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Paramteric Analysis of a Gasoline Fueled Premixed Spark-Ignition (SI) Race Car Engine by using Methanol and Methanol-Water Blends

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***Abstract*— Numerical simulations were performed in a port injected spark ignition (SI) race car engine with pentroof geometry. The study was performed for a premixed case with one-step global reaction mechanisms for gasoline, gasoline-methanol and gasoline-methanol-water blends of 1.8%, 3.6% and 5.4% at stoichiometric conditions. The purpose of the study was to analyze the combustion and thermal efficiency as well as the indicated power and emissions with and without methanol/water enrichment. The model was tested using a numerical simulation code that solves compressible, turbulent, three-dimensional transient equations. These equations apply to reacting multicomponent gas mixtures with flow dynamics of an evaporating liquid spray. The simulations performed provided a comparative analysis between the gasoline, the gasoline-methanol and the gasoline-methanol-water global mechanisms. The engine geometry used in this study was ø101.6 mm bore and 88.4 mm stroke, running at 6500 rpm. Earlier joint computational experimental studies performed by the researchers have shown that injection of a secondary fuel in small quantities in conjunction with the base fuel could lead to marked combustion process improvements and thermal efficiency. The secondary fuel had only a small contribution to the total engine heat release, but, it improved engine efficiency by increasing flame speed and ensuring a more complete combustion process for the base fuel. Secondary fuel enrichment in a small amount to the air-fuel charge results in efficient engine operation with lean air-fuel mixture. Results showed aggressive burning of gasoline while using methanol and methanol-water blends. Addition of methanol encouraged complete burn of the fuel. Increase in indicated power, and thermal efficiency were observed with an increase in methanol percentage. The trend was visible during all methanol enrichments. With the increase in the methanol concentration, the average temperature and maximum temperature inside the cylinder slightly decreases. This is significant because as the percentage of methanol increases in the fuel, the result is higher efficiencies, complete combustion, and slightly lower temperatures inside the cylinder with fewer emissions.**

Keywords*— *IC Engine; KIVA-3V; Spark Ignition (SI); Race Car; pentroof geometry; combustion efficiency; thermal efficiency; indicated power; NOx emissions; reaction mechanism; gasoline-methanol; gasoline-methanol-water.

I. INTRODUCTION

Understanding the thermo-chemical phenomena involved in spark-ignition combustion is quite challenging. Considerable insight into the in-cylinder combustion dynamics can be achieved through computational simulations in conjunction with either collateral experiments or in comparison with published experimental data. The overall combustion-related performance of the engine is highly dependent on the in-cylinder fuel distribution and equivalence ratio [1]. Instead of pure gasoline powered engines, the concept of fuel enrichment to petroleum-based fuels for use in internal combustions engines generates greater interest. The methanol enrichment does not require any major engine design changes and involves fewer modifications to the engines and their fueling system. The proposed computational study will quantify the effect of different levels of methanol enrichment starting at zero percent up to the maximum percent allowable.

A natural aspirated spark ignited gasoline race car engine operating at full load conditions will be investigated in this study.

With 10%, 20% and 85% of methanol by volume, Methanol-gasoline blends of were used to investigate the engine performance parameters of engine power, thermal efficiency and emissions using a port-fuel injection spark ignition (SI) engine [2]. Methanol enrichment improved the brake thermal efficiency of the engine. The use of 85% methanol decreased CO and NO_x formation. A comparative analysis in-cylinder pressure of these blends against gasoline was also performed that showed a decrease in peak in-cylinder pressure by using the methanol/gasoline blends.

Dimethyl ether (DME) and methanol blends were tested to investigate the effects of DME addition on a SI engine performance under idle conditions [3]. Modifications in engine design were made to simultaneously inject both fuels into the intake ports. DME fractions were varied to investigate and analyze the effects of indicated thermal efficiency and pollutant emissions. With the increase in the DME fraction, hydrocarbon (HC) emissions dropped to 50% while the NO_x and CO emissions increased.

As shown in the previously in the study performed by Arshad et. al [1] for hydrogen enrichment, hydrogen enrichment improved SI engine performance. Laminar flame speed correlation for hydrogen-methanol blends was developed to correctly predict the laminar flame speeds [4]. The predicted data was validated against the experimental data which showed the validity of the developed correlation. The study also validated engine performance parameters of heat release and in-cylinder pressure with the experimental data to demonstrate the correlation's suitability.

Experimental investigation was performed of hydrogen blended port injected methanol SI engine [5]. Hydrogen was injected at volume fractions of 0 and 3% to investigate the combustion and emissions performance. The investigation showed an improvement in brake thermal efficiency. Also, a decrease in carbon monoxide (CO) and hydrocarbon (FC) emissions was observed.

Due to downsizing and down-speeding techniques employed in the design of latest SI engines to reduce pollutant emissions, brake mean effective pressure has increased in the latest designs causing increased risk of knock and abnormal combustion [6, 7]. To suppress knock, the engines are operated at high fuel enrichments and delayed spark advances. The studies aimed to decrease the risk of knock by port-injecting water/methanol and water to replace high gasoline enrichment. It was found that pure water injection with gasoline was the best case to suppress the knock. Similar results were obtained with low methanol cases. A similar study was performed to investigate the effects of water/methanol blends on the highly downsized port injected GDI engine [8].

A study was performed to investigate the effects of hydrogen boosting on SI engine running on gasoline methanol and gasoline ethanol blends [9]. Comparative analysis was performed to investigate the effects of hydrogen boosting of various fuels: gasoline, gasoline-methanol, and gasoline-ethanol, on the engine performance parameters of brake power, brake thermal efficiency and emissions.

