# IC Engine Term Paper


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## Topic

To study power, HRR, HR, emission gases etc. for a CI Engine using Methanol-Diesel as fuel at two different loads ($0$ and $30Nm$).

## Abstract


The generation and utilization of coming-up age fuels for vehicle consumption is highly encouraged due to rising pollution and depleting availability of petroleum and like fuels. And among these non-conventional fuels oxygenated fuels are gaining popularity and can be used as a substitution for diesel based compression ignition engines. Hence, an experimental research was conducted to determine the impact of various types of alcohols on combustion phenomenon, heat release rate, peak value cylinder pressure, combustion phasing, pressure trace and combustion duration which defines the combustion related characteristics of the fuel. Moreover, research has been conducted on controlled emission. A Twin cylinder CI engine was utilized to perform tests with diesel, M10 (10% oxygenated fuel m/m respectively). This test was done at a speed of 2000 rpm under three distinct loading situations (0 Nm and 30 Nm). Lower load circumstances lead to longer ignition delay (compare to diesel), high load conditions resulted in faster combustion and similar centroid CA50 (as observed in diesel) for oxygenated fuel. The presence of oxygen in the fuel contributes to its high peak pressure and facilitates fast burning.



**Keywords:** Combustion phasing, oxygen percentage variation, methanol fuel, in-cylinder pressure


## Introduction

The diesel based engines are extremely used in the heavy duty construction and transportation works because for theses certain working condition s they are the most efficient, reliable and can last long. But on the contrary these engines are well known for their particulate matter

(PM) and NOx emissions. On a positive side alternative liquid fuels such as biodiesels, alcohol, etc. have been able to control some of these emissions more specifically the particulate matter (PM). These oxygenated liquid fuels can be produced by bio-resources hence increasing the sustainability. Among the oxygenated fuels Methanol is an extensively studied alternative diesel alternative for internal combustion (IC) engines keeping its easy processing, availability and low cost production in eyes. Methanol has high heat of vaporization and elevated inherent oxygen content 50% (m/m), hence it is believed that it will be useful in minimalizing NOx and PM[1][2]. Blending of methanol with mineral diesel is one of the common methods for utilization in diesel engines. In some cases a variety of additives can be required to stabilize the blend and to improve the ignition quality.[3][4] Several prior researchers have looked into the usage of methanol as a diesel blend or applying methanol in IC engine with fumigation techniques. An experimental investigation to discover the impact of varying fuel injection pressure in a common rail CI engine running on dual fuel (diesel and methanol).[3] A report was prepared which mentioned NOx emissions were increased and the opacity decreased with surge in rail pressure for dual fuel experiments, but dual fuel method reduced NOx (by 23.14%) and emissions of HC significantly (49.11%).[3] In a different study using methanol-diesel blends, it was observed that by advancing injection timing (25 CAD BTDC) reduced smoke (up to 22%), HC (50%) and CO emissions (up to 52%) while increased the emission of NOx (up to 69%).[5] Injecting methanol in intake manifold resulted in charge cooling (evaporation of methanol occurred due to the exchange of energy with the charge) and decrease in the exhaust gas temperature in IC engine in the experiments performed by.[6] The impacts of air inlet temperature and fuel injection timing were investigated by using a customized diesel engine.[4] Higher air-intake temperature (35 °C to 115°C) and advanced injection time (17.4 BTDC to 4.6 ATDC.) have shown to be beneficial in improving thermal efficiency (higher by 7.3%).[4] It was tested that methanol for partially premixed method of combustion in a high-power engine and perceived that varying injection pressure had no discernible impact on PM emissions. However, increasing the intake increase specific PN emissions for methanol fuel. Methanol was applied in a reactivity-controlled compression ignition and found that methanol delayed start of combustion with rising premixing ratio and delivered more consistent combustion. Neat methanol was injected with port and methanol blended with ethyl hexyl nitrate and di-tert-butyl peroxide through direct injection in the dual fuel operation. It was obtained that injector settings for direct injection had little effect on combustion phasing since methanol direct injection caused significant charge cooling. It also mitigated the increased reactivity caused by above mentioned cetane improvers. Turbocharging, port injection engine with internal, high-pressure loop and dedicated exhaust gas recirculation setups for neat methanol and high-octane fuel were tested. It was found that dedicated recirculation was evidenced to be the most beneficial and increased the engine's maximum load by 2 bar BMEP. It has been observed that methanol has been extensively investigated in terms of varying blend ratio and by adapting 2 different injection strategies. The present study has been performed by keeping the oxygen content (m/m) in the blends of methanol and mineral diesel. The effect of varying mass fraction of oxygen has been investigated for combustion parameters such as in-cylinder pressure, heat release rate, combustion duration, combustion phasing etc. The emission characterization was also performed for gaseous emission like CO, HC and NOx along with smoke measurements

