# Exergy Analysis of SI Engine during Combustion and Compression processes Based on Alternative Fuels as Methanol, Ethanol

**Santosh kumar Gupta**[1]

[1]*Department of Mechanical Engineering, Madan Mohan Malviya University of Technology, Gorakhpur, Uttar Pradesh -273010, India*
[1]*Mechanical Engineering, Department of Technical Education, Uttar Pradesh-India*

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***Abstract -*** *It is well known that the destruction of availability or exergy results from irreversible processes called combustion. For an adiabatic, constant volume system, the cycle's destruction of availability (energy) is investigated. This analysis was conducted without the use of experimental measures. For ethanol, methanol, propane, and octane-air combinations, the fraction of fuel availability that is destroyed by the irreversible processes is determined as a function of temperature, pressure, and equivalency ratio. When operating at greater temperatures, the combustion process generally results in less destruction of the fuel's available energy. Because they include some oxygen, methanol and ethanols differ greatly from gasoline. This study examines the fuel-air cycle and performs estimates for energy destruction during the compression, combustion, expansion, and exhaust process, taking into account changes in the equivalency ratio. With the adjustment in the equivalency ratio, the efficiency of the first and second laws is also determined for propane, ethanol, methanol, and iso-ocatne. Furthermore, depending on the specific operating conditions, equivalency ratio might have a large impact on the destruction of availability. In particular, the destroyed availability as a result of the combustion process varied from around 5 to 25% of the initial reactant availability under the study's conditions. There is a description of how these findings affect internal combustion engine combustion processes.*

***Key Words***: **Availability, Petrol Engine, Equivalent Ratio, Air Fuel Ratio, Methanol, Ethanol**

## 1.INTRODUCTION

The availability of a given system is defined as the maximum useful work that can be obtained in a process in which the system comes to equilibrium with the surroundings or attains a dead state. The availability of energy conversion systems is one direct result. The second law of thermodynamics is a potent exposition of pertinent physical facts with many applications in engineering and energy conversion system operation. For example, the second law computes the maximum performance of thermal systems, establishes equilibrium conditions, determines the direction of processes, and identifies the components of processes which are used to determine overall performance. The framework provided by thermodynamics second type law allows for a deeper comprehension of combustion processes. thorough understanding of the energy conversion process. [1-3] Numerous investigations on the operation and exhaust emissions of spark-ignition engine (SI engine) based on blends of fuel like gasoline and alcohol have been conducted; result show that the emissions are reduced [8-13]. For example, Ceviz and Yüksel [14] looked into how blends of gasoline without ethanol affected emissions and cyclic variability in a spark-ignited engine. According to the study's findings, utilizing fuel mixes of ethanol-free gasoline decreased the coefficient of variation in the displayed mean effective pressure, as well as the concentrations of CO and HC emissions, while raising the CO2 content in the fuel blend up to 10% ethanol by volume[6]. In a gasoline engine, Çelik [18] employed ethanol as fuel at a high compression ratio. In comparison to running with pure gasoline fuel, he discovered that using E50 fuel at a high compression ratio enhanced engine power while lowering specific fuel consumption (sfc) and exhaust pollutants. In a different investigation, Shenghua et al. [35] ran three-cylinder Spark Ignition engine at full load using different methanol percentages in gasoline (10%, 15%, 20%, 25%, and 30%). They discovered that when the fuel blend's methanol content increased, the engine's power and torque dropped and the brakes' thermal efficiency increased. Gasoline-methanol mixes were also used as fuels in a port fuel injection (PFI) gasoline engine by Fan et al. [36] without any modifications. According to the study's findings, methanol-gasoline blended fuels had no effect on engine performance, and there was no discernible change in the cylinder pressure or heat release rate as the blended fuel's methanol concentration increased [37,38].

Ethanol, methanol, and propane are the alternative fuels under consideration. An engineer is interested in determining the most useful work that can be obtained from a system in a given state. [20,27,28] Obviously, the greatest amount of work will be produced only when the system's the ultimate condition is in harmony with its environment. In other words, until the system's interaction with the environment reaches a point where it can no longer function in any way. A system's availability is determined by the maximum amount of useful work which may be find throughout a process that ends when the system reaches with state of inactivity or equilibrium with its surroundings.[4] It is obvious that a system's availability is dependent upon both its internal and external conditions. The entropy law of thermodynamics, a comprehensive and potent statement that related physical investigations with numerous implications for the engineering design and operation of energy conversion systems, directly leads to availability. [5].

### 1.1 Research Gap and Novelty

The aim of the present investigation was to evaluation and analysis in detail the destruction of availability for processes like combustion and compression. The compression process in adiabatic, constant volume combustion system was studied. The novelty of the research is changing the equivalency ratio. As the ratio of equivalency changed, for computations and analysis. Ethanol, Methanol are very different from gasoline as it contains some amount of oxygen in it. In this paper, a fuel – air actual cycle is being considered and calculations are being done for energy destruction during compression, combustion, expansion and exhaust process with the change in equivalence ratio.

## 2. Methodology

### 2.1 Engine Description for analysis

For the analysis, a real four-stroke fuel-air cycle for spark ignition has been taken into account. The investigation only looks at the destruction of energy during combustion. **Figure 1** depicts the entire SI engine cycle. The process 1-2s is an example of reversible adiabatic compression, whereas the process 1-2 is actual compression process. Processes 2-3 and 3-4 are examples of reversible and irreversible adiabatic processes, respectively, and combustion and expansion, respectively. Process 4-1's exhaust is running at a steady volume. It is assumed that the process of constant volume combustion is adiabatic. There is no energy transmitted as a result of heat transfer because the process is adiabatic. Since the system volume is closed, mass flows do not result in the transmission of energy. Consequently, during a work transfer, no energy is exchanged. Therefore, the combustion process alone is responsible for any change in exergy during a change of state. This study's analytical methodology is based on related work that employed engine cycle calculations with compression changes.

