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The Effect of E100 Water Content on High Load Performance of a Spray Guide Direct Injection Boosted Engine

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ABSTRACT

Ethanol as a renewable fuel is employed in either the hydrated or anhydrous states. The production of anhydrous ethanol requires an additional and costly processing step, and is less advantageous with regard to Life Cycle Inventory. The use of hydrated ethanol may then be preferred for high blend and pure fuels, and future engine technologies designed for ethanol may need to accommodate either form.

In the current study a spark ignited ethanol direct injection (EDI) turbocharged engine, proposed for efficient delivery of high specific output, is evaluated for performance at high load with anhydrous and hydrated ethanol as fuel. Test data show the EDI engine may be operated at high load on either fuel with the same output and efficiency. The key differences arising from fuel water content are reduced burn rate requiring advance in ignition timing, a decrease in engine emissions of NOx and increase of HC, and higher potential for increase of compression ratio and output.

INTRODUCTION

It is generally acknowledged that sustainable alternatives to crude oil derived fuels should be deployed to address issues of the environment, and longevity and security of energy supply. Ethanol is widely considered as a renewable alternative fuel although in many markets there remain concerns on both the supply and demand sides regarding viability and level of benefit.

These concerns may in part be addressed through demand side vehicle technologies. Especially, overall engine operating efficiency may be significantly enhanced by utilising the particular properties of ethanol as a fuel [1-8]. Such approaches by necessity require the use of high content including pure ethanol fuels, either within a dedicated or dual-fuelling strategy, and typically depend upon a combination of increased compression ratio and specific output in relation to a spark ignited gasoline engine.

High ethanol content fuels may employ ethanol in the anhydrous or hydrated azeotropic states, whilst due to phase separation only the former is utilised for ethanol blended with gasoline at supplementary levels [9]. Ethanol is manufactured in either state although the anhydrous form typically requires an additional and costly processing step to remove water [9]. In addition, the Life Cycle Inventory is adversely affected for anhydrous ethanol [10]. Hydrated ethanol fuel is at present used in Brazil [11] and Sweden [12], and offers potential for use in other markets.

Early stage development of engine technology targeted for use with ethanol may therefore consider the effect of water content in the fuel. Water may be considered primarily as an inert diluent, and may present both challenges and opportunities. On one hand the required mass delivery increases to achieve the same energy input, and the challenge for fuel preparation is greater. On the other hand the charge cooling effect will be higher and charge dilution will be increased. Whilst alternative fuel deployment is evolving worldwide, consideration of such issues may enable improved alignment between supply and demand sides for the transportation sector.

This study is an experimental evaluation of engine performance in response to varying water content of ethanol fuel. In particular, the test engine is an SI multi-cylinder turbocharged unit incorporating centrally mounted spray guide direct injection. This is a configuration with potential for very high specific output, and an enabler for an aggressive engine down-sizing strategy. The study builds on evaluation of output potential conducted previously [8], and is focussed on high load operation. Evaluation of compatibility and durability, effect on lubrication, emission speciation and cold start capability are subjects for further studies.

LITERATURE REVIEW

The potential effect of water content on performance of ethanol fuel has been noted for at least 100 years [13], and the performance and efficiency gains that may be achieved with the introduction of water to a spark ignited internal combustion engine for somewhat longer [14]. Studies which directly evaluate the effect of ethanol or alcohol water content on operating characteristics of a spark ignited engine are, however, limited. In the following text attention is given to studies conducted for spark ignited engines which evaluate the effect of water as a charge diluent.

Heywood [15] has documented the effect of diluent addition on SI engine operation with particular reference to recirculation of exhaust gas (EGR). Emissions reduction of NOx and increase of HC, combined with reduced burn rate and increased efficiency at part load are noted. In particular the reduction of NO is ascribed to reduction in peak combustion temperature arising from the diluent heat capacity, allowing some degree of comparison with water. Recently, several researchers have investigated the dilution of boosted engines at high load with either cooled EGR or excess air in order to suppress knock [16-18]. However, ethanol exhibits no knock in the current application, and it is considered the characteristics of water and these diluents are somewhat different, hence these sources are not considered further.

