ÇUKUROVA UNIVERSITY
INSTITUTE OF NATURAL AND APPLIED SCIENCES

MSc THESIS

Erdi TOSUN

ENERGY AND EXERGY ANALYSIS OF A DIESEL ENGINE

DEPARTMENT OF MECHANICAL ENGINEERING

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We certify that the thesis titled above was reviewed and approved for the award of degree of the Master of Science by the board of jury on 31/12/2013

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ABSTRACT

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ENERGY AND EXERGY ANALYSIS OF A DIESEL ENGINE

Erdi TOSUN

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This study deals with assessment of energy and exergy analysis applied to experimental data obtained from a four stroke, four cylinders, naturally aspirated, direct-injected diesel engine by using different fuels. Each test was performed at 2600 rpm engine speed. Air and fuel flow rates, engine speed, emissions, relevant temperatures are needed to be able to perform the first and second law of thermodynamics. With the use of these data and balances of energy and exergy rates for the control volume, 1st and 2nd law (energy and exergy) efficiencies related with quantity and quality were determined for various fuels. All results were compared with each other to see fuel effect on energetic and exergetic performance. It was concluded that diesel fuel showed better energetic and exergetic performance than both biodiesels.

Key Words: Energy, Exergy, Internal Combustion Engine, Alternative Fuel

ÖZ

YÜKSEK LİSANS TEZİ

BİR DİZEL MOTORUNUN ENERJİ VE EKSERJİ ANALİZLERİ

Erdi TOSUN

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Bu çalışma, dört zamanlı, dört silindirli, doğal emişli, direk enjeksiyonlu bir dizel motorunda, farklı yakıtlar kullanılarak, elde edilen deneysel verilere uygulanan enerji ve ekserji analizlerinin değerlendirilmesiyle ilgilidir. Tüm testler 2600 d/d motor hızında gerçekleştirilmiştir. Termodinamiğin birinci ve ikinci yasasını uygulayabilmek için hava ve yakıt debilerine, motor hızına, emisyonlara, ilgili sıcaklıklara ihtiyaç duyulmaktadır. Bu verileri ve kontrol hacmi için enerji ve ekserji dengelerini kullanarak, nicelik ve nitelikle ilgili olan 1. ve 2. Yasa verimleri bulunmuştur. Enerji ve ekserji performanslarına yakıt etkisini görebilmek için bütün sonuçlar birbirleriyle karşılaştırılmıştır. Sonuç olarak, dizel yakıtı, biyodizellerden daha iyi enerji ve ekserji performansı göstermiştir.

Anahtar Kelimeler: Enerji, Ekserji, İçten Yanmalı Motorlar, Alternatif Yakıt

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1. INTRODUCTION

The scarcity of known petroleum reserves will make renewable energy resources more attractive. The most feasible way to meet this growing demand is by utilizing alternative fuels (Demirbas, 2007). Exploring new energy resources, such as biodiesel fuel, is of growing importance in recent years. The use of biodiesel as a fuel has been widely investigated. Its commercial use as a diesel substitute began in Europe in the late 1980s (Zhang et. al., 2003).

The increasing industrialization and motorization of the world has led to a steep rise for the demand of petroleum products. Petroleum based fuels are obtained from limited reserves. These finite reserves are highly concentrated in certain regions of the world. Therefore, those countries not having these resources are facing a foreign exchange crisis, mainly due to the import of crude oil. Hence, it is necessary to look for alternative fuels, which can be produced from materials available within the country. In addition, the use of vegetable oil as fuel is less polluting than petroleum fuels (Ramadhas et. al., 2004).

The diesel engines dominate the field of commercial transportation and agricultural machinery due to its ease of operation and higher fuel efficiency. The consumption of diesel oil is several times higher than that of petrol. Due to the shortage of petroleum products and its increasing cost, efforts are on to develop alternative fuels especially, to the diesel oil for fully or partial replacement. It has been found that the vegetable oils are promising fuels because their properties are similar to that of diesel and are produced easily and renewably from the crops (Ramadhas et. al., 2005).

An alternative fuel to petro-diesel must be technically feasible, economically competitive, environmentally acceptable and easily available. This current alternative diesel fuel can be termed as biodiesel. Biodiesel usage may improve emissions levels of some pollutants and deteriorate other. However, for quantifying the effect of biodiesel it is important to take into account several other factors such as raw material, driving cycle, vehicle technology etc. Usage of biodiesel will allow a

balance to be sought between agriculture, economic development and the environment (Demirbas, 2007).

Biodiesel is defined as the monoalkyl esters of vegetable oils or animal fats. The vegetable oils and fats as alternative engine fuels are all extremely viscous with viscosities ranging from 10 to 17 times greater than petroleum diesel fuel. Biodiesel is produced by transesterifying the parent oil or fat to achieve a viscosity close to that of petro-diesel. The chemical conversion of the oil to its corresponding fatty ester (biodiesel) is called transesterification. The purpose of the transesterification process is to lower the viscosity of the oil. Transesterification refers to a reaction between an ester (triglyceride) of one alcohol (glycerin) and a second alcohol (methanol) to form an ester of the second alcohol (methyl ester). Hundreds of scientific articles and various other reports from around the world dealing with vegetable oil-based alternative diesel fuels or biodiesel have appeared in print. Transesterification of triglycerides (vegetable oils and animal fats) to generate monoalkyl esters and glycerin to obtain a new engine fuel (biodiesel) is a well-proved process. Renewable biomass has also been considered as potential feedstock for vegetable oils to produce biodiesel; the economics of present processes prefer the use of nonedible and waste cooking oils. (Demirbas, 2007).

More than 100 years ago, a brilliant inventor named Rudolph Diesel designed the original diesel engine to run on vegetable oil. Dr. Rudolph Diesel used peanut oil to fuel one of this his engines at the Paris Exposition of 1900. Among them, biodiesel, produced from different vegetable oils (soybean, rapeseed and sunflower, for example), seems very interesting for several reasons: it can replace diesel oil in boilers and internal combustion engines without major adjustments; only a small decrease in performances is reported; almost zero emissions of sulfates; a small net contribution of carbon dioxide (CO_2) when the whole life-cycle is considered (including cultivation, production of oil and conversion to biodiesel); emission of pollutants comparable with that of diesel oil. For these reasons, several campaigns have been planned in many countries to introduce and promote the use of biodiesel (Bozbas, 2008).

World biodiesel production trend is increasing day by day due to the depleting petroleum based fuels. Figure 1.1 shows world biodiesel production year by year between 1991 and 2012. It is clearly seen from figure, demand for alternative fuel, biodiesel, is increasing drastically.

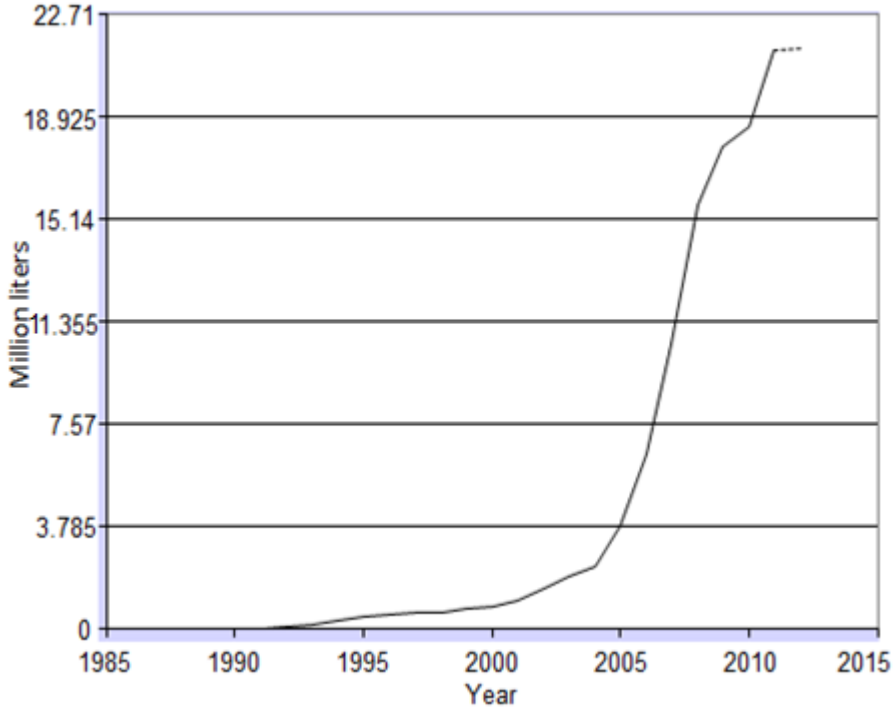

Figure 1.1. World biodiesel production (1991-2012) [Anonymous]

The general European Union (EU) policy objectives considered most relevant to the design of energy policy are: (1) competitiveness of the EU economy, (2) security of energy supply, and (3) environmental protection. All renewable energy policies should be measured by the contributions they make to these goals. Current EU policies on alternative motor fuels focus on the promotion of biofuels. In a proposed biofuels directive the introduction of a mandatory share scheme for biofuels, including as from 2009 minimum blending shares. In the Commission's view mandating the use of biofuels will (i) improve energy supply security and (ii) reduce greenhouse gas (GHG) emissions and (iii) boost rural incomes and employment. Current regulations would preclude a notable negative impact on the rural environment (Bozbas, 2008).

Internal combustion engine simulation modeling has long been established as an effective tool for studying engine performance and contributing to evaluation and new developments. Thermodynamic models of the real engine cycle have served as effective tools for complete analysis of engine performance and sensitivity to various operating parameters. It has long been understood that traditional first-law analysis, which is needed for modeling the engine processes, often fails to give the engineer the best insight into the engine's operation. In order to analyze engine performance that is, evaluate the inefficiencies associated with the various processes second-law analysis must be applied. For second-law analysis, the key concept is 'availability' (or exergy). The availability content of a material represents its potential to do useful work. Unlike energy, availability can be destroyed which is a result of such phenomena as combustion, friction, mixing and throttling. The destruction of availability -often termed irreversibility- is the source for the defective exploitation of fuel into useful mechanical work in a compression or spark ignition engine. The reduction of irreversibilities can lead to better engine performance through a more efficient exploitation of fuel. To reduce the irreversibilities, we need to quantify them. That is we need to evaluate the availability destructions-we need the second-law analysis (Rakopoulos et. al., 2006).

Energy is a key element of the interactions between nature and society. It is also considered a key input for economic development and as everyone knows, there is no source of energy which is absolutely neutral with respect to the environment. Under these circumstances, energy has an interdisciplinary nature, covering sustainable development and environmental impact. In this regard, all energy policies should be sound and make sense in global terms, i.e., become an integral part of the international process of energy system adaptation, which will recognize the very strong linkage existing between energy requirements and emissions of pollutants. The impact of energy resource utilization on the environment and the achievement of increased resource utilization efficiency are best addressed by considering exergy. The exergy of an energy form or a substance is a measure of its usefulness or quality or potential to cause change. The latter point suggests that exergy may be, or provide the basis for, an effective measure of the potential of a substance or energy form to

impact the environment. It is important to mention that in practice a thorough understanding of exergy and the insights it can provide into the efficiency, environmental impact and sustainability of energy systems, are required for the engineer or scientist working in the area of energy systems and the environment. Further, as energy policies increasingly play an important role in addressing sustainability issues and a broad range of local, regional and global environmental concerns, policy makers also need to appreciate the exergy concept and its ties to these concerns (Dincer, 2002).

Renewable energy sources can be a good substitute of the fossil fuels which are being terminated fast. Nowadays biomass and biofuels are considered because of their environment friendly characteristics and their ability of supplying much more energy. An alternative means to select the most efficient and convenient biomass, is exergy analysis. This thermodynamic analysis technique estimates the efficiency of the process and determines the energy quality and usefulness. Exergy analysis makes us able to specify the maximum performance of a system and the sources of the irreversibilities. During the past decades exergy analysis had been used to evaluate the performance of different systems and to improve their efficiencies. The relationship between the exergy analysis and the sustainability shows that diminution the loss of energy quality (exergy) leads to increment of the sustainability of energy use (Saidur et. al., 2012).

Energy resources and their utilization intimately relate to sustainable development. In attaining sustainable development, increasing the energy efficiencies of processes utilizing sustainable energy resources plays an important role. The utilization of renewable energy offers a wide range of exceptional benefits. There is also a link between exergy and sustainable development. A sustainable energy system may be regarded as a cost-efficient, reliable, and environmentally friendly energy system that effectively utilizes local resources and networks. Exergy analysis has been widely used in the design, simulation and performance evaluation of energy systems (Hepbasli, 2008).

In this study, energy and exergy analysis were applied to the experimental data of a diesel engine fuelled with Diesel No:2 and alternative fuels. Canola and

Peanut Biodiesels were used as alternative fuels during the experiments. 1st and 2nd law (energy and exergy) efficiencies related with quantity and quality were determined and compared with each other for various fuels.

2. PRELIMINARY WORK

2.1. Biodiesel Literature Review

In the late 1970's and early 1980's, taking place of fuel and energy crises was a major and need to be tackled problem in the world. Decreasing petroleum-derived fuels day by day accelerated the interest of researching alternative fuels to use as an energy source. Biodiesel, which can be used as an alternative diesel fuel, is made from renewable biological sources such as vegetable oil and animal fats. It is biodegradable, non-toxic and possesses low emission profiles. Also, the uses of bio-fuels are environmentally beneficial. It can be used in compression ignition engines with little or no modifications (Ozcanli, 2009). In this respect, many researchers investigated production methods, fuel properties, performance and emissions characteristics of diesel engines fuelled with various biodiesels.

Ozsezen et. al. (2011) investigated the performance, combustion and injection characteristics of a direct injection diesel engine experimentally when it was fueled with canola oil methyl ester (COME) and waste (frying) palm oil methyl ester (WPOME). In order to determine the performance and combustion characteristics, the experiments were conducted at constant engine speeds under the full load condition of the engine. The results indicated that when the test engine was fueled with WPOME or COME instead of petroleum based diesel fuel (PBDF), the brake power reduced by 4–5%, while the brake specific fuel consumption increased by 9–10%. On the other hand, methyl esters caused reductions in carbon monoxide (CO) by 59–67%, in unburned hydrocarbon (HC) by 17–26%, in carbon dioxide (CO_2) by 5–8%, and smoke opacity by 56–63%. However, both methyl esters produced more nitrogen oxides (NO_x) emissions by 11–22% compared with those of the PBDF over the speed range.

