ENERGY AND EXERGY BASED ANALYSIS OF SI ENGINES USING METHANE, METHANOL AND OCTANE

Md. Mizanuzzaman
Department of Mechanical Engineering
Bangladesh Military Academy (BMA), Bhatiary, Chittagong, Bangladesh
Email: mizan_zaman@yahoo.com

ABSTRACT
In this study, the phenomena of air-fuel mixing and combustion of Methane, Methanol and octane has been studied numerically in order to assess the thermodynamic analysis of the system. For this purpose, a thermodynamic-based engine cycle model is developed without considering geometric features of fluid motion. Here a single cylinder four-stroke spark-ignition (SI) engine is considered as a system. Variable specific heat ratio is considered instead of constant specific heat ratio. Empirical correlations are used to predict heat loss from the engine cylinder. Developed fundamental equations and empirical relations have been used in this model with the help of FORTRAN 95. In this study, friction losses are also modeled to estimate the overall energy losses calculation. The effects of changing some design and operating parameters such as compression ratios and rpm on the variation and destruction of exergy have been investigated through the analysis. It was found that the design and operating conditions have considerable effects on the variation of energy, exergy and irreversibilities as well as the efficiency during investigated parts of the SI engine cycle. 1st law and 2nd law efficiencies were found to increase for all the (three) fuels with increasing engine speed and compression ratios. The Present study recommends the necessity of both energy and exergy analysis to identify the source of work potential losses for different fuels used in SI engine.

Key Words: Energy, Exergy, Alternative fuels, Combustion, Engine performance, Friction, 1st law efficiency, 2nd law efficiency.

1.0 INTRODUCTION

Energy conservation and its efficient uses are nowadays a major issue. The evident reduction in oil reserves combined with the increase in its price, as well as the need for 'cleaner' fuels, have led in the past years to an increasing interest and research in the field of alternative fuels for both compression and spark ignition engines propulsion [1]. Alternative fuels are inherently cleaner than petroleum derived fuels. Alternative fuels are expected to be friendlier to the environment and more sustainable than conventional fuels. Alternative fuels can include bio fuels such as alcohols and biodiesel, hydrogen, LPG, and natural gas. When used with advanced engine and emission control technologies, alternative fuels burn efficiently and release fewer emissions from incomplete combustion. Advanced combustion and emission control technology requires combustion data with certain degree of detailing. Data for conventional fuels are available, however, there is a dearth of combustion data for alternative fuels. Generation of experimental data is very expensive, and therefore more emphasis is given, now-a-days, on modelled results for combustion of non-conventional fuels.

Internal combustion engines are a significant part of energy utilization. They are used in both vehicles and cogeneration plants for electrical power generation. The first law of thermodynamics has widely been used as a primary tool to assess the performance of energy conversion devices such as internal combustion engines. Energy is a thermodynamic property that can be transformed from one form to another but its total amount is conserved. The first law of thermodynamic alone is not capable of providing a suitable insight into the engine operations [13]. For this reasons, the interest in the use of the second law of thermodynamics has been intensified in the field of internal combustion engines. Analysis of a process or a system in terms of the second law of thermodynamics is termed exergy or availability analysis. The application of exergy analysis for engineering system proves to be very useful because it provides quantatives information about irreversibilities and exergy losses in the system. In this way, the thermodynamic efficiency can be quantified, poor efficiency areas can be identified, and thus processes can be designed and operated to be more efficient [1]. Therefore, exergy analysis is a suitable tool for researchers to investigate processes and systems. The second law of thermodynamics asserts that energy has quality as well as quantity. It provides an alternative and revealing mean of assessing and comparing processes and systems rationally and meaningfully by introducing exergy.

Exergy is defined as the maximum theoretical work that can be obtained from a system as it comes to equilibrium with a reference environment. The exergy content of a natural material input can be interpreted as a measure of its quality or potential usefulness, its ability to perform work. In recent years the exergy analysis method has become widely used in the design, simulation and performance assessment of thermal system.

As the exergy of fuel is destroyed during irreversible processes, the entire fuel energy cannot be converted to useful work even when combustion is complete. Considering this exergy destruction, exergetic efficiency provides more realistic criteria for evaluating the effectiveness of the energy conversion during combustion.

Exergy based analysis yields a true measure of how nearly actual performance approaches the ideal, and clearly identifies the causes and the sources thermodynamic losses and consequent impact on the environment. Exergy analysis can assist in improving and optimizing design [2].