With regard to addition of water, various researchers have conducted studies from the late 1960's driven primarily by the requirement to reduce emissions of NOx [19-23], and these studies are of most interest. The use of water as a diluent is somewhat less practical than that of EGR and hence these studies are limited in number. Some recent work for water addition has been directed at control of abnormal combustion and NOx emissions in hydrogen fuelled internal combustion engines [24], although the combustion of hydrogen differs somewhat from that of ethanol and gasoline so these sources are not considered further. Other studies focusing on either internal engine cooling [25] or knock suppression for low octane fuels [26] are less pertinent and also not considered further.

Nicholls et al. [19] conducted a theoretical and experimental study on a single-cylinder CFR engine at a compression ratio of 8.5:1. The engine was operated at WOT, varying equivalence ratio, and ignition timing at 28°BTDC; engine speed is unclear but believed to be at 1100 rpm. Water was injected continuously as liquid into the induction pipe at quantities varying from 0 to 1.25:1 water/fuel ratio (WFR). The work showed a maximum NO reduction ~90% with the trend directly related to quantity of water added, and also a moderate improvement in efficiency. The latter finding differs from other studies.

Quader [20] conducted an experimental study on a propane fuelled single-cylinder Waukesha engine at a compression ratio of 10.0:1. The engine was operated at engine speed of 1600 rpm, ~50kPa MAP, 0.98 equivalence ratio, and ignition timing at either 27.5°BTDC or MBT. Water was supplied as steam into the intake system at up to 12.5% by volume of mixture, and other inert diluents were evaluated: CO_2, nitrogen, exhaust gas, helium and argon. The work showed a maximum reduction in NO emissions ~90% at 27.5°BTDC which was less at MBT, with reduction directly related to the heat capacity of the diluents. Also demonstrated for water were reduced efficiency at fixed spark timing, similar efficiency at MBT, and increased HC emissions.

Lestz et al. [21] conducted an experimental study on an iso-octane fuelled single-cylinder CFR engine at a compression ratio of 9.0:1. The engine was operated at an engine speed of 900 rpm, WOT, varying equivalence ratio and ignition timing at either 30°BTDC or MBT. Water was directly injected into the cylinder at up to 1.5 WFR, and with injection angles of 340, 45, 25 and 10°BTDC. The work showed a maximum reduction of ~90% for NO at ignition 30°BTDC which was reduced at MBT. The rate of reduction is indicated to be greater than for manifold addition of water, and whilst NO reduces directly with water quantity at early injection timings, the relationship is skewed with compression stroke injection. Water injection does not affect CO, increases HC, and reduces power at fixed ignition timing which could be recovered at MBT.

Peters et al. [22] conducted an analytical and experimental study, the latter involving both a gasoline fuelled single-cylinder CFR engine at a compression ratio of 8.0:1, and testing in vehicle. The engine was operated at an engine speed of 1200 rpm, 80 kPa MAP, varying equivalence ratio and ignition timing at MBT. Water was added in the form of emulsion with fuel at up to 40% by weight, and through addition to the intake manifold. The work showed the method of water addition had no effect, other than that of the emulsifying agent. Water addition increased knock limit, advanced MBT ignition timing, reduced emissions of NO, increased emissions of HC, and left emissions of CO relatively unchanged. At MBT, water addition was also observed to have minimal effect on efficiency.

Harrington [23] conducted an experimental study on an indolene fuelled single-cylinder Waukesha engine at a compression ratio of 8.0:1. The engine was operated at engine speeds of 600, 1000 and 1500 rpm, 50~110kPa MAP, varying equivalence ratio, and varying ignition timing. Water was supplied to the intake system in the form of either injected liquid or steam. The work showed the effect of liquid water to be greater than that of steam, indicating a contribution from latent heat of vaporisation. Water addition increased knock limit, slowed burn rate, advanced MBT ignition timing, and decreased efficiency at higher flow rates. Emissions of NO were significantly reduced by water addition, and more so with liquid water. Effects on HC and CO were small, the HC result differing somewhat with other studies.

