Numerical Simulation of Ethanol-based Fuels in an F1 Power Unit

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Abstract

Formula 1 vehicles have transitioned from E5 to E10 fuel for the 2022 season to reduce carbon emissions and by 2026 the vehicles are required to use 100% sustainable fuels. The aim of this paper is to identify the operating envelope of the F1 power unit for E10-E100 fuel and the resulting emission levels for these fuel compositions using numerical simulations. To achieve this aim an F1 engine model has been developed in GT-Suite with reference to the FIA 2022 Technical Regulations. The combustion model has been validated using data obtained from literature relating to laminar and turbulent flame speed, friction and heat transfer characteristics within the combustion chamber. One of the main challenges of using ethanol-based fuels is the increased levels of formaldehyde in the tailpipe. This paper presents the operating window for achieving the optimum engine performance with ethanol fuel blends ranging from the current E10 to E100, in keeping with the current 2022 FIA F1 regulations and beyond 2026 where all fuel must be fully sustainable. The study showed that the estimated formaldehyde levels from 2026 Formula 1 engine is significantly higher than the current emission levels of automotive vehicles. This paper highlights the required regulatory changes to ensure the engine out aldehyde emissions meet WHO air quality standards.

Background

Formula 1 (F1) is the set of regulations that governs the highest level of motorsport competition in the world. Modern F1 racing events feature 10 teams and 20 drivers who compete for the constructor's and driver's championship respectively over 20+ race events per year. F1 developed from its simple beginnings in the 1950's to establish itself as one of the most technically demanding businesses on the planet, with each team spending hundreds of millions of pounds each year to outcompete each other [1]. The competitive aspect of Formula 1 has led to numerous innovations which have found applications outside the sport, such as communications technology – now used to improve public transport connectivity, pit stop techniques adopted to improve efficiency in the manufacturing industry and advanced structural materials and manufacturing techniques now used in the automotive industry and in other sports to reduce mass and increase mechanical and aerodynamic performance [2].

One of the most impactful and constantly evolving technical areas in F1 is the power unit, which in 2014 transitioned from normally aspirated to a highly efficient turbocharged unit, resulting in thermal efficiency increasing from 29% to 40% and a reduction in CO_2 levels by up to 26%. By 2019 thermal efficiencies increased over 50% and the engines are outperforming their old normally aspirated predecessors whilst using significantly less fuel. Due to regulated fuel consumption requirements in the current power unit regulations, the power unit's efficiency is a key component in maximising on track performance, as the engine that can deliver most power per unit mass of fuel will be more competitive [2].

Formula 1 announced in November 2019 that it is planning to have a net-zero carbon footprint by 2030 to become a sustainable sport and to globally promote and accelerate the development of technologies to eliminate carbon emissions. All factors of the sport will be improved upon to meet this goal, but a key component is the internal combustion engine used by all competing cars [3][4]. To reduce the carbon emissions from the internal combustion engines, the fuels used have started to incorporate increasingly higher percentages of biofuel content, moving from 5% to 10% sustainably sourced ethanol fuel (E10) for 2022 and with the goal of reaching a 100% sustainable fuel content in the future, where there is a high likelihood, this will be achieved with further increased fuel ethanol content [5]. The current generation power units are hybrid, and as well as the new sustainable fuel regulations, an increase in electrical power output is to come in 2026. These new regulations are crucial for the survivability of Formula 1 as they attract more manufacturers looking to improve their powertrain technology, who in turn accelerate the development and adoption of highly efficient consumer vehicles.

Due to the differing chemistry of the higher ethanol-content fuels in comparison to the previous fuel used there are major ramifications for the engine performance and emissions. The higher percent ethanol fuel is now less prone to the undesirable phenomena of engine knock but also has less work output potential, and whilst there are fewer carbon-based emissions there are now also greater aldehyde emissions [6][7][8]. These factors necessitate the need for reoptimizing the engines for maximising performance and also warrant an evaluation of the engine-out emissions levels.

The fuel mass flow rate of Formula 1 engines is currently limited at all operating speeds by the sport's technical regulations, and they therefore have to run at lean conditions to extract the maximum performance [9]. Lean combustion of fuels produces lower cylinder temperatures which results in fewer losses due to heat transfer, a greater polytropic index, greater thermal efficiencies and fewer emissions [6]. A downside of the reduced cylinder temperatures is a slower laminar flame speed which can in turn negatively affect the engine thermal efficiency. This can be combated by increasing the compression ratio which reduces pumping losses and increases flame speed [10]. A high compression ratio in turn presents its own risk of engine knock. Knock is a phenomenon where the cylinder chamber gas mixture is at a high enough temperature and pressure to automatically ignite before a propagating flame reaches the mixture. This uncontrolled combustion has negative ramifications for emissions, performance and fuel consumption and can catastrophically damage all components in the combustion chamber, causing issues such as melted pistons, gasket damage, cylinder wall scuffing and cylinder head degradation [11].

The thermal efficiency of spark ignition (SI) combustion engines is limited in large part by end knock at high compression ratios and finding a means to reduce the likelihood of knock or its ability to occur is paramount to their viability and relevance for continued development in creating a net zero carbon future [12]. Fuels such as ethanol have a higher octane number than gasoline which increases their resistance to end knock, allowing for higher cylinder pressures and leaner combustion [7]. Owen and Coley [13] found that when the compression ratio of an ethanol fuelled single cylinder engine was increased from 8 to 18 (matching the maximum geometric compression ratio prescribed in the F1 Technical Regulations Article 5.4.6 [9]) performance gains of 16% were achievable. In testing lean combustion with a cooperative fuel research engine, Ran et al. [7] demonstrated that pure ethanol fuel exhibited greater combustion efficiency than E10 fuel and could operate at leaner conditions. There is however a downside to ethanol fuel regarding the work output in comparison to gasoline. Due to the lower calorific value of ethanol in comparison to gasoline, to produce a comparable work output more ethanol fuel needs to be consumed [6]. The allowable maximum fuel flow rates for the Formula 1 power unit prescribed by the technical regulations have remained unchanged between 2021 and 2022 meaning that there is a reduced rate of useful energy available for combustion this year, reducing the potential work output. It is generally understood at this point that Formula 1's new E10 fuel produces less power than the outgoing E5 [14][15].