When performing constant volume adiabatic combustion, the following presumptions are made.

1.The closed chamber content is the thermodynamic system.
2. It is believed that the cylinder's contents are homogeneous in space and only occupy one zone.
3. It has been included to vary the specific heat with temperature at various process points.
4. There is quick combustion.
5. It is considered that all of the fuel has evaporated and combined with the reactant air.
6. There is spatial uniformity in the thermodynamic properties, such as temperature and pressure.

Table 1 showing the engine specification which are used in analysis of exergy analysis during various operations like compression and expansions etc.

**Table 1** Engine specification

| | |
|---|---|
| Dia of Bore | 120 mm |
| Inlet valve time opening | 35° Before TDC |
| Inlet valve time closes | 45° Before TDC |
| Equivalence ratio | Various |
| Fuel | Different |
| Compression Ratios | Varying |
| Spark ignition Time | 30°Before TDC |
| Stroke length | 80mm |
| Engine Speed 3000rpm | 3000 rpm |
| Length of connecting rod | 180 mm |

## 3 Thermodynamics Approach

### 3.1 Availability

A thermodynamic property of a system, availability is also known as exergy or exergy (essence of energy) by various authors. It measures the maximum amount useful work that any system achieve when it is allowed to reversibly change into a condition of thermodynamic equilibrium with its environment.[23][24]

### 3.2 Restricted Dead State

The local environment's circumstances are characterized by the confined dead state. This was selected because, when any system reaches perfect balance with its immediate surroundings, it can no longer generate any more valuable work and is deemed to be "dead." This dead state is incomplete and called "restricted" because the contents of the system cannot mix or react with the surroundings.[25][26]

For the dead state conditions, 298.15 K and 101.325 kPa which is 1 atm are the recommended temp and pressure. The elements that make up the dead state needs to be specified in addition to the temperature and pressure of dead state.

### 3.3 Irreversibility

According to the second law, the entropy of a system plus its surroundings, or an isolated system, can never decrease. The second law states:[26]

$$(\Delta S)_{System} + (\Delta S)_{surr.} = 0, \text{ where } \Delta = \text{final} - \text{initial}$$

$> 0$ irreversible for real world

$= 0$ reversible frictionless, ideal

If Q is the heat from a source at T, then in an ideal situation, its availability or the maximum amount of work which it can produce is $Q(1-T_0/T)$,
where $T_0$ is the ambient temperature. It will always be less than this amount. We call the difference as irreversibility.

Availability (Exergy) = Maximum possible work-Irreversibility

$$W_{useful} = W_{rev} - I$$

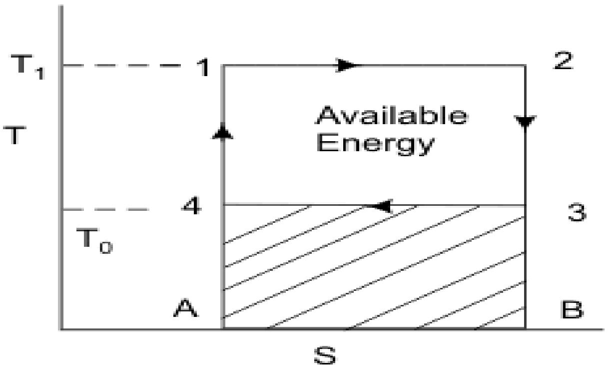

**Figure 1** T-S diagram of Carnot Cycle [21][22]

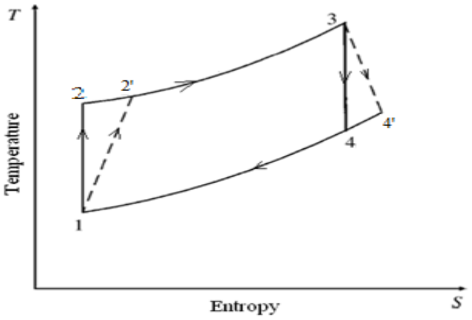

**Figure 2** T – S Diagram of SI Engine [7]

The differential amount of work done by the engine is given by

$$dW = dQ\ \eta = dQ\ (1 - To/T) \qquad (1.2)$$

If the Carnot engine works till the temperature of the body attains a value T2 (**Figure 2**) the total work done by the reversible engine is given by

$$W = \int Q\ \eta = \int_{T1}^{T2} dQ(1 - \frac{To}{T}) = \int dQ - To|\Delta S| = Q - To|\Delta S| \qquad (1.3)$$

### 3.4 Determination of Availability

After determining the thermodynamic characteristics of a specific set of circumstances, determining availability (Exergy) is comparatively straightforward. The potential and kinetic energies are ignored in this development (and can be shown to be negligible). All the way through the system, every time. [29-31][39]

$$a = (u-u_0) + p_0\,(v-v_0) - T_0(s-s_0) \qquad (1.4)$$

the availability balance reduces to

$$a_{dest} = a_1 - a_2 \tag{1.5}$$

change in entropy due to irreversibility's as follows:

$$a_{dest} = T_0 S_{gen}/m = I/m \tag{1.6}$$

where $S_{gen}$ is the difference in entropy due to irreversibility

### 3.5 Exergy Concept

A system's exergy is the maximum shaft work it can accomplish within a specific reference environment. The reference environment is thought to be infinite, harmonious, and encompasses all other systems.[17]