The Brazilian experience with hydrated ethanol is extensive [11]. Dedicated E93h vehicles were developed and used widely from the late 1970's onwards, although declined in popularity in the late 1980's for a variety of reasons. A new wave of development in Flex Fuel Vehicle technology offers significant promise for the widescale take up of new vehicles which use E93h alongside Brazilian gasoline [27-30]. For high content fuels Flex Fuel powertrain programs have typically been developed around hydrated ethanol, whilst research activity outside Brazil has typically focussed on anhydrous ethanol [1-8], and engine studies have not considered the relative performance of the fuels.

SUMMARY – It is found that little literature exists for spark ignition engines directly comparing the performance of anhydrous and hydrated ethanol as a fuel. The literature is extensive with regard to dilution by exhaust gas, and includes some useful sources regarding the addition of water. These sources refer to experimental data collected with hydrocarbon based fuels, with manifold or port injection of fuel, at lower engine speed and with non-boosted operation. In conclusion, it is considered that a study offering data in this area will be of some value.

EXPERIMENTAL APPARATUS

ENGINE - The turbocharged direct injection engine is described in a previous study [8]. For reference the specification is shown in Table 1.

Table 1 – Engine Specification.

Engine	In-line 4 Cylinder
Bore	86 mm
Stroke	86 mm
Displacement	500 cc / cylinder
Number of valves	4
Compression ratio	10.4:1
Fuel System	Air-assist Direct Injection
Pressure Charging	Garrett M53
EVO (ATDC$_f$)	139
EVC (ATDC$_f$)	375
IVO (ATDC$_f$)	351
IVC (ATDC$_f$)	582

The direct injection system employed is a centrally mounted spray guide type, and utilises air to assist in preparation and delivery of the fuel. The system is known in the market and is described more extensively in previous work [8, 31].

FUEL – Anhydrous and three blends of hydrated ethanol were evaluated in this study, the hydrated blends being produced by addition of de-mineralised water to E100. The fuels are shown in Table 2 together with water for reference. The specification of anhydrous ethanol is given in the Appendix.

Table 2 – Fuel Specification.

	E100	E93h	E87h	E80h	H_20
Ethanol by Mass (%)	100	93.5	87.0	80.0	0
Water by Mass (%)	0	6.5	13.0	20.0	100
AFR (:1)	9.00	8.41	7.83	7.20	---
LHV (MJ / kg)	26.8	25.0	23.3	21.4	---
Latent Heat[1] (kJ/kg)	855	n/a	n/a	n/a	2260

[1] Latent heat of vaporisation for blends not cited.

TEST PROCEDURE – The test program and reporting of results has been divided into three main sections, each involving an experimental comparison of ethanol fuelled engine operation as follows.

1. Effect of Ignition Timing: evaluation of varying ignition timing at constant speed and intake manifold pressure.

2. Effect of Load: evaluation of varying intake manifold pressure at constant speed and MBT ignition timing.

3. Effect of Speed: evaluation at varying engine speed at fixed engine output with MBT ignition timing.

Note that stoichiometric fuelling was maintained under all conditions.

RESULTS & DISCUSSION

EFFECT OF IGNITION TIMING – To illustrate the typical effects of ignition timing, data are presented at an engine speed of 2000 rpm, lambda 1, and an intake manifold pressure of 100 kPa. Anhydrous ethanol and three different compositions of hydrated ethanol are examined.

Combustion Characteristics – Figure 1 illustrates the effect of ignition timing on engine behaviour with four levels of hydration, and shows a typical response for each fuel as ignition timing is varied. At fixed ignition timing, the increasing addition of water reduces engine output, and with ignition later than MBT reduces engine efficiency and combustion stability. The MBT ignition timing advances as water content increases. At MBT, output reduces slightly as water content is increased, as a function of reduced efficiency and also airflow for E80h.

