

# Exergy Analysis of Ideal Otto Cycle with Oxy-Fuel and Exhaust Gas Recirculation

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## Abstract

Exergy analysis of ideal Otto cycle with oxy fuel combustion for different exhaust gas recirculation percentage (20% to 60%) at fixed compression ratio 9 and air fuel equivalence ratio 1 is carried out. Octane ( $C_8H_{18}$ ) is considered as fuel. Air fuel mixture Otto cycle is taken as reference cycle to compare oxy-fuel with and without exhaust gas recirculation (EGR) Otto cycles. Exergy at all thermodynamic state points are evaluated to calculate exergy efficiency for different Otto cycles and the results are compared. It is found that the exergy efficiency of oxy fuel Otto cycle without EGR is 28% less than exergy efficiency of air fuel Otto cycle. It is also found that EGR improves the exergy efficiency of oxy fuel Otto cycle. Exergy efficiency of 20% EGR Otto cycle is 0.5% higher than the air fuel Otto cycle. Proposed oxy-fuel EGR Otto cycle is suitable for large stationary engines where Oxy-fuel combustion is clean which releases  $CO_2$  and  $H_2O$  as exhaust and  $NO_x$  emission is completely eliminated.

**Keywords:** Exergy analysis, Otto cycle, Oxy-fuel, EGR.

## Introduction

With the increasing automobile population, vehicle pollution becomes major concern. Government and regulating agencies are putting more pressure on automobile manufacturers to reduce the emission levels. On the other side the development of alternate technologies like electric vehicles, hybrid vehicles and alternate fuels are taking place. An Otto cycle is an ideal thermodynamic cycle that describes the functioning of a typical spark ignition engine.

Theoretical work of Noam Lior George J. Rud<sup>4</sup> provides exergy analysis of ideal Otto cycle. In this analysis, different compression ratio and equivalence ratio were used to compare the energetic efficiency.

Jerald A. Caton<sup>3</sup> discuss about the importance and significance of thermodynamic analysis of heat engines in his work with examples. Exergy analysis is very much helpful in identifying the areas where the useful energy is lost<sup>2,5,8</sup>. Exergy analysis of oxy fuel power plant was investigated by Shiyi Chen et. al.<sup>1</sup>. They found exergy efficiency of oxy fuel power plant with chemical looping is higher than the conventional power plants.

Limitation of thermodynamic first law energy analysis is that it will not provide inefficiencies of process/component which are not related to energy loss but to exergy destruction. Exergy analysis can be used as common measure of quality of a process/component along with quantity, and to improve system efficiency. Aim of this theoretical analysis is to find the energy efficiency of ideal Otto cycle with oxy fuel combustion process and EGR. Advantage of oxy fuel combustion is that it will produce only  $H_2O$  and  $CO_2$  during combustion process.  $CO_2$  can be separated from the exhaust gas and can be stored at super critical state which is not covered in this work.

**Exergy Analysis:** Ideal Otto cycle is illustrated in the figure 1 using pressure-volume(PV) and temperature-entropy(TS) diagram. Ideal Otto cycle consists of 5 processes. First process(5→1) is intake stroke. During intake stroke piston moves outwards and air fuel mixture is drawn into cylinder. Second process is compression stroke(1→2). In this stroke, mixture is isentropically compressed as piston moves inwards and inlet valve is closed when this stroke starts. At the end of compression, combustion of mixture(2→3) takes place instantaneously and at constant volume after introduction of spark. After combustion, pressure and temperature of products are increased and the products start to expand which pushes the piston to move outwards(3→4) by producing useful mechanical work. At the end of expansion process, exhaust valve opens and exhaust gases are released into atmosphere(4→5).

Ideal cycle analysis is preliminary step to evaluate the oxy fuel combustion with exhaust gas recirculation(EGR). Further detailed analysis is required. General steps followed to determine the different thermodynamic state points are described below<sup>4</sup>.