A study [10] was performed on a hydrogen-methanol engine to investigate effects of spark timings on engine performance parameters i.e. combustion and emissions. The engine was installed with a hydrogen port injection system. Hydrogen intake was kept at volume fractions of 0, 1.5 and 3%. With the increase of spark advance, brake thermal efficiency first increased and then decreased, flame propagation period shortened and flame development period prolonged. Upon investigation of emissions, HC and CO emissions decreased. It was learned that NO_x can be reduced by retarding the spark timing.

In an effort to reduce the reliance on fossil fuel energy and reduce the damage to the environment, a 20% hydrogen and 80% methane mixture known as hythane was used in the study [11]. The study observed that CO, CO_2 and HC emissions reduced by use of the mixture. The paper investigated the effects of hythane on gasoline-methanol blends and gasoline-ethanol blends as well. Apart from emissions, brake power and brake thermal efficiency were evaluated. NOx emissions decreased in gasoline-methanol and gasoline-ethanol blends while thermal efficiency and brake power increased.

Effects of compression ratio were investigated in a stratified-charge methanol engine for various engine performance parameters of power, brake thermal efficiency, emissions, and torque [12]. The results were then compared with its diesel counterpart. With a decrease in compression ratio, it was observed that the power and torque decreased, and the brake thermal efficiency increased (at low load) and decreased (at high load). The HC and NO_x emissions increased with increase in compression ratio while CO emissions decreased.

The following reaction mechanism was considered when using hydrogen enrichment [1]:

$$4C_8H_{17} + 49O_2 \rightarrow 32CO_2 + 34H_2O \quad (1)$$

$$2H_2 + O_2 \rightarrow 2H_2O \quad (2)$$

A CFD analysis showed a higher efficiency using hydrogen and gasoline as fuel as compared to pure gasoline for the considered engine [1].

II. Computational Model

The numerical simulations code used in the current study is KIVA-3V [13] developed by Los Alamos National Laboratory for numerical calculation of transient, two- and three-dimensional chemically reactive fluid flows with sprays.

The paper investigates premixed air-fuel mixture and the effects of small amounts of methanol and methanol/water in the reactant gases of a gasoline spark ignition (SI) race car engine with pentroof geometry. A substantial amount of experimental research has been done on methanol injection in gasoline engines to increase the combustion and thermal efficiency. KIVA-3V was used to perform a comparative parametric analysis between the gasoline, gasoline/methanol and gasoline/methanol/water global mechanisms. A mesh independent study was performed for the engine geometry of ø101.6 mm bore and 88.4 mm stroke, running at 6500 rpm. The number of mesh cells used in the present study was 230,000. Table I and Fig. 1 below show the engine geometry.

The analysis was performed at varying methanol and methanol-water inlet concentrations, i.e. 0%, 1.8%, 3.6% and 5.4%, with gasoline for an equivalence ratio of 1.0.

TABLE I. ENGINE GEOMETRY SPECIFICATIONS

Compression Ratio	9.5
Bore [mm]	101.6
Stroke [mm]	88.4
Displacement	5.7L
Engine Speed [rpm]	6500
Ignition	30º bTDC; 25º bTDC
Ignition Duration	3º

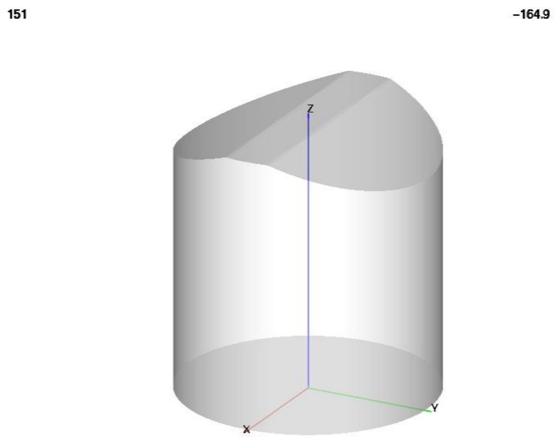

FIG 1. ENGINE GEOMETRY

Global mechanism [14] uses one-step fuel reaction mechanism along with the three reactions from the Zeldovich mechanism for prediction of NO_x formation. The chemical reaction equation for the gasoline global mechanism is:

$$4C_8H_{17} + 49O_2 \rightarrow 32CO_2 + 34H_2O \quad (3)$$

When methanol is added to the gasoline, the global of methanol is:

$$2CH_3OH + 3O_2 \rightarrow 2CO_2 + 4H_2O \quad (4)$$

In case of water addition to methanol, the reaction of methanol-water is:

$$2CH_3OH + 3O_2 + 2H_2O \rightarrow 2CO_2 + 6H_2O \quad (5)$$

The global mechanism [14] uses a one-step reaction mechanism for the fuel, one-step mechanism of methanol, and a one-step mechanism of methanol-water with three reactions from the Zeldovich mechanism.

The chemical reaction equations for the Zeldovich mechanism are:

$$2N_2 + O_2 \rightarrow 2N + 2NO \quad (6)$$

$$N_2 + 2O_2 \rightarrow 2O + 2NO \quad (7)$$

$$N_2 + 2OH \rightarrow 2H + 2NO \quad (8)$$

The analysis helped in determining the correct reaction mechanism to understand the chemical kinetics. Comparison analysis was performed between global gasoline, gasoline/methanol and gasoline/methanol/water mechanism for engine parameters: average temperature, fuel concentration thermal efficiency, indicated power and emissions.

