Comparison of the Performances of Biodiesel, Diesel, and Their Compound in Air Standard Diesel-Atkinson Cycle

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Abstract: Biodiesel is a fuel which due to its environmental friendly and renewability properties has established a proper place among researchers. There have been several researches on this fuel. But it has not been observed in a certain research on biodiesel to demonstrate, by using numerical simulators, the behavior of this fuel in an engine which performs a specific cycle. In this research it is considered to review the irreversible Diesel-Atkinson cycle of biodiesel fuel and its compounds by means of thermodynamics laws and finite time thermodynamics when the biodiesel fuel is applied as the operative fluid inside the cycle. The results from numerical simulation showed that applying biodiesel fuel and its compounds in this cycle proved to have similar or in some cases even better results from the traditional diesel fuel.

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1. Introduction

Finite-time thermodynamics is an extension of conventional thermodynamics relevant in principle across the entire span of the subject, from the most abstract level to the most applied. The approach is based on the construction of generalized thermodynamic potentials (Hermann 1973) for processes containing time or rate conditions among the constraints on the system (Salamon et. al., 1977) and on the determination of optimal paths that yield the extreme corresponding to those generalized potentials (Mozurkewich et. al., 1981). Many significant achievements have been made since introduction of finite-time thermodynamics in order to analyze and optimize the performances of real heat-engines (Chen et. al., 1999; Bejan 1996 and Chen et. Al, 2004).

The troubles of using fossil fuels have made researchers to think of finding replacement fuels. Since biodiesel has similar properties to the traditional diesel fuel and less environmental troubles occur from using it, is one of the main candidates for replacing diesel fuel in engines. Several researchers are experimenting on making this fuel applicable. Biodiesel is a proper fuel which is generally extracted from plant oils and animal fat along with methanol and by Trans-esterification method (Shaoyang et. al., 2011; Jie et. al., 2011; and Atadashi et. al., 2010).

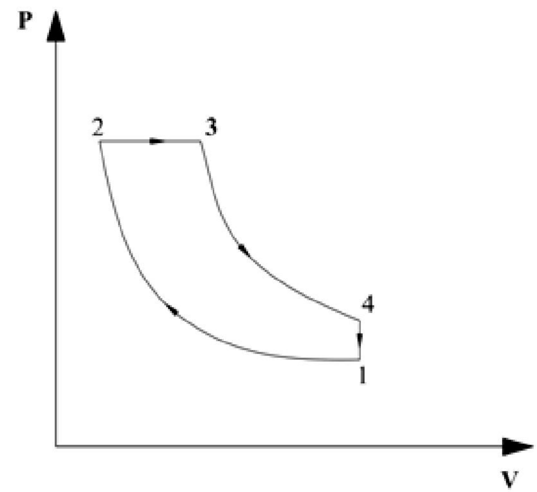
There have been conducted vast experiments on performance of an engine while using biodiesel fuels.

Cumali Ilkilic et, al evaluated a single cylinder diesel engine while consuming biodiesel compounds derived from sunflower oil with traditional diesel. In the end it was clarified that using biodiesel compounds in comparison to use pure diesel fuel will slightly increase the brake specific consumption but at the same time greatly decrease pollutants such as CO, PM (Cumali et. al., 2011). The environmental friendly trait is one the most important properties of biodiesel which is mentioned in many experiments (Jason et. al., 2011; Lin et. al., 2011; and Yoon et. al., 2011).