In this study, exergy analysis was performed for the compression, combustion and expansion phases of an SI engine cycle. A two-zone thermodynamic-based combustion model was developed for this purpose. The details of the model and the analysis were given in the following sections.

2.0 DESCRIPTION

In the present study, governing equations of the SI engine are numerically solved for each time step, the time step used is for 0.1 degree crank angle. The calculation for a cycle stops at the end of power stroke when the exhaust valve opens (EVO) and values of P_e and T_e are noted to calculate the residual mass fraction, f (Eq. 12) and T_{IVC} (Eq. 13). The calculation cycle is repeated with estimated values of f and T_{IVC} until converged solution is achieved.

3.0 SIMULATION AND ENERGY-EXERGY ANALYSIS

3.1 Basic Equations of SI Engine Cycle

The cylinder volume is the function of crank angle (θ). The governing equation of volume is as follows [3]:

$$V(\theta)=\frac{V_D}{R_c-1}+\frac{V_D}{2}\left[\left(\frac{2L_C}{L_S}\right)+1-cos\,\theta-\sqrt{\left(\frac{2L_C}{L_S}\right)^2-\sin^2\theta}\right] \quad (1)$$

Where the displacement volume $V_D=\pi/4\ B^2\ L_S$, B is bore and L_S is the stroke length, R_C is the compression ratio and L_C is the connecting rod length.

The infinitesimal change of pressure in cylinder, dp can be express as follows [3]:

$$dP(\theta)=-KP(\theta)\left[\frac{dV(\theta)}{V(\theta)}\right]+(k-1)\left[\frac{\delta Q_{net}(\theta)}{V(\theta)}\right] \quad (2)$$

Where the specific ratio, $k=C_p/C_v$ and

$\delta Q_{net}(\theta)=\delta Q_{fuel}-\delta Q_{loss}(\theta)$

From the 1st law of thermodynamics

$$\delta Q_{net}(\theta)=dU(\theta)+\delta w(\theta) \quad (3)$$

Where $\delta w=PdV$ and $dU=mC_v dT$, also assumed $PV=mRT$ for ideal gas.

In the present study, experimental data of [11] are used as the reference, and engine specification is reported in Table 1.

TABLE 1. Engine specification [10][11]

Bore dia	Stroke length	Connecting rod length	Start of Combustion	Combustion duration	Compression ratio
76.3 mm	111.1 mm	160.0 mm	330^0	60^0	7

3.2 Wiebe Function as Fuel Burn Profile

In ideal SI engine cycle, fuel is assumed to burn instantaneously to result in constant volume combustion at TDC. In actual engines, finite time is required to burn the fuel-air mixture. In the present study, mass fraction burnt as a function of crank angle is calculated using the Wiebe function [4]:

$$y_b(\theta) = \begin{cases} 1 - exp[-a(\theta - \theta_s / \Delta\theta_b)^b] & if\ \theta_s \le \theta \le \theta_s + \Delta\theta_b \\ 0 & if\ \theta < \theta_s, \theta > \theta_s + \Delta\theta_b \end{cases} \quad (4)$$

where,

$y_b(\theta)$ mass fraction burnt at a crank angle

θ_s crank angle at the start of combustion

$\Delta\theta_b$ total combustion duration ($y_b = 0$ to 0.99)

a 'Wiebe efficiency factor'

b 'Wiebe form factor'

3.3 Energy Analysis

Heat release from the combustion of fuel δQ_{fuel} can be written as:

$$\delta Q_{fuel}(\theta) = Q_{LHV} . y_s . dy_b . (1 - f) \quad (5)$$

Where LHV is the lower heating value, f is the residual gas fraction and y is the mass fraction of fuel in the stoichiometric fuel-air pre-mixture and is related to stoichiometric fuel-air ratio, $(F/A)_S$:

$$y_s = \frac{(F/A)_s}{(F)\ A)_s + 1} \quad (6)$$

The heat loss from the gases to the cylinder walls, $\delta Q_{loss,}$ at an engine speed, N, can be determined with a Newtonian convection equation:

$$\delta Q_{loss}(\theta) = h_g(\theta) . A_w(\theta) . [T(\theta) - T_w] . \frac{d\theta}{2\pi N} \quad (7)$$