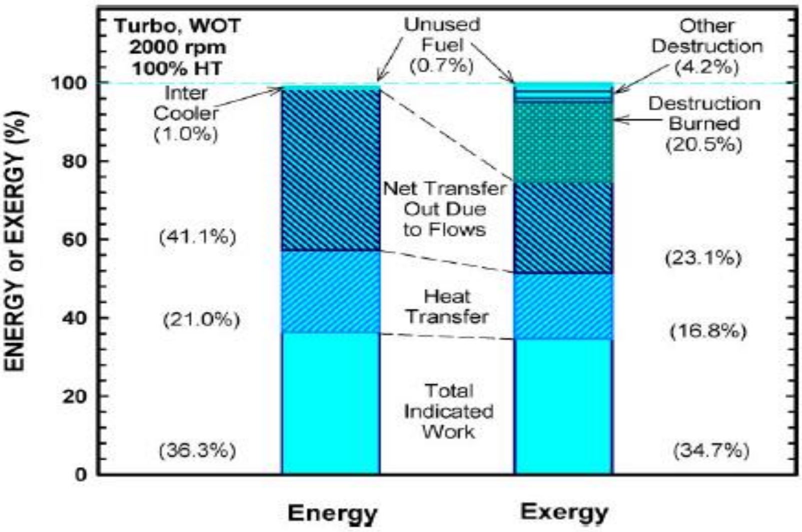

**Figure 3:** Energy destruction of various components at full load [45-47]

It indicates energy destruction of various components was examined at full load. It is very essential to analyzing heat transfer level and work transfer (**Figure 3**). After examines engine performance thermodynamically chemical balance are necessary for different fuel use.

## 4 Analysis

### 4.1 Chemical balance equations

Table 2 shows the properties of Fuels Used in Analysis. This table 2 is used to analysis of chemical balance equation.

**Table –2** Property of Different Fuels [15,16]

| Property of substance | Methanol | Ethanol | Methane | Iso Octane | Propane |
|---|---|---|---|---|---|
| Chemical formula of fuel | $CH_3OH$ | $C_2H_5OH$ | CH4 | $C_8H_{18}$ | $C_3H_8$ |
| Fuel Molecular weight (Kg/kmol) | 32.04 | 46.07 | 16.04 | 114.228 | 44.14 |
| oxygen present in fuel (wt%) | 49.9 | 34.8 | - | - | - |
| (AFR) Stoichiometric air/fuel ratio | 6.47 | 9.0 | 17.19 | 15.2 | 15.6 |
| LCV of the fuel (KJ/Kg) | 20000 | 26900 | 50016 | 44300 | 46350 |
| Energy of fuel (KJ/Kg) | 21100 | 28400 | 51056 | 45500 | 47100 |

Chemical Equations for Propane [11] [40-44]
$0.8C_3H_8 + 5O_2 + 18.8N_2 = 2.4CO_2 + O_2 + 3.2H_2O + 18.8N_2$
$0.9C_3H_8 + 5O_2 + 18.8N_2 = 2.7CO_2 + 0.5O_2 + 3.6H_2O + 18.8N_2$
$1.1C_3H_8 + 5O_2 + 18.8N_2 = 2.3CO_2 + CO + 4.4H_2O + 18.8N_2$
$1.2C_3H_8 + 5O_2 + 18.8N_2 = 1.6CO_2 + 2CO + 4.8H_2O + 18.8N_2$
Chemical Equations for Iso-Octane
$0.8C_8H_{18} + 12.5O_2 + 47N_2 = 6.4CO_2 + 2.5O_2 + 7.2H_2O + 47N_2$
$0.9C_8H_{18} + 12.5O_2 + 47N_2 = 7.2CO_2 + 1.25O_2 + 8.1H_2O + 47N_2$
$1.1C_8H_{18} + 12.5O_2 + 47N_2 = 6.3CO_2 + 2.5CO + 9.9H_2O + 47N_2$
$1.2C_8H_{18} + 12.5O_2 + 47N_2 = 4.6CO_2 + 5CO + 10.8H_2O + 47N_2$
Chemical Equations for Methanol
0.8CH3OH + 1.5O2 + 5.64N2 = 0.8CO2 + 0.3O2 + 1.6H2O + 5.64N2
0.9CH3OH + 1.5O2 + 5.64N2 = 0.9CO2 + 0.15O2 + 1.8H2O + 5.64N2
1.1CH3OH + 1.5O2 + 5.64N2 = 0.8CO2 + 0.3CO + 2.2H2O + 5.64N2
1.2CH3OH + 1.5O2 + 5.64N2 = 0.8CO2 + 0.3CO + 2.2H2O + 5.64N2
Chemical Equations for Ethanol
0.8C2H5OH + 3O2 + 11.28N2 = 1.6CO2 + 0.6O2 +2.4H2O + 11.28N2
0.9C2H5OH + 3O2 + 11.28N2 = 1.8CO2 + 0.3O2 +2.7H2O + 11.28N2
1.1C2H5OH + 3O2 + 11.28N2 = 1.6CO2 + 0.6CO +3.3H2O + 11.28N2
1.2C2H5OH + 3O2 + 11.28N2 = 1.2CO2 + 1.2CO +3.6H2O + 11.28N2
All of the above equations were derived from the analysis section's equating and balancing [32-34].