Altin et. al. (2001) investigated the effect of vegetable oil fuels and their methyl esters (raw sunflower oil, raw cottonseed oil, raw soybean oil and their methyl esters, refined corn oil, distilled opium poppy oil and refined rapeseed oil) on a direct injected, four stroke, single cylinder diesel engine performance and exhaust

emissions. The results showed that both vegetable oils and their esters are promising alternatives as fuel for diesel engines. Vegetable oils still have some problems due to high viscosity, drying with time and thickening in cold conditions such as flow, atomization and heavy particulate emissions.

Graboski et. al. (1998) reviewed that the status of fat and oil derived diesel fuels with respect to fuel properties, engine performance, and emissions. The fuels considered are primarily the methyl esters of fatty acids derived from a variety of vegetable oils and animal fats, and referred to as biodiesel. The major obstacle to widespread use of biodiesel is the high cost relative to petroleum. Economics of biodiesel production are discussed, and it is concluded that the price of the feedstock fat or oil is the major factor determining biodiesel price. Biodiesel is completely miscible with petroleum diesel fuel, and is generally tested as a blend. The use of biodiesel in neat or blended form has no effect on the energy based engine fuel economy. The lubricity of these fuels is superior to conventional diesel, and this property is imparted to blends at levels above 20 vol%. Emissions of PM can be reduced dramatically through use of biodiesel in engines that are not high lube oil emitters. Emissions of NO_x increase significantly for both neat and blended fuels in both two- and four-stroke engines. The increase may be lower in newer, lower NO_x emitting four-strokes, but additional data are needed to confirm this conclusion. A discussion of available data on unregulated air toxins is presented, and it is concluded that definitive studies have yet to be performed in this area. A detailed discussion of important biodiesel properties and recommendations for future research is presented. Among the most important recommendations is the need for all future studies to employ biodiesel of well-known composition and purity, and to report detailed analyses. The purity levels necessary for achieving adequate engine endurance, compatibility with coatings and elastomers, cold flow properties, stability, and emissions performance must be better defined.

Canakci et. al. (2008) presented both biodiesel production from various feedstocks and their effects on the fuel properties. Even though biodiesel has some superiorities, the greatest barrier to biodiesel use is its cost compared to conventional diesel fuel. Therefore, it is necessary to economize biodiesel production to improve

its marketability. Compared to high-quality vegetable oils, less expensive feedstocks, such as used frying oils, soapstocks, fats and greases, can be used to lower the cost of biodiesel. However, for these feedstocks, transesterification cannot be applied directly due to their high level of FFA. The cold flow properties of the biodiesel, especially produced from fats are unacceptably worse since they have high saturation level. However, when branched-chain alcohols are used, positive results in the cold flow properties can be obtained. Therefore, the studies should focus on improvement of the cold-flow properties of the biodiesels produced from the low-cost feedstocks.

Canakci et. al. (2003) investigated the effect of the biodiesel produced from high free fatty acid feedstocks on engine performance and emissions. Two different biodiesels were prepared from animal fat–based yellow grease with 9% free fatty acids and from soybean oil. The neat fuels and their 20% blends with No. 2 diesel fuel were studied at steady–state engine operating conditions in a four–cylinder turbocharged diesel engine. Although both biodiesel fuels provided significant reductions in particulates, carbon monoxide, and unburned hydrocarbons, the oxides of nitrogen increased by 11% and 13% for the yellow grease methyl ester and soybean oil methyl ester, respectively. The conversion of the biodiesel fuel's energy to work was equal to that from diesel fuel.

Ozcanli (2009) used Ricinus Communis oil, Pistacia Terebinthus oil and waste chicken oil as raw materials for the biodiesel production in his study. In addition to biodiesel production, engine performance and exhaust emission experimental studies were performed. The results showed power and torque values of all biodiesels tended to decrease at high engine speeds. Due to lower calorific value of biodiesel fuels, higher specific fuel consumption values were measured with the blend fuels. CO and CO_2 emissions decreased, NO_x emissions increased with usage of biodiesel.

Tuccar (2011) investigated growth characteristics of microalgae and to determine the effect of microalgae biodiesel to engine performance parameters and exhaust emissions. The engine performance and exhaust emission measurement experiments showed that the power and torque output of engine fuelled with Microalgae Biodiesel (MB) decreased by 6% and 5.3% respectively, the emission

values were improved with MB usage, namely CO,CO_2 and NO_X emissions decreased by 9.4%, 4.8% and 11.7%, respectively.

Demirbas (2008) compared transesterification methods for production of biodiesel from vegetable oils and fats. Biodiesel is obtained from a chemical reaction called transesterification (ester exchange). The reaction converts esters from long chain fatty acids into mono alkyl esters. Chemically, biodiesel commonly is a fatty acid methyl ester. Vegetable oils can be transesterified by heating them with a large excess of anhydrous methanol and an acidic or basic reagent as catalyst. A catalyst is usually used to improve the reaction rate and yield. In a transesterification reaction, a larger amount of methanol was used to shift the reaction equilibrium to the right side and produce more methyl esters as the proposed product. Several aspects including the type of catalyst (alkaline, acid or enzyme), alcohol/vegetable oil molar ratio, temperature, purity of the reactants (mainly water content) and free fatty acid content have an influence on the course of the transesterification. A non-catalytic biodiesel production route with supercritical methanol has been developed that allows a simple process and high yield because of the simultaneous transesterification of triglycerides and methyl esterification of fatty acids. In the catalytic supercritical methanol transesterification method, the yield of conversion rises to 60–90% for the first 1 min.

2.2. Energy and Exergy Literature Review

Thermodynamic analysis plays an important role for designing, assessing and defining inefficiencies of a thermal system. Therefore, these analyses give important information about the system and help to improve system by reducing inefficiencies.

Thermodynamics permits the behavior, performance and efficiency to be described for systems for the conversion of energy from one form to another. Conventional thermodynamic analysis is based primarily on the first law of thermodynamics, which states the principle of conservation of energy. An energy analysis of an energy-conversion system is essentially an accounting of the energies entering and exiting. The exiting energy can be broken down into products and

wastes. Efficiencies are often evaluated as ratios of energy quantities, and are often used to assess and compare various systems. (Dincer and Rosen, 2007).

In recent years, exergy analysis method has been widely used in the design, simulation and performance assessment of various thermal systems. In this regard, exergy method may be applied to various types of engines for identifying losses and efficiencies (Caliskan et. al., 2009).

Canakci et. al. (2006) made comparative energy and exergy analyses of a four-cylinder, turbocharged diesel engine using two different biodiesel fuels, petroleum diesel fuel and blends of the biodiesels with petroleum diesel. Utilizing experimental data obtained from steady-state tests, balances of energy and exergy rates for the engine were determined. Then, various performance parameters of the engine were evaluated for each fuel operation. It was found that the tested biodiesels offer almost the same energetic performance as petroleum diesel fuel, while exergetic performance parameters usually follow similar trends with the corresponding energetic ones.

Sayin et. al. (2006) presented comparative energy and exergy analyses of a four-cylinder, four-stroke spark-ignition engine using gasoline fuels of three different research octane numbers (RONs), namely 91, 93 and 95.3. Each fuel test was performed by varying the engine speed between 1200 and 2400 rpm while keeping the engine torque at 20 and 40 Nm. Then, using the steady-state data along with energy and exergy rate balance equations, various performance parameters of the engine were evaluated for each fuel case. It was found that the gasoline of 91-RON, the design octane rating of the test engine, yielded better energetic and exergetic performance, while the exergetic performance parameters were slightly lower than the corresponding energetic ones. Furthermore, this study revealed that the combustion was the most important contributor to the system inefficiency, and almost all performance parameters increased with increasing engine speed.

Gokalp et. al. (2009) performed comparative energy and exergy analyses of a No. 2 diesel fuel, marine fuel, and four various biodiesel fuels for a four-cylinder, four-stroke, direct injection diesel engine. The engine was used to obtain the experimental data at full-load conditions for constant 1600 rpm speed. It was found

that the first and second-law efficiencies increased with biodiesel addition and the minimum exergy destruction was obtained at 100% biodiesel fuels. On the other hand, exhaust exergy was found to be inversely proportional to the biodiesel addition. It is also shown with extensive emission analysis that the addition of biodiesel fuels to the standard diesel fuel enhances the emission characteristics of diesel engines.

Tat (2011) studied the energy, exergy and heat release analysis of a John Deere 4045 T 4.5 L, four-stroke, four-cylinder, turbocharged diesel engine. The engine was run with four different types of fuel: yellow grease methyl ester (YGME); soybean oil methyl ester (SME); and soybean oil methyl ester containing either 0.75 or 1.5 w/w % of the cetane improver 2-ethylhexyl nitrate (SME-0.75%EHN and SME-1.5%EHN, respectively). The engine was tested at 1400 1/min under a full load of 352 Nm. For reliability, the fuels were tested three times, and the mean values were compared using different statistical techniques. The objective in this study was to determine the effect of cetane number and ignition delay on the energy and exergy efficiencies of an internal combustion engine and to compare the results for the types of fuel stated earlier. The average thermal efficiency was approximately 40.5%, and the exergetic efficiency was approximately 37.3%. The mean exergetic efficiencies of the fuels were in the order $\psi_{SME} > \psi_{SME-0.75\%EHN} > \psi_{SME-1.5\%EHN} > \psi_{YGME}$. There were significant differences among the mean values according to Student's t-tests. It is concluded that a lower cetane number, a longer ignition delay period and a higher level of premixed combustion may increase the exergetic efficiency of a diesel engine.

Caliskan et. al. (2009) investigated the effect of varying dead state temperatures on exergy efficiency of a high-oleic methyl ester (HOME) fuelled internal combustion engine (ICE). This engine is a 4.5L, four-stroke, four-cylinder, turbocharged, 66.5 kW maximum power capacity John Deere 4045T diesel engine run with HOME, which is genetically modified with a high-oleic soybean oil methyl ester. The test speed is 1400 min^{-1} at a full load. In the analysis, actual operational data obtained from a study conducted by one of the co-authors at Iowa State University, USA are used. Exergy efficiency values at various dead state

temperatures are calculated for comparison purposes since these types of engines may be operated under different outdoor air conditions. The results obtained are discussed from the exergetic point of view. It was found that exergetic efficiency increased as dead state temperature decreased. As a result, exergy efficiency values ranged from 29.78% to 34.93% based on dead state temperatures between -5 ^{o}C and 30 ^{o}C.

Alasfour (1997) studied the availability analysis of a spark-ignition engine using a butanol-gasoline blend has been experimentally investigated. A Hydra single-cylinder, spark-ignition, fuel-injection engine was used over a wide range of fuel/air equivalence ratios (ϕ = 0.8-1.2) at a 30% volume butanol-gasoline blend. The goal of this research was to study the effect of using a butanol-gasoline blend in a spark-ignition engine in terms of first- and second-law efficiency. In addition, the optimal engine conditions of energy utilization were investigated. Results show that, at ϕ = 0.9, when a butanol-gasoline blend is used, the energy analysis indicates that only 35.4% of the fuel energy can be utilized as an indicated power, where 64.6% of fuel energy is not available for conversion to useful work. The availability analysis shows that 50.6% of fuel energy can be utilized as useful work (34.28% as an indicated power, 12.48% from the exhaust and only 3.84% from the cooling water) and the available energy unaccounted for represents 49.4% of the total available energy.

Sekmen et. al. (2011) employed the first and second laws of thermodynamics to analyze the quantity and quality of energy in a four-cylinder, direct injection diesel engine using petroleum diesel fuel and biodiesel fuel. The experimental data were collected using steady-state tests which enable accurate measurements of air, fuel and cooling water flow rates, engine load, and all the relevant temperatures. Balances of energy and exergy rates for the engine were determined and then various performance parameters and energy and exergy efficiencies were calculated for each fuel operation and compared with each other. The results of tested biodiesel offer similar energetic performance as petroleum diesel fuel. In addition to this, the exergetic performance parameters usually follow similar trends according to the energetic performance parameters.

Caliskan et. al. (2010a) presented energy and exergy analyses of a John Deere 4045T diesel engine run with no. 2 diesel fuel, Soybean oil Methyl Ester (SME) and High-Oleic soybean oil Methyl Ester (HOME) at 1400 1/min. It was aimed at determining energy and exergy efficiencies, energy losses and exergy destructions of the combustion process and comparing exergetically the fuels used. The specific exergy of the fuels was calculated to be $\varepsilon_{fuel,No.2\ Diesel} > \varepsilon_{fuel,HOME} > \varepsilon_{fuel,SME}$, while energy (thermal) and exergy efficiencies were 40.5% and 37.8%, respectively.

Rakopoulos et. al. (2006) surveyed the publications available in the literature concerning the application of the second-law of thermodynamics to internal combustion engines. The availability (exergy) balance equations of the engine cylinder and subsystems are reviewed in detail providing also relations concerning the definition of state properties, chemical availability, flow and fuel availability, and dead state. Special attention is given to identification and quantification of second-law efficiencies and the irreversibilities of various processes and subsystems. The latter being particularly important since they are not identified in traditional first-law analysis. In identifying these processes and subsystems, the main differences between second- and first-law analyses are also highlighted. A detailed reference is made to the findings of various researchers in the field over the last 40 years concerning all types of internal combustion engines, i.e. spark ignition, compression ignition (direct or indirect injection), turbocharged or naturally aspirated, during steady-state and transient operation. All of the subsystems (compressor, aftercooler, inlet manifold, cylinder, exhaust manifold, turbine), are also covered. Explicit comparative diagrams, as well as tabulation of typical energy and exergy balances, are presented. The survey extends to the various parametric studies conducted, including among other aspects the very interesting cases of low heat rejection engines, the use of alternative fuels and transient operation. Thus, the main differences between the results of second- and first-law analyses are highlighted and discussed.

Kopac et. al. (2005) applied energy and exergy analysis to the experimental data of an internal combustion engine operating on the conventional Otto cycle. The data are collected using an engine test unit which enables accurate measurements of

fuel flow rate, combustion air flow rate, engine load, engine speed and all the relevant temperatures. Energy and exergy efficiencies are calculated for different engine speeds and compared. Results indicate that energy efficiency is maximum at a speed of 2040 rpm whereas exergy efficiency is maximum at a speed of 2580 rpm.