# Mathematical work

### Engine Specification

| | |
|---|---|
| Make | **Super carry CI engine** |
| Engine type | 4 stroke |
| No. of cylinders | 2 |
| Compression ratio, $CR$ | 15:1 |
| Cylinder bore (mm), $d$ | 77.0 |
| Stroke length (mm), $\Delta x$ | 85.1 |
| Connecting rod length, $l$ | 138.0 |
| adiabatic constant, $\gamma$ | $1.36 - 1.38$ |

### Variation of adiabatic constant of air with respect to temperature (source: link)

| Temperature, $K$ | $C_P$, $(kJ/kg.K)$ | $C_v$ , $kJ/kg.K$ | $k$ |
|---|---|---|---|
| 250 | 1.003 | 0.716 | 1.401 |
| 300 | 1.005 | 0.718 | 1.400 |
| 350 | 1.008 | 0.721 | 1.398 |
| 400 | 1.013 | 0.726 | 1.395 |
| 450 | 1.020 | 0.733 | 1.391 |
| 500 | 1.029 | 0.742 | 1.387 |
| 550 | 1.040 | 0.753 | 1.381 |
| 600 | 1.051 | 0.764 | 1.376 |
| 650 | 1.063 | 0.776 | 1.370 |
| 700 | 1.075 | 0.788 | 1.364 |

| | | | |
|---|---|---|---|
| 750 | 1.087 | 0.800 | 1.359 |
| 800 | 1.099 | 0.812 | 1.354 |
| 900 | 1.121 | 0.834 | 1.344 |
| 1000 | 1.142 | 0.855 | 1.336 |
| 1100 | 1.155 | 0.868 | 1.331 |
| 1200 | 1.173 | 0.886 | 1.324 |
| 1300 | 1.190 | 0.903 | 1.318 |
| 1400 | 1.204 | 0.917 | 1.313 |
| 1500 | 1.216 | 0.929 | 1.309 |

Considering the variation of temperature inside the engine, value of adiabatic constant to be taken for the analysis is $1.36$ to $1.38$.

### Equations and constants to be fed to the Excel sheet

- Swept volume, $V_s$

$$V_s = \frac{\pi d^2}{4}\Delta x$$

$$V_s = \frac{\pi 0.077^2}{4}0.0851$$

$$V_s = 0.000396m^3 = 396cc$$

- Clearance volume, $V_c$

$$CR = \frac{V_s + V_c}{V_c}$$

$$V_c = \frac{V_s}{CR - 1}$$

$$V_c = \frac{0.000396}{15-1} = 2.83 \times 10^{-5}m^3 = 28cc$$

- The ratio, $R = \frac{2l}{\Delta x}$

$$R = \frac{2*l}{\Delta x}$$

$$R = \frac{2*138}{85.1} = 3.24$$

- Volume as a function of $\theta$:

$$V(\theta) = \frac{V_s}{CR-1} + \frac{V_s}{2}(R + 1 - cos\theta - \sqrt{R^2 - sin^2\theta})$$

$$V(\theta) = 0.0000283 + 0.000198(4.24 - cos\theta - \sqrt{10.5 - sin^2\theta})$$