### 4.2 Thermodynamic Relations

In order to define the composition and fuel air equivalency ratio, it is useful to know about fuel air equivalency ratio, which is defines

$$\Phi = (F/A)_{actual}/(F/A)_{stoichiometric} \quad (4.1)$$

The molecular weights of the reactants and products were calculated using the following relations.

$$MR = \sum ni\ Mi\ /N \quad (4.2)$$

The quantity of reactant or product moles per working fluid mass unit is denoted by ni. The species has a molecular mass of Mi and a total molecular mass of N. The relationship is used to determine the specific heat of the reactant and product.

$$R = C_p - C_v \quad (4.3)$$

$$C_v = \sum n_i\ C_{vi} / N \quad (4.4)$$

$$C_p = \sum n_i\ C_{pi} / N \quad (4.5)$$

The value of Cp and Cv were evaluated at two points at the start of the compression processes
At the average temperature (temperature following compression plus adiabatic flame temperature) 2 during the combustion process equations like the following are used to compute the values of Cp and Cv of various fuels. The representation of specific heats as a function of temperature is $Cp = a + bT + cT^2 + dT^3$ (4.6)
(T in K, Cp in kj / $k_{mol}$ K)
The values of the constants a, b, c and d are taken from the table 3.
The compression efficiency can be defined as (For the reversible adiabatic process 1- 2s)

$$\eta_c = T_{2s} - T_1/\ T_2 - T_1 \quad (4.7)$$

$$T_{2s}/T_1 = (V_1/V_2)^{\gamma-1} \quad (4.8)$$

Where γ is the adiabatic index and its value is Cp/Cv
Now polytropic index is calculated by using the following equation

$$T_2/T_1 = (V_1/V_2)^{n-1} \quad (4.9)$$

where V represents the gas in the cylinder's volume.
A balance equation for availability was used to determine the mixture's availability during compression. (1–2).[39]

$$a_2 - a_1 = ( u_2 - u_1) + P_0 (v_2 - v_1) - T_0(s_2 - s_1) \quad (4.10)$$

The energy with work transfer is represented by the second term in this equation, and the energy destruction is represented by the third term.

During process of compression, entropy generation is given by

$S_{gen} = (C_p \ln T_2/T_1 - R \ln P_2/P_1 + Q/T_0)$ (4.11)

In this cycle, the choice of constant volume adiabatic combustion allows for the combustion process to be held responsible for any variations in availability. Since combustion is adiabatic, there won't be any availability transfer as a result of heat loss through the cylinder wall. Undesirably high temperatures are reached at the maximum cycle temperature when all of the fuel's energy is deployed.

$Q_{in} \times \eta_c \times LHV = m_f C_v (T_3 - T_2)$ (4.12)

Whereas $m_f$ is mass flow rate of the air fuel (AF) mixture.

The ratio of the net chemical energy released to the total energy released during combustion is known as combustion efficiency.[22]

$\eta_c = H_R (T_A) - H_P(T_A) / m_f \times LHV$ (4.13)

Where $H_R (T_A) - H_P (T_A) = m (\sum n_i \Delta h_{fi})_R - (\sum n_i \Delta h_{fi})_P$ (4.14)

$\Delta h_{fi}$ is the species i at standard enthalpy of production at room temperature $T_A$.

For every system going through any process, including reacting systems, the entropy balance is expressed as [25][26]

$S_{in} - S_{out} + S_{gen} = \Delta S_{sys}$ (4.15)

$S_{in} - S_{out}$ is the mass and heat-induced net entropy transfer. $S_{gen}$ is generation of entropy and $\Delta S_{sys}$ is the entropy change.

$\sum Q_k/T_k + S_{gen} = S_{prod} - S_{react}$ (4.16)

Where Tk is the boundary temperature where Qk crosses it.

Nevertheless, entropy generation for an adiabatic process (Q = 0) is provided by

$S_{gen} = S_{prod} - S_{react}$ (4.17)

The ideal gas entropy change relation provides absolute entropy values for any temperature T at pressures other than 1 atm.[19]

$\bar{s}\,(T, P) = \bar{s}^0(T,P_o) - \bar{R} \ln P/P_o$ (4.18)

$\bar{s}\,(T,P_i) = \bar{s}^0(T,P_o) - \bar{R} \ln y_i P_m/P_o$ (4.19)

Where $P_i$ is partial pressure, and $y_i$ is mole fraction of the component, $P_m$ is mixture's total pressure.

Exergy destruction during combustion process is provided by [19-20][48].

$E_{dest} = T_0 S_{gen}$ (4.20)

Percentage of energy destruction is defined as Energy destruction in the process of combustion/ Chemical energy of the fuel (4.21)

TABLE – 3 Constant Values in the Eqn. (4.6) [21]

| Substance chemical | A | b×10-3 | c×10-5 | d× 10-9 |
|---|---|---|---|---|
| Iso-octane | -0.053 | 6.75 | -0.367 | -0.39 |
| Propane | -4.04 | 304.8 | -15.72 | 31.74 |
| Methanol | 19.0 | 91.52 | -1.22 | -8.039 |
| Ethanol | 19.9 | 209.6 | -10.38 | 20.05 |

Since the equivalency ratios of different fuels vary, the equivalency ratios used in the analysis are 0.8, 0.9, 1.1, and 1.2. This is because the entire exergy analysis is based on these differences. This is so because these discrepancies form the foundation of the entire exergy analysis. Consequently, all chemical equations are balanced when the equivalency ratio values above are taken. This research contains a tabular representation of the calculated results for every parameter that was used in the analysis (such as the percentage of energy destruction during compression and combustion, the change in availability during compression and combustion, and the energy destruction during compression and combustion). Additionally, this research provides a thorough explanation along with the necessary graphs and curves.