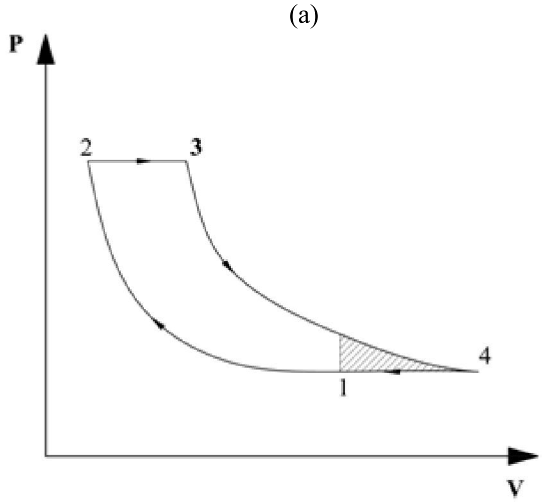
In most experiments generally in order to approximate the biodiesel properties to the traditional diesel's, the mixture of these two is applied, and it seems that 20% volume of biodiesel along with 80% of traditional diesel fuel is determined to be one of the most proper compounds of biodiesel in order to use for engines (Farahani et. al., 2011, Usta et. al., 2005, and Lin et. al., 2007).

The applicable aspect of biodiesel compounds in engines has been the main objective of most researches (Xue et. al., 2011). Practical experiments usually have great costs and are time-taking, which seems to be inapplicable for most researchers. One the experimental fields of biodiesel, which is used less than other fields, is the use of numerical simulation theories and cases for predicting performance engines which are working with a special cycle, while using biodiesel and its compounds. In the past two decades, using finite time thermodynamics for optimizing reversible and irreversible cycles of thermal engines has been expanded (Ebrahimi 2010 and 2009). In this experiment it is considered to evaluate the performance of standard air irreversible cycle while using this compound as an operative fluid by means properties of a special biodiesel compound and numerical simulations.

Chen et. al. (2003) used output numerical simulation and examined and compared the performance of reversible and irreversible cycles of diesel. This cycle was also evaluated by Ebrahimi (2009) under variable compression ratio and stroke length. The results showed that if the compression ratio decreases from a certain limit, by increasing stroke length, the cycle output work will increase first and then decreases. Therefore, using numerical and thermodynamic simulation of fuels behavior in special cycles is a method which without requiring equipments and tools can evaluate the performance of two fuels in an engine.



(a)



(b)

Figure 1. Cylinder pressure based on volume in a) an air standard Diesel cycle, b) an air standard Diesel-Atkinson cycle

Hou (2007) Compared performances of air standard Atkinson and Otto cycles with heat transfer considerations. Based on this study, an air standard Atkinson cycle has a better performance than the air standard Otto cycle. So it is suggested that a Diesel cycle with an exhaust process like an Atkinson cycle one be utilized instead of common Diesel cycle (named Diesel-Atkinson cycle).

This study is aimed to analysis of the performance of an irreversible Diesel-Atkinson cycle with losses arising from heat resistance, friction and the temperature variation of the specific heat of the working fluid. Also, its show comparison of the performances of biodiesel, diesel and their compound in diesel-Atkinson air standard irreversible cycle.

2. Thermodynamics simulation of air standard Diesel and Diesel-Atkinson cycles

Figure 1, a and b show the pressure-volume diagrams of thermodynamic process of an air standard Diesel and Diesel-Atkinson cycles which all four phases of these irreversible cycles are considered. Diesel cycle is first introduced by Rudolf Diesel based his engine which is known as Diesel engine today. Difference of the mentioned cycles is shown in this figure.

In an air standard Diesel, the compression process 1→2 is isentropic, heat is added to the cycle during process 2→3 which is an isobaric process, throughout isentropic process 3→4 the expansion is occurred, and the heat rejection process 4→1 is an isochoric process. In an air standard Diesel-Atkinson processes 1→2, 2→3, 3→4 are the same as Diesel cycle, but unlikely, the process 4→1 stage is an isobaric process. As is usual in finite-time thermodynamic heat-engine cycle models, there are two instantaneous adiabatic-processes, namely 1→2 and 3→4. For the heat addition and heat rejection (2→3 and 4→1 stages, respectively), it is assumed that heating occurs from state 2 to state 3 and cooling ensues from state 4 to state 1 and proceed according to isothermal rates, as shown in Eq. (1):

$$\frac{dT}{dt} = \frac{1}{c_1} \quad \text{for } (2\rightarrow3)$$
$$\frac{dT}{dt} = \frac{1}{c_2} \quad \text{for } (4\rightarrow1) \qquad (1)$$