The cylinder wall temperature, T_w is the spatial-averaged mean temperature of the exposed cylinder wall, the head and the piston crown. The exposed cylinder area, $A_w(\theta)$ is the sum of the cylinder bore and head area, and the piston crown area:

$$A_w(\theta) = A_{wall} + A_{head} + A_{piston}$$

$$= 2V_D \left[\frac{1}{L_s} + \frac{1}{B}\left\{\left(\frac{2L_c}{L_s}\right) + 1 - cos\theta - \sqrt{(2L_c/L_s)^2 - sin^2\theta}\right\}\right] \quad (8)$$

The heat transfer coefficient, $h_g(\theta)$ is the instantaneous area averaged heat transfer coefficient, and is related to engine parameters in Woschni correlation [3] as:

$$h_g(\theta) = 3.26[P(\theta)]^{0.8} . [U_g(\theta)]^{0.8} . B^{-0.2} . [T(\theta)]^{-0.55} \quad (9)$$

U_g is the characteristic gas velocity which is proportional to the mean piston speed during intake, compression and exhaust. During combustion and expansion, it is assumed that the gas velocities are increased by the pressure rise resulting from the combustion, so the characteristic gas velocity is affected by both mean piston speed, $\bar{U}_p = 2NL_S$, and cylinder pressure, $(P-P_m)$,. Hence,

$$U(\theta) = 2.28\bar{U}_p + 0.00324T_0\left[\frac{V(\theta)}{V_D}\right]\left[\frac{P(\theta) - P_m(\theta)}{P_0}\right] \quad (10)$$

Motoring pressure Pm is obtained by simulating the code without heat release and its value is estimated using [5]:

$$P_m(\theta) = P_0\left[\frac{V_0}{V(\theta)}\right]^{1.3} \quad (11)$$

At the end of the exhaust stroke, when the engine exhaust valves open (at $\theta = \theta_{EVO}$), gas pressure P_{EVO} is greater than exhaust pressure, P_e. Hence, gas blow-down and gas displacement occur. Since, the engine cylinder have a finite clearance volume, some of the exhaust gases are left in the clearance volume. This residual gas will mix with the incoming fresh-charge giving rise to its temperature while reducing engine volumetric efficiency. Hence, residual gas fraction, f is the ratio of the gas mass in the cylinder at the end of the exhaust stroke to the mass of the charge at the time of inlet valve close ($\theta = \theta_{IVC}$), and its values can be estimated using [3]:

$$f = \frac{1}{R_c}\left[\frac{P_e}{P_{EVC}}\right]^{\frac{1}{k}} \quad (12)$$

For unthrottled engine, P_e =1 atm. Hence, temperature at the beginning of the cycle, T_{IVC} is related to f in [3] as:

$$T_{IVC} = (1-f).T_i + f.T_e.\left[1-\left(1-\frac{P_i}{P_e}\right)\left(\frac{k-1}{k}\right)\right] \quad (13)$$

The efficiency is defined in order to be able to compare different engine size applications or evaluate various improvements effects from the perspective of either the First or the second law [11]. By the 1st law (energy-based) efficiency is defined as [15][16]:

$$\eta_{I,ind} = \frac{Energy\ out\ (as\ Indicated\ work, W_{ind})}{Energy\ in\ (m_f\ Q_{LHV})} \quad or$$

$$\eta_{I,brk} = \frac{Brake\ work, W_{brk}}{m_f\ Q_{LHV}} \quad (14)$$

in this study, brake thermal efficiency is used as 1st law efficiency.

3.4 Equilibrium Composition and Thermodynamic Properties

For practical chemical equilibrium, stoichiometric combustion of a fuel, $C_\alpha H_\beta O_\gamma N_\delta$ in dry air containing 21% oxygen and 79% nitrogen, by volume can be written as [3]:

$$\epsilon \overbrace{C_\alpha H_\beta O_\gamma N_\delta}^{fuel} + \overbrace{0.21O_2 + 0.79N_2}^{air} = \nu_1 N_2 + \nu_2 O_2 + \nu_3 CO$$
$$+\nu_4 H_2 + \nu_5 CO_2 + \nu_6 H_2O + \nu_7 H + \nu_8 O + \nu_9 NO + \nu_{10} OH$$

Here ε is the molar fuel ratio and is related to stoichiometric fuel-air ratio. By using atom balance and equilibrium constant, number of mole numbers and mole fraction of 10 specified product can be estimated. NASA polynomial are also used to find out the specific heat ratio and Gibbs free energy. Then all other thermodynamic properties like entropy, enthalpy, internal energy etc can be estimated easily.

