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Analysis of performance parameters of the energy and specific heats of alternative fuels in spark-ignition engine-A Technical Review.

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Abstract :

This paper presents a fundamental thermodynamic approach to study spark - ignition engine. A thermodynamic model is developed to study the energy distribution of alternative fuels that is methanol, ethanol, iso- octane and liquefied petroleum gas (LPG). The energy of the fuels are distributed with work, heat transfer through the wall and energy with the exhaust gases. In addition to this, specific heats of air fuel mixture of alternative fuels are calculated with the change in compression ratios during compression, combustion, expansion and exhaust. This study shows that the major portion of energy goes waste with the exhaust. The results also showed that the energy with the exhaust gases decreases with the increase in compression ratio and energy with work and heat transfer increases with the increase in compression ratio for all the fuels examined. The specific heats of all the fuels increase from compression to the combustion and decreases slightly during exhaust stroke. This variation in specific heats is taken into consideration while calculating energy distribution. The specific heats of hydrocarbon fuels are lower than the oxygenated fuels during compression, combustion and exhaust. Energy with work for methanol and LPG are higher about 2.21% from ethanol and iso-octane because of their higher adiabatic flame temperature. Energy with exhaust gases of methanol are higher about 4%, 4.78% and 3.83% for iso-octane, LPG, and ethanol respectively. Energy with heat loss of iso-octane are higher about 1.78%, 19.86% and 1.36% for LPG, methanol and ethanol respectively.

Keywords **- Energy, Compression Ratio, Ethenol,Methenol, LPG,Iso-Octane.**

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1. INTRODUCTION

Today's energy crises and environmental problems have concentrated the investigations on alternative fuels for decreasing the consumption of exhaustible petroleum reserves and minimizing the concentration of toxic components. Alcohols can be considered as suitable alternative fuels because they can be made from renewable resources, such as various grown crop sand even waste products.[1,2] Moreover, alcohols reduce the harmful emissions, such as carbon monoxide (CO) and unburned hydrocarbon (UHC) emissions, by supplying leaner combustion because of the oxygen content in their molecular structures.[3]Methanol and ethanol are commonly used alcohols as engine fuels or fuel additives because of their fuel properties. containing the knowledge from the rules. Recently the KBANN system [4], addresses the problem by using domain knowledge to select a promising configuration for a neural network[4] The fuel properties of isooctane, methanol, and ethanol and LPG are given in Table1.5-7 Alcohol fuels have simple molecular structures. They burn efficiently and improve combustion efficiency. High octane numbers of methanol and ethanol allow for the use of higher compression ratios and improve thermal efficiency of the engine. Methanol and ethanol have a higher latent heat of vaporization in comparison to isooctane. This provides more mass into the cylinder by cooling the inducted air and increases engine power.[9]For these reasons, numerous experimental and theoretical studies have been performed on the use of oxygenated fuels in internal combustion engines.[7,8]

Table 1.0 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	114.228	44.14	58.17
Oxygen present (wt %)	49.9	34.8	-	-	-
Density (g cm^{-1})	792	789	700	2.458×10^{-3}	1.865×10^{-3}
Freezing point at 1 atm (^{0}C)	-97.778	-80.0	-107.378	-	-
Boiling temperature at 1 atm (^{0}C)	64.9	74.4	99.224	231	273
Auto-ignition temperature(^{0}C)	463.889	422.778	257.23	482	-
Latent heat of vaporization at 20^{0}C (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	

2. SYSTEM DESCRIPTION:

The cycle considered for analysis is a complete representation of the four stroke SI engine including the compression, combustion, expansion and exhaust processes and as shown in Figure.1. Process 1-2s is a reversible adiabatic compression, while process 1-2 is an irreversible adiabatic process that takes into account the internal irreversibilities in the real compression process. The heat addition is a constant volume process 2-3 process 3-4s is a reversible adiabatic expansion, while 3-4 is an irreversible adiabatic process that takes into account the internal irreversibilities in the real expansion process. The heat rejection is at a constant volume process 4-19-11.