Where T is the absolute temperature and t is the time; C_1 and C_2 are constants. Integrating Eqs. (1) yield:

$$t_1 = c_1(T_3 - T_2) \quad \text{and}$$
$$t_2 = c_2(T_4 - T_1) \qquad (2)$$

Where t_1 and t_2 are heating and cooling periods, respectively. Then, the cycle period is

$$\tau = t_1 + t_2 = c_1(T_3 - T_2) + c_2(T_4 - T_1) \qquad (3)$$

Specific heat generally is considered constant by most researchers in diesel standard air cycle, but in an actual engine this amount is not constant and generally the specific heat is considered only as a function of temperature. Since in engine work range, the temperature varies between 300 through 2200 Kelvin, these changes could be considered linear with temperature (Ganesan 1993). Therefore, these two equations are introduced for determining the specific heat:

$$C_V = b + kT \quad (4)$$

$$C_p = a + kT \quad (5)$$

Where, a, b, and k are constant and C_v and C_p are mole specific heat at constant volume and pressure, respectively. It is obvious that gas mole constant could be calculated from the following equation:

$$R = C_p - C_v = a - b \quad (6)$$

During the process of 2→3 which is constant pressure process, the fluid is heated which could be calculated from the following equation:

$$Q_{in} = M\int_{T_2}^{T_3} C_p dT = M\int_{T_2}^{T_3}(a+kT)dT = M\left[a(T_3-T_2)+0.5k(T_3^2-T_2^2)\right] \quad (7)$$

And during the 4→1 process which is a constant volume process, the fluid inside the cylinder will release an amount of heat which output heat from this system can be determined from the following equation for Diesel-Atkinson cycle:

$$Q_{out} = M\int_{T_1}^{T_4} C_p dT = M\int_{T_1}^{T_4}(a+kT)dT = M\left[a(T_4-T_1)+0.5k(T_4^2-T_1^2)\right] \quad (8)$$

Where, M is the mole number of the working fluid.

Since above equations are only utilized for reversible conditions, which the adiabatic amount is considered constant. But in this experiment, because of irreversible conditions, the specific heat amounts depend on the fluid's temperature. So, direct utilization of reversible adiabatic equations directly is not possible. In order to make these equations applicable for irreversible conditions, an engineering approximation was used. If an irreversible process was considered as an infinite amounts of small processes, the adiabatic ratio of each phase (for example i to j) could be determined as constant. Since in each considered small phase the temperature and volume change as much as dT and dV, thus the temperature-volume adiabatic equation for an irreversible process (with variable adiabatic ration) can be determined from the following equation:

$$TV^{r_c-1} = (T+dT)(V+dV)^{r_c-1} \quad (9)$$

For each small phase (for example i to j):

$$k(T_j - T_i) + b\,ln(\frac{T_j}{T_i}) = -R\,ln(\frac{V_j}{V_i}) \quad (10)$$

So, the isentropic process of 1→2 and 3→4 can be rewritten based on Eq. (10) as follows:

$$k(T_2 - T_1) + b\ln\left(\frac{T_2}{T_1}\right) = R\ln r_c \quad (11)$$

$$k(T_3 - T_4) + b\,ln(\frac{T_3}{T_4}) = R\,ln(\frac{V_4}{V_3}) \quad (12)$$

Where, in these equations r_c is compression ratio, and $r_e = V_4/V_3 = r_c(T_4/T_1)$.

In this study, in order to make conditions closer to the reality, the heat losses from heat transfer to outside of the cycle were noted. It could be presumed that heat loss from the cylinder wall is proportionate to the average temperature of fluid inside the cylinder and chamber wall. Therefore, the heat given to a fluid during an actual cycle can be calculated from the following equation (Chen et. al., 2003):

$$Q_{in} = M\left[A - B(T_2 + T_3)\right] \quad (13)$$

Where, in this equation A and B are constant amounts which are related to the heat transferred to walls and combustion.