3.5 Exergy Analysis

Exergy of a system emerges as the sum of two contributions: thermomechanical exergy, EX_{tm} and chemical exergy, EX_{ch} [2]. Hence,

$$EX_{tot} = EX_{tm} + EX_{ch} \quad (15)$$

Thermomechanical exergy, EX_{tm} is the work which is obtained by taking the system by means of reversible physical processes, from its initial state to the state of restricted equilibrium with the environment, i.e., to P_0 and T_0. In the case of restricted equilibrium the system is kept separate from the environment by a physical boundary to prevent mixing and chemical interaction with the environment [6].

At the restricted dead-state, the control mass is only in thermomechanical equilibrium with the environment. In principle, the difference between the compositions of the system at the restricted dead state and the environment can be used to obtain additional work. Chemical exergy, EX_{ch} is the work which is obtained by taking the system by means of reversible processes, from the state of restricted equilibrium with the environment to unrestricted equilibrium with the environment. The maximum additional work obtained in this way is called the chemical exergy [1]. Hence, the expression for these exergies are given by [2] as:

$$EX_{tm} = u + P_0 v - T_0 s - \sum_{i=1}^{n} y_i \mu_{i,0} \quad (16)$$

$$EX_{ch} = \sum_{i=1}^{n} y_i(\mu_{i,0} - \mu_{i,00}) \quad (17)$$

The quantity $\mu_{i,o}$ is the chemical potential of species i in the restricted dead state, whereas $\mu_{i,oo}$ represent the value in the environmental dead state. The subscript 0 is used to identify properties of the thermomechanical or restricted dead state, and the subscript 00 is used to represent the properties at the environmental dead state.

The chemical exergy of a system can be divided into the diffusion exergy and the reactive exergy, which in turn can be split up into the oxidation exergy and the reduction exergy. As a result, the following equation can be derived:

$$EX_{ch} = EX_{diffusion} + EX_{reactive}$$
$$= EX_{diffusion} + EX_{oxidation} + EX_{reduction} \quad (18)$$

The total exergy content within fuel-air premixture is consumed as the exergy transfer due to work and heat, and exergy destruction due to combustion and irreversibility. Exergy due to work transfer, EX_w is defined as the exergy of the system to do actual work on a changing control volume against its surroundings. The maximum work obtainable from fuel combustion is defined as the fuel exergy, EX_{fuel}. Equations required to estimate various exergy values are as follows.

Fuel exergy: $EX_{fuel} = -(\Delta g)_{T_0,P_0}$ (19)

Exergy transfer as work: $EX_w = \int \delta w = \int Pdv$ (20)

Exergy transfer as heat : $EX_Q = \int(1 - T_0 /T) \delta q$ (21)

Various second law efficiencies (exergetic or availability efficiency, or effectiveness) have been define in the literature. In this study the following definition is used for second law efficiency [15] [16]:

$$\eta_{II} = \frac{Exergy\ out\ \ as\ work}{Maximum\ extractable\ exergy\ \ as\ work}$$

$$= \frac{W_{ind}}{W_{max}} = \frac{W_{ind}}{W_{ind} + I} \qquad (22)$$

The second law efficiency gives a finer understanding of performance than the first law efficiency. Moreover, the second law efficiency stresses both exergy transfer losses and internal exergy destruction due to the need of dealing with irreversibilities to improve performance.

3.6 Friction Calculation

For estimating the performance of a SI engine, friction calculation is one of the requirement. Friction is a totally irreversible process. It will reduce the exergy of a system as well as the potentiality to do the work. There are several ways to calculate the friction. All governing equation and calculation described in ref [7], [8] & [9], the Total friction are occurred in various parts in SI engine, that is :

Total Friction Loss =Friction losses in (piston+ valve train+crank shaft+pumping+accessory)

4.0 RESULT AND DISCUSSION

In recent years, exergy analysis method has been widely used in the design, simulation and performance assessment of various types of engines for identifying losses and efficiencies. Energy analysis of three fuels like methane, methanol and Octane include work output, cumulative heat release rate, net energy destruction, 1st law efficiency while exergy analysis was done to study availability input, exergetic 2nd law efficiency and destruction exergy which is recovered in the water jacket and exhaust gas. Special attention is given to identification and quantification of second law efficiencies and the irreversibility of various processes and subsystems.

Distribution of energy and exergy usage including efficiency in the key processes of SI engine at three different speeds for stoichiometric methane, methanol and Octane fuelling has been shown in Fig. 1, 2 & 3. A portion of exergy is lost due to combustion and other irreversibilities and the rest is exhausted. Energy balance does not take into account of the irreversibilities and consequent degradation of its quality during conversion. Addressing these irreversibilities can play a key role in future design and performance improvement.

