Effect of Fuel, Compression ratios on Energetic and Exergetic efficiency of Spark Ignition (SI) Engine

Munawar Nawab Karimi*, Sandeep Kumar Kamboj

Abstract - In this study, the effect of the change in compression ratios on the energetic and exergetic efficiency is being investigated for methanol, ethanol, iso-octane and LPG fuels in SI engine model at stoichiometric condition. A thermodynamic mathematical model is developed. First and second laws of thermodynamics principle are applied to the cycle model to perform energy and exergy analysis. The exergy of the fuel is being distributed in heat transfer, work transfer, combustion, exhaust, compression and expansion processes. The result shows that the first and second law efficiencies, exergy with heat transfer through cylinder wall increases with the increase in compression ratios. Exergy destruction during combustion and exhaust gases decreases with the increase in compression ratios. The second law efficiency of LPG is 1.67% greater than iso-octane and 3.20% and 5.98% greater than the methanol and ethanol respectively. Exergy with exhaust gases is 2.95%, 4.09%, and 2.5% is lower for the methanol, ethanol, and iso-octane respectively, compared to LPG. Exergy with heat transfer in case of iso-octane and LPG is almost 8.33% greater than the methanol and ethanol.

Keywords: Compression ratio, Energetic Efficiency, Exergetic Efficiency.

I. INRODUCTION

The increased importance of efficient utilization of fossil fuels in engines with reduced emissions continues to drive the need for thorough understanding of thermodynamics of internal combustion engines. Traditional analysis of thermodynamics efficiency of internal combustion engines is performed by applying a mass and energy balance throughout the engine cycle according to the first law of thermodynamics [1, 2, 3]. Comprehensive analysis of idealized internal combustion engines are found in the literature, these studies have served the basis for further optimization of current engine technologies [4]. However, practical systems are far from the thermal efficiencies predicted by the first law because analysis based on first law is often fail to identify the deviation from ideality [5, 6, 7].

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The deviation between the efficiency predicted by the first law and the observed efficiency of the engine is an evidence of the defective exploitation of the fuel, which is due to the presence of internal thermodynamic irreversibilities in the combustion process. First law of thermodynamics cannot identify and quantify the sources of losses (irreversibilities) in the engine, it simply provide the overall efficiency and hence fails to clarify the reasons of deviation between the ideal and actual performance of the engine [8, 9]. On the other hand, second law of thermodynamics offers a new perspective for the analysis of the performance of the engine based on the concept of exergy. The objective of the present study is to evaluate thermodynamic performance of Spark Ignition engine operated on iso-octane, ethanol, methanol and Liquefied Petroleum Gas (LPG) fuels. The effects of type of fuel used and change in compression ratio is observed on first law efficiency, second law efficiency of the engine. Exergy destruction during individual processes for the engine is evaluated for all selected fuels, which quantitatively visualize losses within a system and gives clear trends for optimization.

II. ASSUMPTIONS AND SPECIFICATION

Following are assumptions made and the specification of engine during analyzing the whole cycle:

Assumptions:

1) The thermodynamic system is the closed chamber content.
2) The cylinder content is assumed to be spatially homogeneous and to occupy one zone.
3) The fuel is assumed to be completely vaporized and mixed with the reactant air.
4) The thermodynamic properties (including pressure and temperature) are spatially uniform.
5) The fuel is completely vaporized in the intake ports and mixed with the incoming air.
6) Variation of specific heat with temperature at different process points has been incorporated.
7) Combustion takes place instantaneously.
8) Compression and Expansion process are polytropic.

Specification:

Bore	:	100 mm
Stroke length	:	74 mm
Engine Speed	:	3000 rpm
Length of connecting rod	:	160 mm
Inlet valve opening	:	30^o BTDC

Inlet valve closes	:	40^o BTDC
Equivalence ratio	:	1
Spark Time	:	26^o BTDC

III. THERMODYNAMIC ENGINE MODEL

An entropy generation analysis was applied to SI engine and an entropy generation calculation model was developed. Mathematical formulation of the principle of the non-conservation of entropy for a non- steady flow process in a time interval t_1 to t_2 following the second law of thermodynamics can be written as

$$\sum_i m_i s_i - \sum_e m_e s_e + \sum_r \left(\frac{Q_r}{T_r}\right)_{1\text{-}2} + S_{gen,\,1\text{-}2} = S_2 - S_1 \qquad (1)$$