III. RESULTS & DISCUSSION

Numerical simulations are performed to investigate the variation between the engine and combustion parameters of average temperature, fuel concentration thermal efficiency, indicated power and emissions.

With the spark-ignition (SI) engine, the ignition option is turned on in the input file in KIVA-3V. For premixed case, tests are performed for nominal ignition angle of 25º and 30º bTDC at stoichiometric conditions. Fig. 2 to Fig. 7 shows the fuel concentration before and during the combustion. Since it is a comparison between single component fuel and dual fuel, Fig. 2 to Fig. 7 shows methanol and methanol-water concentration of 0%, 1.8%, 3.6% and 5.4% at equivalence ratio of 1.0.

TABLE II. ENGINE PERFORMANCE PARAMETERS AT VARIOUS METHANOL AND METHNAOL-WATER CONCENTRATIONS AND IGNITION ANGLES

Methanol Conc. (%)	***Fuel***	***Ignition***	***Thermal Efficiency (η_{th})***	***Indicated Power (hp)***
0 %	Gasoline	30 bTDC	35.73	65.52
		25 bTDC	35.45	65
1.8%	Gasoline-Methanol	30 bTDC	37.01	67.85
3.6%			36.79	67.44
5.4%			36.91	67.66
1.8%	Gasoline-Methanol	25 bTDC	36.68	67.25
3.6%			36.38	66.71
5.4%			36.46	66.84
1.8%	Gasoline-Methanol-Water	30 bTDC	36.84	67.55
3.6%			36.5	66.92
5.4%			36.14	66.25
1.8%	Gasoline-Methanol-Water	25 bTDC	36.58	67.06
3.6%			36.17	66.31
5.4%			35.8	65.63

Results in Fig. 2 to Fig. 7 show aggressive burning of gasoline-methanol and gasoline-methanol-water blends at stoichiometric conditions. The combustion efficiency in all cases has been 100% due to the high speed (rpm) of engine. Addition of methanol and methanol-water blends encouraged complete burn of the fuel. Increase in indicated power, and thermal efficiency was observed with increase in secondary fuel concentration. The peak value of average temperature and maximum temperature in the cylinder decreased with the increase in the methanol and methanol-water concentrations. The average temperature and maximum temperature inside the cylinder slightly decreased at the end of the power stroke at higher methanol and methanol-water concentrations. This is significant because as the percentage of secondary fuel increases in the fuel, there are higher efficiencies, and lower temperatures inside the cylinder by the end of the power stroke, which means fewer emissions. Table-II above shows the thermal efficiency along with the indicated power.

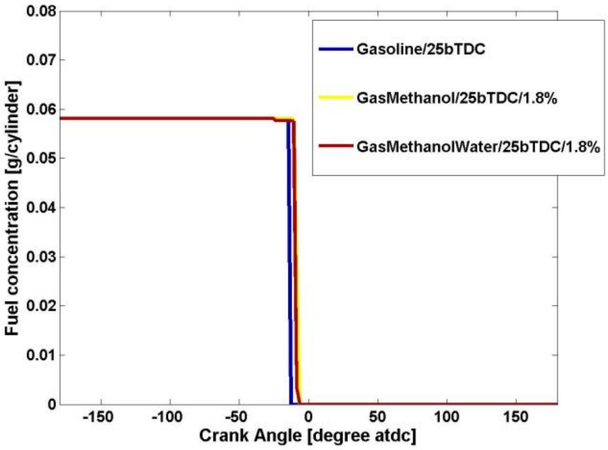

FIG 2. FUEL BURN OF 1.8% BLEND AT 25° BTDC – PREMIXED

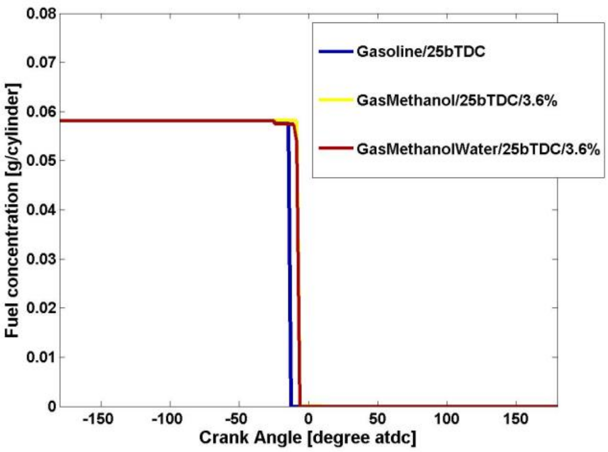

FIG 3. FUEL BURN OF 3.6% BLEND AT 25° BTDC – PREMIXED

Fig. 2 to Fig. 7 shows the comparison of gasoline concentration as a single component fuel, gasoline/methanol and gasoline/methanol/water concentration as a dual fuel. The figures show the fuel concentration before and during the combustion for ignition at 25° bTDC and 30° bTDC for methanol and methanol-water enrichments. For all enrichment levels, the dual fuel burns completely.