Several important features can also be observed with engine speed variations. Increasing engine speed leaves less time for heat transfer from the cylinder, and reduces associated energy and exergy losses. The remaining energy ends up increasing the exhaust temperature and constitutes a higher loss with exhaust at higher engine speeds. Therefore, higher engine speed also results in lower exergy losses associated with heat transfer and higher exergy losses associated with exhaust. Few important parameter like adiabatic temperature, LHV, maximum temperature & pressure of three fuels also validated with ref [10]. However, net work done and work potential values are comparable in various speeds which are reported in Table 2.

TABLE 2. Energy-Exergy Results

fuel	Rc	rpm	Energy work	Energy heat	Energy exhaust	Energy friction	Energy unaval	1st law η	Exergy work	Exergy heat	Exergy exhaust	Exergy friction	Exergy unaval	2nd law η
CH_4	7	1000	34.86	22.48	35.90	3.10	3.66	34.86	34.93	19.09	18.38	3.10	24.50	60.82
		3000	37.24	13.34	40.25	4.93	4.23	37.24	37.31	11.37	21.59	4.94	24.79	63.02
		5000	37.80	11.00	41.40	7.92	1.88	37.80	37.88	9.38	22.45	7.93	22.36	67.20
	11	1000	36.82	28.25	29.29	3.68	1.96	36.82	36.89	23.75	13.83	3.89	21.84	65.12
		3000	41.08	16.22	34.50	5.45	2.75	41.08	41.16	13.71	17.50	5.46	22.17	67.74
		5000	42.11	13.08	35.89	8.49	0.43	42.11	42.19	11.06	18.51	8.50	19.74	71.97
CH_3OH	7	1000	33.10	20.49	33.87	2.84	9.69	33.10	32.47	16.95	16.66	2.79	31.13	53.11
		3000	35.19	12.08	37.83	4.52	10.37	35.19	34.53	10.03	19.48	4.44	31.52	55.28
		5000	35.68	9.94	38.86	7.26	8.25	35.68	35.01	8.26	20.23	7.13	29.37	58.93
	11	1000	35.25	25.64	27.71	3.38	8.02	35.25	34.59	20.99	12.56	3.32	28.54	57.05
		3000	38.98	14.60	32.45	5.00	8.97	38.98	38.25	12.01	15.78	4.91	29.05	59.77
		5000	39.87	11.75	33.70	7.79	6.86	39.87	39.12	9.67	16.66	7.64	26.91	63.47
C_8H_{18}	7	1000	33.16	20.65	33.00	2.80	10.40	33.16	32.35	17.03	16.20	2.73	31.69	52.54
		3000	35.34	12.15	36.99	4.46	11.07	35.34	34.48	10.06	19.02	4.35	32.09	54.75
		5000	35.85	9.99	38.03	7.16	8.98	35.85	34.98	8.28	19.78	6.98	29.99	58.32
	11	1000	35.23	25.75	26.87	3.33	8.81	35.23	34.38	21.02	12.14	3.25	29.21	56.30
		3000	39.08	14.65	31.59	4.93	9.74	39.08	38.13	12.02	15.33	4.81	29.71	59.11
		5000	40.00	11.78	32.85	7.68	7.69	40.00	39.03	9.67	16.20	7.49	27.61	62.75

Figure 1 shows the energy and exergy distribution for methane is slightly different than other fuels. Methane combustion produced a bit less generation of irreversibility which is due to simple molecular structure and better mixing and combustion. However at the present engine design, a lot of exergy is lost in heat transfer and exhaust due to high combustion temperatures. Compared to octane and methanol, methane showed better results in terms of lower exergy loss with irreversibilities and higher work exergy. Work exergy and heat transfer for methanol were very similar to those of octane.

From the analysis of three different fuels considered in the present study, some important results are as follows:

At 1000 rpm, 33.10 to 34.86% of energy contained with fuel is converted to useful work, and the figure changes to 35.68 to 37.80% in case of 5000 rpm at compression ratio 7.

At 1000 rpm, energy loss due to heat transfer is 20.49 to 22.48% and 9.94 to 11.00% at 5000 rpm. However, associated exergy losses are 16.95 to 19.09% at 1000 rpm and 8.26 to 9.38% at 5000 rpm at compression ratio 7.