Taking into account the friction loss of the piston, as deduced Chen et al. (2003), and assuming a dissipation term resulting from the friction force as being a liner function of the velocity, then

$$f_\mu = -\mu v = -\mu\frac{dx}{xt} \quad (14)$$

Where μ is the coefficient of friction, which takes into account the global losses and x is the piston's displacement. Then, the lost power is

$$P_\mu = \frac{dW_\mu}{dt} = -\mu\frac{dx}{dt}\frac{dx}{dt} = -\mu v^2 \quad (15)$$

The piston's mean-velocity is

$$\bar{v} = \frac{x_1 - x_2}{\Delta t_{12}} = \frac{x_2(r_c - 1)}{\Delta t_{12}} \quad (16)$$

Where x_2 is the piston position corresponding to the minimum volume of the trapped gases and Δt_{12} is the time spent in the power stroke.

With given amount of T_1 from Eq. (11), T_2 could be calculated. So by equaling Eq. (7) and Eq. (13), Eq. (17) is derived. Placing T_2 in Eq. (17), T_3 is achieved. Then, by placing T_3 in Eq. (12), T_4 is also achieved.

$$A - B(T_2+T_3) - a(T_3-T_2) - 0.5k(T_3^2 - T_2^2) = 0 \quad (17)$$

Finally, given T_1, T_2, T_3, and T_4, the amount of power output could be calculated from the following equation:

$P_{output} = (W/\tau) - P_\mu$ can be written as

$$P_{output} = \frac{Q_{in} - Q_{out}}{\tau} - b_1(r_c - 1)^2 \quad (18)$$

Where

$$b_1 = \frac{\mu x_2^2}{(\Delta t_{12})^2} \quad (19)$$

And the efficiency of the cycle is

$$\eta_{th} = \frac{P_{output}}{(Q_{in}/\tau)} \quad (20)$$

3. Numerical Simulation and Results

Due to data from resources (Chen et. al., 2003; Lin et. al., 1999; Al-Sarkhi et. al., 2002), the following amounts were selected for numerical analysis (table 1)**:**

Table 1. Data selected for analysis

$A = 60000\frac{J}{mole}$	$B = 20-25\frac{J}{mole.K}$
$b_1 = 21KW$	$M = 1.57\times10^{-5}\ kmole$
$\tau = 33.33\,ms$	$t_1 = t_2 = \tau/2 = 16.6\,ms$
$c_1 = 8.128\times10^{-6}\ s/K$	$c_2 = 18.67\times10^{-6}\ s/K$

Table 2: specific heat constants from diesel, biodiesel, and BIO10 fuels (Fallahipanah et al., 2011).

Fuel Type	a	b	k
Diesel	29.9776	20.1442	0.005372
Biodiesel	28.161	20.3266	0.008175
BIO10	28.0151	20.1807	0.005932

Figure (2) shows variation temperature versus compression ratio for Diesel-Atkinson cycle. Also, in this figure shows variation temperature for diesel, biodiesel and BIO10 fuels. However, we know T_3 is maximum temperate inside cylinder. As can been in figure (2) with increases in compression ratio T_3 increases smoothly and in compression ratio equal 10 that it is a practical compression ratio maximum temperate inside cylinder for BIO10 fuel is more than diesel and biodiesel fuels. But T_4 that is temperate exhaust gas for biodiesel fuel is more than diesel and BIO10 fuels. This is show that the time of ignition for biodiesel fuel is more than ones for diesel and BIO10.