Exergy of the incoming fuel/air or oxy fuel mixture can be evaluated using equation 1:

$$E_{mix} = \sum_i n_i (E_0^i + \Delta E_0^i) \quad (1)$$

Here "i" represents individual components in the air fuel mixture.

In ideal Otto cycle, fuel and air mixture are trapped inside a control volume. So, it can be considered as closed system.

Exergy of closed system is:

$$E_{closed\ mix} = E_{mix} - (p - p_0)v \quad (2)$$

$$E_{closed\ mix} = \sum_i n_i [E_0^i + \Delta E_0^i - (p - p_0)v] \quad (3)$$

$$\Delta E_{0 \rightarrow 1} = (H_1 - H_0) - T_0(s_1 - s_0) \quad (4)$$

$$\Delta E_{0 \rightarrow 1} = C_p(T_1 - T_0) - T_0 \left[ C_p \ln \left( \frac{T_1}{T_0} \right) + R \ln \left( \frac{P_0}{P_1} \right) \right] \quad (5)$$

where  $n_i$  is the molar fraction of  $i$ ,  $P_i$  is the partial pressure of  $i$ ,  $E_0^i$  is the standard chemical exergy of  $i$ ,  $\Delta E_{0 \rightarrow 1}$  is the Gibbs free energy of  $i$  at state 1.

Standard chemical exergies of substances are referred from Exergy Method. Technical and Ecological Applications Szargut, J.<sup>6</sup> According to Dalton's law of partial pressure, partial pressures of the reactants and products can be calculated using the following equation:

$$p_i = x_i p \quad (6)$$

Volume of mixture is calculated by assuming ideal gas behavior.

$$v = \frac{RT}{p} \quad (7)$$

**Process 1→2: Isentropic compression.**

It is assumed that the state 1 is at standard atmospheric pressure 1bar and temperature 300k. Pressure, volume and temperature relations are given in the equation 8 and 9.

$$T_1 V_1^{\gamma-1} = T_2 V_2^{\gamma-1} \quad (8)$$

$$p_2 = p_1 \left( \frac{v_1}{v_2} \right)^{\gamma} \left( \frac{T_2}{T_1} \right) \quad (9)$$

Specific heats are sensible to temperature. Specific heats of reactants and products as a function of temperature are given in the table 1<sup>1</sup>.

Specific heat and specific heat ratio of mixture can be found using the following equations:

$$C_{p, mixture} = \sum n_i C_{pi} \quad (10)$$

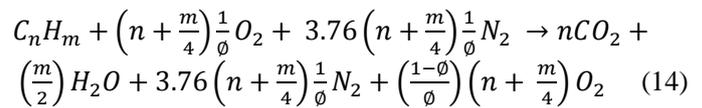
$$C_{v, mixture} = \sum n_i C_{vi} \quad (11)$$

$$\gamma = \frac{C_p}{C_v} \quad (12)$$

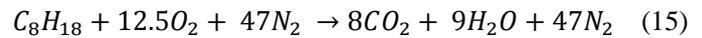
$$\text{For products, } \gamma = \frac{C_p}{C_v} = \frac{0.027438+0.00000827T}{0.019121+0.00000827T} \quad (13)$$

$\gamma$  is calculated using iterative procedure between  $T_1$  and  $T_2$ , Once  $\gamma$  is calculated,  $T_2$  is calculated using equation 8.  $P_2$  can be calculated using equation 9. Now specific heats of reactants, partial pressures and exergies at state 2 can be calculated using the formulae's in table 1 and equations 1, 5 and 6.