At 1000 rpm, energy loss with exhaust is 33.00 to 35.90% and 38.03 to 41.40% at 5000 rpm. Exergy loss with exhaust are 16.20 to 18.38% at 1000 rpm and 19.78 to 22.45% at 5000 rpm at compression ratio 7.

At 1000 rpm, energy loss due to friction is 2.80 to 3.10% and 7.16 to 7.92% at 5000 rpm at compression ratio 7. Exergy losses with friction are nearly same value as energy losses.

At 1000 rpm, 24.50 to 31.69% of exergy contained with fuel is destroyed due to irreversibility and 22.36 to 29.99% is destroyed at 5000 rpm at compression ratio 7.

Engine work output, energy/exergy associated with exhaust and friction loss are found to increase with engine speed, while losses due to heat transfer and irreversibility's (in case of exergy) are reduced.

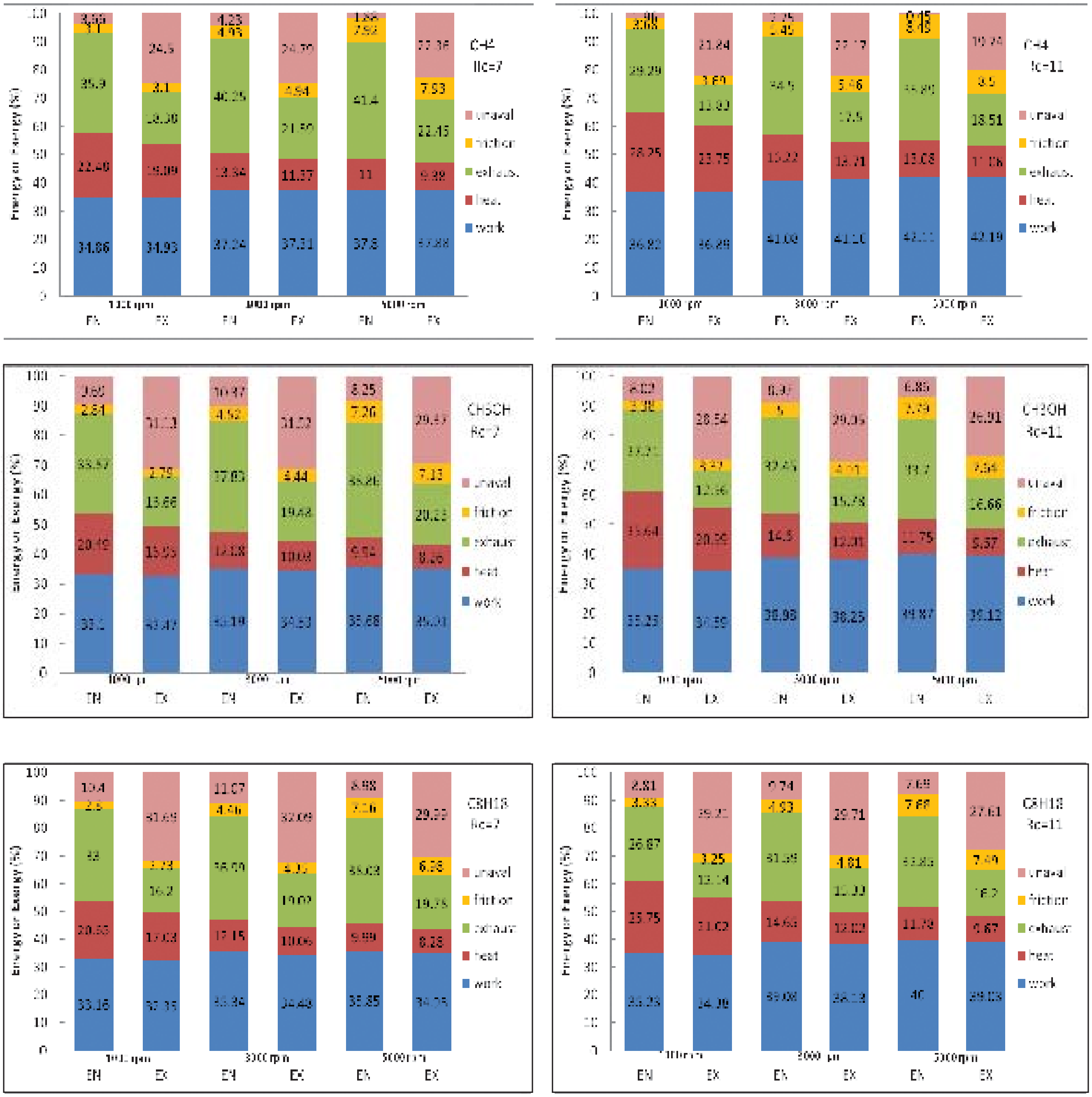

Fig 1. Energy and Exergy distribution for stoichiometric Methane, Methanol and Octane fuelling at different rpm and compression ratios