## 5. Results and Discussion

The all-calculated results of all parameters which have been taken in this analysis (i.e. energy destruction during compression and combustion, change in availability during compression and combustion, percentage of energy destruction during compression and combustion etc.) with respect to equivalence ratio

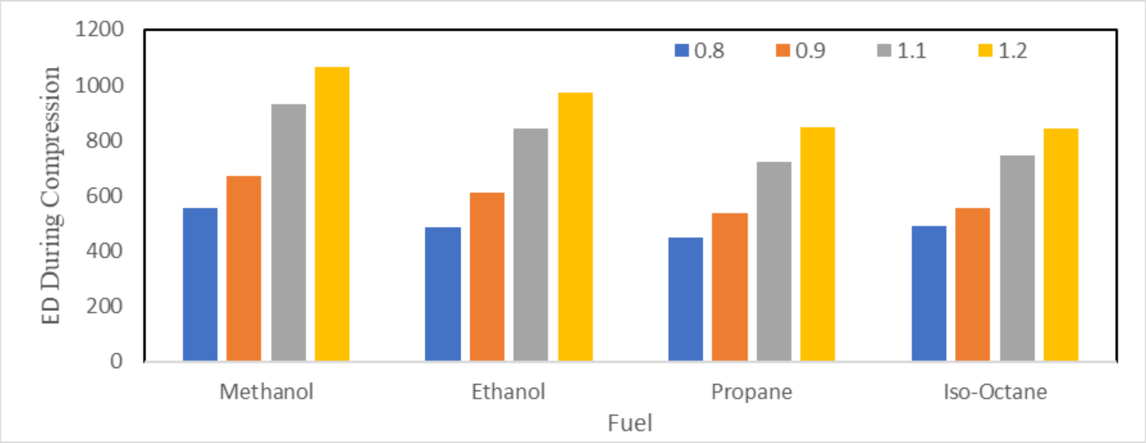

**Figure 4** Exergy destruction during compression vs fuel with varying equivalent ration

**Figure 4** illustrates how the energy dissipation during compression changes the methanol and ethanol equivalency ratios. Energy destruction during compression is demonstrated to decrease up to the stoichiometric condition (Φ = 1), and this is because, for all selected fuels, temperature after compression decreases up to an equivalency ratio of 0.8 to 1. This reduces the amount of heat transfer through the wall. Figure 4 illustrates how energy destruction in the process of compression rises in rich conditions for an equivalency ratio of 1.1 and then falls further for every selected fuels. The temperature at an equivalency ratio of 1.1 is more than temperature for an equivalency ratio 1.2, which explains why. Because temperature is at its lowest during stoichiometric conditions, energy destruction during compression is at its lowest.

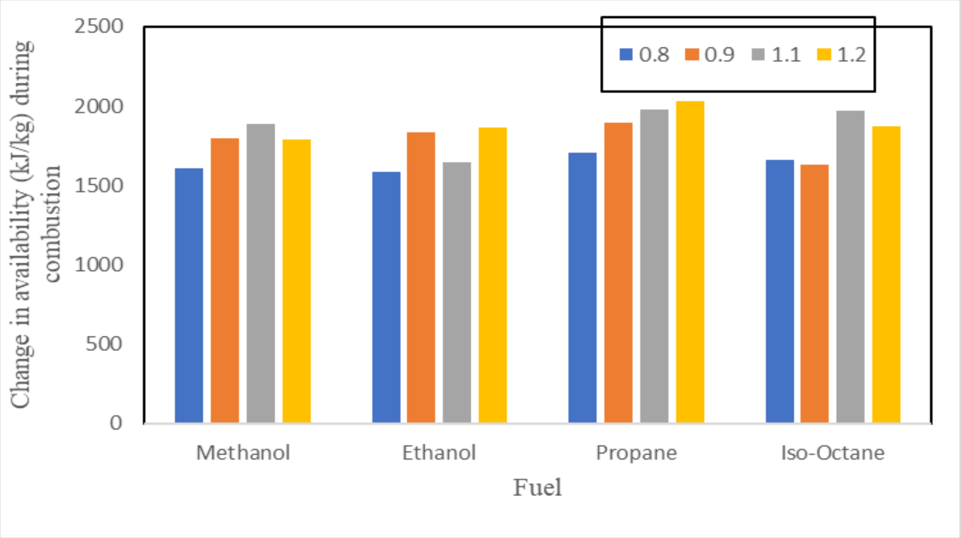

**Figure 5** Exergy destruction during combustion vs fuel with varying equivalent ratio

The energy loss during combustion modifies the methanol and ethanol equivalency ratios, as shown in Figure 5. Because temperature falls up to an equivalency ratio of 0.8 to 1 for all selected fuels, it is depicts that energy destruction in combustion process reduces till stoichiometric condition, which is (Φ = 1).