**Process 2→3: Combustion process - Constant volume, adiabatic, instantaneous and complete.**



For equivalence ratio = 1,



According to Reynolds and Perkins<sup>4</sup>, enthalpy of reactants and products after combustion process are equal.

$$\hat{H}_{reactants} = \hat{H}_{products} \quad (16)$$

Enthalpies of reactants are calculated using the following equation:

$$h_i = h_{io} + C_{pi}(T - T_0) \quad (17)$$

$$\hat{H}_{reactants} = \sum_i x_i h_i \quad (18)$$

$$\hat{H}_{products} = \left( \frac{\text{mole of } CO_2}{\text{mole of products}} \right) h_{CO_2} + \left( \frac{\text{mole of } H_2O}{\text{mole of products}} \right) h_{H_2O} + \left( \frac{\text{mole of } N_2}{\text{mole of products}} \right) h_{N_2} \quad (19)$$

Enthalpies of  $CO_2$ ,  $H_2O$  and  $N_2$  are substituted as a function of temperature. Equating equations 19 and 20 and solving the same will give temperature of combustion.

$$p_3 = p_2 \left( \frac{T_3}{T_2} \right)^{\left( \frac{N_{products}}{N_{reactants}} \right)} \quad (20)$$

After calculating temperature and pressure at state 3, specific heats of products, partial pressures and exergies at state 3 can be calculated using the formulae's in table 1 and equations 1, 5 and 6.

**Process 3→4: Isentropic expansion.**

Work is produced during the expansion process. Pressure, volume and temperature relations for isentropic expansion are given in the equations 21 and 22.

$$T_3 V_3^{\gamma-1} = T_4 V_4^{\gamma-1} \quad (21)$$

$$p_4 = p_3 \left( \frac{v_3}{v_4} \right)^{\gamma} \left( \frac{T_4}{T_3} \right) \quad (22)$$

Specific heats, partial pressures and exergies at state 4 can be calculated using the similar process used in process 1→2. It is assumed that no work is done by exhaust gas and it is released to environment. Exergies at all state points are evaluated. Exergy at state 1 is the exergy possessed by the mixture. During compression process, exergy is added to the mixture. Exergy difference between state 2 and state 1 ( $E_2 - E_1$ ) is the amount of exergy added during compression. Exergy difference between states 3 and 2 ( $E_3 - E_2$ ) is the exergy lost during combustion. During expansion process exergy is extracted in the form of work. Exergy extracted

during expansion is difference between  $E_3$  and  $E_4$ . Some amount of exergy is lost in the exhaust( $E_4$ ).

Exergy efficiency is the ratio of net exergy extracted for useful purpose to the decrease in input fuel exergy. The exergy efficiency can be calculated from the following equation:

$$\text{Exergy Efficiency, } \eta_E = \frac{\text{Net exergy extracted}}{\text{Decrease in input fuel exergy}} \quad (23)$$

Exergy required to separate  $O_2$  from atmospheric air is calculated by assuming air contains 21% of oxygen and 79% of nitrogen by volume. Minimum exergy required for separation process is given in the equation 24:

$$E_{min} = RT_O \sum_i y_i \ln(y_i) \quad (24)$$

Here  $y_i$  is the ratio of mole of substance to the total mole of all substance. Substituting temperature, gas constant and molar ratios in the equation gives the minimum exergy required for separation process.

$$E_{min} = 1.274 \frac{\text{kJ}}{\text{mol}} \text{ of air}$$

$$E_{min} = 1.274 \frac{\text{kJ}}{\text{mol of air}} \times \text{mol of air required per mol of fuel} \quad (25)$$

$$E_{min} = 75.8 \frac{\text{kJ}}{\text{mol of fuel}}$$

Minimum exergy of 75.8 kJ is required to separate oxygen from air for one mole of fuel. In this analysis, air fuel mixture Otto cycle is taken as case 1, Oxy-fuel without EGR Otto cycle is considered as case 2. Totally 9 cases as shown in the

table 2 are evaluated in this study. For each case, combustion equation is modified and exergy at all thermodynamic state points are calculated as described above.

### Results and Discussion

Exergy at all state points for air fuel ideal Otto cycle, Oxy fuel ideal Otto cycle and Oxy fuel ideal Otto cycle with EGR varying from 25% to 60% are calculated and Exergy efficiencies are calculated for all the cases. Exergy required for separation of  $O_2$  from air, exergy at all state points, combustion temperature for all cases are summarized in the table 3. Exergy flows of case 1, 2, 6 and 7 are shown in the figures 2, 3 and 4.