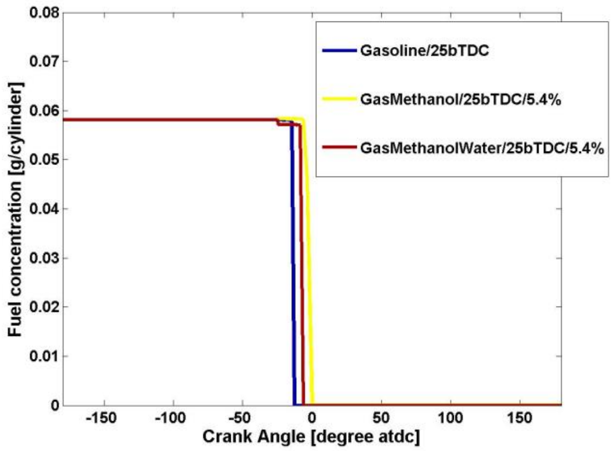

FIG 4. FUEL BURN OF 5.4% BLEND AT 25° BTDC – PREMIXED

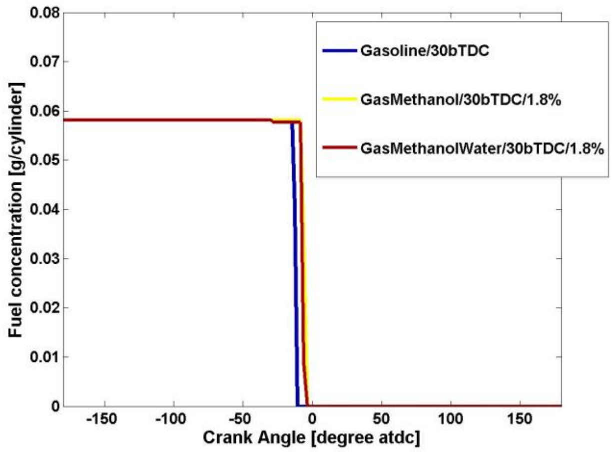

FIG 5. FUEL BURN OF 1.8% BLEND AT 30° BTDC – PREMIXED

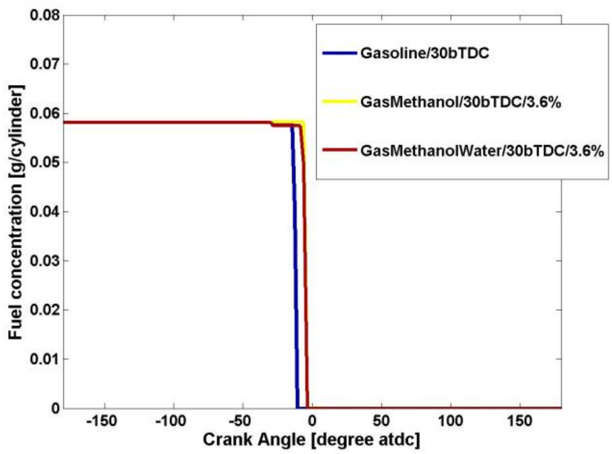

FIG 6. FUEL BURN OF 3.6% BLEND AT 30° BTDC – PREMIXED

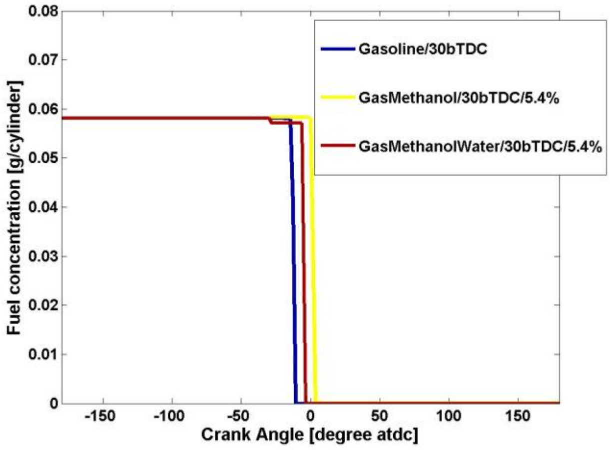

FIG 7. FUEL BURN OF 5.4% BLEND AT 30° BTDC – PREMIXED

The plots from Fig. 8 to Fig. 13 show the work envelope which is used to calculate the indicated power. The PV diagrams obtained show a slight increase in indicated power for the dual-fuels at higher methanol and methanol-water concentrations than the single-component gasoline fuel. The highest indicated power as well as the highest thermal efficiency was achieved for gasoline-methanol for 1.8% blend. This shows that adding methanol proved to be beneficial and will help in the getting more work output from the race car spark ignition engine.