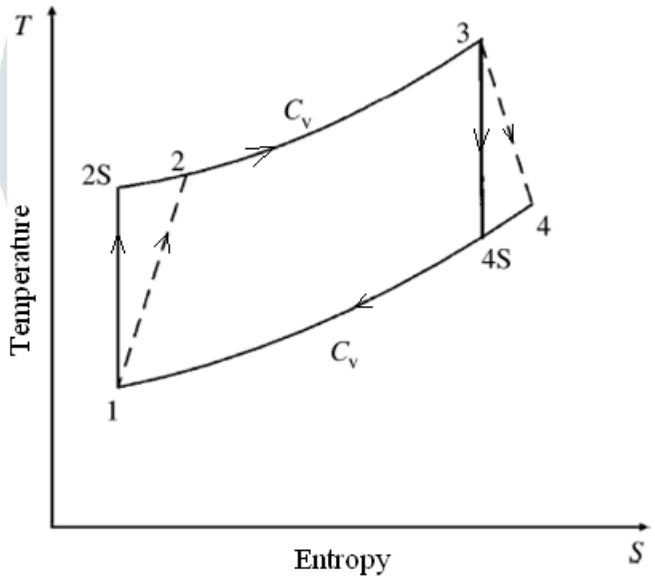


Fig- 1. (Four stroke SI cycle)

2.1 CHEMICAL EQUATIONS:

The following chemical equations are used during combustion of fuels at stoichiometric condition.

(i) $CH_3OH + 1.5O_2 + 5.65\ N_2\ CO_2 + 2H_2O + 5.65N_2$ (methanol fuel)

(ii) $C_2H_5OH + 3O_2 + 11.28\ N_2\ 2CO_2 + 3H_2O + 11.28N_2$ (ethanol fuel)

(iii) $C_8H_{18} + 12.5O_2 + 47\ N_2\ 8CO_2 + 9H_2O + 47N_2$ (iso-octane fuel)

(iv) $0.8C_8H_{18} + 0.2\ C_4H_{10} + 5.3O_2 + 27.32\ N_2\ 3.2CO_2 + 4.2H_2O + 27.32\ N_2$ (LPG fuel)

3. THERMODYNAMIC MODEL:

The specific heat at constant pressure, of the mixture is find out by the relation

$$(C_P)_{mix} = \sum_{i=1}^{n} m_i c_{p_i} \quad (1)$$

The specific heat at constant volume of the mixture is find out by the relation

$$(c_v)_{mix} = \sum_{i=1}^{n} m_i \, c_{v_i} \quad (2)$$

The characteristic gas constant is found out by the relation

$$R = C_p - C_v \quad (3)$$

$$R = \frac{\overline{R}}{M} \quad (4)$$

Where M is the molecular weight of the fuel.

$$m = MN \quad (5)$$

Where m is the mass of the fuel in air fuel mixture.

Temperature at the end of the process is given by

$$T2/T1 = (V1/V2)^{n-1} = (r)^{n-1} \quad (6)$$

Process (2-3):

During process 2-3, combustion of fuels takes place, the of Cp and Cv are calculated at the average temperature (temperature after compression + adiabatic flame temperature) / 2

Values of Cp and Cv of different fuels are calculated using the following equations. Specific heats as a function of temperature are represented as

Cp = a + bT + cT2 + dT3 (T in K, Cp in kj / kmol . k)

Substance	a	b	c	d
Methanol	19.0	9.152×10^{-2}	-1.22×10^{-5}	-8.039×10^{-9}
Ethanol	19.9	20.96×10^{-2}	-10.38×10^{-5}	20.05×10^{-9}
Iso-octane	-0.053	6.75×10^{-3}	-3.67×10^{-6}	-0.39×10^{-9}
Propane	-4.04	30.48×10^{-2}	-15.72×10^{-5}	31.74×10^{-9}
Butane	3.96	37.15×10^{-2}	-18.34×10^{-5}	35.00×10^{-9}

$Q_{in} = mf \times LHV$ of the fuel and

$$m_f \times LHV = m \, Cv \, (T_3 - T_2) \quad (7)$$

Where m is the mass of the mixture (1kg mixture of air and fuel is considered at the stoichiometric condition), m_f is the mass of fuel.