Azoumah et. al. (2009) suggested the use of exergy analysis (as an environmental assessment tool to account wastes and determine the exergy efficiency) combined with gas emissions analysis to optimize the performance of a compression ignition (CI) engine using biofuels such as cottonseed and palm oils, pure or blended with diesel for different engine loads. The results show that the combination of exergy and gas emissions analyses is a very effective tool for evaluating the optimal loads that can be supplied by CI engines. Taking into account technical constraints of engines, a tradeoff zone of engine loads (60% and 70% of the maximum load) was established between the gas emissions (NO and CO_2) and the exergy efficiency for optimal performance of the CI engine.

Caliskan et. al. (2010b) presented a review study on energy and exergy analysis of Otto and Diesel engines. Calculation methods of the analysis are discussed in detail. Previous studies, from 1963 to 2008, are chronologically listed and studied. The test engines had different cylinder numbers, speeds and rated powers. The best exergetic efficiency is achieved with four-stroke, four-cylinder, turbocharged Diesel engines at about 30% excepting to stationary Diesel engine. It is considered that exergetic efficiency can be higher at lower speeds between 1140 1/min and 2200 1/min.

There are lots of studies about application of second law of thermodynamics to various types of internal combustion engines. These works can be summarized in the following table in between dates of 1963-2008 (Caliskan, 2009).

Table 2.1. Literature review

Year	Researcher	Publication	Ignition	Cycle	Cylinder	Power (kW)	Speed (rpm)
1963	Patterson and Van Wylen	SAE	SI	4	1	13	2800
1983	Beretta and Keck	Combust. Sci. Technol.	-	4	-	-	-
1984	Flynn et. al.	SAE	CI-DI	4	6	300	2100
1984	Primus	SAE	CI-DI	4	6	268	1900
1984	Primus et. al.	SAE	CI-DI	4	6	185/220	2100
1986	Primus and Flynn	ASME	CI-DI	4	6	224	2100
1987	Zellat	Entropie	CI	4	NA	109.5/cyl.	338
1988	Alkidas	ASME	CI-DI	4	1	3.14-33	1200/1800
1988	Lior and Rudy	Energy Conv. Mgmt.	SI	-	-	-	-
1988	McKinley and Primus	SAE	CI-DI	4	6	224	2100
1989	Alkidas	SAE	CI-DI	4	1	3.14/33	1200/1800
1989	Lipkea and DeJoode	SAE	CI-DI	4	6	170	2200
1989	Shapiro and Van Gerpen	SAE	SI-CI	4	1	NA	NA
1989	Kumar et. al.	Int. Com. Heat Mass T.	CI-DI	4	1	NA	2000
1990	Van Gerpen and Shapiro	ASME	CI	4	1	NA	NA
1991	Boza et. al.	SAE	CI	4	4	55.6	4500
1991	Zhecheng et. al.	SAE	SI	4	4	76	5500

1991	Sato et. al.	SAE	SI	2	1	NA	2500
1992	Gallo and Milanez	SAE	SI	4	1	NA	2000/5200
1993	Rakopoulos	Energy Conv. Mgmt.	SI	4	1	NA	2500
1993	Rakopoulos and Andritsakis	ASME	CI-DI	4	1	NA	1500/2500
1993	Rakopoulos et. al.	Heat Recov. Syst. CHP	CI-DI	4	1	NA	1500/2500
1994	Velasquez and Milanez	SAE	CI-DI	4	1	NA	3200
1995	Li et. al.	SAE	CI-IDI	4	1	NA	2000
1997	Alasfour	Appl. Therm. Eng.	SI	4	1	4.9	1700
1997	Rakopoulos and Giakoumis	Appl. Therm. Eng.	CI	4	6	236	1500
1997	Rakopoulos and Giakoumis	Energy	CI	4	1	NA	1350/2250
1997	Rakopoulos and Giakoumis	Energy Conv. Mgmt.	CI-IDI	4	6	236	1500
1997	Fijalkowski and Nakonieczny	Proc. Inst. Mech. Engrs.	CI	4	6	NA	2200
1998	Anderson et. al.	SAE	SI-Miller	4	4	6.66	2000
1999	Kohany and Sher	SAE	SI	4	V8	-	1400
1999	Köktürk	YÖK	SI	4	4	11.7/37.54	990/3480
2000	Caton	Energy	Adiab. Const. V.	-	-	-	-
2000	Caton	SAE	SI	4	V8	21.9	700/2800
2001	Kyritsis and Rakopoulos	SAE	CI-DI	4	1	NA	2000

2001	Rakopoulos and Kyritsis	Energy	CI-DI	4	1	NA	NA
2002	Caton	SAE	SI	4	V8	21.9	1400
2002	Nakonieczny	Energy	CI	4	4	52	2850
2002	Abdelghaffar et. al.	ASME	CI	4	4	25-152 Nm	1000/2000
2003	Sobiesiak and Zhang	SAE	SI	4	V8	NA	4000
2004	Rakopoulos and Giakoumis	Energy	CI	4	6	236	1500
2004	Rakopoulos and Giakoumis	SAE	CI	4	6	236	1500
2004	Uçkun	YÖK	DI	4	4	57.1	2100
2005	Kanoğlu et. al.	Energy Conv. Mgmt.	DI	-	18	5808 kW/m^2	514
2005	Parlak	Energy Conv. Mgmt.	CI-IDI	4	1	3.1/6.7	1000/2000
2005	Parlak et. al.	Energy Conv. Mgmt.	CI-DI	4	6	136	2400
2005	Rakopoulos and Giakoumis	Appl. Therm. Eng.	CI-IDI	4	6	236	1500
2005	Caton	SAE	SI	4	V8	-	2500
2005	Rakopoulos and Giakoumis	SAE	CI-IDI	4	6	236	1500
2005	Kopac and Köktürk	Int. Journal of Exergy	SI	NA	NA	103/135 Nm	990/3480
2005	Ozcan and Soylemez	Int. Journal of Exergy	SI	4	4	NA	2000
2006	Rakopoulos and Giakoumis	Energy	CI	4	6	236	1500
2006	Rakopoulos and Kyritsis	Hydrogen Energy	CI-SI	-	-	-	-

2007	Sayin et. al.	Int. Jour. Of Energy Res.	SI	4	4	58.88	3400
2007	Yılbaşı	YÖK	DI	4	4	46	2400
2008	Kanoglu et. al.	Int Journal of Exergy	-	4	4	19000	NA
2008	Abusoglu and Kanoglu	Appl. Therm. Eng.	DI	-	18	5808 kW/m^2	514

3. MATERIAL AND METHOD

3.1. Material

3.1.1 Engine Test Rig

Engine performance experiments were conducted on a four cylinder, four stroke, naturally aspirated, direct injection diesel engine. Technical specifications of the engine were given in Table 3.1. Diesel engine test rig and its schematic representation were shown in Figure 3.1 and Figure 3.2.

Table 3.1. Technical specifications of the engine

Brand	Mitsubishi Canter
Model	4D34-2A
Configuration	In line 4
Type	Direct injection diesel with glow plug
Displacement	3907 cc
Bore	104 mm
Stroke	115 mm
Power	89 kW @ 3200 rpm
Torque	295 Nm @ 1800 rpm
Oil cooler	Water cooled
Air cleaner	Paper element type
Weight	325 kg



Figure 3.1. Diesel engine test rig

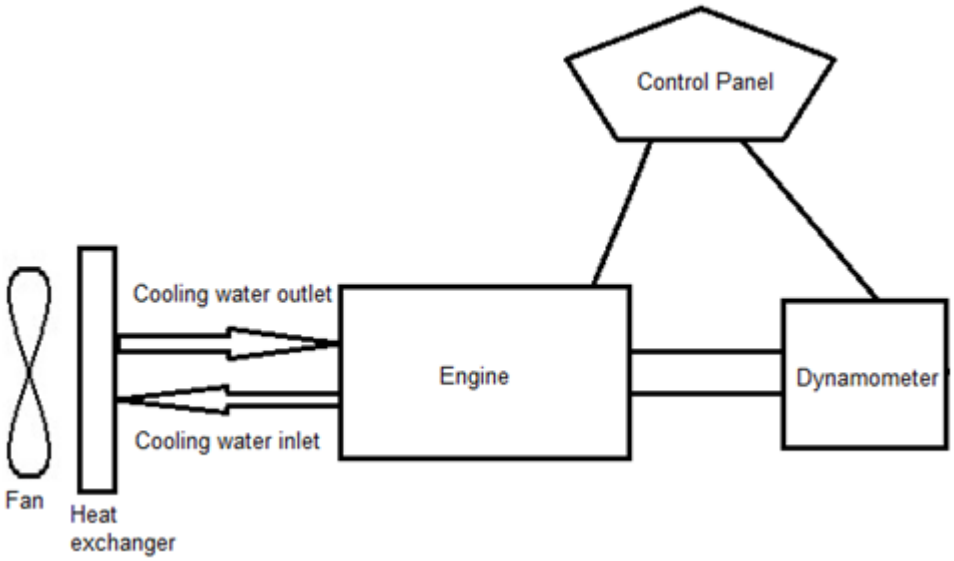

Figure 3.2. Schematic representation of the engine test rig

A hydraulic dynamometer was used for determination of torque and power output. Table 3.2 shows technical specifications of the dynamometer.

Table 3.2. Technical specifications of the dynamometer

Torque Range	0-1700 Nm
Speed Range	0-7500 rpm
Body Weight	45 kgf
Total Weight	110 kgf
Body Diameter	350 mm
Torque Arm Length	350 mm

TESTO 350 XL gas analyzer was used to measure exhaust emissions. Emission data was collected by the help of a computer program. Accuracy of the gas analyzer is ±10 ppm for CO, 1% for CO_2 and ±1 ppm for NO_x. Table 3.3 shows measurement ranges, accuracy and resolution values of emission device.

Table 3.3. Measurement ranges, accuracy and resolution values of emission device

	CO	CO_2	NO	NO_2
Measurement range	0 to +500 ppm CO	0 to +50 Vol. % CO_2	0 to +4000 ppm NO	0 to +500 ppm NO_2
Accuracy	±5% of mv (+40 to +500 ppm CO) ±2 ppm CO (0 to +39.9 ppm CO)	±0.3 Vol. % CO_2 + 1% of mv (0 to 25 Vol. % CO_2) ±0.5 Vol. % CO_2 + 1.5% of mv (>25 to 50 Vol. % CO_2)	±5% of mv (+100 to +1999.9 ppm NO) ±10% of mv (+2000 to +3000 ppm NO) ±5 ppm NO (0 to +99 ppm NO)	±5% of mv (+100 to +500 ppm NO_2) ±5 ppm NO_2 (0 to +99.9 ppm NO_2)
Resolution	0.1 ppm CO (0 to +500 ppm CO)	0.01 Vol. % CO (0 to 25 Vol. % CO2) 0.1 Vol. % CO_2 (>25 Vol. % CO_2)	1 ppm NO (0 to +3000 ppm NO)	0.1 ppm NO_2 (0 to +500 ppm NO_2)

3.1.2. Measurement Devices

The fuel quality measurements were done according to TS EN 14214 and EN 590. The testing equipments and their measuring methods were briefly explained below.

3.1.2.1. Density Measurement

The density measurement was done using Kyoto Electronics DA-130 type density-meter (Figure 3.3). This device uses the resonant frequency method to measure the densities. The measurement interval of the device is 0 to 2 g/cm^3 and 0 to 40 ºC. The device has a sensitivity of ±0.001 g/cm^3, and a stability of 0.0001 g/cm^3. The device is measuring according to the standards of TS 6311, ASTM D 4052-96.



Figure 3.3. Kyoto Electronics DA-130 portable densimeter

3.1.2.2 Cetane Number Measurement

Cetane number is a fuel property which indicates ignition quality of fuel. High cetane number means short ignition delay which cause faster combustion in combustion chamber.

Cetane Number and indexes were measured by Zeltex ZX440 (Figure 3.4), which works under the close infrared spectrometer (NIR) principal. With the help of this principal Cetane number measurement became very fast and cheap with only 3% error compared to the time consuming expensive motor tests.



Figure 3.4. Zeltex ZX440

3.1.2.3. Flash Point Measurement

The flash point is the lowest temperature at which a combustible mixture can be formed above the liquid fuel. It is important for fire safety considerations and depends on both the lean flammability limit of the fuel as well as the vapour pressure of the fuel constituents (Challen and Baranescu, 1999).

Tanaka Automated Pensky-Martens Closed Cup Flash Point Tester with APM-7 model number (Figure 3.5) was used to determine flash point of test fuels. It is conforming the standarts, ISO 2719, ASTM D93, IP 34, ETC. Measuring range is from ambient temperature to 370 ^{o}C.



Figure 3.5. Tanaka APM-7

3.1.2.4. Cold Flow Measurement

A fuel property that is particularly important for the low temperature operability of diesel fuel is the cloud point. The cloud point is the temperature at which a cloud of wax crystals first appears in a liquid upon cooling. Therefore, it is an index of the lowest temperature of the fuel's utility under certain applications. Operating at temperatures below the cloud point for a diesel fuel can result in fuel filter clogging due to the wax crystals. As described in ASTM D 2500, the cloud point is determined by visually inspecting for a haze in the normally clear fuel, while the fuel is cooled under carefully controlled conditions.

A second measure of the low temperature performance of diesel/biodiesel fuels is the pour point. The pour point is the lowest temperature at which a fuel sample will flow. Therefore, the pour point provides an index of the lowest temperature of the fuel's utility for certain applications. The pour point also has implications for the handling of fuels during cold temperatures. The standard procedure for measuring the pour point of fuels is ASTM D 97.

Tanaka MPC-102 (Figure 3.6) was used to determine pour point of fuels. Pour Point measurement is made utilizing a new ASTM test method, namely "Air Pressure Method" (ASTM D6749 on "Standard Test Method for Pour Point of Petroleum Products (Automatic Air Pressure Method 1)), which yields eventually no bias against the conventional test method, repeatability/reproducibility of $1^{o}C$ /$2^{o}C$

and 2-3 times faster determinations. The device can measure according to the ASTM D6749/D97, ISO 3016 (PP), ASTM D2500, ISO 3015 (CP). Measuring range is +51 ^{o}C to – 40 ^{o}C with tap water of 20 ^{o}C and +51 ^{o}C to – 65 ^{o}C with cooling liquid of -35 ^{o}C. Amount of applied air pressure for pour point detection, to accommodate different sample types: L (low) for diesel fuels, H (high) for lube oils, VH (very high) and UH (ultra high) for residual fuels and similar samples. Testing intervals: 1.0 ^{o}C, 2.5 ^{o}C, 3.0 ^{o}C.