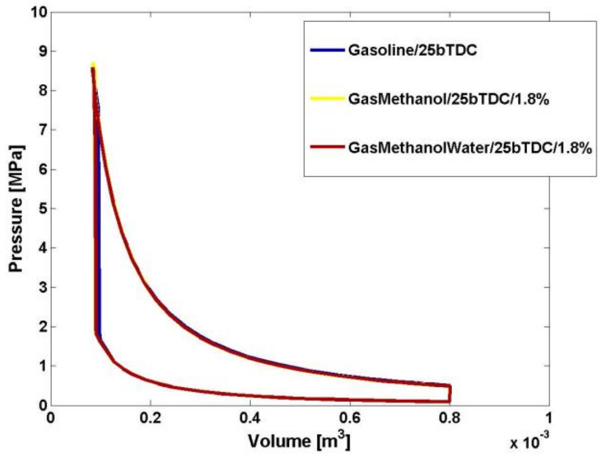

FIG 8. INDICATED POWER PRODUCED BY 1.8% BLEND AT 25° BTDC – PREMIXED

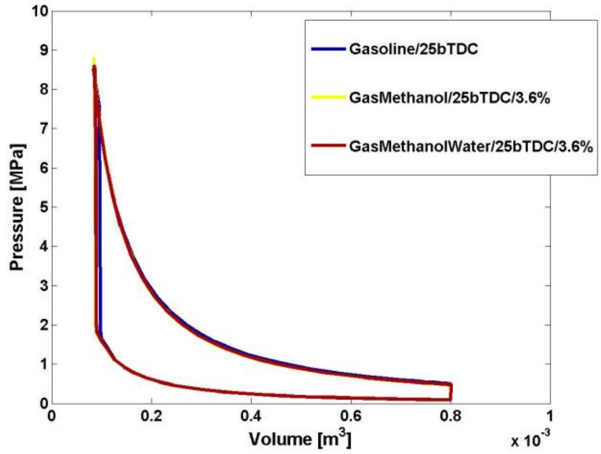

FIG 9. INDICATED POWER PRODUCED BY 3.6% BLEND AT 25° BTDC – PREMIXED

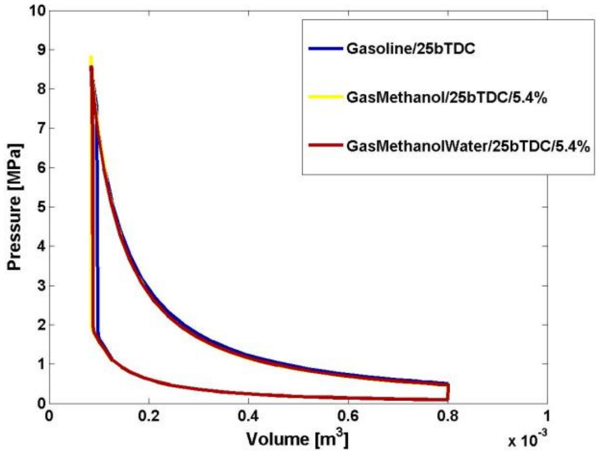

FIG 10. INDICATED POWER PRODUCED BY 5.4% BLEND AT 25° BTDC – PREMIXED

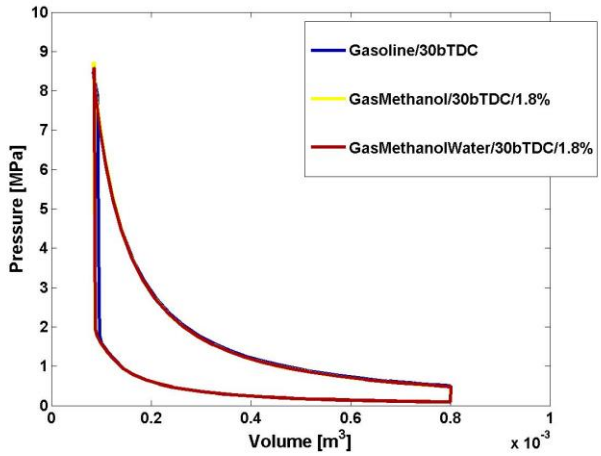

FIG 11. INDICATED POWER PRODUCED BY 1.8% BLEND AT 30° BTDC – PREMIXED

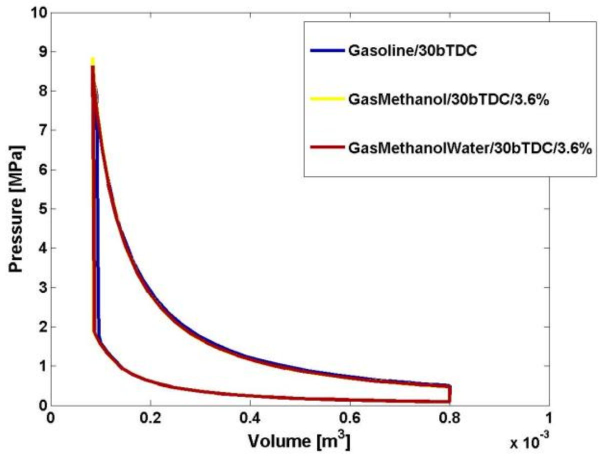

FIG 12 INDICATED POWER PRODUCED BY 3.6% BLEND AT 30° BTDC – PREMIXED

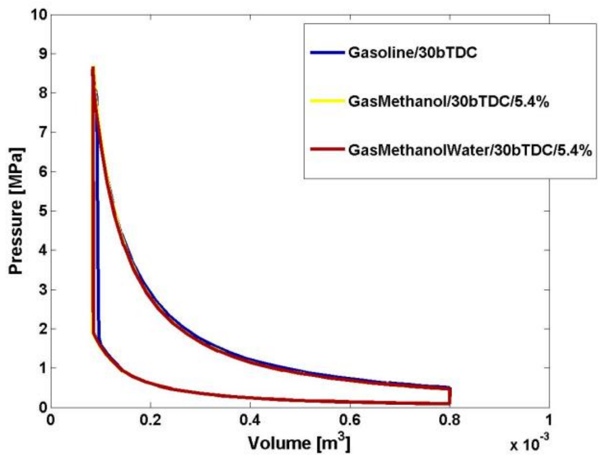

FIG 13. INDICATED POWER PRODUCED BY 5.4% BLEND AT 30° BTDC – PREMIXED

It was also observed that increase in the methanol and methanol-water concentrations helped in reducing NO_x emissions. As previously mentioned that the reduction in average temperature in the engine was observed which resulted in the reduction of NO_x emissions as shown the Fig. 14 to Fig. 16.