Experiments to verify the performance of ethanol fuel in comparison to gasoline have been conducted previously but not in the context of the turbo-hybrid power units used in Formula 1 or in a duty cycle to simulate an ethanol fuelled F1 car on a circuit. Ran et al. [6] published the results of an experimental procedure which compared the combustion performance and emissions of E10 and 100% ethanol fuel at lean conditions. Whilst this gave a direct comparison between high and low ethanol concentration fuels, the experiments were carried out with a compression ratio of 8 and an engine speed of 1200rpm, far below F1's compression ratio limit of 18 and optimal engine speed of 10,500rpm where the maximum fuel consumption rate is reached [9].

Of the few publications evaluating the performance characteristics for validated F1 power units, none have studied the effects of the new ethanol-based fuels. Bopaiah and Samuel [16] produced and validated an F1 power unit and vehicle model to the 2019 regulations with GT-Suite to optimise the car's performance, but the focus of their study was the optimisation of the energy recovery system and therefore the only fuel considered was E5. Hassan and Samuel conducted a study as recently as 2021 [17] to optimise the control strategy for an F1 car's energy flow which similarly had less emphasis on the fuels used. One of the primary aims of this project is to evaluate methods to maximise the performance in F1 engines fuelled with high percentage ethanol blends used in 2022 and onwards. It is possible to analyse future fuel performance with a high degree of certainty because the engine regulations are frozen until 2025 [18].

The reason for the introduction of ethanol-based fuels into Formula 1 is to reduce the levels of carbon emissions[5]. Whilst ethanol fuels have the potential to reduce carbon emissions, they produce much greater quantities of aldehydes than gasoline. Acetaldehyde and formaldehyde (HCHO) are examples of such aldehydes and are produced during ethanol combustion [19]. Acetaldehyde and formaldehyde (HCHO) are carcinogenic chemicals which, along with other products such as carbon-based emissions and nitrogen oxides (NOx), are the highest released emissions from ethanol combustion, with 125 - 204% more formaldehyde and 4500 - 8200% more acetaldehyde produced from E85 in comparison to E5 fuel [8][20][21]. HCHO emission standards were first implemented in the US for heavy duty trucks and buses in 1998 as a voluntary certification where inherently low emission vehicles and ultra low emission vehicles had to produce less than 0.05g/bhp.hr and 0.025g/bhp.hr respectively. From 2004, HCHO emissions were regulated for light duty vehicles and cars and from 2024 onwards heavy-duty Otto cycle engines will have to produce less than 0.01g/bhp.hr – demonstrating the increasing awareness of the danger of HCHO emissions over time [22][23]. Short term exposure to formaldehyde presents minimal risk, but long term side effects include the development of nasal tumours and leukaemia, whilst those with breathing issues such as asthma may experience allergic reactions [24]. However, very limited information on emissions produced by F1 engines is available in the published domain especially in relation to aldehydes. Elmagdoub and Samuel [25] estimated the exhaust emissions produced by F1 cars at eight different race tracks and proposed regulatory changes to improve the air quality based on the emissions levels of CO, THC, NOx and particulate matter (PM). Mourao [26] estimated only CO_2 emissions from F1 cars throughout its history. Similar to Ran et al.'s 2019 and 2020 reports [6][7], Roso et al. [27] examined the emissions of ethanol-based fuels at lean conditions but did not investigate aldehyde emissions. Therefore, the primary objectives of this paper are to develop the overall performance envelop of the Formula 1 power unit for varying levels of Ethanol content in the fuel and resulting Formaldehyde emission levels.

Modelling F1 Engine with Ethanol fuel blends

An F1 engine model was built in GT-Suite engine simulation software based on 2022 FIA F1 Technical Regulations which can run on varying ethanol fuel content. Laminar flame speed, friction factors, combustion chamber working temperatures and laminar to turbulent flame transition characteristics are obtained from literature for validating the combustion model. The formaldehyde emissions were estimated using both the GT-Suite F1 engine model output data and literature. Table 1 shows the input parameters required to build the GT-Suite model and Figure 1 shows a schematic diagram of the GT-Suite engine model.

Table 1: Input parameters for the GT-Suite F1 engine model [9].

Engine type	4-Stroke, SI engine
Cubic capacity	1600cc (+0/-10cc)
Maximum fuel mass flow rate	100kg/h (10500rpm – 15000rpm)
Pressure charging	Sole single stage compressor and sole single stage turbine
Number of cylinders	6, equal capacity
Valves	2 inlet, 2 exhaust per cylinder
Cylinder bore	80mm (+/-0.1mm)
Crankshaft main bearing journal diameter	Minimum 43.95mm
Crankshaft crank pin bearing journal diameter	Minimum 37.95mm
Compression ratio	Maximum 18
MGU-H rotational speed	Maximum 125,000rpm