Exergy added during compression process increases with EGR, this is because of the mass increase due to EGR. Exergy lost during combustion for air fuel Otto cycle is 2.02MJ/mol of fuel and for oxy fuel Otto cycle is 2.879MJ/mol of fuel. Exergy lost during combustion decreases with EGR percentage. Theoretical combustion temperature for oxy fuel combustion is 200% of air fuel combustion. When EGR increases combustion temperature decreases.

Exergy efficiency of air fuel Otto cycle is 40.07% and for oxy-fuel Otto cycle is 28.82%. When EGR is used exergy efficiency is increased. At 20% EGR oxy-fuel Otto cycle, exergy efficiency is increased by 0.5% when compared to oxy-fuel Otto cycle. When EGR percentage is increased, exergy efficiency decreases slightly as shown in the figure 5. Theoretical combustion temperature of all cases is shown in figure 6. Combustion temperature decreases with increase in EGR percentage. At 60% EGR, theoretical combustion temperature is 7.2% higher than the theoretical combustion temperature air fuel cycle i.e. case 1.

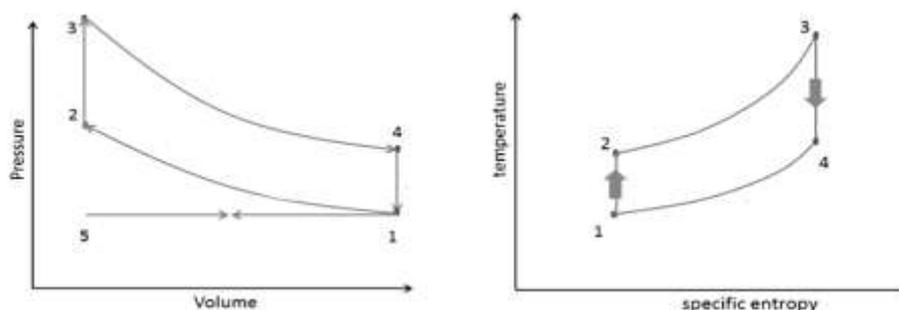


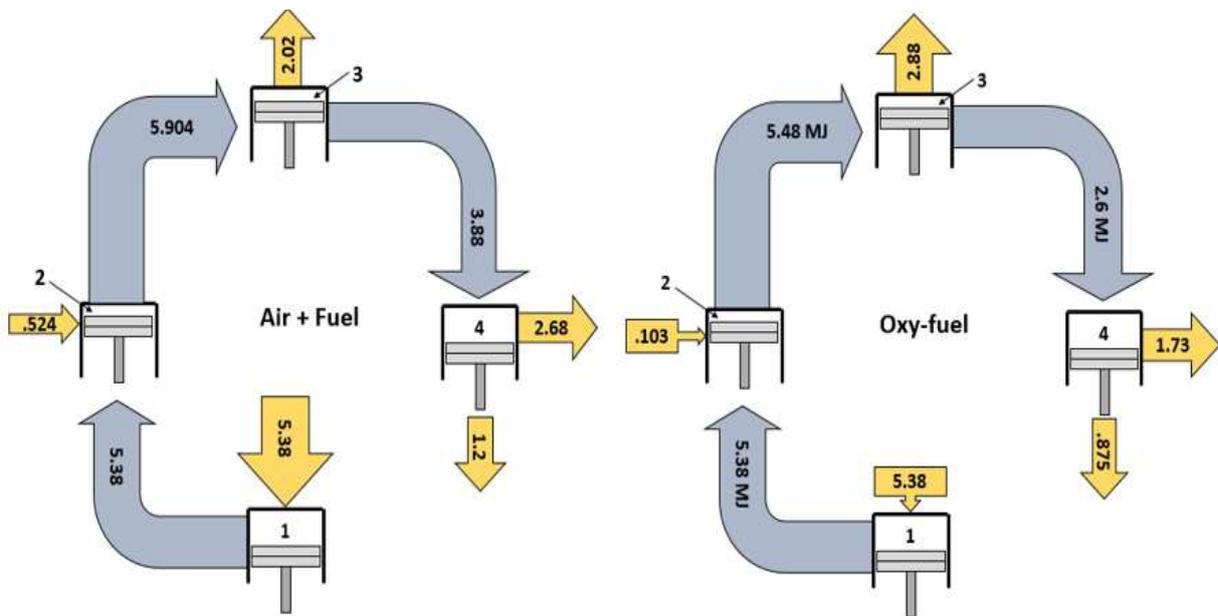
Figure 1: PV and TS diagram of Ideal Otto cycle