Where m_i and m_e respectively denote the amount of mass across input port i and exiting across port e; $(Q_r)_{1\text{-}2}$ denotes the amount of heat transferred into the control volume r on the control surface. $S_{gen\ 1\text{-}2}$ denotes the amount of entropy

TABLE 1
COMPARISON OF SELECTED FUELS PROPERTIES

Property	Methanol	ethanol	Iso-octane	Propane	Butane
Chemical formula	CH_3OH	C_2H_5OH	C_8H_{18}	C_3H_8	C_4H_{10}
Molecular weight (Kg/kmol)	32.04	46.07	114.22	44.14	58.17
Oxygen present (wt %)	49.9	34.8	-	-	-
Density (g cm^{-1})	792	789	700	0.0024	0.001865
Freezing point at 1 atm (^0C)	-97.778	-80.0	-107.4	-	-
Boiling temperature at 1 atm (^0C)	64.9	74.4	99.224	231	273
Auto-ignition temperature(^0C)	463.889	422.778	257.23	482	-
Latent heat of vaporization at 20^0C (KJ/Kg)	1103	840	349	-	-
Stoichiometric air/fuel ratio (AFR)	6.47	9.0	15.2	15.6	15.34
Lower heating value of the fuel (KJ/Kg)	20000	26900	44300	46350	45710
Research octane number (RON)	111	108	100		
Motor octane number (MON)	92	92	100	104	

generated in the control volume and S_1 and S_2 are the amounts of entropy in the control volume at t_1 and t_2.

Fuel properties of methanol, ethanol, iso-octane and LPG (A mixture of 75 % propane and 25 % butane) are given in table 1[11, 12, 13,14].

The entropy generation during the four key processes of the SI engine is calculated after applying the equation (1) and the consequent exergy destruction is evaluated after using the Gouy –stodola theorem as

$$E_{destruction} = T_0 S_{generation} \qquad (2)$$

Process (1-2):

The fuel air mixture (charge) is assumed to be an ideal gas, and hence for the constant value of specific heats, the specific exergy destruction during the process 1-2 which is a compression process is given by

$$E_{Dest,(1\text{-}2)} = T_o \left(C_P \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1}\right) \qquad (3)$$

Where, the temperature T_2 of the charge at the end of the compression is given by:

$$T_2/T_1 = (V_1/V_2)^{n-1} = (r)^{n-1}$$

Process (2-3):

The exergy destruction during the process of heat addition of constant volume (from heat source to the fuel) is given by:

$$e_{Dest\ 2\text{-}3} = To\ \left\{C_V \ln \frac{T_3}{T_2} - \frac{Qin}{T\ source}\right\} \qquad (4)$$

Where,

$$Q_{in} = m_f \times (\text{LHV of the fuel})$$
$$\text{and } m_f \times LHV \times \eta_{comb} = m\, C_v (T_3 - T_2) \qquad (5)$$

The exergy destruction during the subsequent combustion, which is the most concerned irreversible process, may be obtained after applying the exergy balance over the engine combustion chamber as

$$e_2 + e_Q = e_3 + e_{Dest,com} \qquad (6)$$

Where, e_Q is the amount of exergy associated with the heat transfer q_{in} and is given by:

$$e_Q = (1 - T_o/T)\ m_f \times LHV$$

e_2 is the physical exergy of the charge before combustion, and e_3 is the sum of physical and chemical exergy of the fuel air mixture after combustion. The equation (8) may further be elaborated as:

$$e_{Dest,comb} = e_{2,physical} - e_{3physical} - e_{3,chemical} + e_Q \qquad (7)$$
$$e_{Dest,comb} = (e_{2,} - e_3)_{physical} - e_{3chemical} + e_Q$$

The term $(e_{2,} - e_3)_{physical}$ for a closed system engine combustion chamber is given by:

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$(e_2\text{-}e_3)_{physical} = u_2\text{-}u_3 + p_o(v_2\text{-}v_3) - T_0(s_2\text{-}s_3)$ (8)

The term (e_3) chemical is the chemical exergy of the fuel which is given by equation (8). The first two terms of the equation (9) is the reactive chemical exergy which is the work produced due to reversible transition of the species of the mixture from restricted dead state to the ultimate dead state (environmental state), and the third term of the equation (9) is the diffuse chemical exergy which is the maximum work produced due to the difference of partial pressures of the ambient species of mixture between the restricted dead state and the partial pressures of the same species at the environment state.