The adiabatic flame temperature T_3 is calculated after using the energy balance over the heat addition process as

$$H_P = H_R \quad (8)$$

Where Hp and HR are the enthalpy of products and reactants respectively.

3.1 WALL HEAT TRANSFER CALCULATIONS:

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[12-16].

$$h_c = 3.26 \, p^{0.8} \, B^{-0.2} \, U^{0.8} \quad (9)$$

where, U is the characteristic gas velocity and is given by

$$U = 2.285 \, _{sp} + 0.00324 \, T_o \, V_d/V_o \times \Delta p/p_o$$

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

$$A_w(\theta) = A_{head} + A_{piston} + A_{cy(\theta)} \quad (10)$$

Acy(θ) is the area of the cylinder, and at given crank angle θ it may be presented as

$$A_{cy(\theta)} = \Pi bl\{R+(1-\cos\theta) - (R^2-\sin 2\theta)^{1/2}\} \quad (11)$$

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

$$Q_w = h_w A_w \Delta T \quad (12)$$

Process 3- 4:

Work done during expansion process is calculated by:

$$W_{3\text{-}4} = mR(T_3\text{-}T_4) / n\text{-}1 \quad (13)$$

Where n is the polytropic index and R is the characteristic gas constant.

Process 4-1:

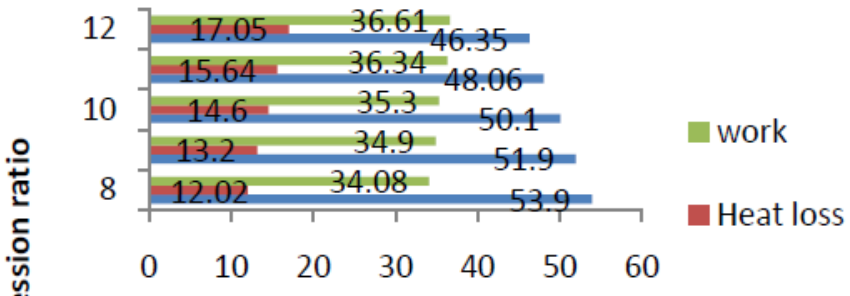
The energy lost with the exhaust gases are calculated by using the following equation.

$$Q_{out} = m C_v (T_4 - T_1) \quad (14)$$

4.0 RESULTS AND DISCUSSIONS:

4.1 ENERGY DISTRIBUTIONS OF ISO-OCTANE:

Figure 2 shows effects of the change in compression ratio on the energy with work, energy with exhaust gases and heat loss for the iso-octane. The results shows that the energy with the exhaust gases decreases with the increase in compression ratios and energy with work and heat transfer increases with the increase in compression ratio for the iso-octane. It is because of the reason that pressure and temperature increases which results in increased heat loss through the cylinder wall and more indicated power is produced during expansion. The exhaust gases cooled down with the increase in compression ratios because most of the energy goes with work and heat transfer through wall which results in less energy transfer with the exhaust gases.