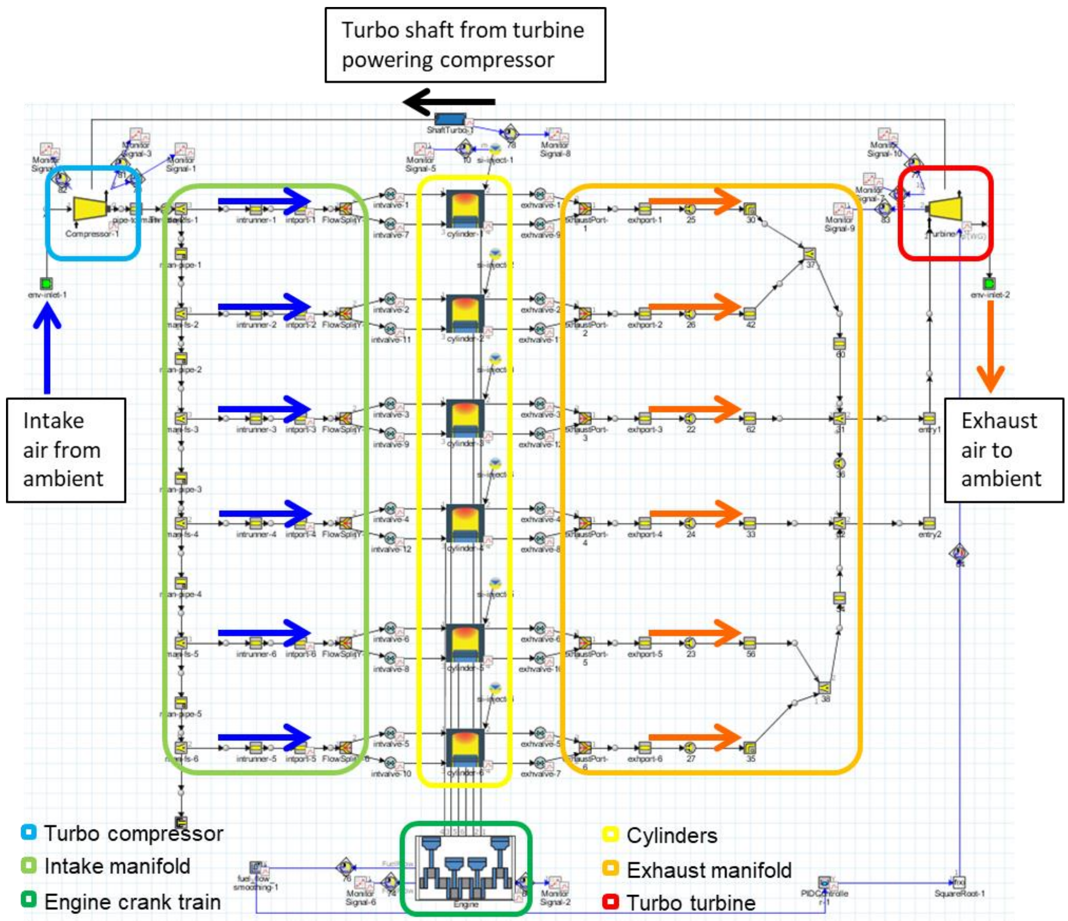

Figure 1: Schematic diagram of F1 engine modelled in GT-Suite.

Modelling

Laminar flame speed

No elements of the piston cup geometry are specified within the 2022 technical regulations [9] and the simple cylindrically shaped indentation used in this project is reasonable based on the rough dimensions of appropriate pistons for this type of application [28]. The spark location was set at the top-centre of the cylinder. Equation (1) is used to calculate the laminar flame speed of single ingredient fuels [29] and Equation (2) is used to calculate the laminar flame speed of the ethanol – isooctane fuel blend. Isooctane fuel is used as it complies with Article 16.3 for a fuel with minimum (RON+MON)/2 value of 87 [9].

$$S_L = S_{L,o}\left(\frac{T_u}{T_{ref}}\right)^{\alpha}\left(\frac{P}{P_{ref}}\right)^{\beta} \tag{1}$$

$$S_{L,o} = B_m + B_{\Phi}(\Phi - \Phi_m)^2$$

$$\alpha = 2.18 - 0.8(\Phi - 1)$$

$$\beta = -0.16 + 0.22(\Phi - 1)$$

$$S_{L,blend} = \frac{1}{\sum \frac{x_i}{S_{L,i}}} \tag{2}$$

Friction

The engine friction model [30] used was developed from the testing of over 300 motorbike engines of varying displacements, cylinder geometries, cylinder numbers and configurations (including V configuration engines such as the V6 used in F1), capable of operating at speeds between 8000rpm – 16000rpm, covering the speed range of F1 engines. The friction model is based on the condition of wide-open throttle and is therefore ideal for racing engine application [30]. Equation 3 shown below is used to calculate the friction mean effective pressure (FMEP) at different engine speeds. The cylinder number coefficient, K_c, is was assumed to be equal to 0.7 as the resulting friction matches very closely with the well-established Chen-Flynn friction model at lower speeds.

$$FMEP = \left\{\Phi_1\left(\frac{V_s N_e}{Z}\right)^2 + \Phi_2 N_e^2 + \Phi_3 \nu + \Phi_4\right\} \times \frac{\sqrt{S \cdot D_{CM}}}{B} \tag{3}$$

$$D_{CM} = \frac{K_c\{\sum_1^m D_{cj} + \sum_1^n D_{cp}\}}{m + n}$$

Cylinder working temperature

Articles 5.17 and 5.18.1 state that pistons may not be manufactured from titanium alloys or ceramics as well as a wide variety of other alloys [9]. It is assumed that the pistons are either made from iron or aluminium based alloys as these alloys have been used in previous F1 engines [31]. Article 5.18.8 states that static components such as engine crankcases and cylinder heads must be manufactured from aluminium or iron alloys [9]. Article 5.18.6 states that valves must be manufactured from intermetallic alloys or iron, nickel, cobalt or titanium alloys [9]. The GT-Suite model requires a single operating temperature for the entire ceiling surface of the combustion chamber and therefore it is assumed the valves are made from the same material as the cylinder head in this study.