- rate of change of volume with respect to the crank angle:

$$\frac{dV}{d\theta} = \frac{V_s}{2}sin\theta\left(1 + \frac{cos\theta}{\sqrt{R^2 - sin^2\theta}}\right)$$

$$\frac{dV}{d\theta} = \frac{0.000396}{2}sin\theta\left(1 + \frac{cos\theta}{\sqrt{3.24^2 - sin^2\theta}}\right)$$

$$\frac{dV}{d\theta} = 0.000198 \times sin\theta\left(1 + \frac{cos\theta}{\sqrt{10.5 - sin^2\theta}}\right)$$

- Heat release rate equation:

$$HRR = \frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma - 1}.P.\frac{dV}{d\theta} + \frac{1}{\gamma - 1}.V.\frac{dP}{d\theta}$$

$$HRR = \frac{dQ_{net}}{d\theta} = \frac{1.37}{1.37 - 1}.P.\frac{dV}{d\theta} + \frac{1}{1.37 - 1}.V.\frac{dP}{d\theta}$$

$$HRR = \frac{dQ_{net}}{d\theta} = 3.7P.\frac{dV}{d\theta} + 2.7V.\frac{dP}{d\theta}$$

### Determining calorific value of M10-diesel fuel

The fuel used in the experiment is M10 diesel fuel, i.e. diesel fuel mixed with methanol such that the mixture contains 10% oxygen.

| Properties | Diesel | Methanol | M10 fuel |
|---|---|---|---|
| Density g/liter | 822 | 790 | |
| C.V. MJ/kg | 45.6 | 20.27 | |
| O2 % m/m | 0% | 50% | 10 |

Thus, methanol $w/w\%$ in the final blend is $20\%$. Now, the calorific value of the M10, $C.V_{M10}$ is

$$C.V_{M10} = 0.8 * 45.6 + 0.2 * 20.27$$

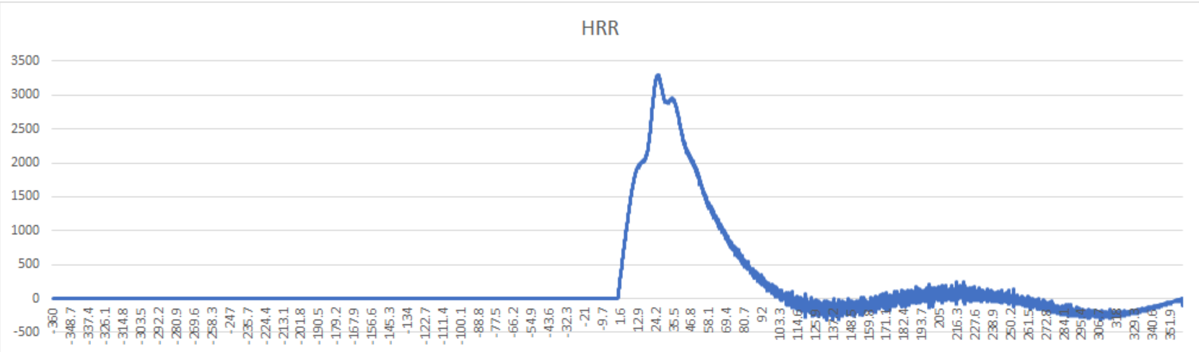
$$C.V_{M10} = 40.534\ MJ/kg$$

Using this calorific value we can determine, fuel consumption rate as a function of crank angle.

