\frac{GPa}{m^2}\right)$
0.15	0	8.9084	4363	2.5	2.1645	61.26	100	28.0
	0.1	8.9666	4425	2.5	2.2021	61.19		28.2
	0.2	9.0233	4484	2.5	2.2387	61.12		28.4
	0.3	9.0778	4542	2.6	2.2740	61.06		28.5
	0.4	9.1294	4596	2.6	2.3072	61.00		28.7
	0.5	9.1767	4646	2.6	2.3377	60.94		28.9
	0.6	9.2171	4689	2.7	2.3637	60.90		29.0
	0.7	9.2458	4719	2.7	2.3821	60.86		29.1
	0.8	8.2519	4726	2.7	2.3861	60.86		29.1
	0.9	9.2056	4677	2.7	2.3563	60.91		29.0

Table 4. Engine performance parameters for $\psi = 0.15$ in real conditions $n = 1.5$.

ψ	λ	P_{max} (Mpa)	T_{max} (K)	W_{ret} (kJ)	Imep (MPa)	$\eta_{1st,kne}$ (%)	$\eta_{2st,kne}$ (%)	$\xi\left(\frac{GPa}{m^2}\right)$
0.15	0	10.8445	4718	2.4	2.1134	59.81	59.94	34.16
	0.1	10.9027	4779	2.4	2.1506	59.76	60.15	34.34
	0.2	10.9594	4839	2.5	2.1867	59.70	60.36	34.52
	0.3	11.0139	4897	2.5	2.2215	59.65	60.56	34.69
	0.4	11.0656	4951	2.6	2.2544	59.60	60.76	34.86
	0.5	11.1128	5001	2.6	2.2844	59.55	60.95	35.00
	0.6	11.1532	5044	2.6	2.3103	59.52	61.14	35.13
	0.7	11.1819	5074	2.6	2.3283	59.49	61.34	35.22
	0.8	11.1880	5080	2.6	2.3322	59.48	61.55	35.24
	0.9	11.1417	5031	2.6	2.3028	59.53	61.81	35.10

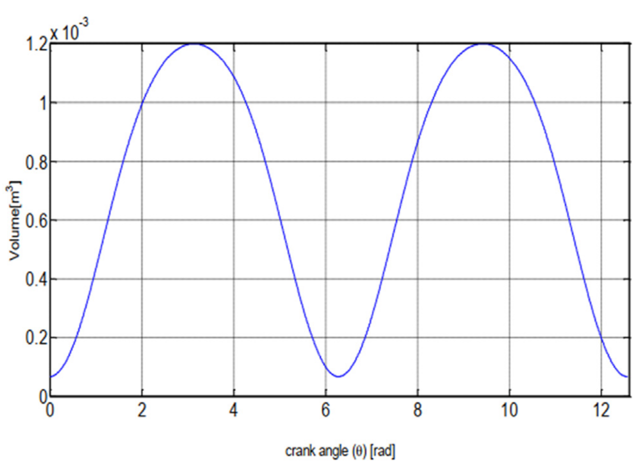

Fig. 5. Diagram of changes in cylinder volume in relation to crank angle.

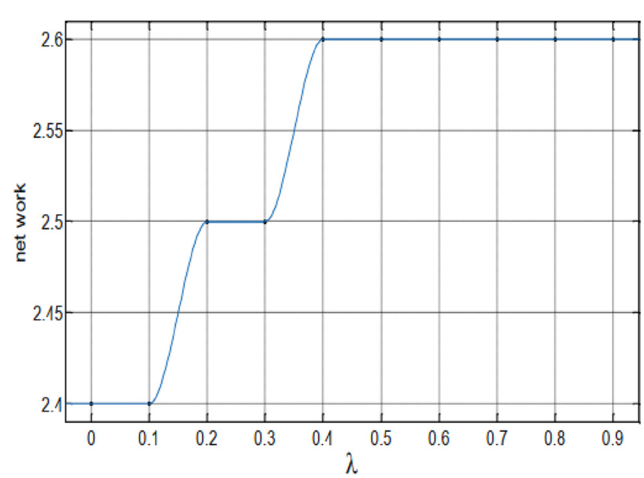

Fig. 6. How the net work changes per λ at $\psi = 0.15$.

the engine, the efficiency of the second law of thermodynamics improved. Considering that the second law of thermodynamics can be an interpretation of the concept of exergy. The findings of this research show that the combustion of natural gas will destroy a smaller amount of exergy than diesel. In other words, the main contribution to the loss of workability is caused by the combustion of diesel fuel, which is completely consistent with the laboratory results of other researchers [22].

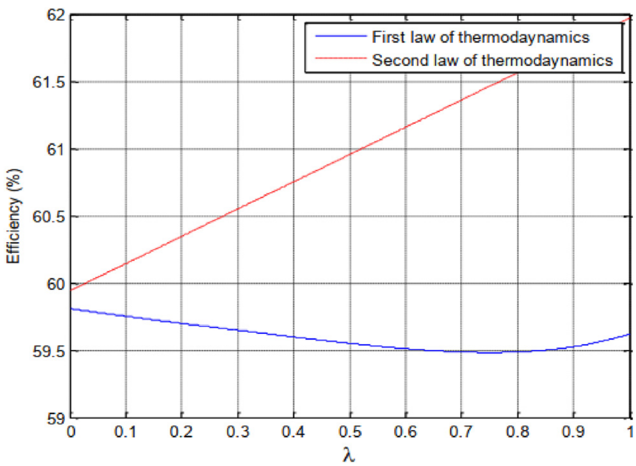

Fig. 7. Change in the efficiency of the first and second laws of thermodynamics per λ at $\psi = 0.15$.

Table 5. Information about the genetic algorithm.

Population	50
Maximum Repetition	200
$\eta_{2nd,law}(\lambda) = 59.482$	$\lambda = 0.751$
$\eta_{1nd,law}(\lambda) = 61.464$	

6. Conclusion

In the current research, the changes caused by the increase of natural gas injection in the diesel engine on its performance parameters and exergy analysis and efficiency of the second law of thermodynamics were investigated. To achieve this goal, dual cycle relations for diesel engine were written in MATLAB software and its output was discussed in the article. The results of this research can be summarized as follows:

1) The amount of net work of the system was slightly improved by increasing the values, and it seems that these changes are not very sensitive to the amount.
2) With the increase of gas injected into the diesel engine, the efficiency of the first law of thermodynamics decreased to = 0.75 and then increased again, which confirms the laboratory results of other researchers.
3) Contrary to what was extracted about the efficiency of the first law of thermodynamics, the efficiency of the second law of thermodynamics improved in a completely upward manner with the increase in the amount of injected gas.
4) It can be said that diesel combustion has the main contribution in destroying the exergy of the dual fuel diesel engine. Therefore, the smaller the ratio of diesel fuel to natural gas, the less exergy is destroyed from the system.
5) With the increase in pressure, the maximum cycle also increased, which is not desirable in general, because it leads to the phenomenon of quiche in the engine.
6) The optimization results show that the optimal performance of the engine takes place according to the efficiency of the first and second laws of thermodynamics $\lambda \geq 0.7, \psi = 0.15$. Therefore, in a dual fuel engine, for maximum efficiency, the engine must operate under the conditions of the first and second laws of thermodynamics.

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