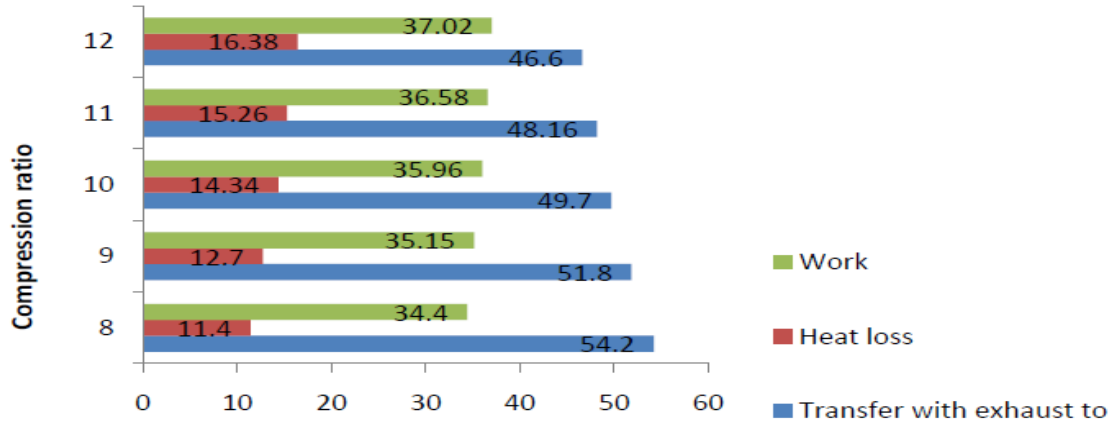


	8	9	10	11	12
work	34.08	34.9	35.3	36.34	36.61
Heat loss	12.02	13.2	14.6	15.64	17.05
Transfer with exhaust to enviroment	53.9	51.9	50.1	48.06	46.35

Fig.2 Energy distribution for Iso -octane

4.2 ENERGY DISTRIBUTION OF LPG:

Figure 3 shows effects of the change in compression ratio on the energy with work, energy with exhaust gases and heat loss for the LPG. The results shows that the energy with the exhaust gases decreases with the increase in compression ratios and energy with work and heat transfer increases with the increase in compression ratio for the LPG. It is because of the reason that pressure and temperature increases which results in increased heat loss through the cylinder wall and more indicated power is produced during expansion process. The exhaust gases cooled down with the increase in compression ratios because most of the energy goes with work and heat transfer through wall which results in less energy transfer with the exhaust gases.

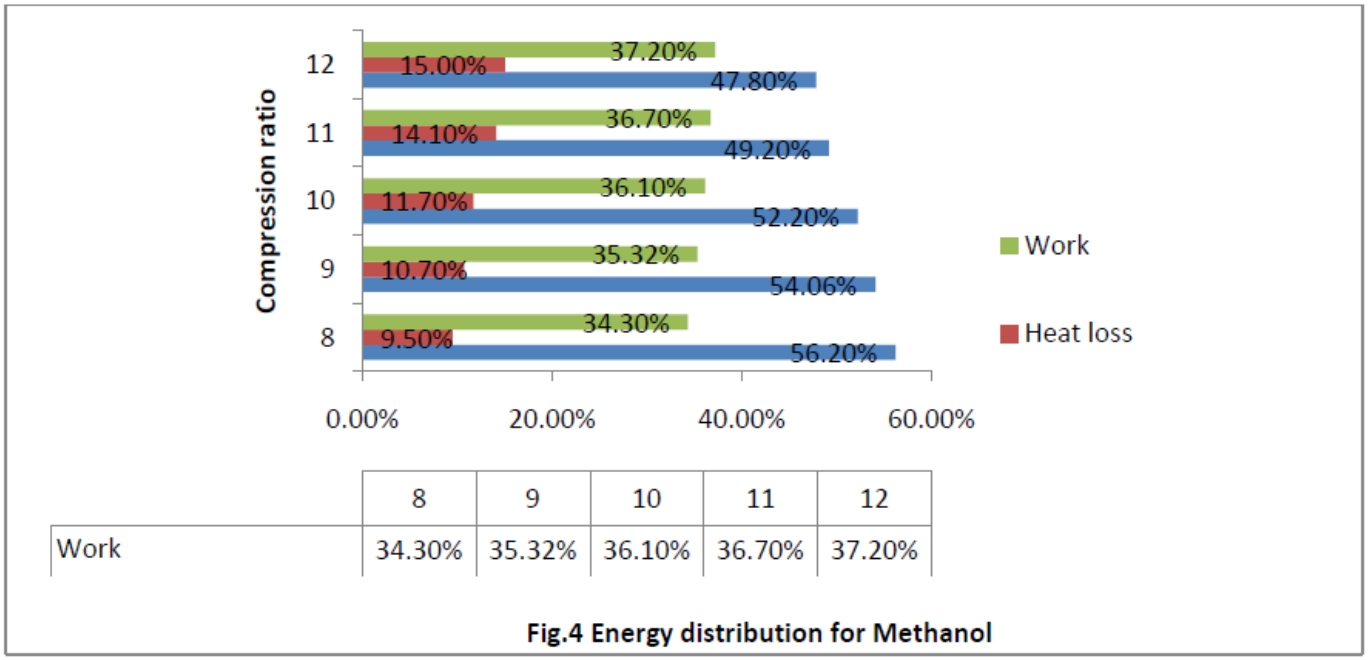


	8	9	10	11	12
Work	34.4	35.15	35.96	36.58	37.02
Heat loss	11.4	12.7	14.34	15.26	16.38
Transfer with exhaust to enviroment	54.2	51.8	49.7	48.16	46.6