$E_{3,chememical} = E_{Re} + E_{Diff}$

$$E_{3,chem} = \sum_{i=1}^{n} N_i g_i(T_0 p_i) - \sum_{j=1}^{n} N_j g_j(T_0 p_j) + \sum_{j=1}^{n} N_j \bar{R} \; T_o \ln p_j / p_{jo} \quad (9)$$

In equation (9), the index i denotes the individual species present in the mixture at restricted dead state and the index j denotes the individual ambient species present in the mixture that were formed from the species i through a series of oxidation /reduction reactions, g_i and g_j are the Gibbs free energies, N_i and N_j are the number of moles, p_i and p_j are the partial pressure of the species denoted by the species i and j respectively. p_{jo} is the partial pressure of the ambient species of the environmental state and $\bar{R}$ is the universal gas constant.

The specific chemical exergy of the mixture may be obtained as:

$e_{3,\,chem} = E_{3,chemical} / \sum M_j N_j$ (10)

Where M_j is the molecular weight and N_j is the number of moles of species j of the combustion products of the mixture.

Wall heat transfer calculations [4]

The wall heat losses in SI engine are different for different fuels depending upon the thermal conductivity and buring rates in addition to the quenching distances.
In general the instantaneous convective heat transfer coefficient for gas to wall heat exchange is modeled by using Annands & Woschnis correlation

$h_c = 3.26p^{0.8}B^{-0.2}U^{0.8}$ (11)

Where, U is the characteristic gas velocity and is given by:

$U = 2.285s_p + 0.00324T_oV_d/V_o \times \Delta p/p_o$ (12)

The surface area of the engine combustion chamber exposed to the heat at the given crank angle is:

$A_w(\theta) = A_{head} + A_{piston} + A_{cy(\theta)}$ (13)

$A_{cy(\theta)}$ is the area of the cylinder, and at given crank angle θ it may be presented as:

$A_{cy(\theta)} = \pi BL(R+1-\cos\theta - (R^2-\sin^2\theta)^{1/2})$ (14)

Where R = 2L/B and L is the stroke length and B is the bore.
Therefore using the above parameters the amount of heat lost from gas to the wall heat transfer in the combustion chamber is given by:

$Q_w = h_w A_w \Delta T$ (15)

The amount of exergy associated with the wall heat transfer is given by:

$E_w = Q_w(1-T_o/T_w)$ (16)

Process (3-4)

The exergy destruction during the expansion process (3-4) is calculated similar to the process of compression, and may be presented as

$e_{Dest,expansion} = T_o(C_p \ln T_4/T_3 - R \ln p_4/p_3)$ (17)

Process (4-1)
The exergy lost during the exhaust process is calculated after using the entropy generation principle and is given by:

$e_{Dexhaust} = T_o(C_v \ln T_1/T_4 - Q_{out}/T_{sink})$ (18)

The first law efficiency or thermal efficiency of the SI engine is given by:

η_{th} = net work done/m_f × LHV of the fuel (19)

The second law efficiency or exergy efficiency of the engine is given by:

η_{exergy} = Net work done/chemical exergy of the fuel (20)

IV. RESULT AND DISCUSSION

The variation of exergy destruction during combustion with the change in compression ratio of the engine is shown in figure 1

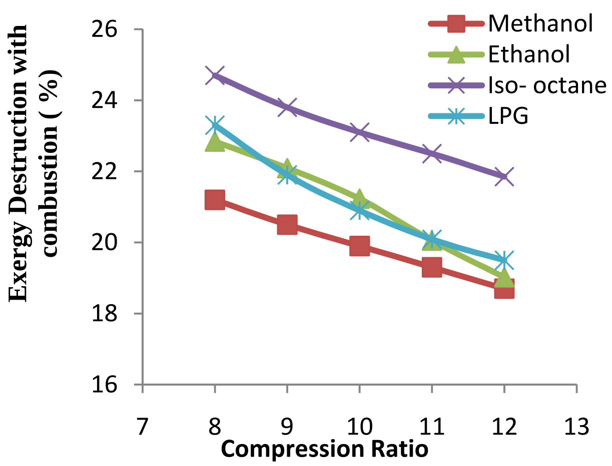

Fig 1: Variation in Exergy destruction with combustion of various fuels

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It is found that exergy destruction decreases significantly with the increase in compression ratio for all four selected fuels.