Figure 3.6. Tanaka MPC-102L

3.1.2.5. Viscosity Measurement

The viscosities of the fuels were measured with Saybolt Universal Viscosimeter (Figure 3.7) produced from Ubbelohde tube with ASTM D88 standards. The measurement results were recorded in seconds. Then using a conversion table the results were converted from SSU (Saybolt Universal Second) to centistokes (cSt) unit. The measurements were done at 40 ºC according to the TS EN 14214 standard.



Figure 3.7. Saybolt Universal Viscosimeter

3.1.3. Biodiesel Resources

Vegetable oils are becoming a promising alternative to diesel fuel because they are renewable in nature and can be produced locally and in environmentally friendly ways. Edible vegetable oils such as canola and soybean oil in the USA, palm oil in Malaysia, rapeseed oil in Europe and corn oil have been used for biodiesel production and found to be good diesel substitutes. Non-edible vegetable oils, such as Pongamia pinnata (Karanja or Honge), Jatropha curcas (Jatropha or Ratanjyote) and Madhuca iondica (Mahua) have also been found to be suitable for biodiesel production (Yusuf et. al., 2011). Biodiesel is also produced from animal fats (e.g. beef tallow, pork lard) and used cooking oils (e.g. yellow grease) (NREL, 2009).

In this study, canola and peanut oil were used as biodiesel resource. Canola oil was supplied from Sunar Mısır Corporation, a local oil factory at Cukurova Region. Peanut oil was supplied from Osmaniye, south region of Turkey.

Production and tests of biodiesels were carried out in Petroleum Research Laboratories of the Department of Automotive Engineering in Cukurova University. Table 3.4 shows the specifications of diesel and biodiesel fuels used in the analyses. It also shows standards covering requirements for fatty acid methyl esters (TS EN 14214) and standards covering requirements for diesel fuel (EN 590). As can be seen

from the table, all properties of both diesel fuel and biodiesels (canola and peanut) were in the standard ranges.

Table 3.4. Fuel specifications and standards covering requirements for biodiesel and diesel fuel (Ozgur, 2011)

Properties	Diesel fuel No:2	Canola biodiesel	Peanut biodiesel	TS EN 14214	EN 590
Density (kg/m^3)	833	896	880	860–900	820-845
Cetane Number	56	53.4	53.2	Min 51	Min. 51
Flash Point (^oC)	58.5	>180	>180	Min. 120	Min. 55
Pour Point (^oC)	-10	-9	5	Summer < 4.0 Winter < -1.0	-
Kinematic Viscosity (mm^2/s)	2.85	5.09	5.33	3.5 - 5.0	2-4.5
Heating value (kJ/kg)	42980*	37300*	36768	-	-

*(Allen et. al., 2011)

3.2. Methods

3.2.1. Transesterification Method

Improvement of fuel properties of oils start with reducing their viscosities. In this way, there are two methods available.

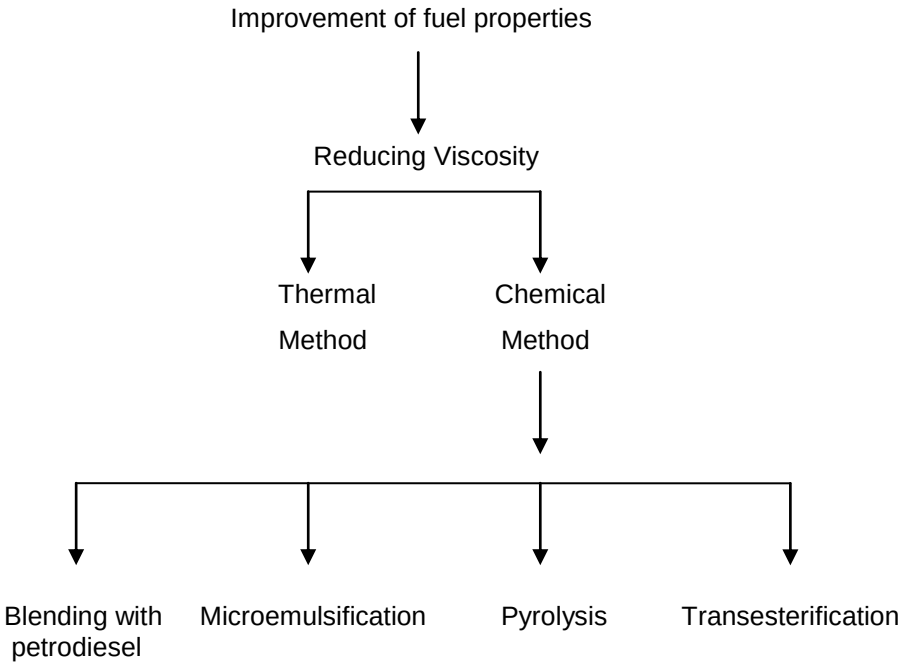

Figure 3.8. Improvement of fuel properties (Alptekin et. al., 2006)

Common vegetable oils or animal fats are esters of saturated and unsaturated monocarboxylic acids with the trihydric alcohol glyceride. These esters are called triglycerides (Leung et. al., 2010). Transesterification method is reaction of these triglycerides with an alcohol in the presence of a catalyst. At the end of this reaction glycerol and methyl esters are produced (Figure 3.9). Methanol and NaOH (sodium hydroxide) are usually used as alcohol and catalyst.

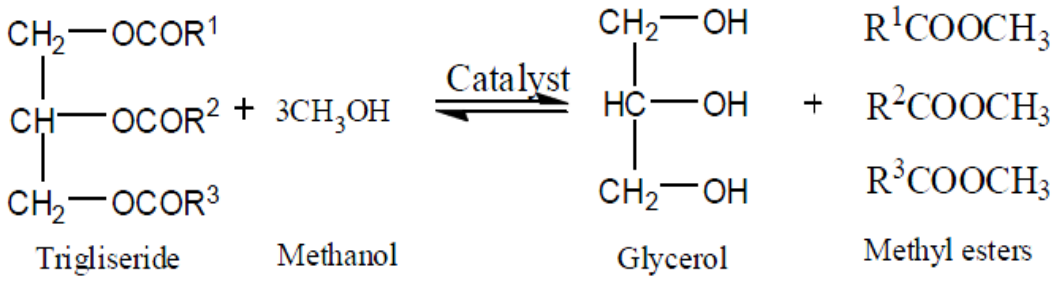

Figure 3.9. Chemical structure of transesterification reaction

Molar ratio of alcohol to oil should be 3:1 for a stoichiometric reaction. At the end of reaction 3 moles methyl ester (biodiesel) and 1 mol glycerol are produced. In order to increase production conversion, molar ratio of alcohol to oil can be increased. For example, production conversion is % 89,7 when molar alcohol to oil ratio is 3:1 and % 98,9 when ratio is 6:1 (Alptekin et. al., 2006).

Experimental studies were conducted in Petroleum Research Laboratories of Department of Automotive Engineering in Çukurova University, Adana, Turkey.

Transesterification reaction (Figure 3.9) was carried out in a spherical glass reactor equipped with reflux condenser, stirrer and thermometer (Figure 3.10). Sodium hydroxide (NaOH) used as catalyst and methanol (CH_3OH) was chosen as alcohol. At the transesterification reaction, mol ratio of alcohol and oil was 6:1. 100% excess methanol was used to ensure that the reaction goes to completion. In order to obtain sodium methoxide, methanol and NaOH were mixed. Then sodium methoxide and oil were mixed in the small scale reactor. The mixture was heated up to 62-63 ^{o}C and kept at this temperature for one hour; during this period mixture was stirred continuously. At the end of the reaction period, the crude methyl ester was waited at separating funnel for 8 hours. And then, deep phase containing crude glycerin was separated from crude methyl esters. Then the crude methyl ester was washed three times with warm water and all the other substances precipitated at the bottom. After washing process crude methyl ester was dried at 105 ^{o}C for one hour. After the drying process, methyl ester let to be cool down. In order to neglect the possibility of impurities, the methyl ester drained from paper filter. This production process was repeated for 2 times and each party collected in the same vessel to get a

homogenous biodiesel (Ozgur, 2011). Flow diagram of biodiesel production is shown in Figure 3.11.

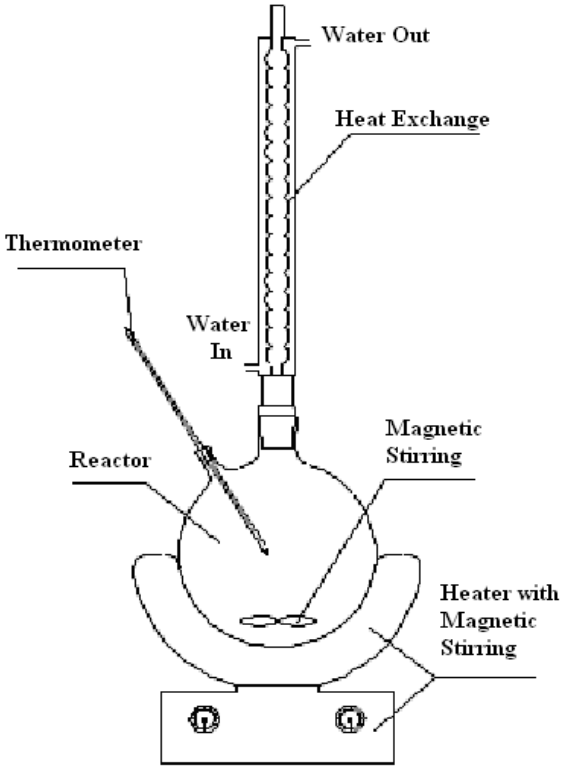

Figure 3.10. Small scale reactor

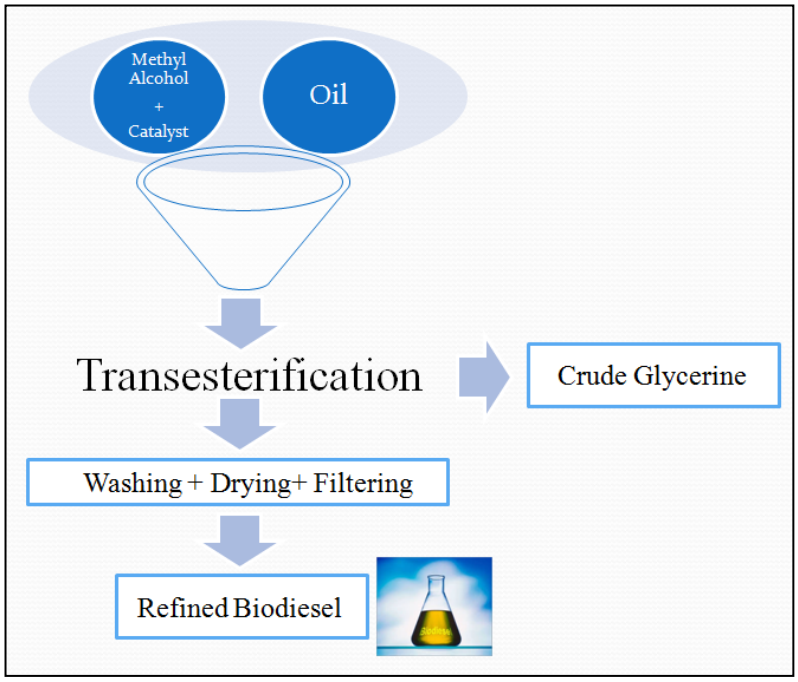

Figure 3.11. Biodiesel production flow diagram

3.2.2. First and Second Laws of Thermodynamics

The first law of thermodynamics, also known as the conservation of energy principle, provides a sound basis for studying the relationships among the various forms of energy and energy interactions. Based on experimental observations, the first law of thermodynamics states that energy can be neither created nor destroyed during a process; it can only change forms. Therefore, every bit of energy should be accounted for during a process.

We all know that a rock at some elevation possesses some potential energy, and part of this potential energy is converted to kinetic energy as the rock falls (Figure 3.12). Experimental data show that the decrease in potential energy exactly equals the increase in kinetic energy when the air resistance is negligible, thus confirming the conservation of energy principle for mechanical energy.

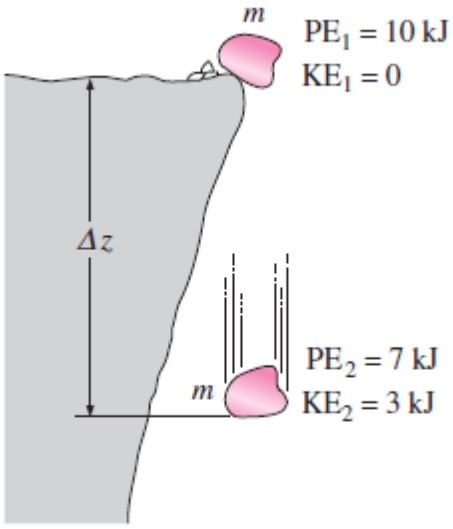

Figure 3.12. Conservation of energy principle

Consider a system undergoing a series of adiabatic processes from a specified state 1 to another specified state 2. Being adiabatic, these processes obviously cannot involve any heat transfer, but they may involve several kinds of work interactions. Careful measurements during these experiments indicate the following: For all adiabatic processes between two specified states of a closed system, the net work done is the same regardless of the nature of the closed system and the details of the process. Considering that there are an infinite number of ways to perform work interactions under adiabatic conditions, this statement appears to be very powerful, with a potential for far-reaching implications. This statement, which is largely based on the experiments of Joule in the first half of the nineteenth century, cannot be drawn from any other known physical principle and is recognized as a fundamental principle. This principle is called the first law of thermodynamics or just the first law (Cengel and Boles, 2005).