Fig.3 Energy distribution for LPG

4.3 ENERGY DISTRIBUTION OF METHANOL:

Figure 4 shows effects of the change in compression ratio on the energy with work, energy with exhaust gases and heat loss for the methanol. The results shows that the energy with the exhaust gases decreases with the increase in compression ratios and energy with work and heat transfer increases with the increase in compression ratio for the methanol. It is because of the reason that pressure and temperature increases which results in increased heat loss through the cylinder wall and more indicated power is produced during expansion. The exhaust gases cooled down with the increase in compression ratios because most of the energy goes with work and heat transfer through wall which results in less energy transfer with the exhaust gases.

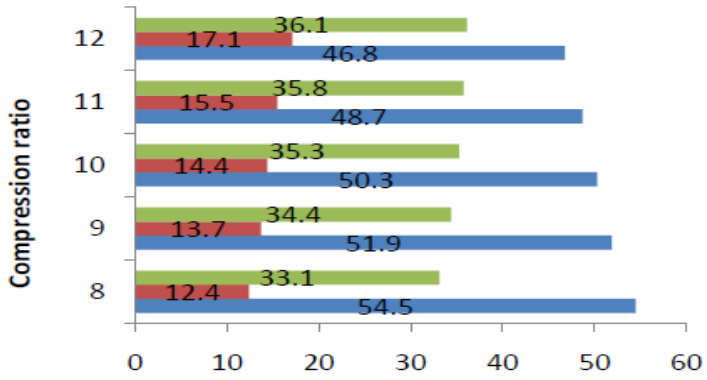


	8	9	10	11	12
Work	34.30%	35.32%	36.10%	36.70%	37.20%

Fig.4 Energy distribution for Methanol

4.4 ENERGY DISTRIBUTION OF ETHANOL:

Figure 5 shows effects of the change in compression ratio on the energy with work, energy with exhaust gases and heat loss for the ethanol. The results shows that the energy with A National Conference on "Advances In Mechanical Engineering & Emerging Technologies" AMEET-2012 the exhaust gases decreases with the increase in compression ratios and energy with work and heat transfer increases with the increase in compression ratio for the ethanol. It is because of the reason that pressure and temperature increases which results in increased heat loss through the cylinder wall and more indicated power is produced during expansion. The exhaust gases cooled down with the increase in compression ratios because most of the energy goes with work and heat transfer through wall which results in less energy transfer with the exhaust gases.



	8	9	10	11	12
Work	33.1	34.4	35.3	35.8	36.1
Heat los	12.4	13.7	14.4	15.5	17.1
Transfer with exhaust to enviroment	54.5	51.9	50.3	48.7	46.8

Fig.5 Energy distribution for Ethanol

4.5 SPECIFIC HEATS OF DIFFERENT FUELS:

Figure 6, 7, 8, and 9 shows the change in specific heats at constant pressure and constant volume for the ethanol, methanol, iso-octane and LPG during compression, combustion, expansion and exhaust processes for the air fuel mixture. The values of specific heats increases with the increase in temperature of the air fuel mixture of all the fuels examined which results in more specific heats during combustion which is calculated at mean temperature of $T_2+T_3/2$. The values of specific heats for the iso-octane and LPG are significantly lower than the ethanol and methanol during compression, combustion and exhaust. The specific heat of ethanol during compression is 0.59% higher than methanol and 1.9% greater than the iso-octane and LPG. The specific heats of methanol during combustion are 0.6%, 4.59% and 2.16% higher than ethanol, iso-octane and LPG respectively. The specific heats of methanol during exhaust are 2.2%, 5.4% and 4.1% higher than ethanol, iso-octane and LPG respectively.