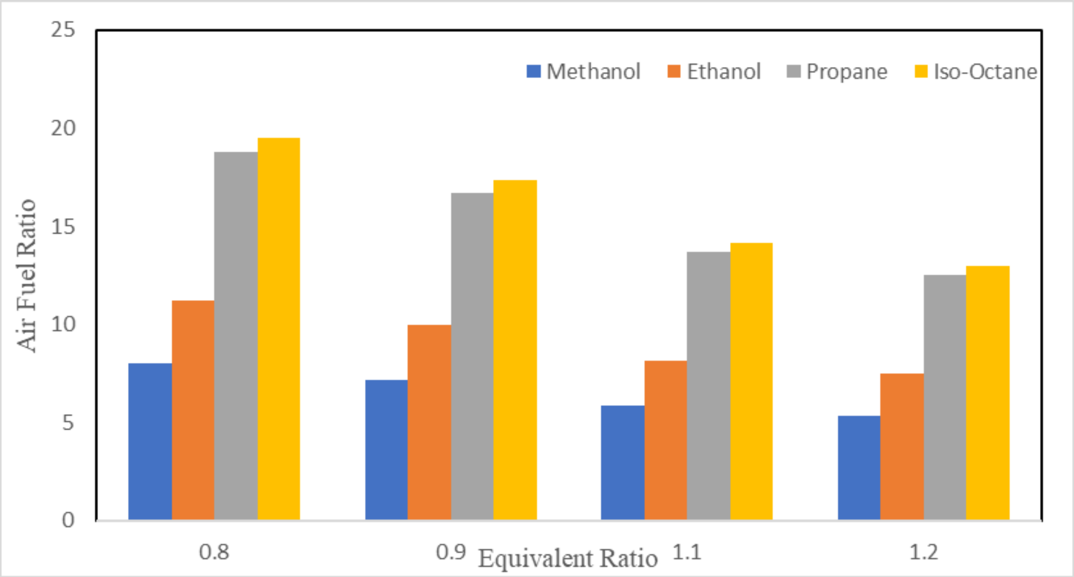

**Figure 6** Air Fuel ratio Vs Equivalent Ratio

Figure 6 explains the Air fuel Ratio vs Equivalence ratio for the all-selected fuels. It is shown that changes in air fuel ratio for the all-selected fuels.

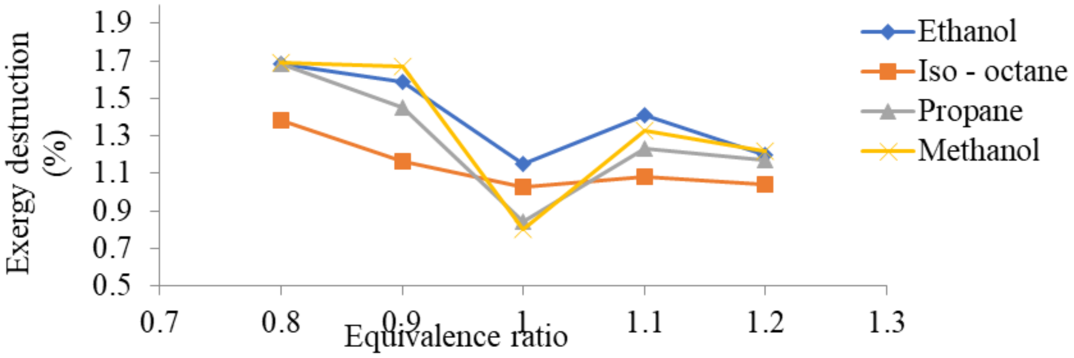

**Figure 7** Percentage Exergy Destruction vs Equivalence Ratio

Exergy Destruction in percentage is shown in **Figure 7.** On changing the Equivalent ration availability of each of the chosen fuel's changes from time to time, It decreases upto equivalent ratio1 after that it increases.

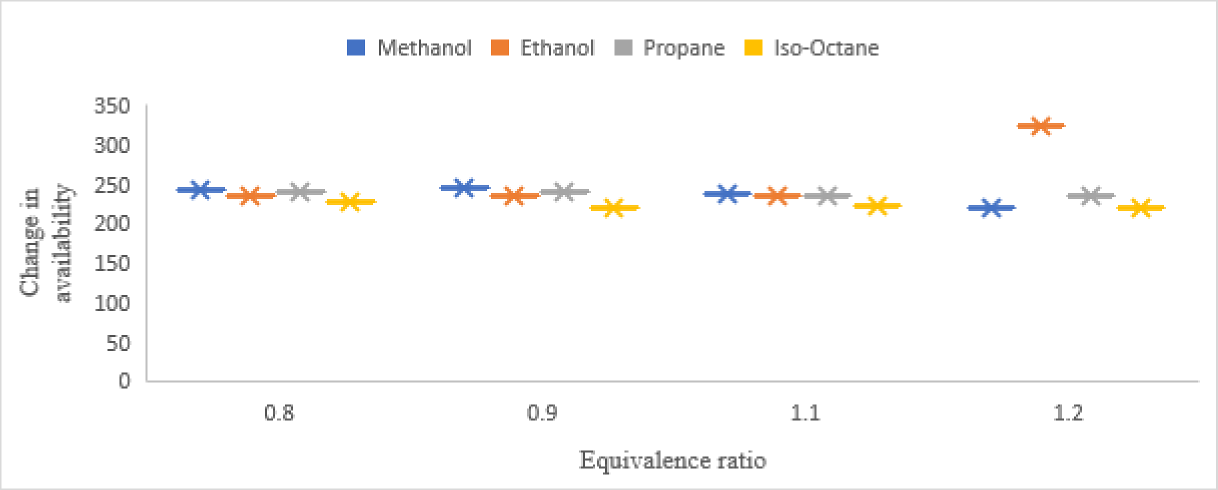

**Figure 8** Change in availability with the change in equivalence ratio

Change in availability with the change in equivalence ratio in the **Figure 8.** It is demonstrated that availability increases for all selected fuels following compression, and this is because work is supplied during this process. Additionally, it is demonstrated that the trends for methanol and ethanol differ.