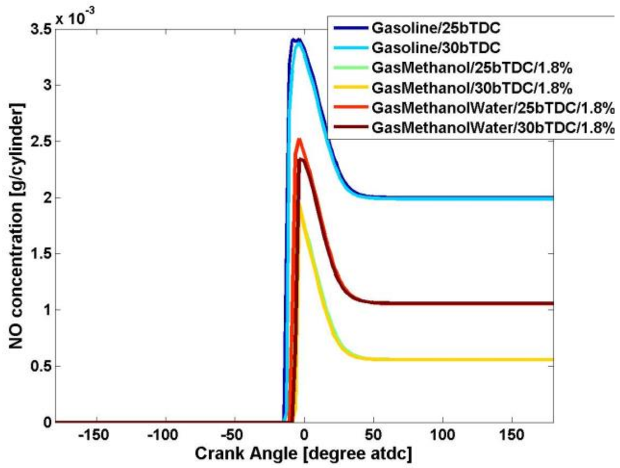

FIG 14. COMPARISON OF NO_x EMISSIONS: 1.8% BLEND AT 25° AND 30° BTDC – PREMIXED

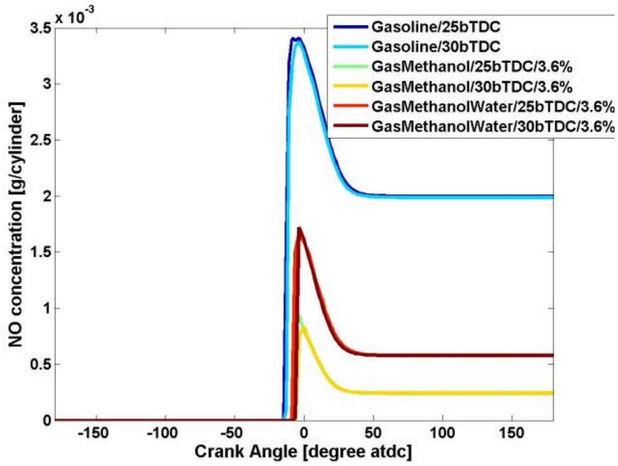

FIG 15. COMPARISION OF NO_x EMISSIONS: 3.6% BLEND AT 25° AND 30° BTDC – PREMIXED

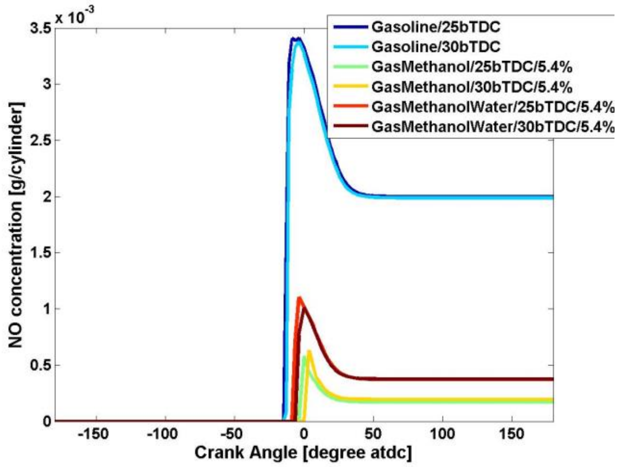

FIG 16. COMPARISON OF NO_x EMISSIONS: 5.4% BLEND AT 25° AND 30° BTDC – PREMIXED

A. Temperature Plots - Comparison of Global Gasoline Mechanism, Gasoline/Methanol Mechanism and Gasoline/Methanol/Water Mechanism at $\phi = 1.0$:

The simulation results obtained using KIVA-3V are post-processed to construct temperature plots. The plots in *Fig. 17* to *Fig. 24* show a comparison of the combustion results from start of ignition to complete burn for the premixed case at equivalence ratio of 1.0 for both single component gasoline and dual fuel reaction mechanisms.

IV. CONCLUSION

A comprehensive study was performed for a detailed analysis of combustion efficiency, thermal efficiency, indicated power, fuel concentration, emissions, and a study of the temperature plots. The criterion of this study was to ensure increased combustion and thermal efficiencies with high power output and less emissions. This study provided detailed information on a premixed single-component fuel and on dual-fuel cases. As compared to the base case of single-component fuel (i.e. gasoline), addition of methanol and methanol-water in the premixed case showed a trend of increase in the thermal efficiency, indicated power and encouraged a complete burn of the fuel mixture in the engine. The temperature decrease at high concentrations of methanol and methanol-water implies low emissions. It shows that if a small amount of methanol is premixed with gasoline, it will encourage a complete burn with increased power output and thermal efficiency that is beneficial for a race car engine, thereby producing more power by the race car engine. Methanol injection makes it a cost-effective process as well and will help in reducing the NO_x emissions.

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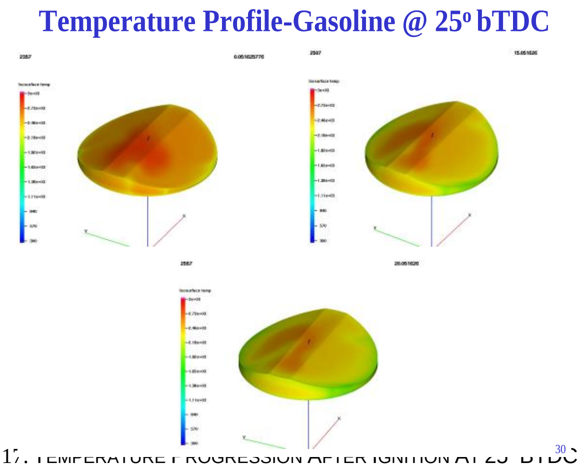