The maximum service temperature for aluminium and iron alloys is ~920K and ~1670K respectively [32]. A parametric study is performed in GT-Suite where the working temperatures of the piston heads, cylinder walls and cylinder ceiling are adjusted to determine the effect on the engine's efficiency. The combination of operating temperatures which gives the greatest indicted efficiency is selected based on the maximum working temperature limits.

Turbulent flame characteristics

To obtain a valid turbulent combustion model, the physical engine parameters used by Qi and Lee [33] were replicated in a GT-Suite model as well as the operating conditions from the experiment. The turbulent flame parameters were adjusted to make the mass fraction burned (MFB) vs crank angle plot from the GT-Suite model match the literature. The replicant GT-Suite engine model was made to the experimental specification shown in Table 2. The Engine was run at an equivalence ratio of 1 at 2500rpm as in the experiment.

Table 2: Engine specification used to obtain turbulent flame characteristics [33].

Engine type	Water-cooled, SI engine
Number of cylinder	3
Firing order	1-3-2
Bore (mm)	68.5
Stroke (mm)	72
Compression ratio	9.4:1
Displacement (ml)	796
Maximum power (kW at 5000 r/min)	26.5
Maximum torque (N m at 3000–4000 r/min)	60.5

Turbocharger and intercooler

A high performance EFR9180 Borgwarner turbocharger was selected as the basis for the optimisation process as it can provide the required mass flow rates and pressures for F1 engine operation [34] since the Borgwarner turbo is designed for engine ratings similar power to F1 engines. Due to the high air compression ratio of the compressor and resulting high temperature of exiting air towards the cylinders, an intercooler was implemented to accompany the turbocharger. The intercooler's effectiveness was varied to determine the effectiveness which would provide the greatest efficiency and power gains for the F1 engine.

Emissions modelling

Experimental data for formaldehyde emission levels were obtained from literature for estimating formaldehyde emission levels from F1 engines. Formaldehyde emission rates (mg/mile) for a 2004 Volkswagen Golf GTI running on 4 different ethanol fuel blends on the LA92 drive cycle were obtained from literature [35] and used as the base dataset for predicting the F1 engine emissions. The average power required for the Golf on the LA92 cycle is calculated with Equation (4) [36], where the vehicle speed is the average speed during the cycle.

$$\dot{W}_r = \frac{1}{\eta_t}\left(\frac{1}{2}\rho C_d A V^2 + mg(f + KV^2)\right)V \tag{4}$$

The rolling coefficient, $f + KV^2$ is calculated with Equation (5) [37], where the vehicle speed is the average speed during the cycle and the tyre pressure is taken as the mean between front and rear tyres.

$$f + KV^2 = 0.005 + \frac{1}{p}\left(0.01 + 0.0095\left(\frac{v}{100}\right)^2\right) \qquad (5)$$

The transmission efficiency was taken from the results of a front wheel drive (FWD) vehicle test by Irimescu, Mihon and Pădure's [38] on transmission efficiency measurement. It was assumed the average transmission efficiency over a range of tested engine speeds from the paper is a suitable value for the Golf GTI on the LA92 cycle. The average transmission efficiency from the tested FWD vehicle is 91 %.

The Golf GTI formaldehyde emissions in terms of mg/mile given by Knoll et al. [35] are for fuel blends ranging between E0 and E20. A trendline was extrapolated from these points to predict the emissions up to E100. The total formaldehyde emissions of the Golf GTI for the 4 ethanol fuel blends and beyond were calculated with knowledge of the LA92 cycle's length (11.04 miles), and with the average required power calculated with Equation 4, the emissions rate was calculated. Using the emissions rate calculated from the Golf GTI, the emissions levels from the F1 engine around F1 circuits is obtained based on qualification laps. The pole position time was used for calculations as the highest performing vehicle usually has the highest performing engine. The California emissions standards are used for comparison as they are more stringent than the standard US formaldehyde requirements [23]. California formaldehyde emissions standards are measured in grams per mile (g/mi). The emissions rate obtained from the Golf GTI and used for scaling to the F1 engine is in units: mg/kWh, therefore the F1 engine power, pole lap times and lap distances are required. F1-tempo [39] was used to obtain pole lap telemetry of 2022 season F1 cars. Engine speed telemetry plots were used to obtain the average engine speed during the lap, which can be used to estimate the lap's average power output. The fuel mass flow rate increases linearly up to 10500rpm where it is capped at 100kg/h. As the fuel mass flow rate has a linear relationship with engine speed [9], it can be assumed power output also has a linear relationship, where 0rpm equates to 0kW output and 10500rpm is equal to the output calculated in GT-Suite at 10500rpm. It is assumed that power does not increase above this calculated GT-Suite output as the fuel mass flow rate is constant above 10500rpm. The lap distances and times are obtained from Formula 1® - The Official F1® Website [40].

Results

Laminar flame speed

Table 3 shows the laminar flame speeds calculated by GT-Suite and the literature method for 10 – 100% ethanol fuel blends. The results align very closely with an average difference of 1.075%, with the largest discrepancy being 2.084% for E90.

Table 3: Calculated laminar flame speed vs GT-Suite predicted flame speed.

Fuel	GT Suite laminar flame speed (m/s)	Calculated laminar flame speed (m/s)	Flame speed difference (%)
E10	2.187	2.161	1.182
E20	2.336	2.324	0.526
E30	2.473	2.473	0.024
E40	2.600	2.611	0.447
E50	2.706	2.735	1.062
E60	2.820	2.852	1.133
E70	2.919	2.961	1.403
E80	2.989	3.023	1.128
E90	3.102	3.168	2.084
E100	3.141	3.197	1.765

Friction

Figure 2 shows the FMEP calculated with Yagi, Ishibasi, and Sono's [30] high speed friction model for wide open throttle conditions in an F1 engine at different operating speeds in comparison to the standard Chen-Flynn friction model. The calculated FMEP has a steadily inclining, near linear curve. The Chen-Flynn plot, whilst not as smooth, also increases somewhat linearly between 4000 – 10500rpm, and at first matches very closely to the high speed model, but remains constant between 11000rpm – 15000rpm which is unrealistic. This demonstrates that the Chen-Flynn model is not suitable for predicting friction in this engine at high speeds and is the reason for using Yagi, Ishibasi, and Sono's [30] high speed friction model for the F1 engine.