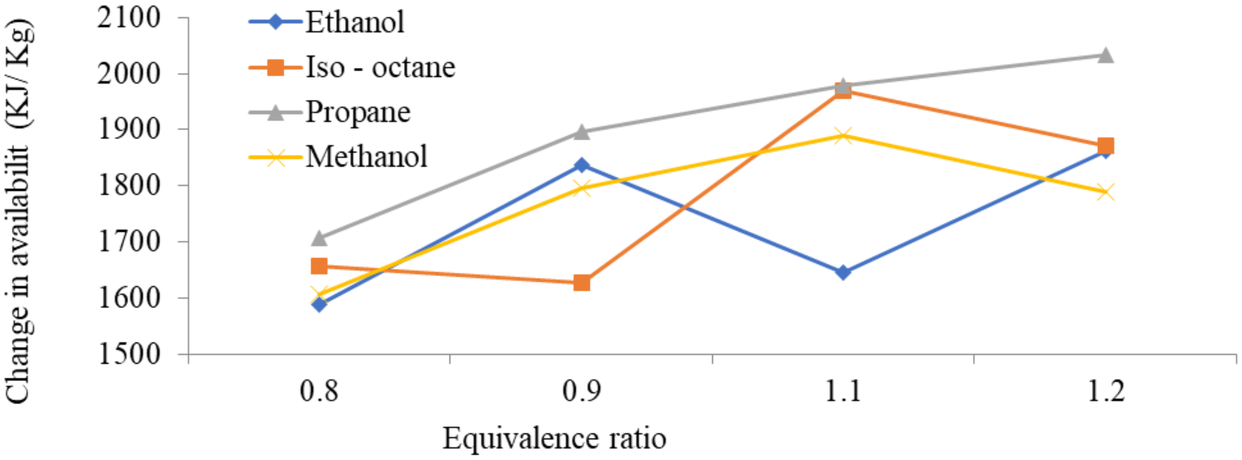

**Figure 9** Change in availability during combustion

Figure 9 illustrates how the equivalency ratio changes in relation to availability during combustion. It has been demonstrated that for propane fuels, availability changes as the equivalency ratio rises. However, the availability of iso-octane and ethanol changes in a wavy pattern during this process, which is because the air fuel ratio and combustion efficiency play a role in the availability changes. The air fuel ratio and combustion efficiency both drop when the equivalency ratio rises. It is additionally demonstrated that under lean conditions (Φ = 0.8 and 0.9), availability changes when the equivalency ratios for propane, methanol, and ethanol grow.

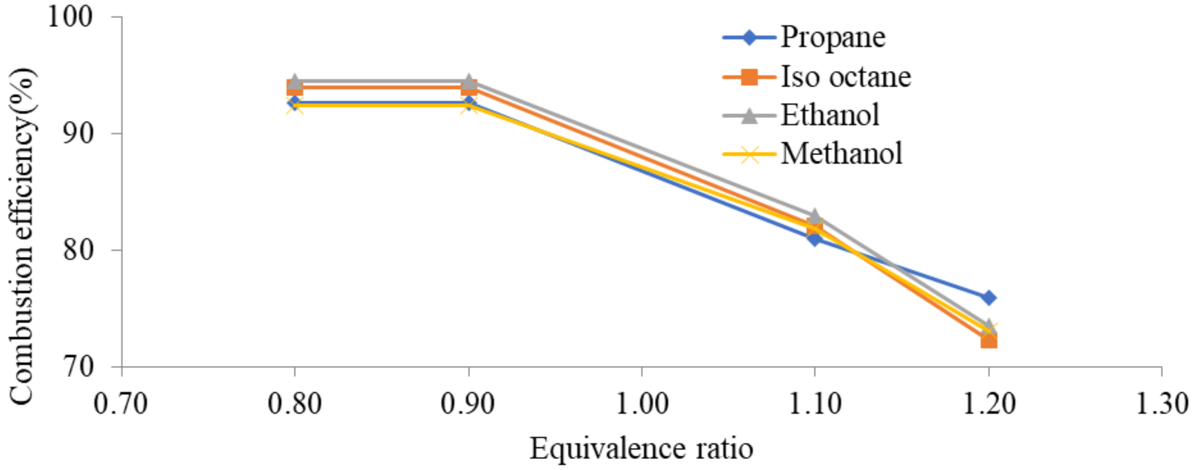

**Figure 10** Combustion efficiency with the change in equivalence ratio

For ethanol, methanol, propane, and iso-octane, **Figure 10** illustrates how the equivalency ratio changes in relation to the variance in combustion efficiency. It is demonstrated that, for all of the fuels that were chosen, combustion efficiency falls as the equivalency ratio rises. This is because, at higher equivalency ratios, fuel does not receive enough oxygen to burn, leaving some fuel unburned. It is also demonstrated that methanol has a minimum combustion efficiency and ethanol a maximum. Furthermore, it is noted that the difference in combustion efficiency is lowest for propane (17.99%) and largest for iso-octane (23.04%).