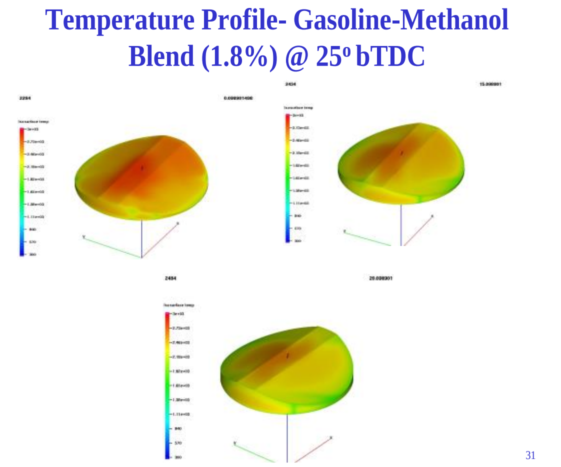
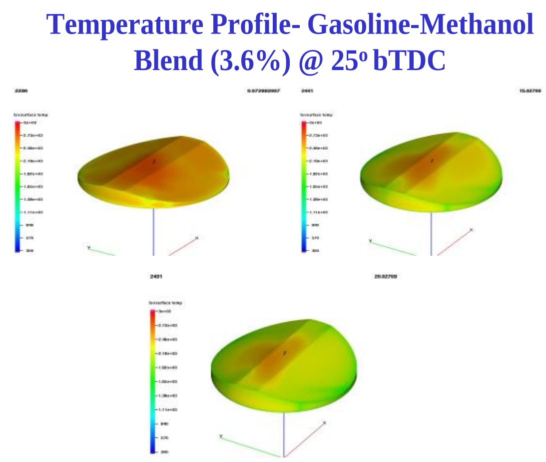


FIG 17. TEMPERATURE PROGRESSION AFTER IGNITION AT 25° BTDC



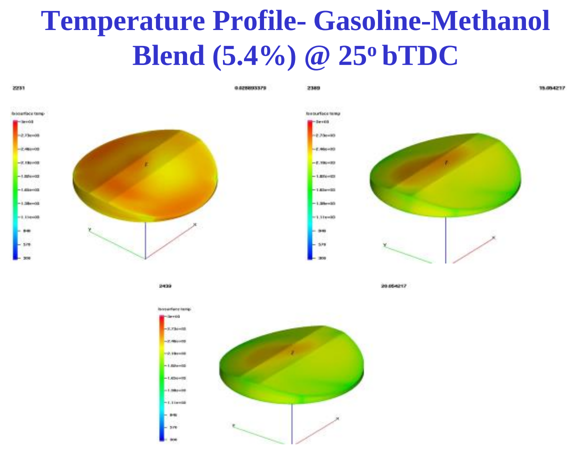
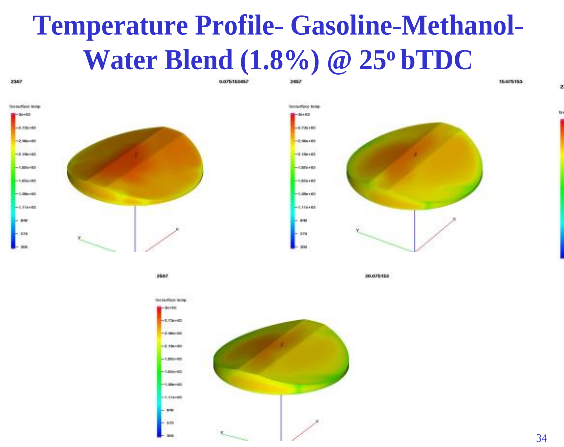


FIG 18. TEMPERATURE PROGRESSION AFTER IGNITION AT 25° BTDC



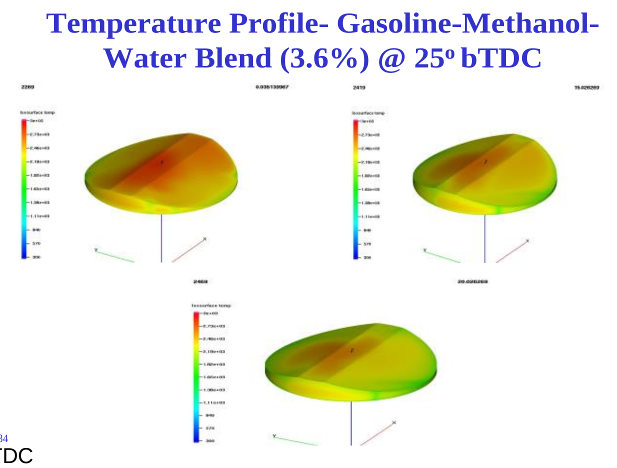


FIG 19. TEMPERATURE PROGRESSION AFTER IGNITION AT 25° BTDC

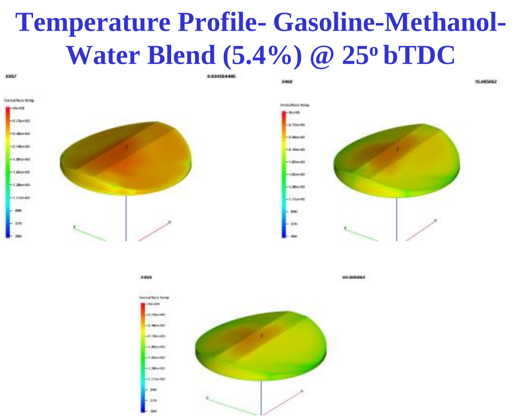
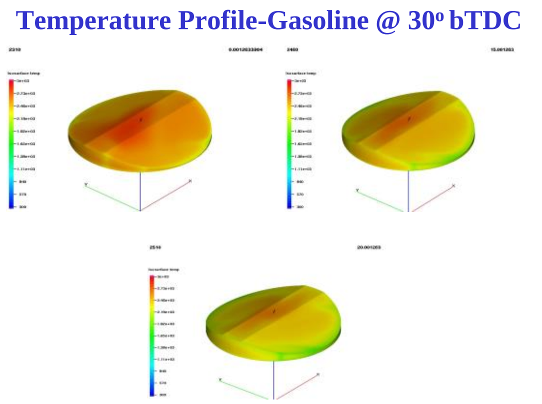