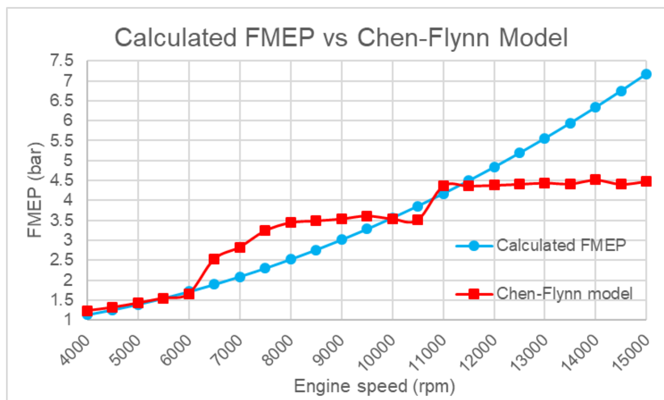

Figure 2: FMEP vs Engine speed using Yagi, Ishibasi, and Sono's [30] model for high speed engine friction prediction and the Chen-Flynn friction model.

Cylinder working temperature

The maximum service temperatures of aluminium and iron alloys are ~920K and ~1670K respectively [32]. Therefore, to achieve a greater efficiency an iron-based alloy was selected in this study. Article 5.5 specifies that the overall mass of the power unit must be a minimum of 150kg and the pistons have a minimum mass of 300g [9], meaning that there is not an advantage to be gained in using the lower density aluminium as the components will be optimised to meet the same minimum mass.

Turbulent flame characteristics

Three parameters in the GT-Suite model are used to define the flame's turbulent characteristics: Flame kernel growth multiplier, turbulent flame speed multiplier and Taylor length multiplier.

Figure 3 shows experimentally obtained plots for MFB by Qi and Lee [33]. Overlaid on the plots from literature are plots obtained from GT-Suite F1 engine simulation. The engine geometric and operating parameters were obtained and modelled as per the specification in Table 2 and the peak cylinder pressure and temperature were matched to the experimental findings. The 3 turbulent parameters were adjusted until the 10%, 50% and 90% MFB points matched the experimental findings very closely, as these are the defining points which determine the in-cylinder flame characteristics and therefore engine performance. The matching MFB curve, peak cylinder pressure and temperature indicate the turbulent flame characteristics of the fuel has been correctly modelled and these characteristics can be applied to the F1 engine.

The process of combustion environment replication was implemented to match the 10%, 50% and 90% MFB points for both E10 and E20 fuels. It was found that the turbulent characteristics used for the E10 fuel were also applicable for E20 and produced results

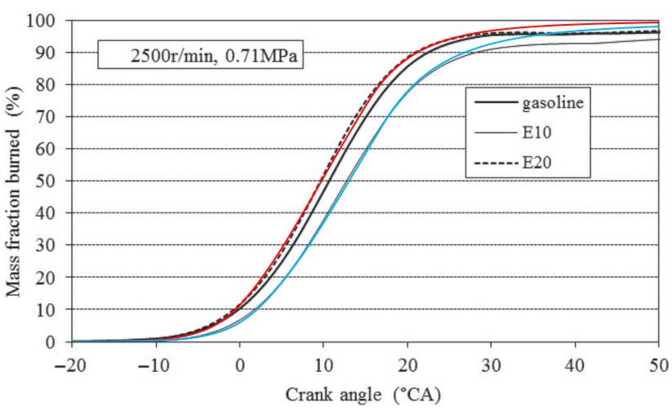

Figure 3: GT-Suite model vs literature plots for MFB against crank angle [33]. Blue trace is for E10, red trace is for E20.

The same turbulent characteristics used for E10 and E20 fuel blends were also implemented for E30 – E100 and produced the MFB plot shown in Figure 4. The results matched very closely with those observed by Oh, Bae and Min [41] and Worm, Michalek, and Naber [42], where greater fuel ethanol content results in faster flame speed and shorter MFB periods.

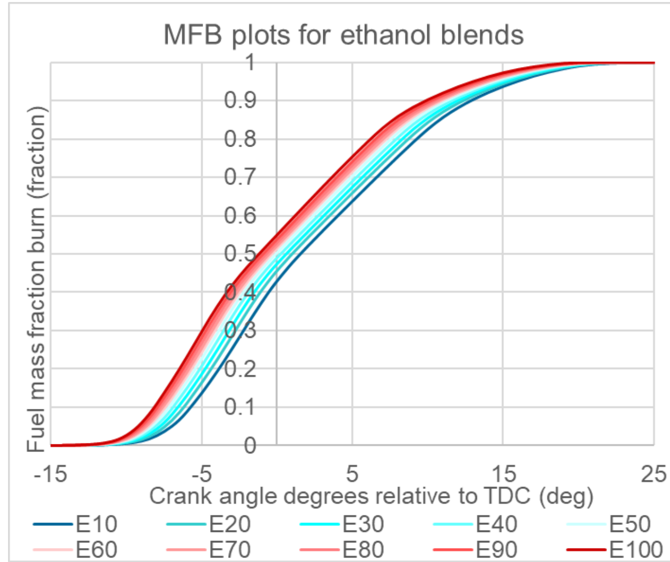

Figure 4: MFB plots for ethanol fuel blends E10 – E100.