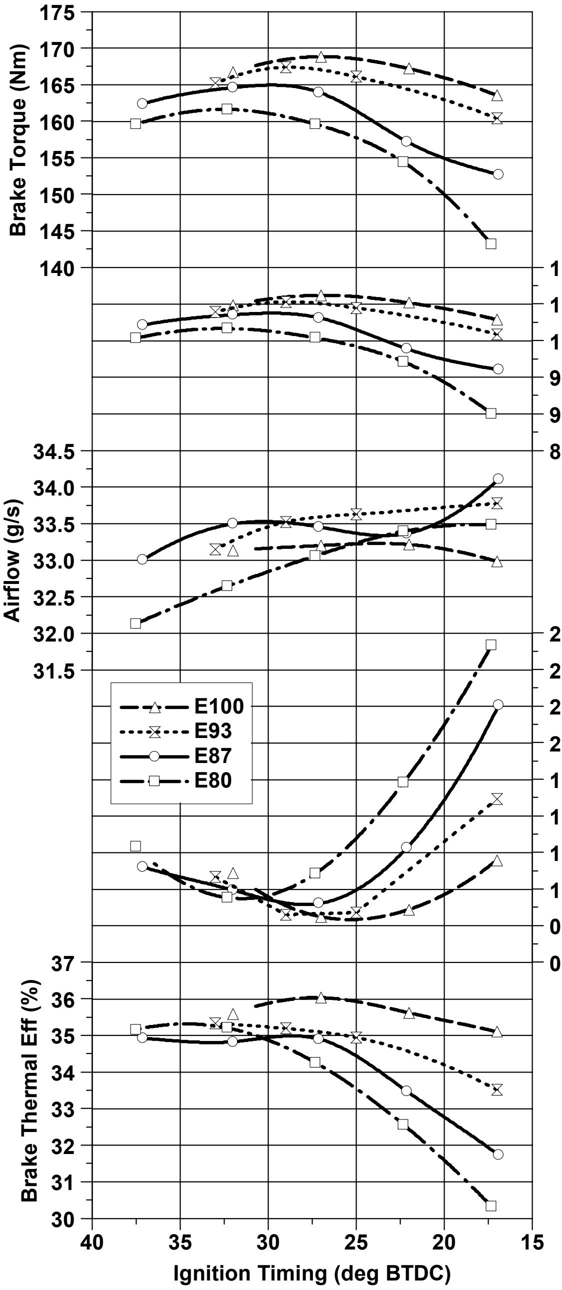

Figure 1 – Engine Output, Airflow, Combustion Stability and Efficiency at 2000 rpm & 100kPa MAP.

The trends for output and efficiency are generally consistent with those in the literature as observed for non-oxygenated fuels, for both manifold addition [20, 22, 23] and direct injection of water [21]. An exception is the work of Nicholls et al. [19], in which a reduction of BSFC with increase of water / fuel ratio at fixed ignition timing was both observed and predicted. The reason for this discrepancy is not pursued here.

The reduction in efficiency at fixed ignition timing may be attributed to a reduction in burn rate and also combustion stability, as would be anticipated with the addition of a diluent [15]. Reduction in burn rate results in later combustion and an advance of MBT ignition timing, required to achieve optimum combustion phase. Combustion stability typically falls within the 1% threshold desired for ideal engine operation, including E80h at MBT. Further details regarding burn rate are presented in the following Figures.