Although a spontaneous process can proceed only in a definite direction, the first law of thermodynamics gives no information about direction; it merely states that when one form of energy is converted to another, the quantities of energy involved are conserved regardless of feasibility of the process. Thus, processes can be envisioned that do not violate the first law of thermodynamics but do violate the second law of thermodynamics (SLT), e.g., transfer of a certain quantity of heat from

a low-temperature body to a high-temperature body, without the input of an adequate external energy form like work. However, such a process is impossible, emphasizing that the first law of thermodynamics is itself inadequate for explaining energy processes. The second law of thermodynamics establishes the difference in the quality of different forms of energy and explains why some processes can spontaneously occur while other cannot. The second law of thermodynamics is usually expressed as an inequality, stating that the total entropy after a process is equal to or greater than that before. The equality only holds for ideal or reversible processes.

The SLT defines the fundamental quantity entropy as a randomized energy state unavailable for direct conversion to work. It also states that all spontaneous processes, both physical and chemical, proceed to maximize entropy, i.e., to become more randomized and to convert energy to a less available form. A direct consequence of fundamental importance is the implication that at thermodynamic equilibrium the entropy of a system is at a relative maximum; i.e., no further increase in disorder is possible without changing the thermodynamic state of the system by some external means (such as adding heat). A corollary of the second law of thermodynamics is the statement that the sum of the entropy changes of a system and that of its surroundings must always be positive. In other words, the universe (the sum of all systems and surroundings) is constrained to become forever more disordered and to proceed toward thermodynamic equilibrium with some absolute maximum value of entropy (Dincer and Rosen, 2007).

Although there are various formulations of the second law of thermodynamics, two are particularly well known (Dincer and Rosen, 2007):
Clausius statement: It is impossible for heat to move of itself from a lower-temperature reservoir to a higher temperature reservoir. That is, heat transfer can only occur spontaneously in the direction of temperature decrease. For example, we cannot construct a refrigerator that operates without any work input.

Kelvin–Planck statement: It is impossible for a system to receive a given amount of heat from a high-temperature reservoir and to provide an equal amount of work output. While a system converting work to an equivalent energy transfer as

heat is possible, a device converting heat to an equivalent energy transfer as work is impossible. Alternatively, a heat engine cannot have a thermal efficiency of 100%.

3.2.3. Energy and Exergy Analysis

Energy is inevitable part of life and industrialization. Therefore, usage of energy sources effectively and efficiently is very important issue. For welfare and composure of human, there is need for energy usage on every sector.

Energy is playing a key role on country policies. The countries which have capable of producing their own energy minimize their dependency to other countries.

In an internal combustion engine, fuel and air enters to combustion chamber then explosion occurs. Finally exhaust gases are formed. Total amount of air and fuel have to be equal total amount of exhaust gases according to 1st law of thermodynamics since its being quantity based.

1st law of thermodynamics (energy) analysis tells something about system efficiency. However, it is inadequate for evaluating some features of energy resource utilization. The thermodynamic details of the operation of a thermal system can be best understood by performing not only an energy analysis but also an exergy analysis of the system. The exergy analysis reveals the locations, causes and magnitudes of energy resource waste in the system (Sayin et. al., 2006).

Some definitions should be given before performing exergy analysis to an internal combustion engine:

- Exergy: It is defined as the maximum amount of work which can be produced by a stream of matter, heat or work as it comes to equilibrium with a reference environment (Dincer, 2000).
- Dead state: A system is said to be in the dead state when it is in thermodynamic equilibrium with the environment. At the dead state, a system is at the temperature and pressure of its environment (in thermal and mechanical equilibrium); it has no kinetic or potential energy relative to the environment (zero velocity and zero elevation above a reference level) and it

does not react with the environment (chemically inert) (Cengel and Boles, 2005).

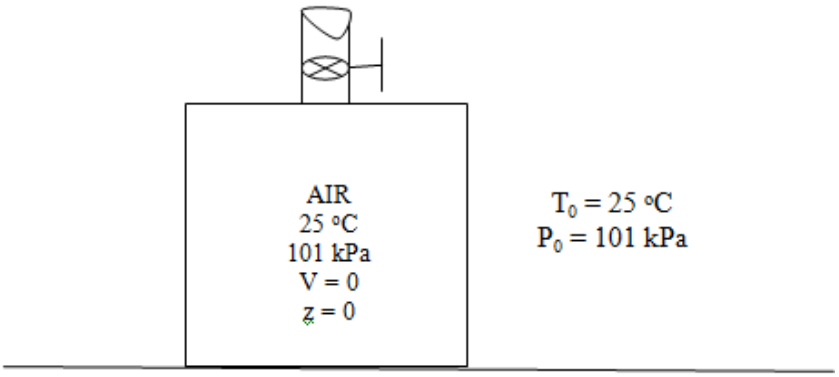

Figure 3.13. A system in dead state

- Entropy: The property entropy is a measure of molecular disorder or randomness of a system and second law of thermodynamics states that entropy can be created but it can't be destroyed (Cengel and Boles, 2005).
- Reversible work: It is defined as the maximum amount of useful work that can be produced (or minimum work that needs to be supplied) as a system undergoes a process between the specified initial and final states (Cengel and Boles, 2005).
- Irreversibility: Any difference between the reversible work and the useful work is due to irreversibilities present during the process, and the difference is called irreversibility (Cengel and Boles, 2005).

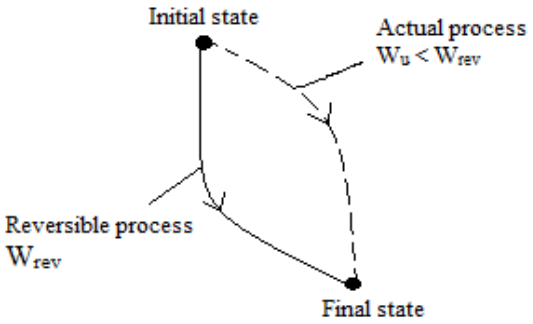

Figure 3.14. The difference between reversible work and actual useful work

To be able to perform energy and exergy analyses, a chemical reaction was considered which occurs theoretically when there was combustion of air-fuel mixture. General form of reaction equation is:

$$aC_xH_yO_z + b(O_2 + 3.76N_2) \rightarrow cO_2 + dCO + eCO_2 + fNO + gNO_2 + hN_2 + jH_2O \quad (3.1.)$$

By applying conservation of mass principle and using emission data obtained from experiments, all coefficients (a,b,c,d,e,f,g,h,j) were found.

Molecular formula of canola biodiesel, diesel fuel and peanut biodiesel considered as $C_{18.82}H_{34.96}O_2$, $C_{14}H_{25}$ (Allen et. al., 2011) and $C_{18}H_{34}O_2$ (Demirbas, 2009), respectively.

The assumptions made in the energy and exergy analyses are listed below:

- Considered control volume (Figure 3.15.) is whole engine and it runs as a steady state open system,
- The combustion air (79% N_2, 21% O_2) and exhaust gas mixtures are ideal,
- The kinetic and potential energy effects of combustion air, fuel stream and exhaust gases are neglected,
- The reference environment is defined as in Table 3.5, P_o=1 atm and T_o=25 ^oC (Moran and Shapiro, 2000).

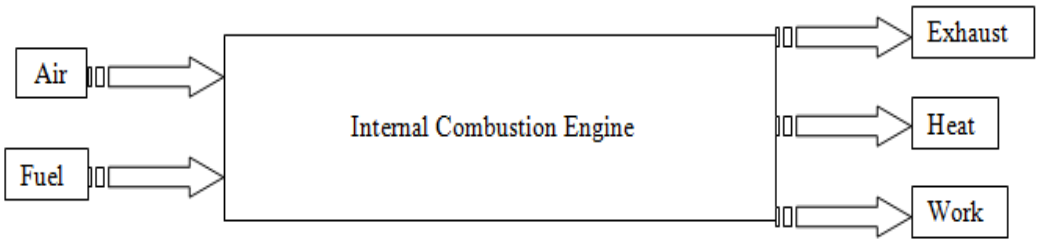

Figure 3.15. Control volume of the system (internal combustion engine)

Table 3.5. Mole fractions of reference environment

Component	Mole fraction (%)
N_2	75.67
O_2	20.35
CO_2	0.0345
H_2O	3.03
CO	0.0007
SO_2	0.0002
H_2	0.00005
$Others$	0.91455

The mass and energy balances for the control volume, steady state open system can be written as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{3.2.}$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out}\, h_{out} - \sum \dot{m}_{in}\, h_{in} \tag{3.3.}$$

where subscript in and out represents inlet and outlet, $\dot{Q}$ (kW) represents heat transfer rate through control volume, $\dot{W}$ (kW) represents work rate, $\dot{m}$ (kg/s) represents mass flow rate, h represents specific enthalpy. Equation (3.2) denotes that total amount of mass entering control volume through system boundary equals to total amount of mass leaving control volume (mass balance). Equation (3.3) denotes that total energy entering control volume through system boundary equals to total energy leaving control volume (energy balance).

Energy input rate (E_{fuel}) can be found using mass flow rate of fuel ($\dot{m}_{fuel}$) and lower heating value of the fuel (H_u) as follows:

$$E_{fuel} = \dot{m}_{fuel} H_u \tag{3.4.}$$

Heat losses are evaluated as the difference between energy input rate (E_{fuel}) and work rate ($\dot{W}$) by:

$$\dot{Q}_{loss} = E_{fuel} - \dot{W} \tag{3.5.}$$

Energy efficiency (1st law efficiency) is calculated as the ratio of work rate ($\dot{W}$) to energy input rate of fuel (E_{fuel}) from:

$$\eta_1 = \frac{\dot{W}}{E_{fuel}} \tag{3.6.}$$

When energy analyses have been determined, exergy analyses can be defined. Exergy balance of a control volume is obtained using following relation (Caliskan et. al., 2010b):

$$\sum \dot{m}_{in}\varepsilon_{in} = \sum \dot{m}_{out}\varepsilon_{out} + \dot{Ex}_{heat} + \dot{Ex}_{work} + \dot{Ex}_{dest} \tag{3.7.}$$

where subscript in and out represents inlet and outlet, $\dot{Ex}_{heat}$ (kW) represents exergy transfer rate related with heat loss from the engine (control volume) to the environment, $\dot{Ex}_{work}$ (kW) represents exergy work rate, $\dot{Ex}_{dest}$ (kW) represents exergy destruction due to irreversibilities, ε (kJ/kg) represents specific flow exergy.

Input exergy rate contains only chemical exergy of the fuel. Thermomechanical exergy of fuel and total exergy of air can be neglected (Tat, 2011).

$$\dot{Ex}_{in} = \dot{m}_{fuel}\varepsilon_{fuel} \tag{3.8.}$$

where $\dot{m}_{fuel}$ (kg/s) represents mass flow rate of fuel (fuel consumption), ε_{fuel} (kJ/kg) represents specific flow exergy of fuels.

$$\varepsilon_{fuel} = H_u \varphi \quad (3.9.)$$

where, φ represents chemical exergy factor.

For liquid fuels, the chemical exergy factor (φ) can be calculated as (Kotas, 1995):

$$\varphi = 1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{\alpha}{c}(1 - 2.0628\frac{h}{c}) \quad (3.10.)$$

where h, c, o, α represents mass fractions of hydrogen, carbon, oxygen, sulfur contents.

The output (exhaust) exergy rate consists of both thermomechanical and chemical exergy of the exhaust gases.

$$\dot{E}x_{out} = \sum \dot{m}_i \, (\varepsilon_{tm} + \varepsilon_{chem})_i \quad (3.11.)$$

where i represents i^{th} component of the exhaust gases, ε_{tm} (kJ/kg) represents specific thermomechanical exergy of the exhaust gas, ε_{chem} (kJ/kg) represents specific chemical exergy of the exhaust gas, $\dot{m}$ (kg/s) represents mass flow rate.

Specific thermomechanical exergy (ε_{tm}) of the each exhaust gases can be calculated with the following formula:

$$\varepsilon_{tm} = (h - h_0) - T_0(s - s_0) \quad (3.12.)$$

where 0 represents the dead state values of parameters, s (kJ/kgK) represents specific entropy, h (kJ/kg) represents specific enthalpy.

Specific chemical exergy (ε_{chem}) of the each exhaust gases can be calculated with the following formula:

$$\varepsilon_{chem} = \bar{R} T_0 ln \frac{y_i}{y_r} \quad (3.13.)$$

where, $\bar{R}$ is general gas constant, T_0 is dead state temperature (25 ^{o}C), y_i is the mole fraction of the each exhaust gases, y_r is the mole fraction of the given environment (Table 3.5).

Exergy transfer rate related with heat loss from the engine (control volume) to the environment ($\dot{E}x_{heat}$) can be calculated with the following formula:

$$\dot{E}x_{heat} = \sum(1 - \frac{T_0}{T_{cw}})\dot{Q}_{cw} \quad (3.14.)$$

where T_0 is dead state temperature (25 ^{o}C), T_{cw}, cooling water temperature, $\dot{Q}_{cw}$, heat transfer rate through cooling water. T_{cw} can be calculated as the average of the cooling water inlet temperature ($T_{cw,inlet}$) and cooling water outlet temperature ($T_{cw,outlet}$) (Gokalp et. al., 2009).

$$T_{cw} = \frac{T_{cw,inlet}+T_{cw,outlet}}{2} \quad (3.15.)$$

Output heat from engine to the environment through the cooling water of the engine is given by equation (3.16) (Caliskan et. al., 2010a):

$$\dot{Q}_{cw} = E_{fuel} - (\dot{E}x_{work} + \sum \dot{m}_{out}\Delta h_{out}) \quad (3.16.)$$

$$\Delta h_{out} = h - h_0 \quad (3.17.)$$

where h and h_0 refers the enthalpy of related exhaust gas at exhaust gas temperature and reference state temperature, respectively.