Table 1  
Specific heats of substances

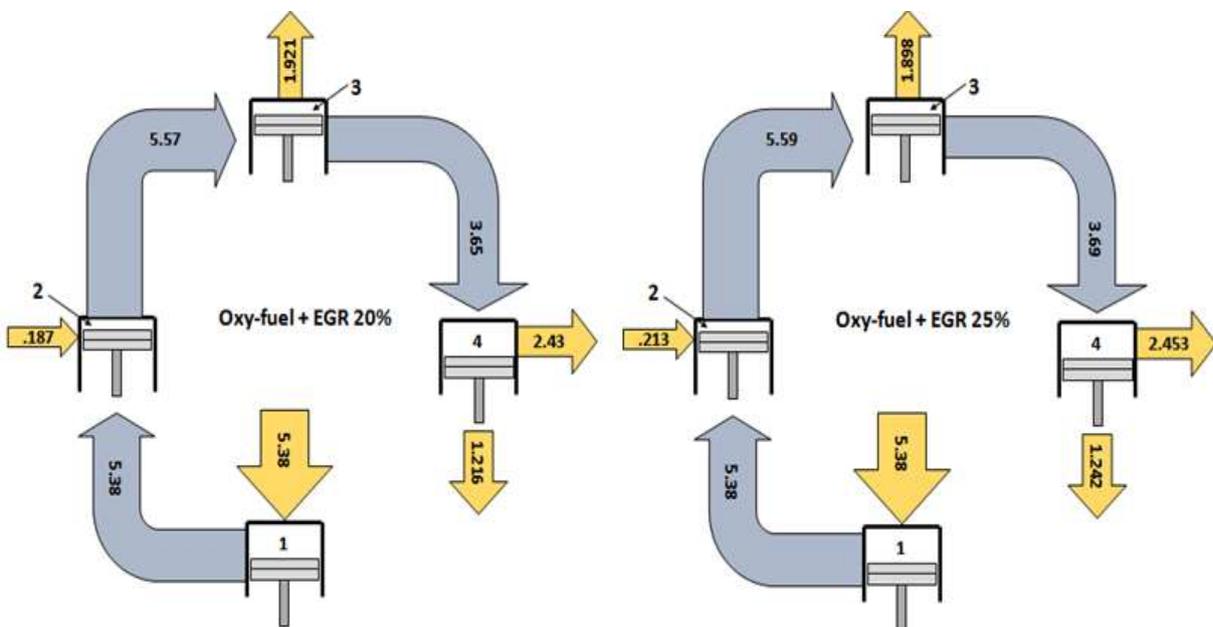
Substances	$C_p$ , KJ/g.mole.K	$C_v$ , KJ/g.mole.K
$C_8H_{18}$	$0.0253 + 0.000162T$	$0.01698 + 0.000162T$
$O_2$	$0.027 + 0.0000079T$	$0.01868 + 0.0000079T$
$N_2$	$0.0276 + 0.0000051T$	$0.01928 + 0.0000051T$
$CO_2$	$0.0274 + 0.0000058T$	$0.01908 + 0.0000058T$
$H_2O$	$0.0305 + 0.0000103T$	$0.02218 + 0.0000103T$
$N_2$	$0.0276 + 0.0000051T$	$0.01928 + 0.0000051T$