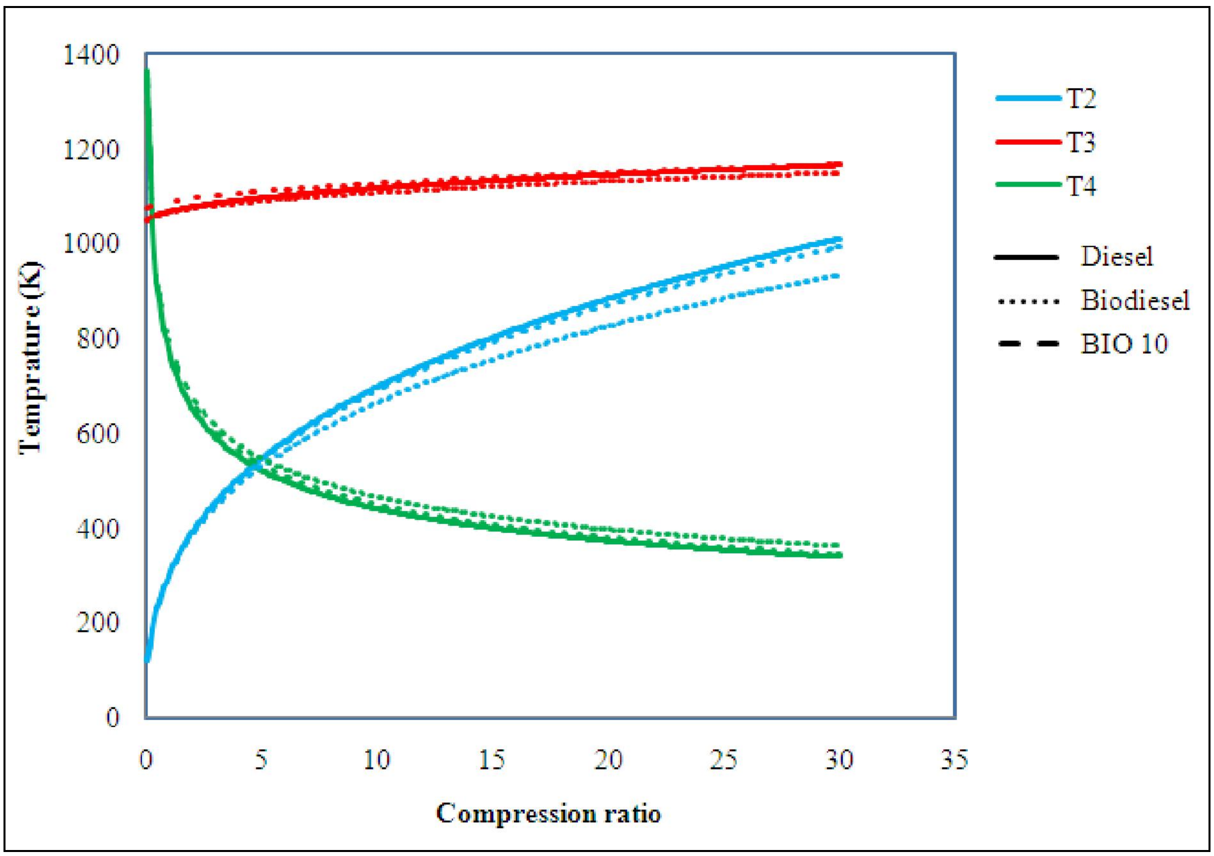


Figure 2. Variation of temperature versus compression ratio for diesel, biodiesel and BIO10 fuels

Figures 3 and 4 show power output and thermal efficiency versus compression ratio for Diesel-Atkinson Cycle with diesel, biodiesel and BIO10 fuels. It's obvious that thermal efficiency and power output of Diesel fuel are higher than biodiesel and BIO10 ones and the points of maximum power output and thermal efficiency of biodiesel occur at the higher compression ratio. The maximum thermal efficiency of diesel, biodiesel and BIO10 fuels is 63.2%, 61% and 662.8% at 7.8, 8 and 7.9 compression ratios, respectively. And the maximum power output of diesel, biodiesel and BIO10 fuels is 5653.7 W, 5521.6W and 5533.2W, respectively.