Net exergy work is equal to the net energy work.

$$\dot{E}x_{work} = \dot{W} \quad (3.18.)$$

Exergy destruction ($\dot{E}x_{dest}$) can be found from exergy balance equation (Equation 3.7) by putting all other known values of equation.

$$\dot{E}x_{dest} = \sum \dot{m}_{in}\varepsilon_{in} - \sum \dot{m}_{out}\varepsilon_{out} - \dot{E}x_{heat} - \dot{E}x_{work} \quad (3.19.)$$

Exergy efficiency (2nd law efficiency) can be found as the ratio of the exergy work rate to the fuel energy input rate.

$$\psi = \frac{\dot{E}x_{work}}{\dot{E}x_{in}} \quad (3.20.)$$

3.2.4. The Ways of Increasing Exergetic Efficiency

The ways of increasing exergetic efficiency can be defined as following (Caliskan, 2009):

- Points which cause exergy loss are defined. Exergy loss generally contains heat transfer, chemical reaction, mixture, friction and expansion of gases.
- The applications which are done with simple assumptions are defined.
- Simplified exergy calculations are used. (ideal gas, incompressibility)
- Exergy losses of the system are decreased by using cogeneration.
- Direct transfer of heat transfer to air, cooling water and cooler must be avoided.
- At lower temperatures in which heat transfer occurs ($T < T_0$), heat transfer with air or heat flow must be avoided.
- When heat exchange is applied at low temperatures, the temperature difference between flows must be minimized.
- At energy exchange via heat transfer between two flows, usage of heat transfer average of the fluids must be avoided.
- We must avoid extremely dense thermodynamic systems because of temperature and pressure differences.

- The temperature and pressure differences of flow mixtures must be minimized.

3.2.5. Calculations for Energy and Exergy Analysis

All equations needed for performing energy and exergy analysis were given in this chapter. Calculations to obtain energetic and exergetic efficiencies were shown only for diesel fuel here.

Some of the data obtained from engine test and other needed information of fuels for equations are given in Table 3.6.

Table 3.6. Data obtained from engine test

Fuels	Heating value (kJ/kg)	Engine speed (rpm)	Work (kW)	Mass flow rate of fuel (kg/s)	Mass flow rate of air (kg/s)
Diesel	42980	2600	55.46	0.00306852	0.10279675
Canola	37300	2600	41.65	0.00346898	0.10223828
Peanut	36768	2600	38.9	0.00318866	0.10399344

3.2.5.1. Energy Input Rate

$E_{fuel} =$ (0.00306852)(42980) = 131.88 kW (Equation 3.4)

3.2.5.2. Total Heat Losses

$\dot{Q}_{loss} =$ (131.88) – (55.46) = 76.42 kW (Equation 3.5)

3.2.5.3. Energy Efficiency

$\eta_1 =$ (55.46) / (131.88) = 0.4205 (%42.05) (Equation 3.6)

3.2.5.4. Input Exergy Rate

Mass fraction ratios of the elements in the fuels were shown in Table 3.7.

Table 3.7. Mass fraction ratios of the elements in the fuels

Fuels	Chemical formula	$\frac{h}{c}$	$\frac{o}{c}$	$\frac{\alpha}{c}$
Diesel	$C_{14}H_{25}$	0.148810	0	0
Canola	$C_{18.82}H_{34.96}O_2$	0.154800	0.141693	0
Peanut	$C_{18}H_{34}O_2$	0.157407	0.148148	0

$\varphi = 1.0401 + 0.1728(0.148810) + 0.0432(0) + 0.2169(0)(1 - 2.0628(0.148810))$ (Equation 3.10)

$\varphi = 1.065814286$

$\varepsilon_{fuel} = (42980)(1.065814286) = 45808.698$ kJ/kg (Equation 3.9)

$\dot{E}x_{in} = (0.00306852)(45808.698) = 140.56$ kW (Equation 3.8)

3.2.5.5. Output Exergy Rate

The output (exhaust) exergy rate consists of both thermomechanical and chemical exergy of the exhaust gases.

For each of the exhaust gases both thermomechanical and chemical exergy values are calculated. Then the total of these two exergy values of each exhaust component were multiplied with their own mass flow rate. All of the mass flow rates of the exhaust gases were calculated from the mass balance.

$$\dot{m}_{air} + \dot{m}_{fuel} = \dot{m}_{exhaust} \quad (3.21.)$$

Heywood (1988) suggested a formula for the mass flow rate of air. It can be calculated as follows:

$$\dot{m}_{air} = \frac{\rho_{air} V_d N}{2.60} \quad (3.22.)$$

Table 3.8. Calculations of the mass flow rates

Fuel	Cylinder volume (m^3)		Air density (kg/m^3)		
	0.0009769		1.225		
	Engine speed (rpm)	Air mass flow rate (kg/s) (1 cylinder)	Air mass flow rate (kg/s) (4 cylinder)	Mass flow rate of fuel (kg/s) ($\dot{m}_f$ was experimentally determined)	Mass flow rate of exhaust gases (kg/s) (total)
Diesel	2600	0.025699	0.102796	0.00306852	0.105865
Canola	2600	0.025560	0.102238	0.00346898	0.105707
Peanut	2600	0.025998	0.103993	0.00318866	0.107182

In air mass flow rate formula, ρ_{air} represents density of air (kg/m^3), V_d, cylinder volume (m^3), N, engine speed (rpm).

Emission results of fuels were given in Table 3.9.

Table 3.9. Emission results of fuels

Emission	Fuel		
	Diesel	Canola	Peanut
O_2 (%)	10.38	10.96	10.43
CO (ppm)	248	283	284
CO_2 (%)	6.02	5.69	5.51
NO (ppm)	946	837	1024
NO_2 (ppm)	50.2	43.4	51.5

By using data in Table 3.8 and Table 3.9, we can find the coefficients of given emission data above in Equation 3.1 (d, e, f, g, h). Rest of the unknown

coefficients (a, b, c, j) can be found with conservation of mass principle for carbon (C), oxygen (O), hydrogen (H) and nitrogen (N).

All of the coefficients are given in Table 3.10.

Table 3.10. Reaction coefficients

$aC_xH_yO_z + b(O_2 + 3.76N_2) \rightarrow cO_2 + dCO + eCO_2 + fNO + gNO_2 + hN_2 + jH_2O$

Coefficient	Diesel	Canola	Peanut
a	1.04129E-5	7.32027E-6	7.51712E-6
b	0.000556	0.000558	0.000542
c	3.43E-4	0.000362047	3.49E-4
d	9.38E-7	1.07E-6	1.09E-6
e	1.45E-4	1.37E-4	1.34E-4
f	3.34E-6	2.95E-6	3.66E-6
g	1.16E-7	9.97E-8	1.2E-7
h	0.002087	0.002095	0.002038
j	0.000130161	0.000127958	0.000127791

With the aid of reaction coefficients, mass fraction of each emission was determined. The total emission flow rate ($\dot{m}_{exhaust}$) of the exhaust gases were also given in Table 3.8. Finally, mass flow rate of each exhaust gases were determined separately (Table 3.11).

Table 3.11. Mass flow rates of each exhaust gas emissions

Emission mass flow rates (kg/s)	Diesel	Canola	Peanut
O_2	0.0148613	0.0155655	0.0156452
CO	3.55068E-5	4.01919E-5	4.26003E-5
CO_2	0.0086189	0.00808999	0.008265074
NO	0.00013544	0.00011887	0.0001536
NO_2	7.1873E-6	6.1637E-6	7.72506E-6
N_2	0.07903829	0.0788011	0.07984874
H_2O	0.00316855	0.00309448	0.00321919

To be able to calculate thermomechanical exergy of exhaust gases we need enthalpy and entropy values of the gases at related temperatures (Dead state

temperature and exhaust gas temperature). McAllister et. al. (2011) gives properties of ideal combustion gases.

Table 3.12 shows the enthalpy and entropy values at stated temperatures.

Table 3.12. Enthalpy and entropy values of exhaust gases (for diesel fuel)

Emissiosns	T_{ex} = 800 K (exhaust gas temperature)			T_0 = 298 K (Dead state temperature)		
	$h - h_{0:}$ (kj/kmol)	$h - h_0$ (kj/kg)	s (kj/kmol.K)	s_0 (kj/kmol.K)	$s - s_0$ (kj/kmol.K)	$s - s_0$ (kj/kg.K)
O_2	15840	495	235.82	205.03	30.79	0.962
CO	15170	541.79	227.17	197.54	29.63	1.058
CO_2	22810	518.41	257.42	213.73	43.69	0.993
NO	15550	518.33	240.99	210.64	30.35	1.012
NO_2	22140	481.30	282.41	239.91	42.5	0.924
N_2	15040	537.14	220.92	191.5	29.42	1.051
H_2O	18000	1000	223.72	188.71	35.01	1.945

Table 3.13. Specific thermomechanical exergy, specific chemical exergy and the total output (exhaust) exergy values (for diesel fuel)

Exhaust gas	Mass flow rate (kg/s)	Specific thermo mechanical exergy (kJ/kg)	Specific chemical exergy (kJ/kg)	The total specific output (exhaust) exergy (kJ/kg) (thermomechanical + chemical exergy)	The total output (exhaust) exergy (kW)
O_2	0.0148613	208.3	123.27	331.57	4.93
CO	3.55068E-5	226.47	1050.34	1276.81	0.05
CO_2	0.0086189	222.51	448.91	671.42	5.79
NO	0.00013544	216.86	-	216.86	0.03
NO_2	7.1873E-6	205.98	-	205.98	0.0014
N_2	0.07903829	224.03	24.67	248.7	19.66
H_2O	0.00316855	420.39	481.31	901.7	2.86

$\dot{E}x_{out}$=$\sum \dot{m}_i \ (\varepsilon_{tm} + \varepsilon_{chem})_i$ = 33.30 kW

3.2.5.6. Work Exergy Rate

$\dot{E}x_{work} = \dot{W} =$ 55.46 kW

3.2.5.7. Exergy Rate Related with Heat Transfer

$\dot{Q}_{cw} =$ 131.88499 – [55.4569 – (0.0148613)(495) + (3.55068E-5)(541.7857) + (0.0086189)(518.4091) + (0.00013544)(518.3333) + (7.1873E-6)(481.3043) + (0.07903829)(537.1429) + (0.00316855)(1000)] (Equation 3.16)

$\dot{Q}_{cw} =$ 18.9065 kW

$T_{cw,inlet} =$ 324.71 K

$T_{cw,outlet} =$ 361.73 K

$T_{cw} =$ (324.71+361.73) / 2 = 343.22 K (Equation 3.15)

$T_0 =$ 298 K (dead state temperature)

$\dot{E}x_{heat} =$ (1 – 298/343.22)(18.9065) = 2.49 kW (Equation 3.14)

3.2.5.8. Exergy Destruction

$\dot{E}x_{dest} =$ 140.5649 – [33.30462 + 2.49097 + 55.4569] = 49. 31 kW (Equation 3.19)

3.2.5.9. Exergy Efficiency

$\psi =$ (55.4569) / (140.5649) = 0.3945 (%39.45) (Equation 3.20)

4. RESULTS AND DISCUSSIONS

The tests were performed on a four stroke, four cylinders, naturally aspirated, direct-injected diesel engine (Figure 3.1) which takes place in Petroleum Research Laboratories of the Department of Automotive Engineering in Cukurova University. Specifications of the engine are listed in Table 3.1. Engine gives 89 kW maximum power at 3200 rpm engine speed and has 295 Nm maximum torque at 1800 rpm engine speed. The engine was operated for 15 minutes with diesel fuel to reach the operation temperature. Test fuels were tested at 2600 rpm at full load condition for performing of energy and exergy analysis.

In the experiments diesel fuel, canola biodiesel and peanut biodiesel are used as fuel. The results from energy (first law) analysis are given both in Table 4.1 and in Figures 4.1 through 4.4.

Table 4.1. Energy analysis results of the fuels used in engine

Fuel	Engine speed (rpm)	Work rate (kW)	Energy input rate (kW)	Total heat loss (kW)	Energy efficiency (%)
Diesel	2600	55.46	131.88	76.42	42.05
Canola	2600	41.65	129.39	87.74	32.19
Peanut	2600	38.9	117.24	78.34	33.18

Figure 4.1 shows work rates of fuels at a certain engine speed comparatively. It can be clearly seen from the figure that work rate of both biodiesels (canola and peanut) were lower than diesel fuel. 24.88% for canola biodiesel and 29.86% for peanut biodiesel power loss at the same engine speed can be explained by their lower heating values. It can also be explained by high viscosity and density values of both biodiesels which cause poor atomization of fuel.

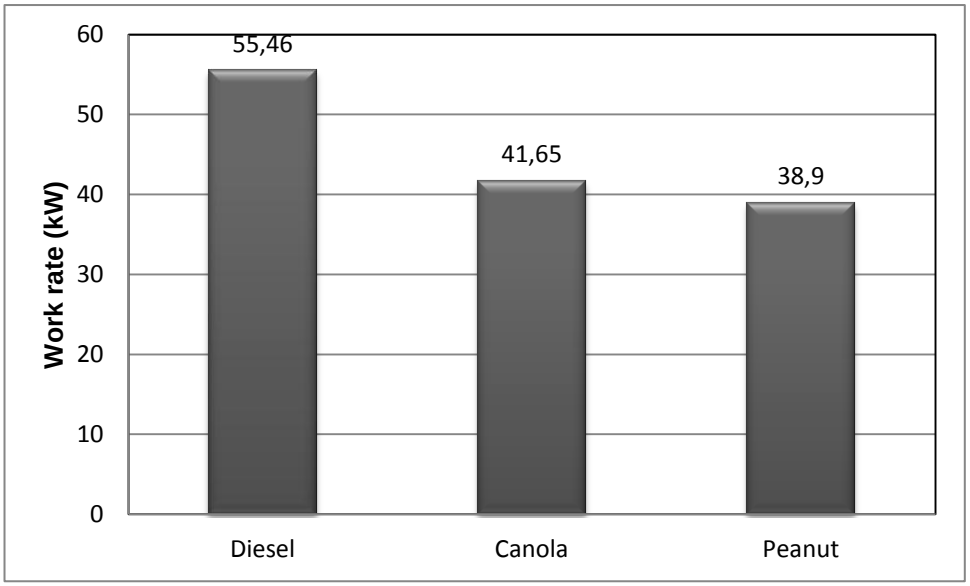

Figure 4.1. Work rate values of diesel fuel and biodiesels

Energy input rate is the energy supplied to the engine which contains heat losses and work rate and calculated by using mass flow rate and lower heating value of fuel. Supplied energy to the control volume is direct proportional with lower heating value of fuel. Therefore, being heating value of diesel fuel is high caused to have higher energy input rate of this fuel. Energy input rate of canola biodiesel and peanut biodiesel were found 1.89% and 11.1% lower than diesel fuel, respectively.