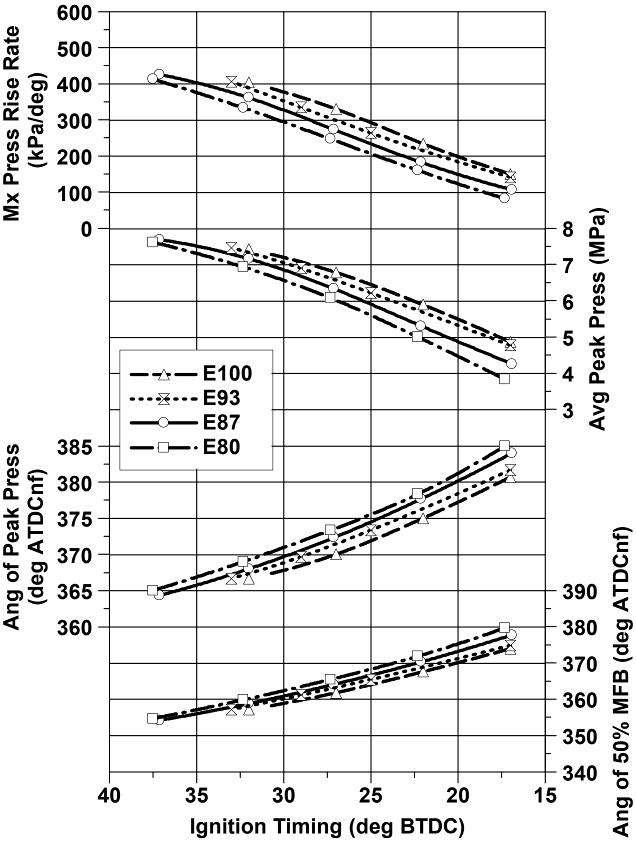

Figure 2 – Cylinder Pressure Metrics & Combustion Phasing at 2000 rpm & 100kPa MAP.

Figure 2 clearly illustrates the effect of ignition timing and water content on some selected cylinder pressure metrics. The advance of ignition timing advances combustion, yielding a higher peak pressure, rate of pressure rise and earlier peak of pressure. At fixed ignition timing the addition of water lowers peak pressure and the maximum rate of pressure rise, and also retards combustion.

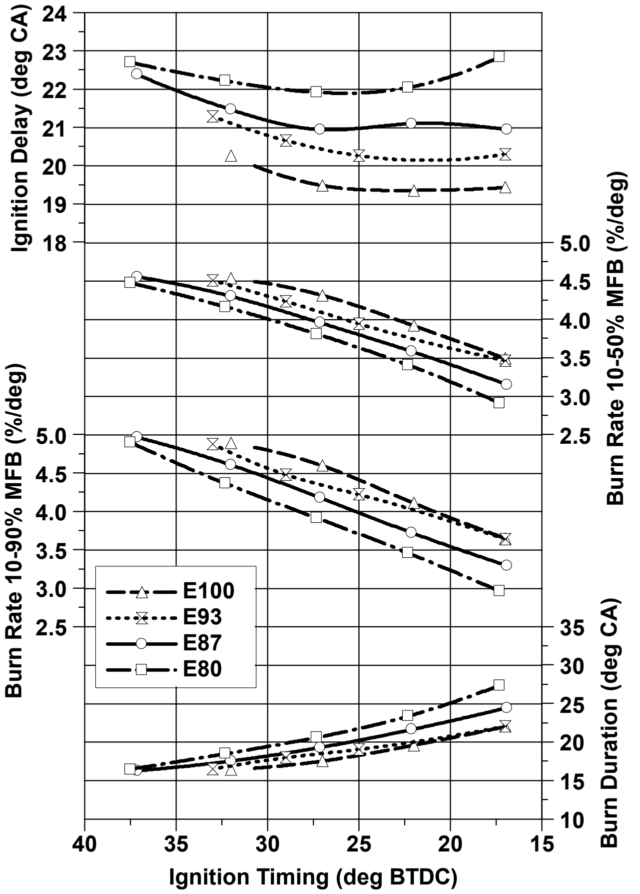

Figure 3 – Burn Rate at 2000 rpm & 100kPa MAP.