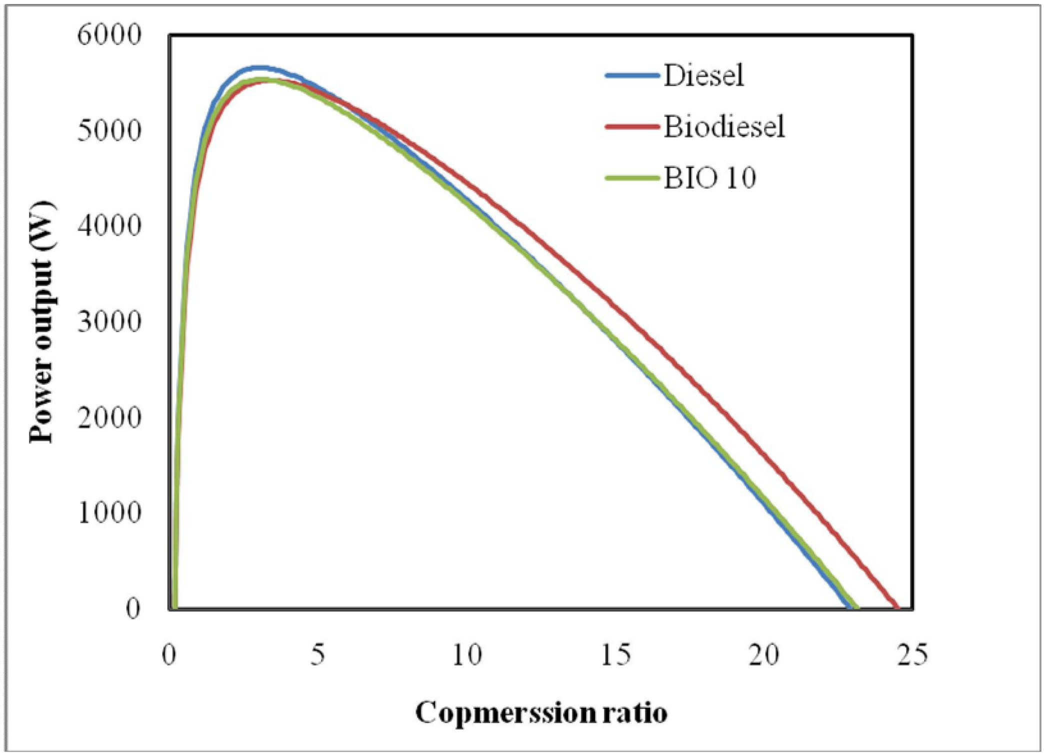


Figure 3. Power output versus compression ratio for diesel, biodiesel and BIO10 fuels

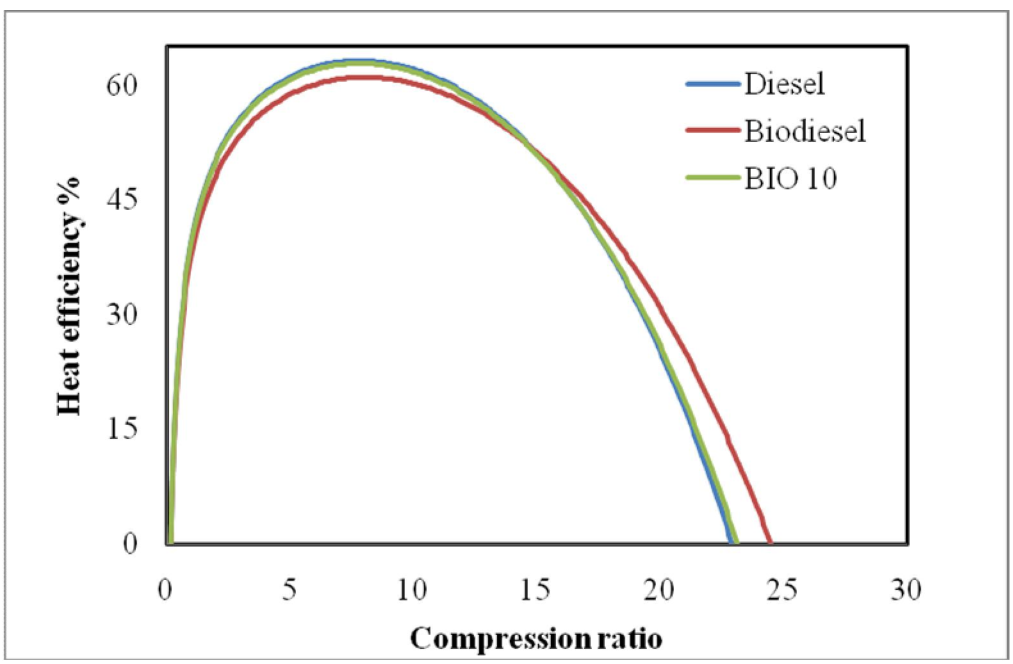


Figure 4. Thermal efficiency versus compression ratio for diesel, biodiesel and BIO10 fuels