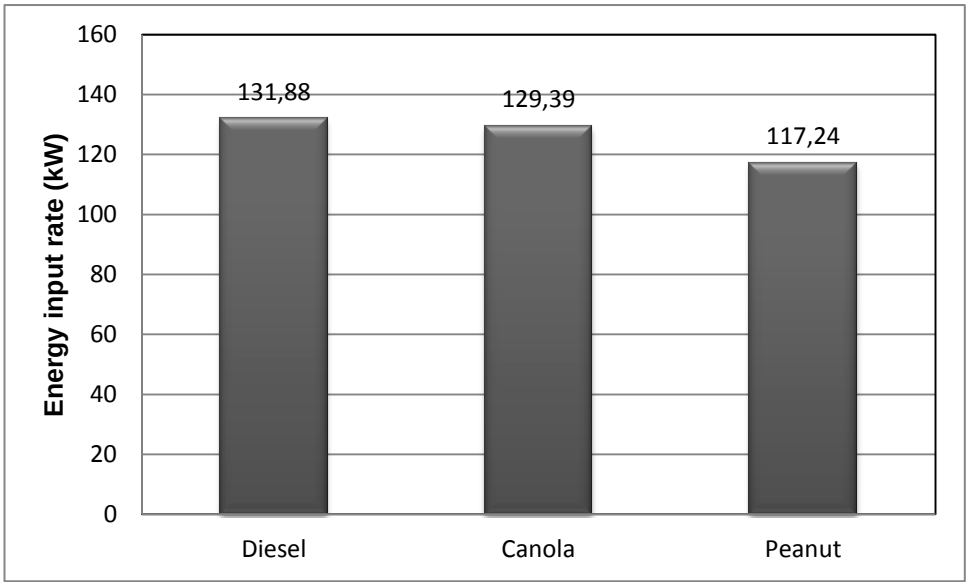

Figure 4.2. Energy input rates (fuel energy) of diesel fuel and biodiesels

When we subtract the useful work (work rate) from supplied energy by fuel (energy input rate), all remaining can be called as heat losses (including from engine to environment, to radiator etc.). Biodiesel has more oxygen content than diesel fuel. This means better combustion and obtaining higher temperature in cylinder. Therefore, rate of heat loss from engine increased when we used biodiesel. It can be clearly seen from the figure that in both biodiesel fuels, there were much more heat losses according to diesel at the same engine speed.

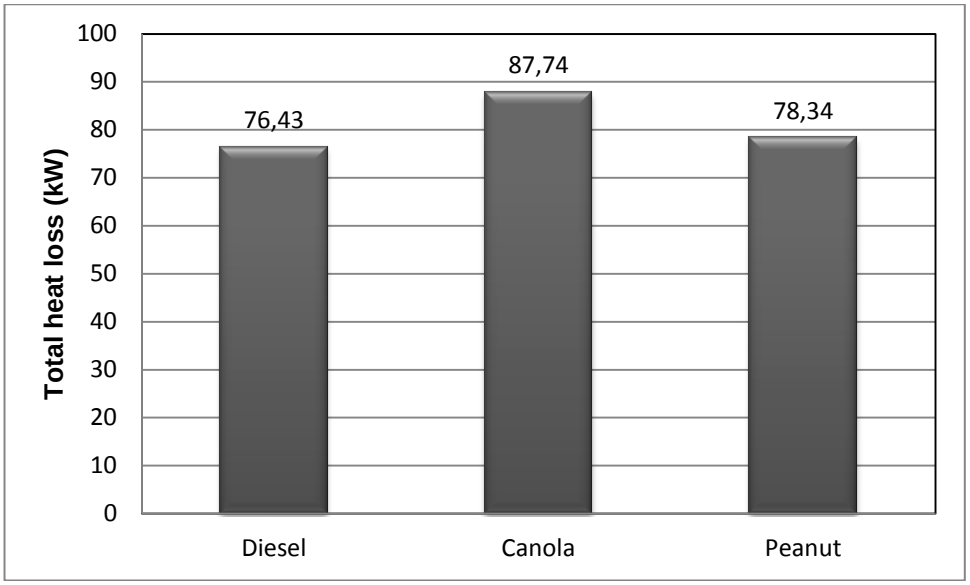

Figure 4.3. Total heat loss values of diesel fuel and biodiesels

Figure 4.4 shows energetic efficiencies (thermal efficiency or 1st law efficiency) of engine by using various fuels at a certain engine speed comparatively. Thermal efficiency is the measure of how efficiently energy input is converted to useful work in engine. This means that the ratio of work rate to energy input rate gives energetic efficiency. As seen from the figure more efficient conversion occurs in engine with diesel fuel usage. Thermal efficiencies of the canola and peanut biodiesels are %23.45 and %21.1 lower than diesel fuel, respectively. Cetane number of diesel fuel, canola biodiesel and peanut biodiesel are 56, 53.4 and 53.2, respectively. Due to the Cetane number is a measure of combustion quality during compression ignition; combustion occurs much more efficiently in fuel which has high Cetane number. This means that more efficient energy conversion in internal combustion engine (chemical to mechanical) and much more thermal efficiency.

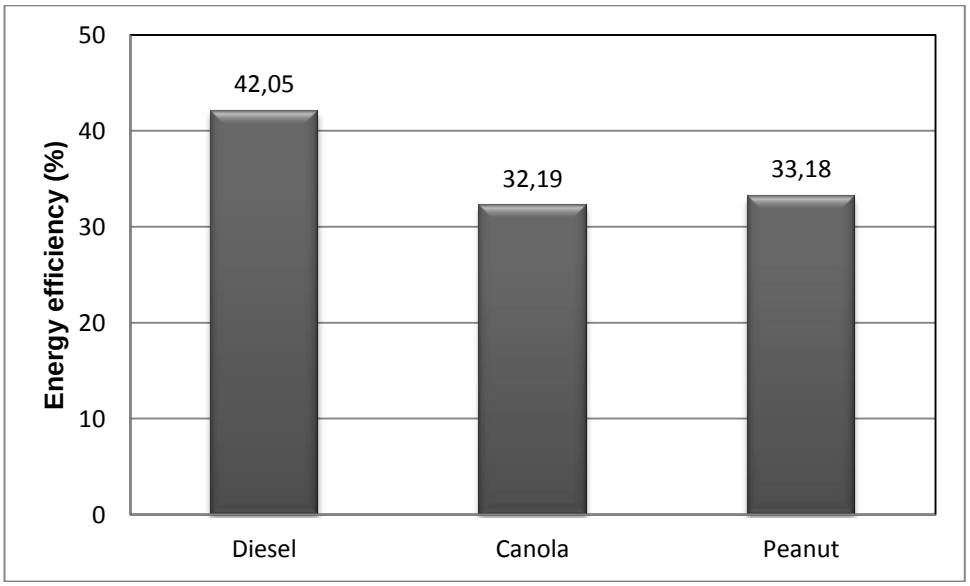

Figure 4.4. Energetic efficiencies of diesel fuel and biodiesels

Table 4.2. Results of second law analysis for various fuels

Fuel	Engine speed (rpm)	Input exergy rate (kW)	Output exergy rate (kW)	Work exergy rate (kW)	Exergy rate related with heat transfer (kW)	Exergy Destruction (kW)	Exergy efficiency (%)
Diesel	2600	140.56	33.30	55.46	2.49	49.31	39.46
Canola	2600	138.83	34.14	41.65	2.77	60.28	30.0
Peanut	2600	125.88	30.59	38.9	3.64	52.76	30.90

The results from exergy (second law) analysis are given both in Table 4.2 and in Figures 4.5 through 4.10.

Figure 4.5 shows input exergy rates of fuels (fuel exergy) at a certain engine speed comparatively. Higher values of lower heating value of diesel fuel causes to obtain high fuel exergy. Since the term specific flow exergy in input exergy rate contains lower heating value of fuel, similar trend with energy input rates of fuels was obtained as expected. Canola biodiesel has 1.23% and peanut biodiesel has 10.44% lower input exergy rate value than diesel fuel.

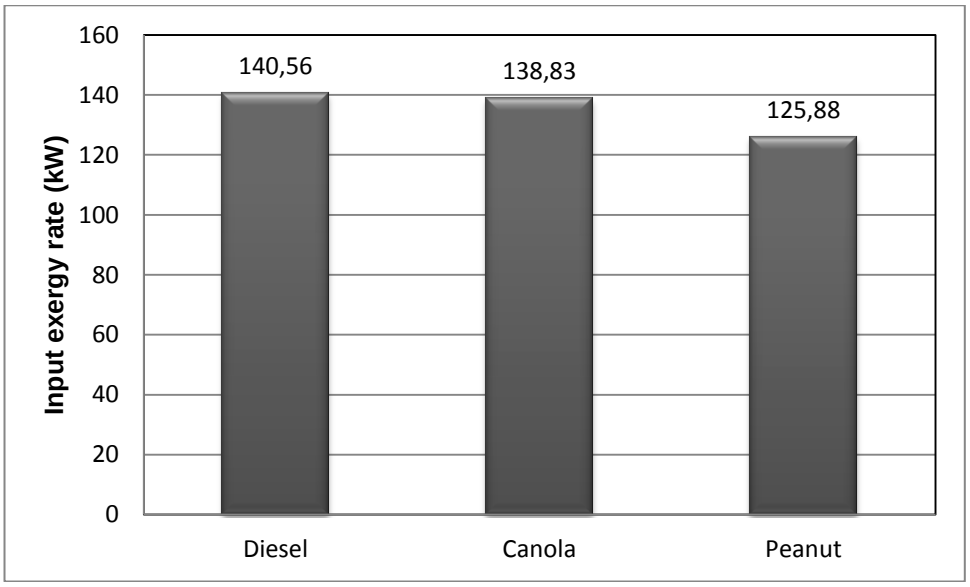

Figure 4.5. Input exergy rate values of diesel fuel and biodiesels

Figure 4.6 shows the output exergy rates of fuels at a certain engine speed. Output exergy contains all of the output exergy of exhaust gases (both thermomechanical and chemical). Output exergy consist of notable amount of input exergy (i.e. 23.69% of input exergy for diesel fuel)

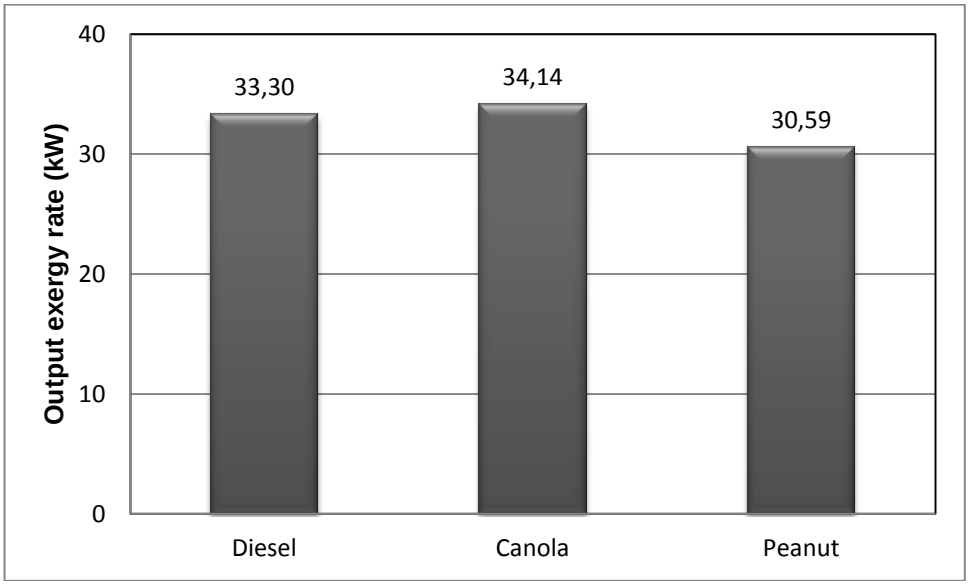

Figure 4.6. Output exergy rate values of diesel fuel and biodiesels

Figure 4.7 shows the work exergy rates of fuels at a certain engine speed. Due to the exergy is defined as the maximum extractable work potential; exergy work rate can be defined as work rate. Therefore, similar graph was given here as showed previously for work rate in exergy analysis.

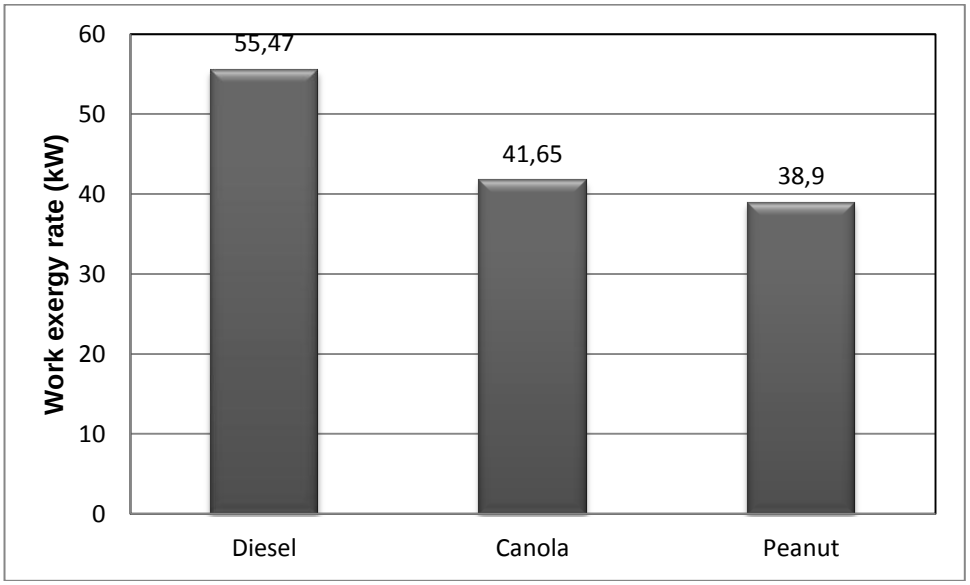

Figure 4.7. Work exergy rate values of diesel fuel and biodiesels

Figure 4.8 shows the exergy rates of fuels related with heat transfer at a certain engine speed. Here, exergy rate related with heat transfer is function of ambient temperature (dead state temperature), cooling water temperature (inlet and outlet) and heat transfer rate through cooling water (Equation 3.16). Highest heat loss exergy due to cooling water was obtained in peanut biodiesel.