# Results

## Heat Release Rate, HRR $(J/s)$

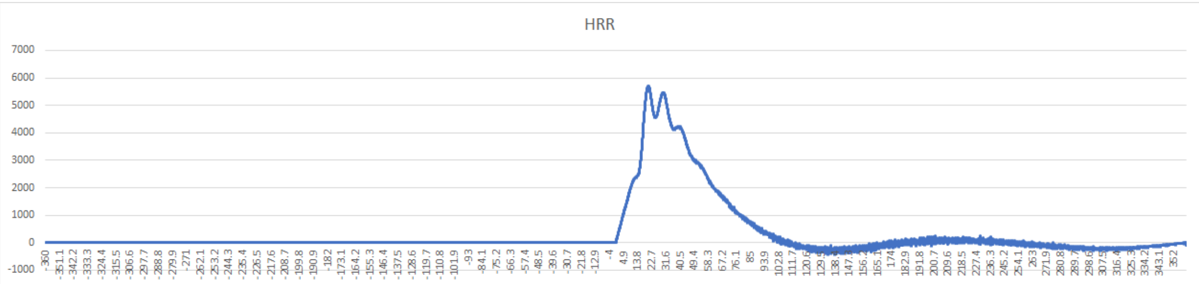
### 0Nm load




Heat release rate profile for 0Nm load

- Maxima of Heat release rate for 0Nm is $3291.726$ which occurs at $25.8$ aTDC.

### 30Nm load

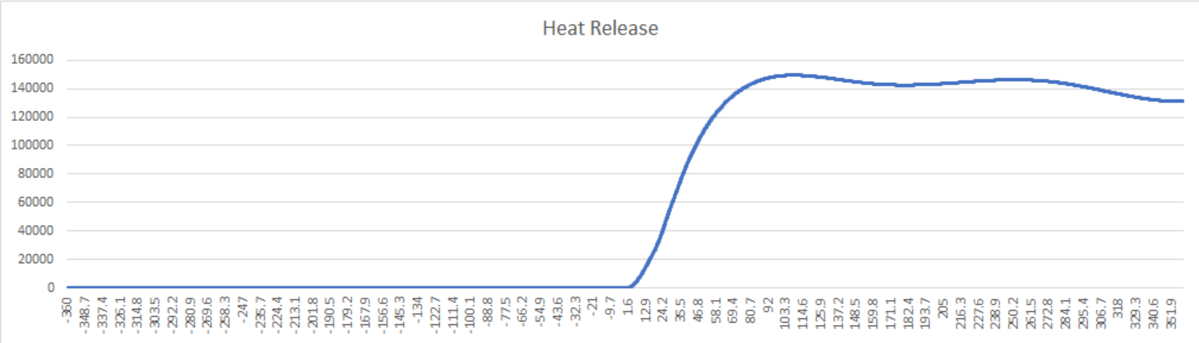



Heat release rate profile for 30Nm load

- Maxima of Heat release rate for $30$Nm is $5712.65$ which occurs at $20.5$ aTDC.

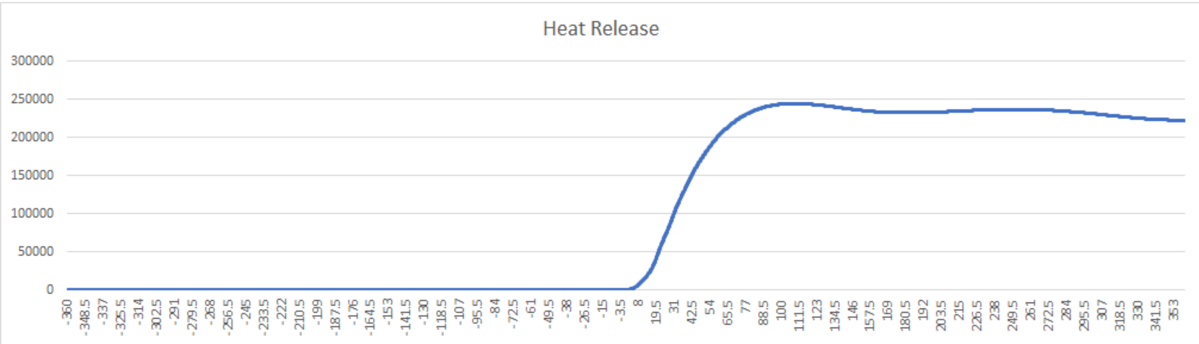
## Heat Released upto $\theta$ crank angle $(J)$

### 0Nm load




Total Heat released in one thermodynamic cycle = max value of HR = $149448.19J$

### 30Nm load




Total Heat released in one thermodynamic cycle = max value of HR = $244286.13J$

## CA5, CA50, CA90

### 0Nm load

- Crank angle upto which 5% of heat (=0.05*149448.19=7,472.40J) has been released: $8.9$ aTDC.
- Crank angle upto which 50% of heat (=0.5*149448.19=74,724.095J) has been released: $35.3$ aTDC.
- Crank angle upto which 90% of heat (=0.9*149448.19=1,34,503.371J) has been released: $68.7$ aTDC.