Lean burning conditions

Figure 5 shows the F1 engine indicated efficiency for E10 – E100 fuels at a flow rate of 100kg/h from lambda 1 – 2. For all fuel blends aside from E100 there is an upward trajectory in efficiency until lambda 1.5 where E10 – E80 peak at 54% indicated efficiency whilst E90 and E100 dip below 53%. The efficiency dips up to 2% for lambda 1.6 for all cases except for E100 and continues to decrease until lambda 2. This is due to the fuels reaching their lean limit and not being able to combust all the injected fuel beyond lambda 1.6.

The brake power plot follows a similar trend to the efficiency plot, where power rises until a peak at lambda 1.5 or 1.6 and rapidly decreases beyond the peak when lean limits are reached. The highest power achieved is 582kW with E10 fuel at lambda 1.6. The power decreased relatively uniformly with increasing ethanol content in the fuel with ~20kW lost per 10% ethanol content gained at equal air-fuel ratios.

Table 4 shows the peak brake power and brake efficiency achieved, with the optimum operating point for all fuel blend types is between lambda 1.5 and 1.6. In 2026, cars are required to use 100% renewable fuel and a greater electric power output of 350kW compared to the current 120kW permitted [3][5]. E100 fuel produces a peak power of 337kW at lambda 1.6, which when combined with the required 350kW electric power for 2026 results in a total of 687kW. The current E10 peak of 601kW in addition to the 120kW currently required results in a total of 721kW which is very similar to the 2026 output. This means similar power can be achieved in the new regulations with environmentally less harmful fuel types.

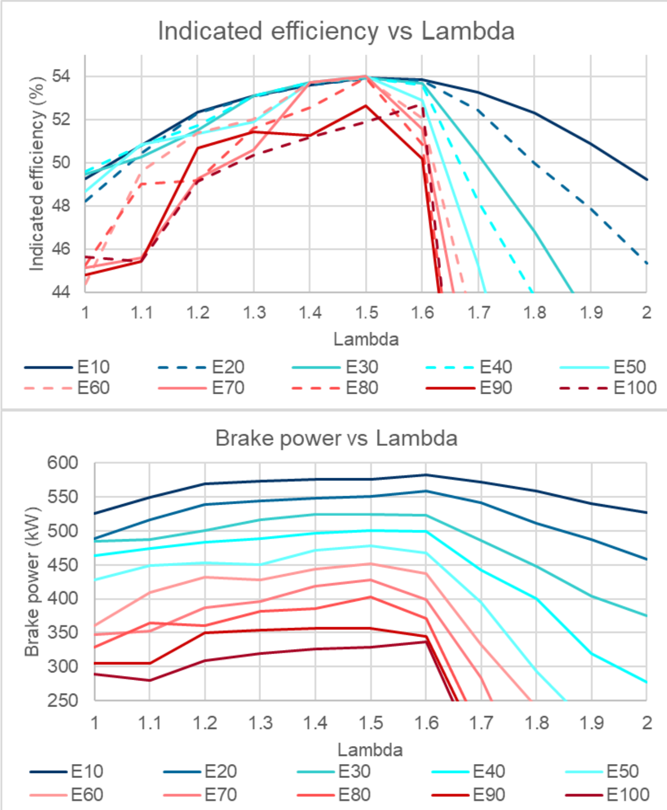

Figure 5: Indicated efficiency and brake power vs lambda for E10 – E100 fuel blends.

Table 4: Peak brake efficiency and brake power and their corresponding lambdas for fuel blends E10 – E100.

Fuel	Peak brake efficiency (%)	Lambda for peak brake efficiency	Peak brake power (kW)	Lambda for peak brake power
E10	50.7	1.6	600.8	1.6
E20	50.5	1.6	576.7	1.6
E30	50.2	1.6	539.7	1.6
E40	49.9	1.6	515.8	1.6
E50	49.7	1.6	490.7	1.6
E60	49.4	1.6	467.9	1.6
E70	48.9	1.6	432.7	1.5
E80	48.5	1.6	407.2	1.5
E90	48.0	1.6	382.3	1.6
E100	45.5	1.6	337.3	1.6

Formaldehyde emissions analysis

Figure 6 shows the 4 measured formaldehyde emissions for E0 to E20 [35] and the resulting trendline made from these points. The red dots represent the predicted emissions for ethanol blends above E20 using the extended trendline. This was used to estimate the total formaldehyde emissions by the Golf GTI for all fuel blends and the emission rates in g/kWh as shown in Table 5. The LA92 cycle average power requirement for the Golf GTI was calculated to be 1883W.

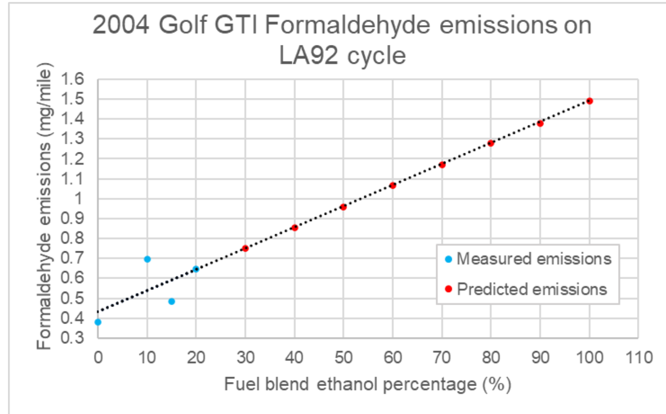

Figure 6: Formaldehyde emissions from experimental findings for E0 – E20 [35] and extrapolated formaldehyde emissions for E30 – E100.

Table 5: Golf GTI Formaldehyde emissions used to calculate F1 engine emissions.