**Table 2**  
**Different Cases**

Case	Inlet Mixture	EGR %
Case 1	Air-fuel	0
Case 2	Oxy-fuel	0
Case 3	Oxy-fuel + EGR	20
Case 4	Oxy-fuel + EGR	25
Case 5	Oxy-fuel + EGR	30
Case 6	Oxy-fuel + EGR	35
Case 7	Oxy-fuel + EGR	40
Case 8	Oxy-fuel + EGR	50
Case 9	Oxy-fuel + EGR	60



**Figure 2: Exergy flow – Case 1&2**



**Figure 3: Exergy flow – Case 3 &4**

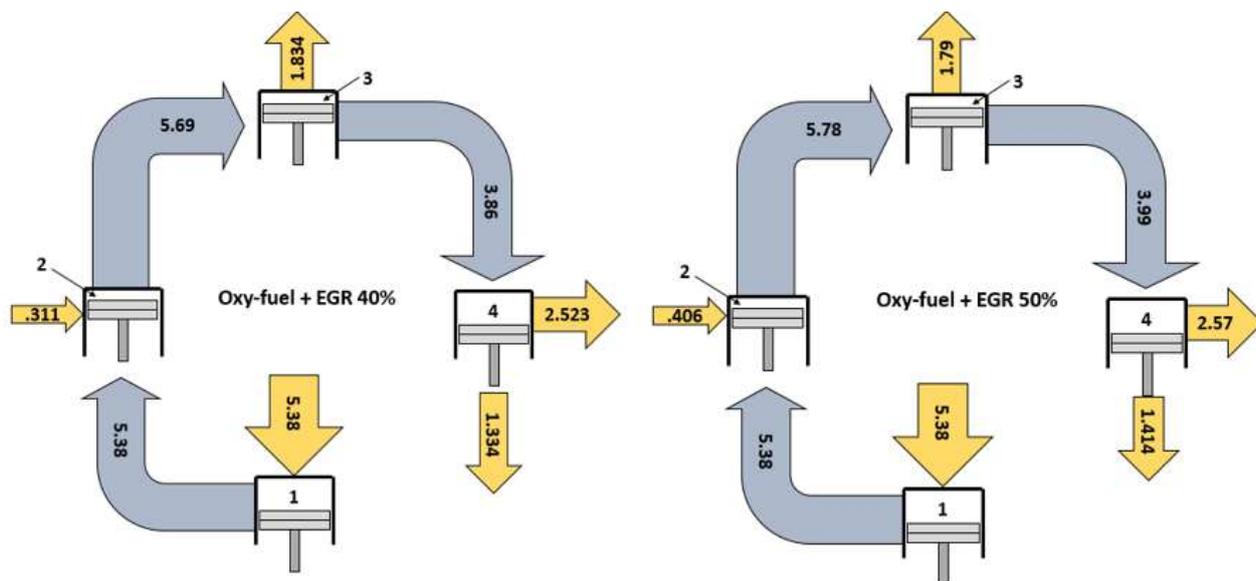


Figure 4: Exergy flow – Case 6 &7

Table 3  
Exergy flow

Case	Exergy, MJ/mol of fuel						Theoretical combustion temperature, k
	Exergy of separation	Fuel air mixture	Added during compression	Lost during combustion	Extracted during expansion	Lost in exhaust	
Case 1	-	5.38	0.524	2.020	2.680	1.205	2398.5
Case 2	75.803	5.38	0.103	2.879	1.729	0.875	4802.2
Case 3	75.803	5.38	0.187	1.921	2.430	1.216	4208.2
Case 4	75.803	5.38	0.213	1.898	2.453	1.242	4049.7
Case 5	75.803	5.38	0.247	1.875	2.479	1.273	3866.0
Case 6	75.803	5.38	0.275	1.855	2.500	1.300	3719.7
Case 7	75.803	5.38	0.312	1.834	2.524	1.334	3545.4
Case 8	75.803	5.38	0.406	1.798	2.574	1.414	3181.5
Case 9	75.803	5.38	0.542	1.766	2.633	1.524	2786.4