Figure 2 shows the variation of the first and second law efficiencies, depending on compression ratio and rpm. variation of first and second law efficiencies are usual and similar trend for compression ratio and rpm. It was found 1st law efficiency for methane is slightly greater than other two fuels with various engine rpm and compression ratios. 1st law efficiency for octane and methanol is nearly same for different engine speed and compression ratios. 1st law efficiency found increased for all the fuels while increasing engine speed and compression ratios.

For similar trend of first law efficiency, second law efficiency also found increased for all fuels while increasing engine speed and compression ratio. Methanol and octane show the similar trend and value for second law efficiency.

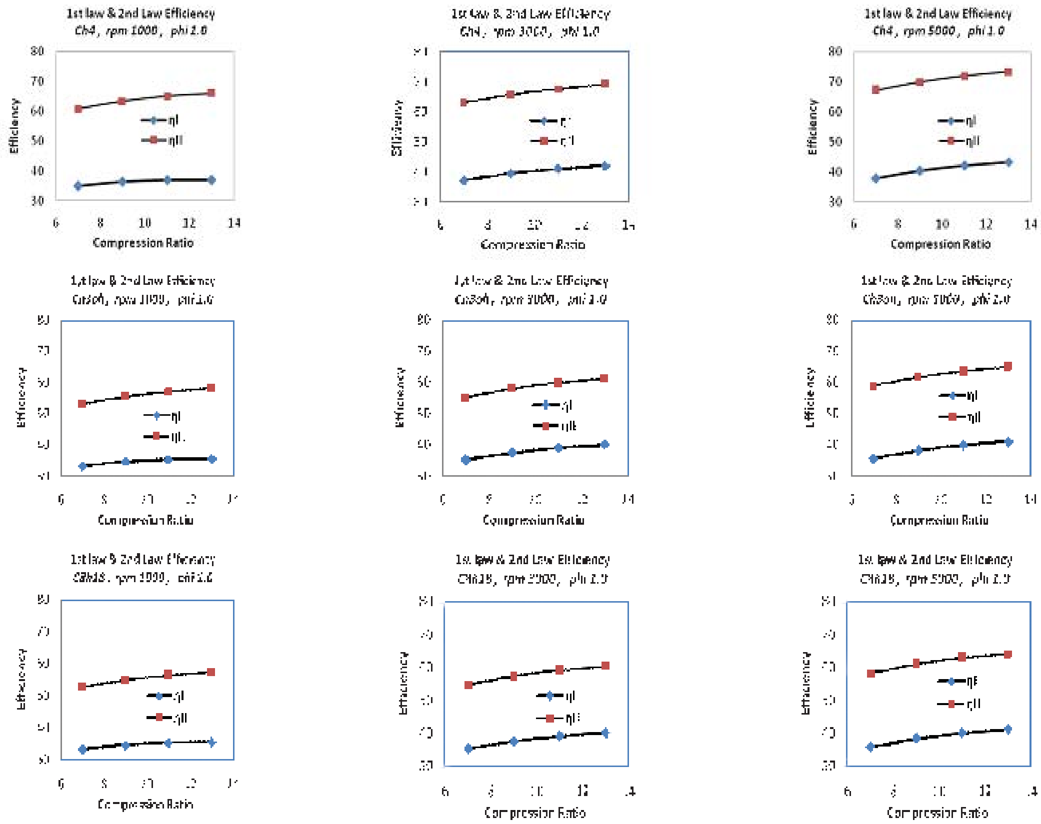

Fig 2. First law and Second law efficiency of Methane, Methanol and Octane at different rpm and compression ratios

Figure 3 shows the comparison of first and second law efficiencies for methane, methanol and octane. When considering 2nd law efficiencies (exergetic) it was found methane showed greater value than methanol and octane fuel with various engine rpm and compression ratios. For methane, total irreversibility (friction plus unavailable exergy) is minimum or less than other two fuels and this value also increase with rpm but slightly decrease with compression ratio. methanol and octane show the similar trend and value for second law efficiency. There is a more scope to reduce the total irreversibility and so to increase the performance for methanol and octane.