FIG 20. TEMPERATURE PROGRESSION AFTER IGNITION AT 25º BTDC



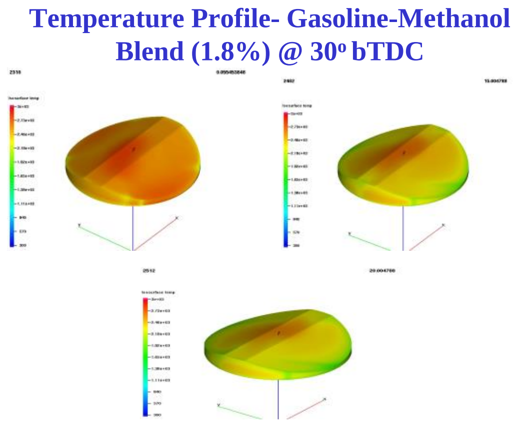
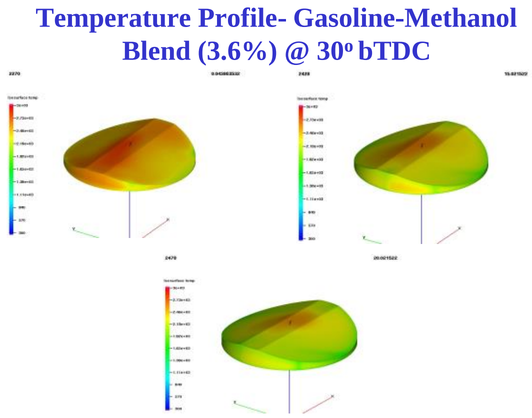


FIG 21. TEMPERATURE PROGRESSION AFTER IGNITION AT 30º BTDC



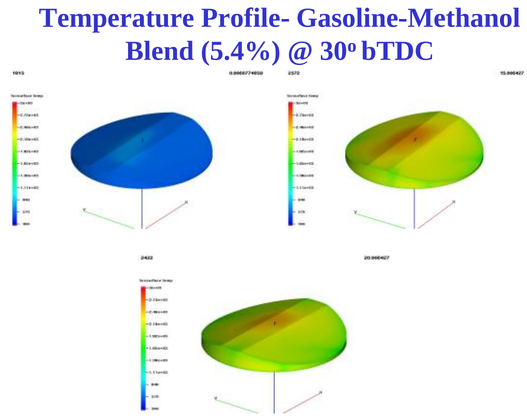


FIG 22. TEMPERATURE PROGRESSION AFTER IGNITION AT 30º BTDC

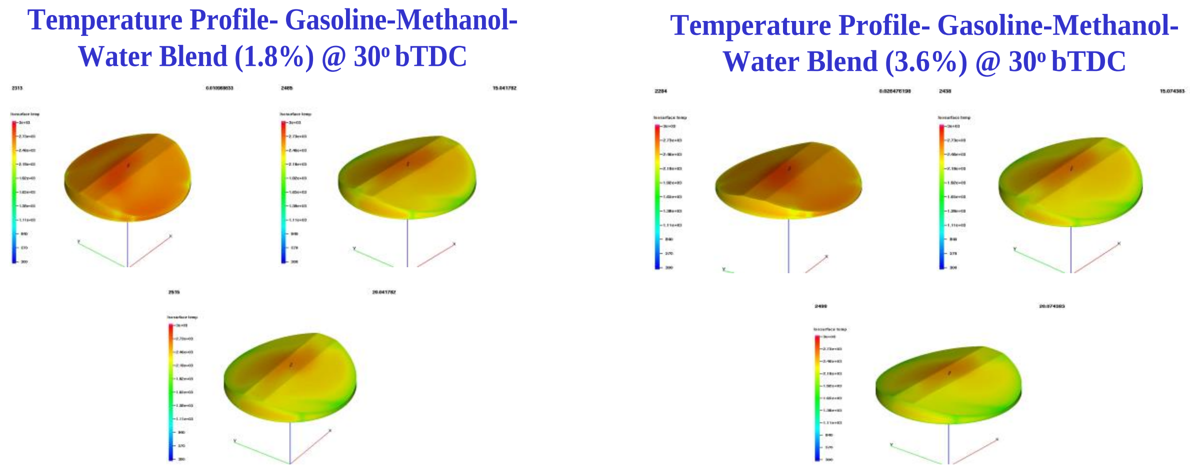


FIG 23. TEMPERATURE PROGRESSION AFTER IGNITION AT 30° BTDC

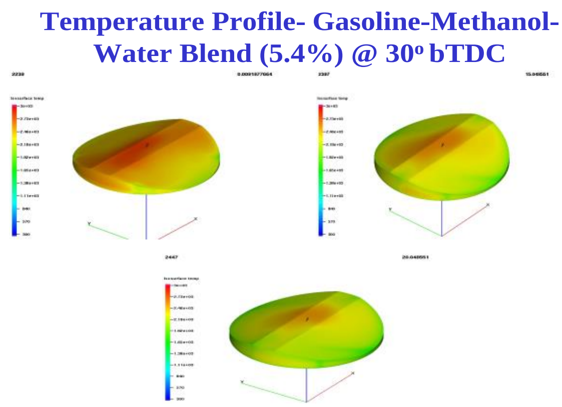


FIG 24. TEMPERATURE PROGRESSION AFTER IGNITION AT 30° BTDC