This is because higher compression ratio increases, the combustion temperature and pressure increases which results in the reduced exergy destruction. It is also observed that among all four fuels, the exergy destruction during combustion of methanol is significantly lower among all the four selected fuels which are due to rich oxygen content.

Similar to this, the exergy destruction during combustion for ethanol is also less than other hydrocarbon and comes next to methanol. Exergy destruction in case of LPG is less than iso-octane and close to ethanol.

This is due to the reason that combustion of LPG in gaseous state mixes faster with the air and hence a lesser entropy would be generated due to mixing which results in the lesser exergy destruction. At the given compression ratio, the exergy destruction during combustion of iso-octane is higher than all other four fuels which is due to its complex molecular structure.

Fig.2 shows the variations of exergy destruction during heat transfer for all four selected fuels with the change in compression ratio. It is shown that exergy destruction via heat transfer for all four fuels increases with the increase in compression ratio.

This is because high compression ratio increases the combustion temperature and pressure which results in great amount of heat transfer from combustion to the cylinder wall. It is also observed that exergy destruction via heat transfer for ethanol and methanol is less than the exergy destruction via heat transfer for LPG and iso-octane. This is due to the reason that the key variables for heat transfer are the convective heat transfer coefficients which depends upon the characteristic gas velocity of the fuels.

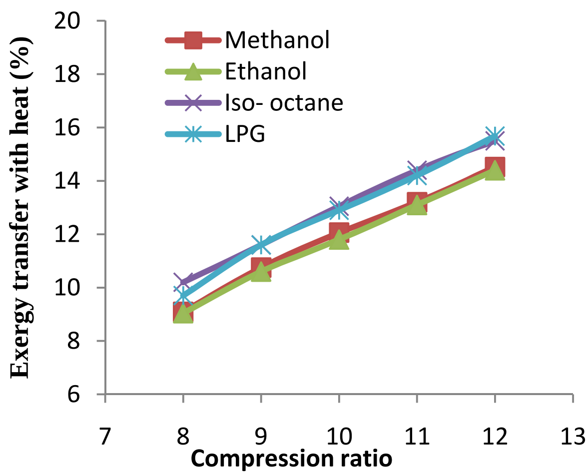

Fig 2: Variation of Exergy transfer with heat

Since the characteristic gas velocity for the ethanol and methanol is less than the characteristic gas velocity of iso-octane and LPG. The exergy destruction via heat transfer in ethanol and methanol is less than the iso-octane and LPG.

Effect of compression ratio on exergy destruction via exhaust of four fuels is shown in Fig.3.

It is found that exergy destruction via exhaust decreases significantly by increase in compression ratio for all fuels. This is because higher compression ratio, results in greater amount of heat transfer from combustion zone to cylinder wall which results in lower exhaust gas temperature and hence the lower exergy lost via exhaust.

The exergy lost through exhaust for oxygenated fuels that is methanol and ethanol is less than the exergy lost via exhaust for iso-octane and LPG. This is because combustion of oxygenated fuel is faster and smoother which results in fast heat release and hence the lower exhaust temperature.

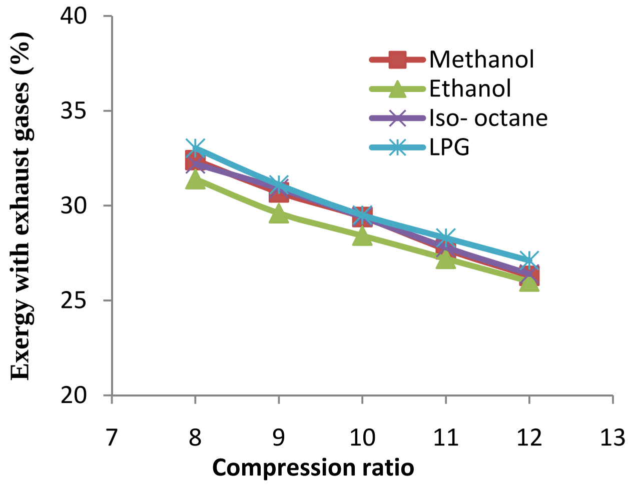

Fig 3: Variation in exergy with exhaust gas of various fuels

The variations of first law efficiency of SI engine running on various fuels with the change in compression ratio are shown in figure 4.