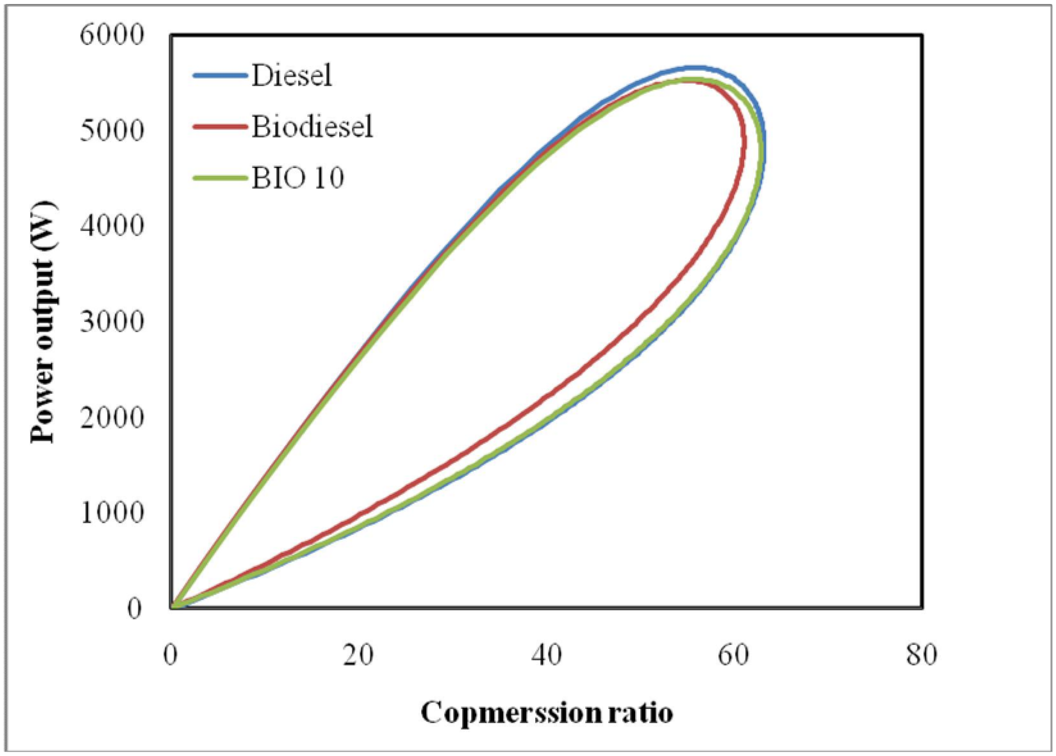


Figure 5. Power output versus thermal efficiency for diesel, biodiesel and BIO10 fuels

4. Conclusion

Numerically, an air-standard Diesel-Atkinson cycle model, assuming a temperature-dependent specific-heats of the working fluid, heat resistance and frictional irreversible losses, have been investigated and the performance characteristics of two cycles were obtained. Then performance of diesel, biodiesel and BIO10 fuels are investigated. Comparison the results show that the power output and the thermal efficiency of diesel are higher than biodiesel and BIO10 fuel ones at their optimum compression ratio. Like previous studies, the results show that there are significant effects of the temperature dependence of the specific heat of the working fluid, friction and heat transfer losses on the performance of the cycles which should be considered in practical-cycle analysis. Also, increases of T_4 for biodiesel fuel results in increases of temperature of exhaust gas which causes to the reduction of NO_x. The results obtained from this research are compatible with those in the open literature, but for other cycles, and may be used with assurance to provide guidance for the analysis of the behavior and design of practical Diesel-Atkinson engines.

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