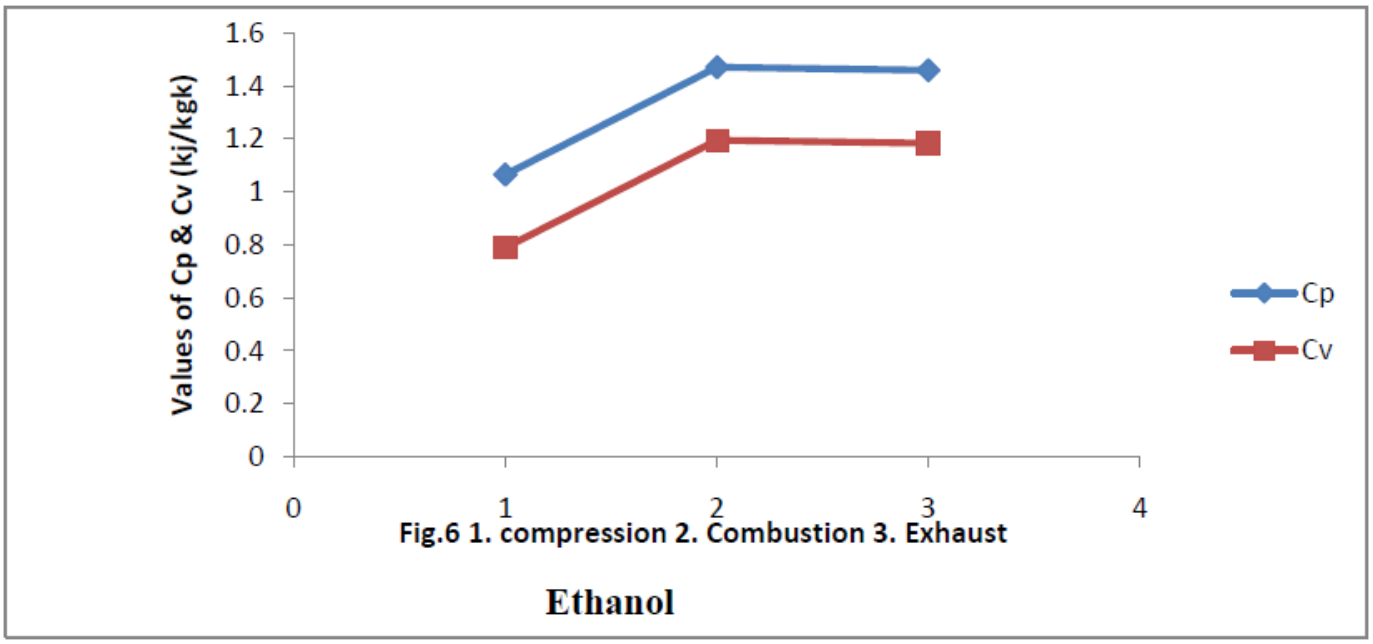


Fig.6 1. compression 2. Combustion 3. Exhaust

Ethanol

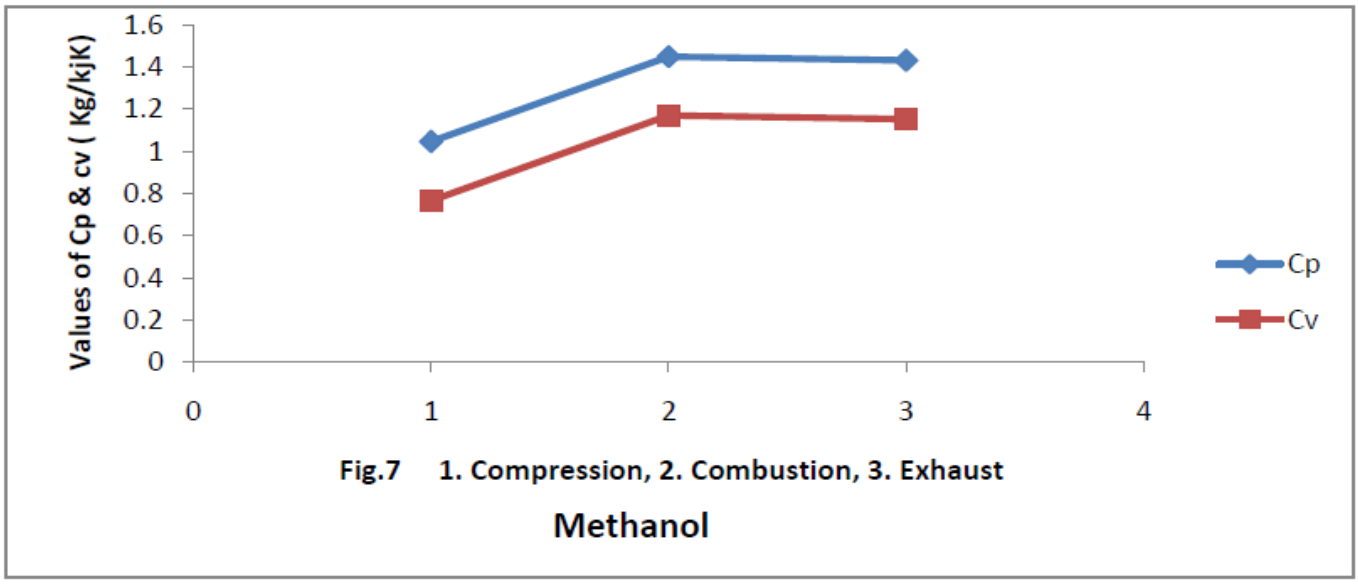


Fig.7 1. Compression, 2. Combustion, 3. Exhaust

Methanol

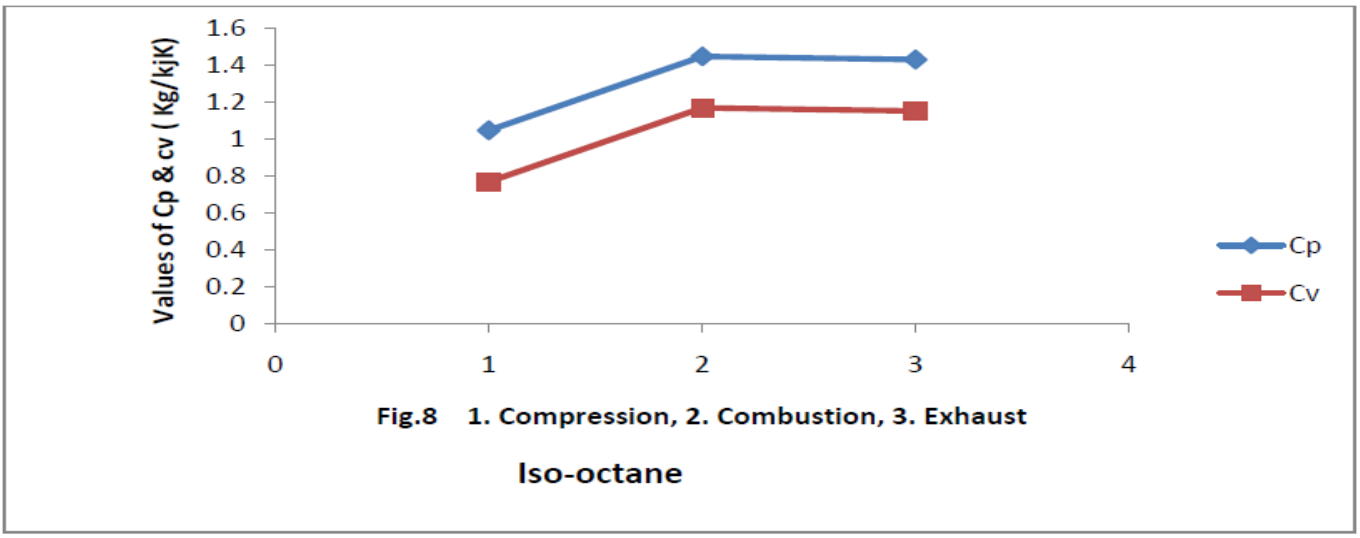


Fig.8 1. Compression, 2. Combustion, 3. Exhaust

Iso-octane

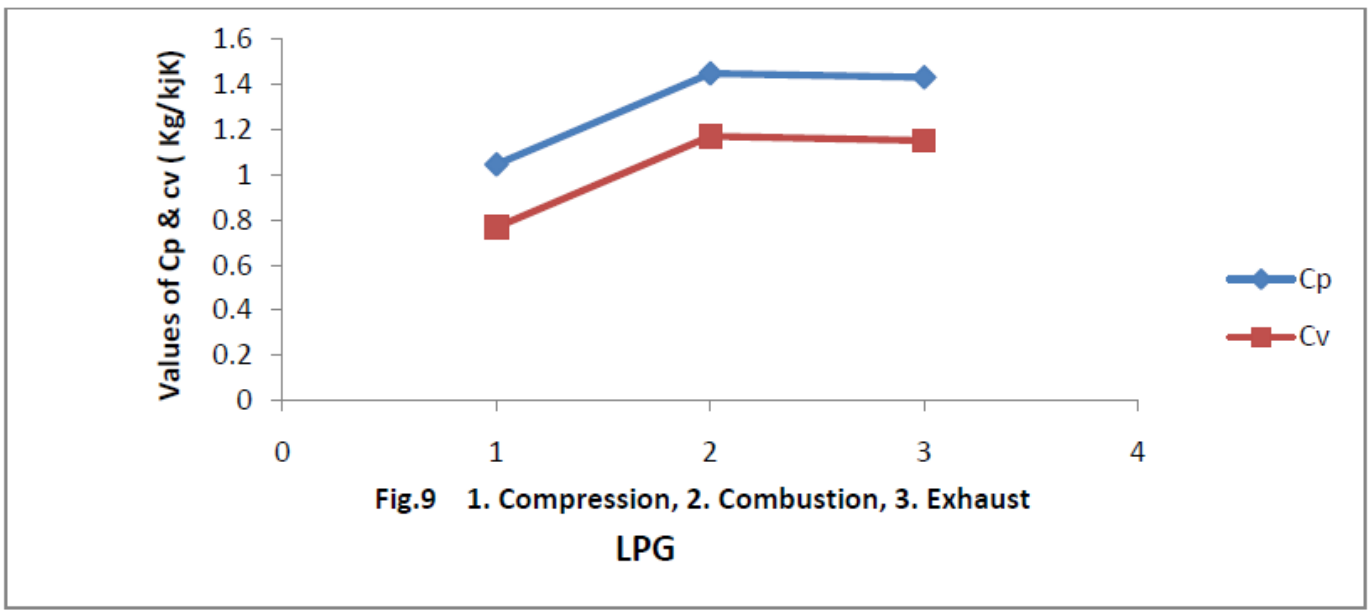


Fig.9 1. Compression, 2. Combustion, 3. Exhaust

LPG

CONCLUSIONS:

It is concluded from this study that the energy with the exhaust gases decreases with the increase in compression ratio and energy with work and heat transfer increases with the increase in compression ratio for all the fuels examined. The specific heats of all the fuels increases from compression to the combustion and decreases slightly during exhaust stroke. This variation in specific heats are taken into consideration while calculating energy distribution in different processes

of otto cycle. Energy with work are almost close to each other for all the fuels examined and energy with work for methanol and LPG are higher about 2.21% from ethanol and iso-octane because of their higher adiabatic flame temperature. Energy with exhaust gases of methanol are higher about 4%, 4.78% and 3.83% for iso-octane, LPG, and ethanol respectively. It is because of the reason that specific heat at constant volume is lower for iso-octane and LPG than the oxygenated fuels. Energy with heat loss of iso-octane are higher about 1.78%, 19.86% and 1.36% for LPG, methanol and ethanol, respectively. Ethanol, methanol, and LPG can be used as alternative fuels in spark ignition engines as a pure fuel or as a blending with iso-octane.

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