Fuel blend	Formaldehyde emissions per unit distance (mg/mile)	Total formaldehyde emissions (mg)	Formaldehyde emissions rate (mg/kWh)
E10	0.695	7.673	8.456
E20	0.645	7.121	7.848
E30	0.750	8.28	9.125
E40	0.855	9.439	10.403
E50	0.960	10.598	11.680
E60	1.065	11.758	12.958
E70	1.170	12.917	14.235
E80	1.280	14.131	15.574
E90	1.380	15.235	16.791
E100	1.490	16.450	18.129

The pole laps of all 2022 dry qualifying sessions up to the Japanese Grand Prix weekend were used for the F1 engine emissions prediction. Wet qualifying sessions were disregarded as they are not representative of the vehicle performance due to lack of traction. Only the qualifying sessions at Imola, Canada and Britain featured wet final sessions, resulting in 15 valid lap times. The average engine speed for each circuit is shown in Table 6, alongside each circuit's length, pole time and power scaling factor. The average engine speed for all circuits ranges between 10233 – 11114rpm, with a mean speed of 10758rpm. This is expected as these speeds are very close to the 10500rpm, maximum fuel mass flow rate point which allows for the greatest engine performance. 13 pole laps had average engine speeds above 10500rpm, meaning that their average engine power can be assumed to be equal to the 10500rpm peaks calculated in GT-Suite. The average engine speeds of Monaco and Singapore were respectively a factor of 0.9965 and 0.9746 of 10500rpm, meaning the average power used over their respective laps can be assumed to be 99.95% and 97.46% of the peak power calculated in GT-Suite for each fuel blend. The formaldehyde emissions of all dry pole laps for all ethanol blends are shown in Figure 7.

Table 6: Average F1 pole lap engine speed and average power multiplication factor used to calculate average power used at each circuit.

Race	Distance (km)	Lap time (s)	Average engine speed (rpm)	Corresponding average power scaling factor
Bahrain	5.412	90.558	11006	1.00
Saudi Arabia	6.174	88.225	10723	1.00
Australia	5.278	77.868	10803	1.00
US	5.412	88.796	10768	1.00
Spain	4.675	78.75	10658	1.00
Monaco	3.337	71.376	10463	0.9965
Azerbaijan	6.003	101.359	10663	1.00
Austria	4.318	64.984	10872	1.00
France	5.842	90.872	10819	1.00
Hungary	4.381	77.377	10802	1.00
Belgium	7.004	103.665	10963	1.00
Netherlands	4.259	70.342	10709	1.00
Italy (Monza)	5.793	80.161	11114	1.00
Singapore	5.063	109.412	10233	0.9746
Japan	5.807	89.304	10779	1.00

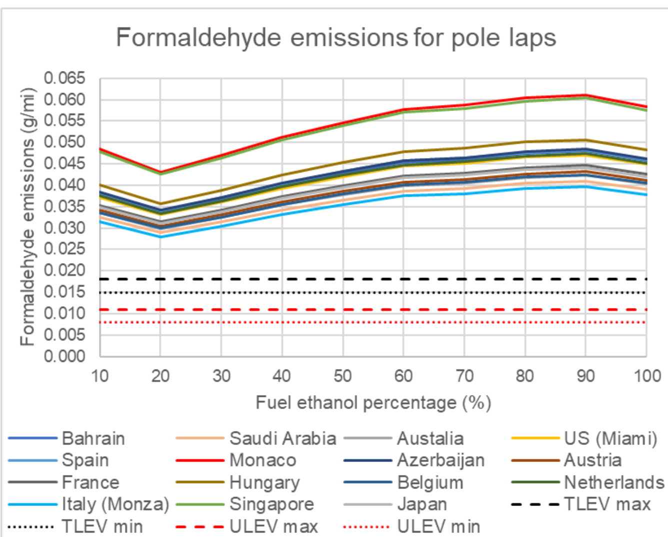

Figure 7: Formaldehyde emissions of F1 engine for 15 2022 season pole laps with fuel blends E10 – E100 alongside California emission standards: TLEV and ULEV.

The trend for all circuits is for formaldehyde emissions to decrease from E10 to E20 and then increase until E90 and decrease again at E100. The emissions behaviour roughly matches that of the projected emission rates of the Golf GTI as shown in Figure 6, but the rate of emissions does not linearly increase between E20 and E100, with emissions starting to decrease after E90. This is due to the emissions being directly related to the engine power output and time taken for the lap, as the conversion from the Golf GTI is in units: mg/kWh. The brake power decreases with increasing fuel ethanol content whilst the pole lap time in calculations stayed the same for each fuel blend. This is a realistic outcome because, as discussed previously, in the future regulations the electric motor component of the F1 power unit will dramatically increase in output power, resulting in similar overall brake power output, which theoretically should result in similar lap times with all else being equal.

The California standards for passenger vehicles and vehicles less than 3750lbs (1705kg) range between 0.008 – 0.015g/mi for 50,000 miles or 5 years and 0.011 – 0.018g/mi for 100,000 miles or 10 years [23]. The F1 engine produces a minimum of 0.028g/mi formaldehyde at Monza with E20 fuel and a maximum of 0.054g/mi at Monaco with E90 fuel. These results are caused by differences in average vehicle speed whilst using similar power outputs. The average speed at Monaco and Monza is 168.3km/h and 260.2km/h respectively, whilst the average power used differs by <1%. At the lowest emission rate – at Monza with E20 fuel, the F1 engine produced 55.6% greater formaldehyde emissions than the least stringent transitional low emission vehicle (TLEV) standard of 0.018g/mi. At the highest rate – at Monaco with E90 fuel, the F1 engine produced 200.0% greater formaldehyde emissions than this TLEV standard. The most stringent ultra-low emission vehicle (ULEV) standard is 0.008g/mi [23]. At Monza with E20 fuel, the F1 engine produced 250.0% greater formaldehyde emissions than the ULEV standard and at Monaco with E90 fuel, the F1 engine produced 575.0% greater formaldehyde emissions than this ULEV standard.