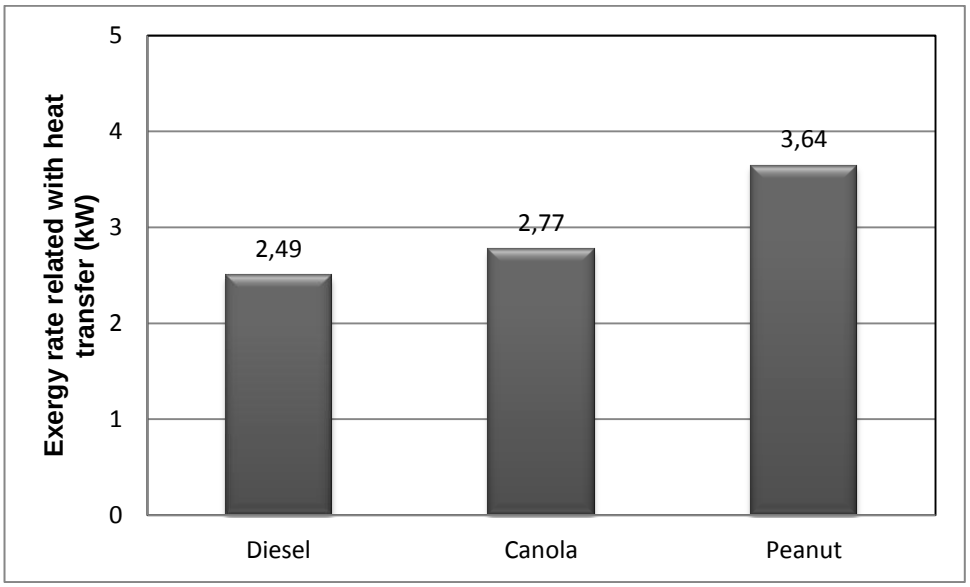

Figure 4.8. Exergy rate related with heat transfer values of diesel fuel and biodiesels

Exergy destruction values were shown in Figure 4.9. In an internal combustion engine; heat transfer, friction, mixing and combustion itself causes destroying of usable, available energy which is named as exergy. This all irreversible processes in internal combustion engine destroy notable amount of input exergy of fuels. This part of exergy can't be converted any useful work. Exergy destruction can be found by subtracting output exergy values from input exergy values. This difference gives us destroyed part of exergy. Otherwise, it can be calculated by multiplying dead state temperature (298 K) with generated entropy (Caliskan, 2009). This means exergy destruction is directly proportional with generated entropy. High temperature in engine means increasing of entropy. Containing more oxygen content of biodiesel than diesel fuel causes high temperature in cylinder. Therefore, in both biodiesels, exergy destruction rates are bigger than diesel fuel.

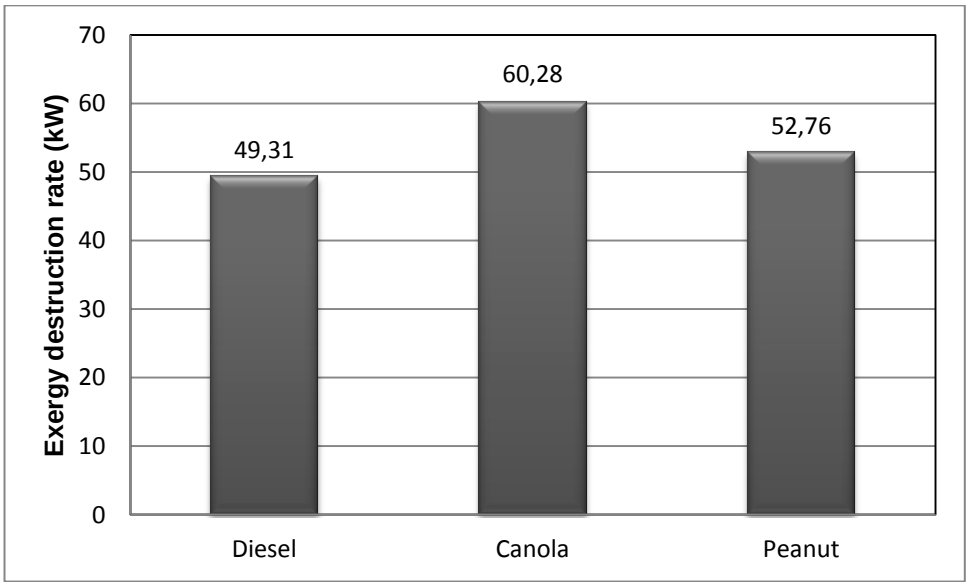

Figure 4.9. Exergy destruction values of diesel fuel and biodiesels

Figure 4.10 shows exergetic efficiencies (2nd law efficiency) of engine by using various fuels at a certain engine speed comparatively. Exergy efficiency can be found as the ratio of the exergy work rate to the fuel energy input rate. Exergy efficiency of the canola biodiesel is %23.97 lower than diesel fuel and exergy efficiency of the peanut biodiesel is %21.69 lower than diesel fuel. There is an opposite trend between exergy destruction and exergetic efficiency. When the amount of exergy destruction increases exergetic efficiency decreases as expected.

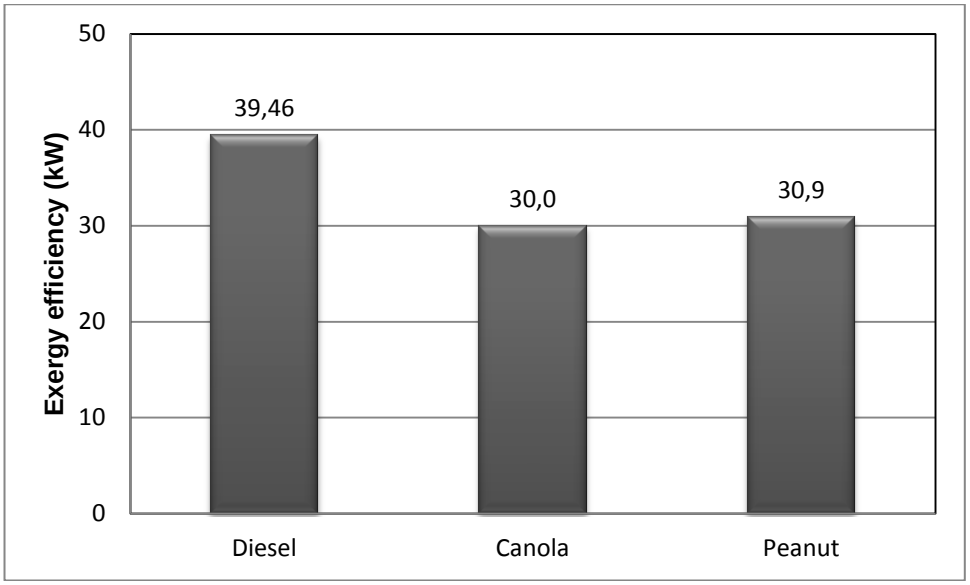

Figure 4.10. Exergy efficiency values of diesel fuel and biodiesels

Breakdown of fuel exergy by percent for each fuel was shown in Figure 4.11, 4.12 and 4.13 above. These demonstrations give considerable information about the conversion of given input fuel exergy to other exergy forms. Output exergy rate, exergy rate related with heat transfer and exergy destruction values of both biodiesel are higher than diesel fuel. Also the work rate of diesel fuel's being higher makes this fuel more usable.

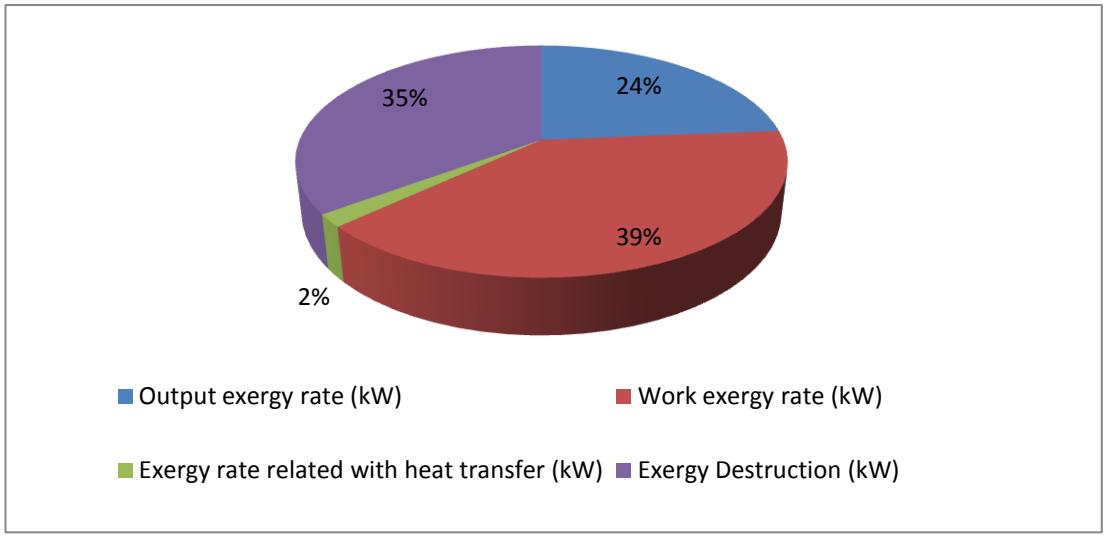

Figure 4.11. Distribution of exergy (diesel)

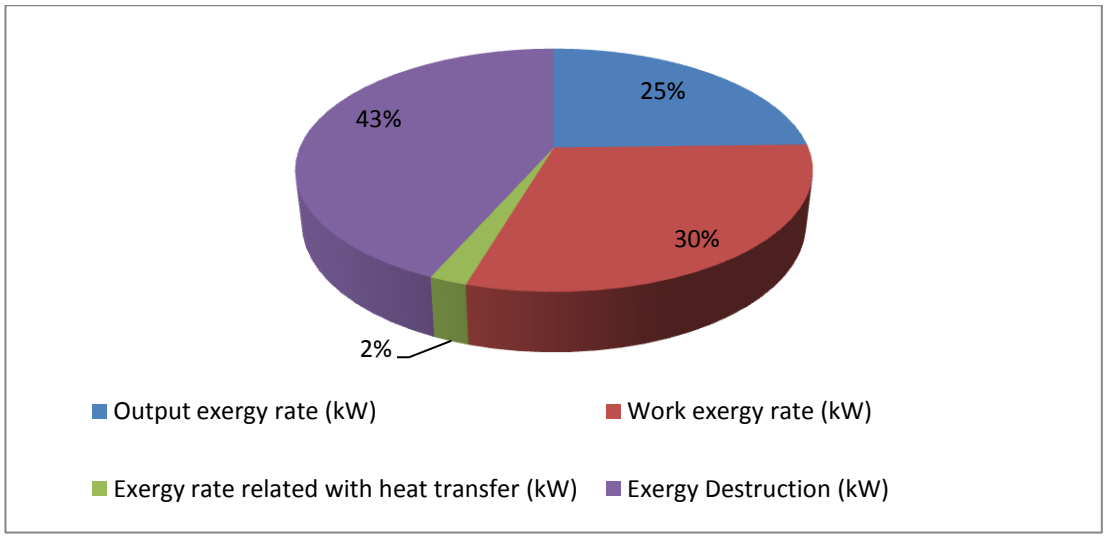

Figure 4.12. Distribution of exergy (canola)

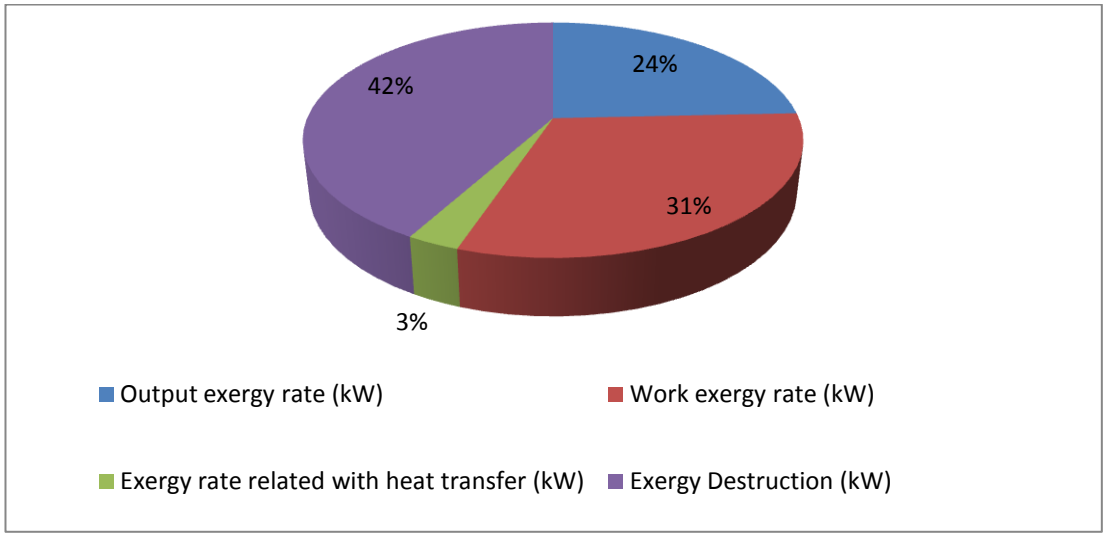

Figure 4.13. Distribution of exergy (peanut)

5. CONCLUSIONS

The aim of thermodynamic analysis is assessment of the system behavior, performance and efficiency. By considering the fundamental of conventional thermodynamics, conservation of energy (1st law of thermodynamics), mass and energy balance equations were formed. 1st law of thermodynamics gives some important information but it does not tell about something about how nearly thermodynamic performance of system approaches to ideal case. At this point, we need entropy balance equations in addition to mass and energy balance equations to be able to perform exergy analysis (2nd law of thermodynamics).

In this study, exergy analysis gave more feasible and realistic process assessment rather than energy analysis. To perform energy and exergy analysis performance and emission results of diesel engine fuelled with various fuels were used.

Various fuels were compared with respect to their energetic and exergetic performance. It was concluded that, notable amount of input exergy can't be used as useful work output due to friction, mixing, irreversibilities etc. in engine. Diesel fuel showed best energetic and exergetic efficiency than biodiesels. There wasn't any significance difference between both canola and peanut biodiesel by means of energetic and exergetic performance.

Lastly, the analysis can be applied to various fuels, fuel blends and can be investigated with change of engine speed.

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