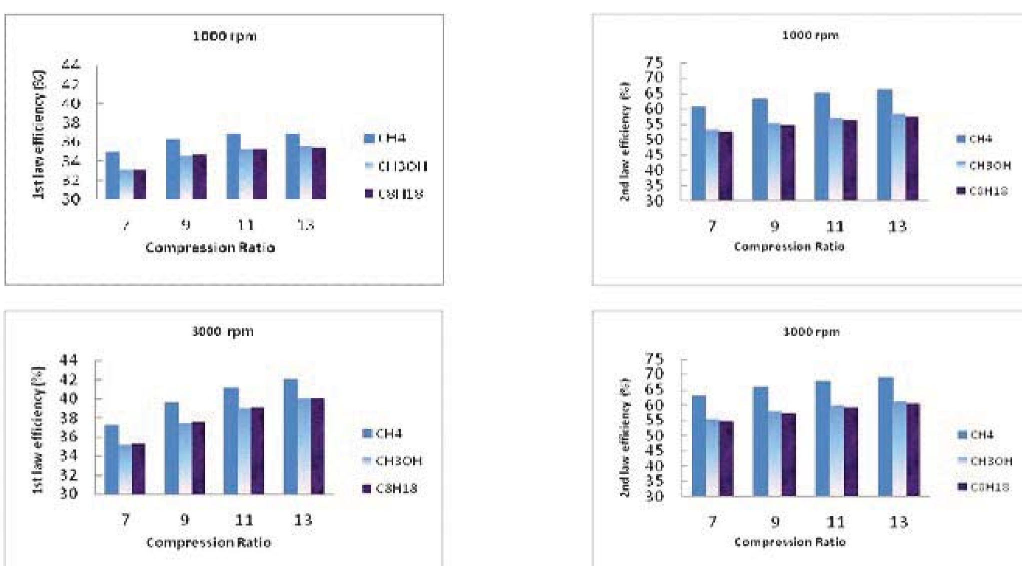

Fig 3. Comparison of 1st law and 2nd law efficiency for methane, methanol and octane at different rpm and compression ratio (considering stoichiometric mixture)

5.0 CONCLUSION

From the thermodynamic point of view, energy and exergy analysis of SI engine are the important tools for finding better energy management. The most important factor of the system inefficiency is the destruction of exergy by irreversible processes. This is mainly occurred by the combustion process. Exergy losses due to the exhaust gas and heat transfer are the other contributors in decreasing order. This study reveals that a combined energy and exergy analysis provides a much better and more realistic answer for the comparison of fuels. The results highlight the importance of exergy based analyses to probe and identify the sources of work potential losses in different phases of the SI engine cycle.

The following general conclusions have been drawn from the results of the study:

Exergetic analysis provides a better understanding of interaction between some design and operating condition, permits the revelation of the magnitude of work potential lost during the cycle in a more realistic way than the first law analysis can; and points to several possible ways for improving engine performance.

While the compression ratio increases, the first and second law efficiencies also increases due to the increasing of combustion speed and extracted work during the expansion, and also decreases in total irreversibilities.

While the rpm increases, the first and second law efficiencies also increases due to the increasing of extracted work during expansion. Friction losses also increases and unavailable irreversibilities (mainly combustion process) slightly reduces due to less time to transfer heat to surrounding by cooling effect, as a result total irreversibilities increases.

6.0 NOMENCLATURE

EX_{ch}	Chemical Exergy (kJ/kg-mix)	Δg	Change of Gibbs energy (kJ/kg-mix)
EX_Q	Exergy associated with heat transfer (kJ/kg-mix)	$\mu_{i,0}$	Chemical potential restricted equilibrium with environment
EX_{tm}	Thermomechanical Exergy (kJ/kg-mix)	$\mu_{i,00}$	Chemical potential unrestricted equilibrium with environment
EX_{tot}	Total Exergy (kJ/kg-mix)	W_{ind}	Indicated work
EX_{fue}	Fuel Exergy (kJ/kg-mix)	W_{brk}	Brake work
EX_w	Exergy associated with work interaction (kJ/kg-mix)	$\eta_{I,brk}$	1st law efficiency (brake thermal efficiency)
y_s	Fuel mass fraction in stoichiometric mixture	$\eta_{I,ind}$	1st law efficiency (Indicated thermal efficiency)
R_c	Compression Ratio	η_{II}	2nd law efficiency (exergetic)
N	Engine Speed, rpm	I	Total Irreversibilities

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