Figure 3 shows more detail concerning burn rate. The ignition delay is defined as the period from ignition until 10% MFB, and the burn duration from 10% until 90% MFB. At fixed ignition timing the addition of water increases ignition delay, suppresses burn rate (more so after 50% MFB), and in consequence increases burn duration. Note that at MBT ignition timing an increased burn rate would be achieved yielding a similar burn duration, however this is accompanied by an increase in ignition delay. The literature is not extensive with regard to quantitative effect of water addition on burn rate, however where this has been reported by Harrington [23] the results are consistent.

For reference Figure 4 illustrates burn profiles for different fuels at peak torque. At this condition the MFB profile, and cylinder pressure rise rate and peak, become consistent for the fuels, albeit with some scatter in phasing. It is observed that the apparent optimum ignition timing at this condition correlates to slightly advanced angles of 50% MFB, and results from the iterative process needed to achieve a target MAP on a boosted and throttled engine. The gradients around MBT are small, and a slight advance of phase is determined to have negligible impact within the context of this study.

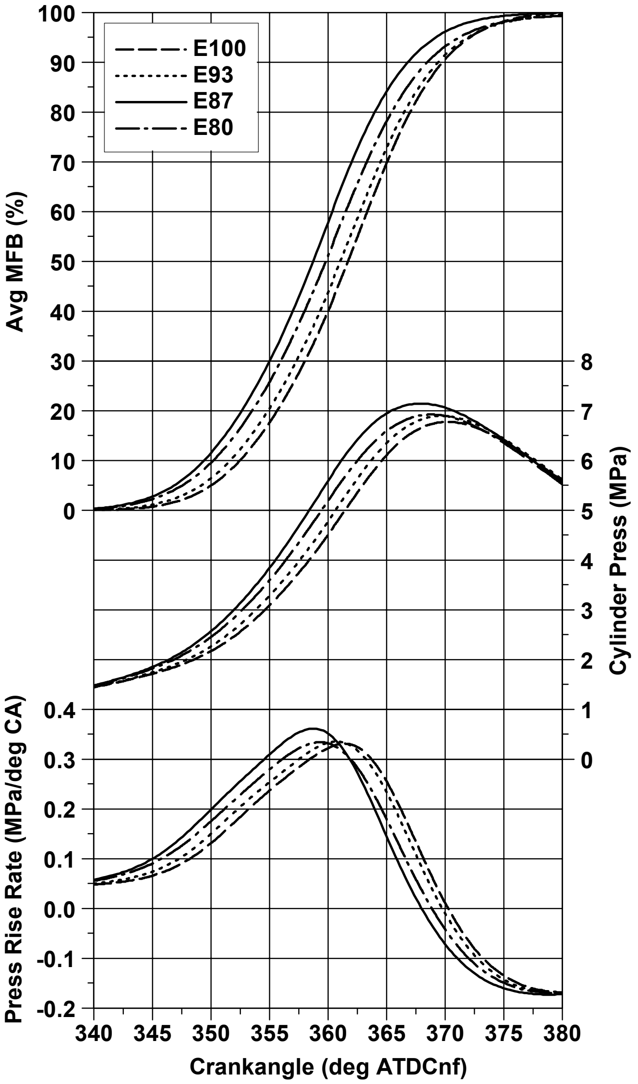

Figure 4 – Maximum Torque Burn Profiles at 2000 rpm & 100kPa MAP.

Exhaust Gas Temperature – With reference to Figure 5, it may be seen that the exhaust gas temperature follows an anticipated trend with ignition timing for all fuels. Note that the lower temperature measured at the exhaust port is primarily attributed to the highly unsteady flow at this point [32]. A difference between the fuels is mostly apparent with retarded ignition timing, under which conditions an increase in water content results in a slight increase in exhaust temperature. It is apparent at MBT that the exhaust temperature may be the same or less for fuels of higher water content.