### 30Nm load

- Crank angle upto which 5% of heat (=0.05*244286.13=$12,214.3J$) has been released: $10.7$ aTDC.
- Crank angle upto which 50% of heat (=0.5*244286.13=$122,143.065J$) has been released: $35.7$ aTDC.
- Crank angle upto which 90% of heat (=0.9*244286.13=$219,857.51J$) has been released: $69.4$ aTDC.

## Fuel Flow rate

- Amount of fuel burnt per unit time (unit: g/s).
- The revolution speed of the engine is 1600rpm, which implies time taken for two revolution (i.e. one thermodynamic cycle) is $0.075s$.

### 0Nm load

In one thermodynamic cycle, we saw that the amount of heat released is $149448.19J$. Now, since the calorific value of the given fuel is $40.534\ MJ/kg$, the amount of fuel consumed during one thermodynamic cycle is $\frac{149448.19}{40.534\times10^6}kg = 3.69g$.

Thus, the fuel flow rate becomes,

$$f.f.r = \frac{3.69}{0.075} g/s$$

$$f.f.r = 49.2 g/s$$

### 30Nm load

The amount of heat released in one thermodynamic cycle is $244286.13J$. Now, since the calorific value of the given fuel is $40.534\ MJ/kg$, the amount of fuel consumed during one thermodynamic cycle is $\frac{244286.13}{40.534 \times 10^6} kg = 6.0267g$.

Thus, the fuel flow rate becomes,

$$f.f.r = \frac{6.0267}{0.075} g/s$$

$$f.f.r = 86.096 g/s$$

## Brake Specific Fuel Consumption (bsfc) for 30Nm load

Brake specific fuel consumption rate is defined as the ratio of fuel consumption per unit time and the brake power delivered.

Brake work done in 1 thermodynamic cycle is = $30Nm \times 2 \times 2\pi = 377J$

And the amount of fuel consumed in one thermodynamic cycle is $6.0267g$.

If the time duration for one thermodynamic cycle is $T$,

$$f.f.r = \frac{6.0267}{T}$$

and the brake power of the engine is

$$bp = \frac{377}{T}$$

Thus, bsfc can be calculate as

$$bsfc = \frac{f.f.r}{bp}$$

$$bsfc = \frac{6.0267/T}{377/T}$$

$$bsfc = \frac{6.0267}{377}$$

$$bsfc = \frac{6.0267}{377} = 0.016 \frac{g}{W}$$

# Analyzing Emission gases

The fuel used in our experiment is diesel mixed with a small fraction of methanol. Now, methanol contains 50% oxygen by weight. Thus, the addition of methanol significantly increases the oxygen content in the combustion cylinder and thus reduces the amount of unburnt by-products such as CO (which in turn increases the amount of $CO_2$ released). CO content in the emission, for example, reduces from an average of 0.73% to 0.03% when 10% Methanol was used with diesel (for 0Nm load) keeping other conditions similar. This reduction is because of better combustion of fuel in the case of M10-diesel. Similar observation can be done for other gases such as HC, NO etc.

# Conclusion

This study has varied the oxygen percentage in the total fuel (m/m) by blending of methanol in diesel fuel in order to investigate the effects of this variation on combustion and emission characteristics. Following observations have been drawn from this study. Diesel shows higher peak of heat release rate in premixed combustion phase, however both methanol blend M10shows higher rate on mixing controlled combustion zone, this indicated the more uniform combustion in No load. For medium and higher loading condition M10 shows higher premixed and mixing controlled heat release rate compare to diesel. Major effects of methanol blending can be seen in the smoke emission, where it has minimized to negligible level at no load and reduced by more than 75% at medium and high load. Higher NO was observed for low and medium load condition and slightly lower NO was observed at higher load condition for 10% oxygen condition.

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