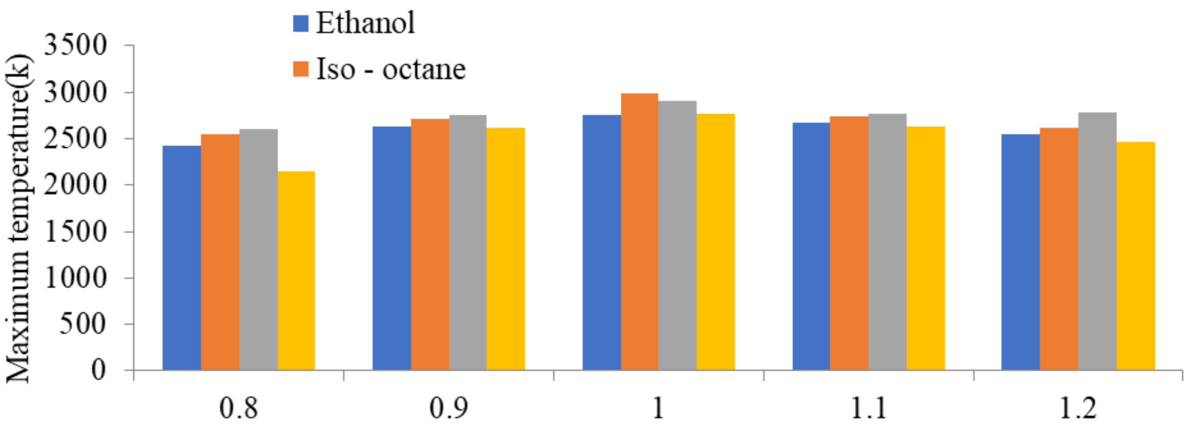

**Figure 11** Maximum temperature in the cycle with the change in equivalence ratio

**Figure 11** illustrates how the temperature changes in relation to the iso-octane, propane, methanol, and ethanol equivalency ratios. The maximum temperature at which the product is found to occur after combustion is found to correspond to an equivalency ratio of one. This is because all fuels burn at stoichiometric conditions, and the maximum temperature for richer fuels decreases because they do not receive enough oxygen to burn. Leaner fuels have enough oxygen to burn, but because they contain less fuel, less heat will be released during combustion, lowering the maximum product temperature above stoichiometric conditions.Additionally, it is noted that the iso-octane in stoichiometric conditions has a maximum temperature.

## 6. Conclusion

By all analysis and calculations, it is concluded that the energy destruction during combustion process is maximum in comparison of compression process. For a constant volume process involving ethanol, methanol, iso-octane, and propane, the impact of the reactant mixture equivalency ratios on the percentage of energy lost through combustion and compression is being assessed with respect to the change in equivalency ratio. The effect of the reactant mixture equivalency ratios on the percentage of energy destroyed by combustion and compression is being evaluated with respect to the change in equivalency ratio for a constant volume process involving propane, ethanol, methanol, and iso-octane. The availability variations during compression and combustion are also examined, in addition to the change in equivalency ratio. Since the combustion efficiency and maximum temperature have an effect on the engine's performance, the impact of the equivalency ratio adjustment is assessed. It is determined that for all of the chosen fuels, the amount of energy destroyed during combustion is minimal at the stoichiometric condition. In the process of burning, ethanol had an energy destruction ranging from 20.8 to 30.1%, iso-octane from 21.2 to 25.05%, propane from 19.54 to 25.1%, and methanol from 14.07 to 29.6%.

The calculations and analysis presented above lead to the conclusion that, in comparison to the compression process, the combustion process destroys the most energy. The effect of the reactant mixture equivalency ratios on the percentage of energy devastated for a constant volume process due to combustion and compression is being evaluated because methanol and ethanol vary with the in-equivalency ratio in different ways. The variation in availability during compression is also analyzed using the shift in the equivalency ratio. The effect of the equivalency ratio change is evaluated because the engine's performance is influenced by both the maximum temperature and combustion efficiency. It is found that at stoichiometric conditions, energy destruction during combustion is negligible for both selected fuels.

## 7. Future Research Scope

It is possible to conduct further research on the two other processes—the expansion and exhaust processes, in particular. After that, the total amount of energy destroyed can be calculated. The energy losses over the course of the full cycle are then estimated. Consequently, we can increase the efficiency of the SI engine or, on the other hand, the efficiency of the entire cycle with the help of the aforementioned computations.

**Nomenclature:**

| | | | |
|---|---|---|---|
| $\eta$ | efficiency | | |
| $\varphi$ | equivalence ratio | | |
| $\gamma$ | ratio of specific heats | | |
| m | mass [kg] | a | air |
| T | absolute temperature [K] | c | combustion |
| s | entropy [kj kg$^{-1}$K$^{-1}$] | f | fuel |
| S | absolute entropy [kj K$^{-1}$] | i | i$^{th}$ species |
| a | availability [kj kg$^{-1}$] | mix | mixture |
| p | pressure [bar] | P | product |
| V | volume [m$^3$] | R | reactant |
| h | specific enthalpy | 0,1,2,3,4 | state points |
| $a_{dest}$ | Availability Destruction | | |
| AFR | Air Fuel Ratio | | |
| $C_p$ | At Constant Pressure Specific Heat | | |
| $C_v$ | At Volume Pressure Specific Heat | | |
| ED | Energy Destruction | | |
| HHV | Higher Heating Value of Fuel | | |
| I | Irreversibility | | |
| LHV | Low Heating value of fuel | | |

| | |
|---|---|
| MR | Molecular Weight |
| mf | The Mass Flow Rate of The Air Fuel Mixture |
| ni | Moles of Working Fluid |
| N | Total Molecular Mass |
| $\eta$ | Efficiency |
| $\eta_c$ | Combustion Efficiency |
| $p_0$ | Initial Pressure |
| Q | Total Heat |
| R | Net Change in Specific Heat |
| $\Delta S$ | Change in Entropy |
| $S_{gen}$ | Entropy Generation |
| $\Delta S_{sys}$ | Change in Entropy of system |
| $T_0$ | Ambient Temperature |
| u | Internal energy |
| $u_0$ | Initial Internal energy |
| v | Volume |
| $v_0$ | Initial Volume |
| W | Work Done by Reversible Engine |

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## BIOGRAPHIES

Santosh Kumar Gupta is a Mechanical Engineering graduate from Vishveshwarya Institute of Engineering & Technology, Post graduate from Kamla Nehru Institute of Technology and pursuing PhD from Madan Mohan Malviya university of Technology. He is working as Head of Department under Technical Education Department with expertise in mechanical engineering specially in Solar Energy and Thermal engg. A dynamic learner and problem-solver, he has experience in 15 years of teaching and 2 years of industrial experience.