Table 4  
Exergy Efficiency

Case	Inlet Mixture	EGR %	Exergy Efficiency, %
Case 1	Air-fuel	0	40.07
Case 2	Oxy-fuel	0	28.82
Case 3	Oxy-fuel + EGR	20	40.28
Case 4	Oxy-fuel + EGR	25	40.23
Case 5	Oxy-fuel + EGR	30	40.07
Case 6	Oxy-fuel + EGR	35	39.91
Case 7	Oxy-fuel + EGR	40	39.70
Case 8	Oxy-fuel + EGR	50	38.90
Case 9	Oxy-fuel + EGR	60	37.45

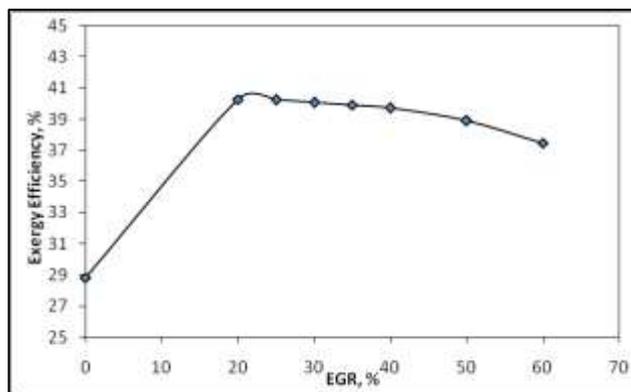


Figure 5: Theoretical combustion temperature of different cases

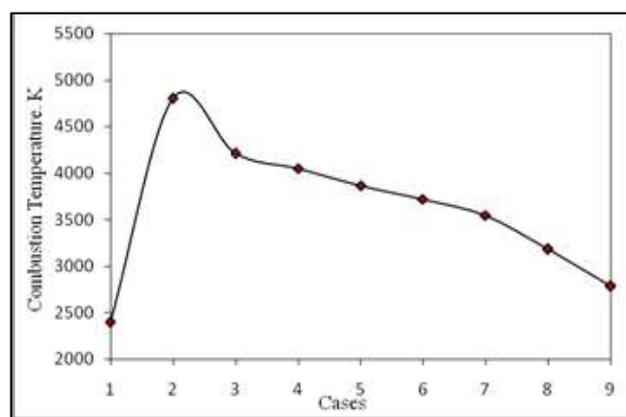


Figure 6: EGR % vs Exergy efficiency

## Conclusion

Exergy analysis of ideal Otto cycle with oxy fuel combustion and EGR (20% to 60%) at fixed compression ratio 9 and air fuel equivalence ratio 1 is carried out. From this theoretical analysis, it is found oxy fuel combustion with EGR is a better alternate method for large stationary air fuel combustion engines. Exergy efficiency of oxy fuel with EGR cycle is comparable with air fuel cycle. Exergy efficiency of 20% EGR cycle is 0.5% higher than the air fuel cycle. But the theoretical combustion temperature is high when EGR percentage is low. At 20% and 60% EGR, theoretical combustion temperature is 75.4% and 7.2% higher than the theoretical combustion temperature of air fuel cycle respectively.

When EGR percentage increases, combustion temperature drops significantly. Oxy-fuel combustion is clean which releases  $\text{CO}_2$  and  $\text{H}_2\text{O}$  as exhaust and  $\text{NO}_x$  emission is completely eliminated.  $\text{CO}_2$  can be separated and stored. This method is more suitable for large stationary engines. Further investigation is necessary to evaluate the exergy required for separating  $\text{CO}_2$  from exhaust and compressing it to super critical state to store it.

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