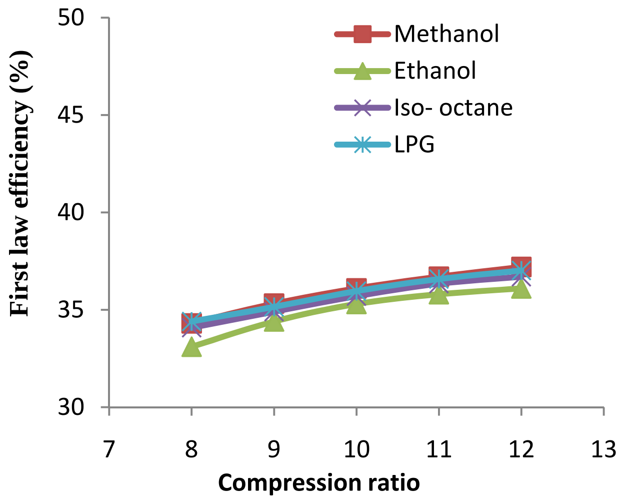

Fig 4: Variation in First law efficiency of various fuels

It is found that first law efficiency increases with the increases in compression ratio for all selected fuels. This is because higher compression ratio increases the combustion temperature and pressure which obviously increases the first law efficiency but not significantly because higher

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combustion temperature results in greater amount of heat transfer due to which first law efficiency suffers. Due to higher combustion temperature , a smaller amount of heat input is required for giving work output of the engine and this effect is dominating over the effect of heat transfer at higher combustion temperature from gases to the cylinder wall therefore an increasing trend for first law efficiency is observed with the increase in compression ratio.

Fig.5 shows the effect of change in compression ratio on second law efficiency of SI engine operated on various fuels. It is observed that the second law efficiency of the engine for all fuels increases with the increase in compression ratio.
This is because higher compression ratio increases the combustion temperature which decreases the exergy destruction during combustion and exhaust gases and hence increases the second law efficiency. It is further shown that second law efficiency for oxygenated fuels is less than the second law efficiency of iso-octane and LPG.

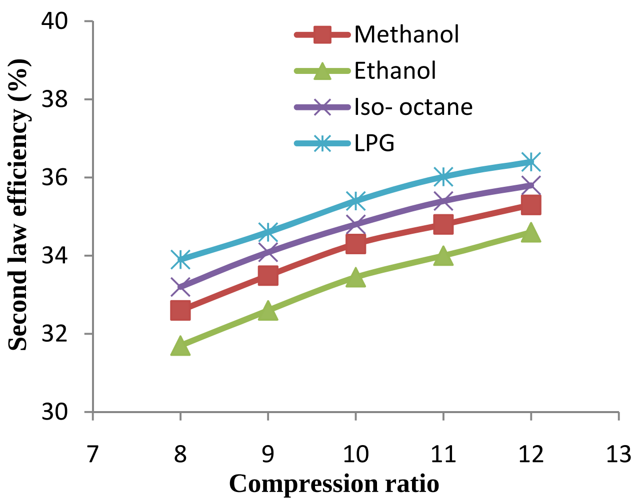

Fig 5: Variation in Second Law efficiency of various fuels

This is because during combustion at constant volume the value of specific heat is higher in oxygenated fuels which lower down the flame temperature that result in higher exergy destruction and hence the lower second law efficiency than first law efficiency.

V. CONCLUSIONS

In this study, the use of alternative fuels that is methanol, ethanol, iso-octane, and LPG in SI engine has been evaluated by using a thermodynamic model of exergy analysis.

The result shows that the first law efficiency and second law efficiency increases with the increase in compression ratio for the all investigated fuels. The results also show that the exergy destruction during combustion and exhaust decreases with the increase in compression ratio for all the examined fuels. But exergy transfer with heat through cylinder wall increases with the increase in compression ratio. Exergy of the products after combustion are lesser than the exergy of the reactants because of combustion irreversibilities. The second law efficiency of LPG is 1.67% greater than the iso-octane and 3.20% and 5.98% greater than the methanol and ethanol respectively at a compression ratio of 10.Exergy with exhaust gases is lowered by 2.95%, 4.09%, and 2.5% for the methanol, ethanol, and iso-octane respectively than the LPG for a compression ratio of 12. Exergy transfer with the iso-octane and LPG is almost 8.33% greater than the methanol and ethanol at a compression ratio of 10.

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