The emissions produced by the F1 engine are substantially higher than the California emissions standards for vehicles under 1705kg. Another factor to be considered is that the Golf GTI emissions used for scaling were measured post-catalyst, meaning that the subsequent F1 emissions were effectively calculated for an F1 engine equipped with a US car and light-duty truck Tier 2 Bin 8 standard catalytic converter (equivalent to California TLEV standards [43]). The F1 engine produced between 55.6 – 575.0% greater emissions than California standards allow, even when equipped with this catalytic converter.

Heavy-duty Otto cycle emissions standards (for vehicles >14,000lbs or 6364kg) require vehicles to produce less than 0.01 g/bhp·h formaldehyde emissions [19]. The F1 engine produces between 0.00585 – 0.01352 g/bhp·h formaldehyde depending on the ethanol fuel blend. This means that for heavy duty Otto cycle standards, fuel blends E10, E20, E30, E40, E50 and E60 would pass, but E70, E80, E90 and E100 would fail. Whilst the vehicle passes for low ethanol content fuels, Formula 1 vehicles are required to use fully sustainable fuels in the future [3], meaning that if ethanol fuel is used, eventually technical changes will still be required further down the line to ensure safety. A noteworthy point is that the heavy-duty Otto cycle emissions standards are for vehicles ~700% heavier than F1 cars. The fact that the F1 engine fails the emissions standards for high ethanol blends despite a huge vehicle mass advantage and the use of a catalytic converter only serves to further emphasise the magnitude of formaldehyde emissions F1 engines can produce and highlights the need for implementation of major technical solutions to ensure the high air quality for people at Formula 1 events.

Summary

It was determined that for ethanol fuel blends E10 – E100, the peak brake efficiency is reached at lambda 1.6, where for fuels E10, E20 and E30, brake thermal efficiencies of over 50% were achievable. The Peak brake power output is reached at lambda 1.6 for all ethanol blends apart from E70 and E80, where peak power is reached at lambda 1.5. There is a linear relationship between brake power and fuel ethanol content, where due to ethanol's lower calorific value in comparison to gasoline, brake power decreases with increasing ethanol content.

Aside from the E10 fuel blend, which produces greater formaldehyde emissions than E20, the trend of the F1 engine for each of the 2022 pole position qualifying laps is to increase in formaldehyde emissions with increasing fuel ethanol content until a peak is reached at E90, where emissions decrease beyond this point. When equipped with a US standard Tier 2 Bin 8 catalytic converter the F1 engine produced a minimum of 0.028g/mi formaldehyde at Monza with E20 fuel and a maximum of 0.054g/mi at Monaco with E90 fuel. Despite being equipped with a catalytic converter, in 2022 F1 qualifying sessions the F1 engine produced 55.6 – 200% greater formaldehyde emissions than the least stringent California TLEV standard for vehicles <1705kg and produced 250 – 575% greater formaldehyde emissions than the most stringent California ULEV standard for vehicles <1705kg. To maintain high power output with ethanol fuel in the new 2026 regulations, fuel mass flow rate will remain high and therefore formaldehyde emissions will remain high, meaning a more advanced exhaust treatment method is required to ensure air quality targets at F1 events.

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Acknowledgments

I would like to thank Stephen Samuel for the vast amount of support he gave me during every stage of the dissertation process. He has provided me with a wealth of knowledge and engineering expertise on racing engine systems and it was incredibly heart-warming that he made me feel that my work and understanding of the subject mattered to him. I would like to thank my close friends: Ewan Little, Mark O'Brien, Ole Ramming and Beth Tinsley, and my family for providing me with support when I was experiencing personal issues. I am in a much better position now than I would be without these people in my life. Finally, I would like to thank Duncan Menzies for Acknowledging me in his acknowledgments.

Definitions/Abbreviations

$\mathbf{B_m}$	Maximum laminar flame speed
$\mathbf{B_\Phi}$	Laminar speed roll − off value
$\mathbf{\Phi}$	In − cylinder equivalence ratio
$\mathbf{\Phi_m}$	Equivelence ratio at maximum speed
$\mathbf{T_u}$	Unburned cylinder gas temperature
$\mathbf{T_{ref}}$	Reference temperature = 298K
$\boldsymbol{\alpha}$	Temperature exponent
$\mathbf{P}$	Unburned cylinder gas pressure
$\mathbf{P_{ref}}$	Reference pressure = 101325Pa
$\mathbf{x_i}$	Fuel species mole fraction
$\mathbf{S_{L,i}}$	Fuel species laminar flame speed
$\mathbf{\Phi_1}$	60×10^{-9}
$\mathbf{\Phi_2}$	1.1×10^{-9}
$\mathbf{\Phi_3}$	0.0011
$\mathbf{\Phi_4}$	0.14
$\mathbf{V_s}$	Displacement of single cylinder
$\mathbf{N_e}$	Engine speed
$\mathbf{Z}$	Effective valve opening area
$\boldsymbol{\nu}$	Lubricant kinematic viscosity
$\mathbf{S}$	Stroke
$\mathbf{B}$	Bore
$\mathbf{K_c}$	Cylinder number coefficient
$\mathbf{D_{cj}}$	Crank journal diameter
$\mathbf{D_{cp}}$	Crank pin diameter
$\mathbf{m}$	number of crank journals
$\mathbf{n}$	number of crank pins
$\boldsymbol{\eta_t}$	Transmission efficicency
$\boldsymbol{\rho}$	Air density
$\mathbf{C_d}$	Drag coefficient
$\mathbf{A}$	Vehicle frontal area
$\mathbf{V}$	Vehicle speed
$\mathbf{m}$	Vehicle mass
$\mathbf{g}$	Acceleration due to gravity
$\mathbf{f + KV^2}$	Rolling coefficient
$\mathbf{p}$	Tyre pressure
$\mathbf{v}$	Vehicle speed