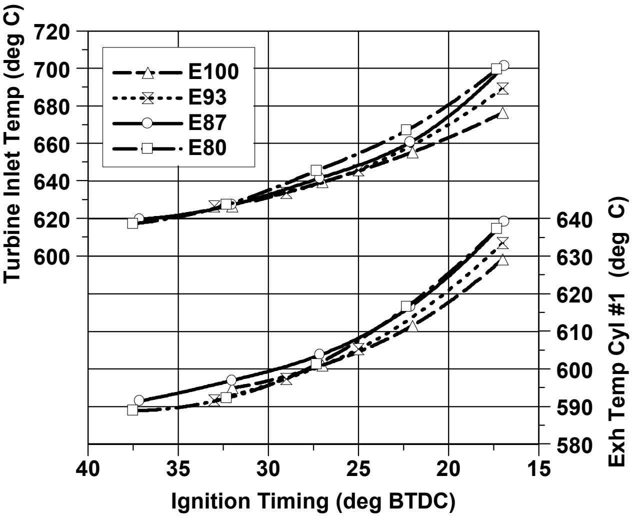

Figure 5 – Exhaust Gas Temperature at 2000 rpm & 100kPa MAP.

These trends are in general agreement with the literature where exhaust gas temperature is reported [20, 21, 23]. In particular Lestz et al. [21] utilised direct injection of water, and observed that intake stroke injection at 340°BTDC yielded exhaust temperature increase whilst compression stroke injection at 55°BTDC and later gave a reduction. An earlier direct injection timing would be expected to correlate most closely with results observed for manifold injection reported elsewhere, as is the case. For the tests described here an injection timing 320°BTDC was employed.

At fixed ignition timing the increase in exhaust temperature may be attributed primarily to a delay in combustion, and is consistent with reduction in efficiency. At MBT, optimum combustion phase is recovered and exhaust temperatures are similar for the fuels. Where a reduction in temperature is noted at increased water addition, this may be attributed to reduced combustion temperatures arising from increased dilution and heat capacity of the charge, and also the latent heat of vaporisation.

Exhaust Emissions – Figure 6 illustrates the response observed for HC, CO and NOx emissions with variation in ignition timing and fuel type. Emissions of CO are consistent at approximately 0.5% with both variation in water content and ignition timing. At fixed ignition timing, emissions of hydrocarbons increase as water content increases, and decrease as ignition timing is advanced. Emissions of NOx decrease with increasing water content at fixed ignition timing, and increase as ignition timing is advanced.

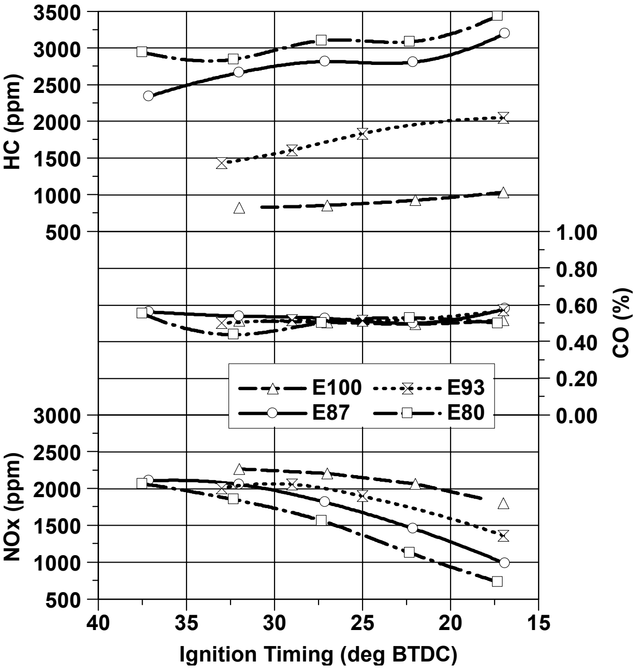

Figure 6 – Exhaust Emissions at 2000 rpm & 100kPa MAP.

The gross trends in emissions are typical of the effects of dilution and ignition timing [15]. The mechanism of CO formation is clearly insensitive to water addition at lambda 1 and entirely consistent with the literature where reported [21-23], therefore is not discussed further. The trends for both NOx and HC emissions deserve further comment, and to assist interpretation data are shown as a function of water flow rate in Figures 7 and 8.

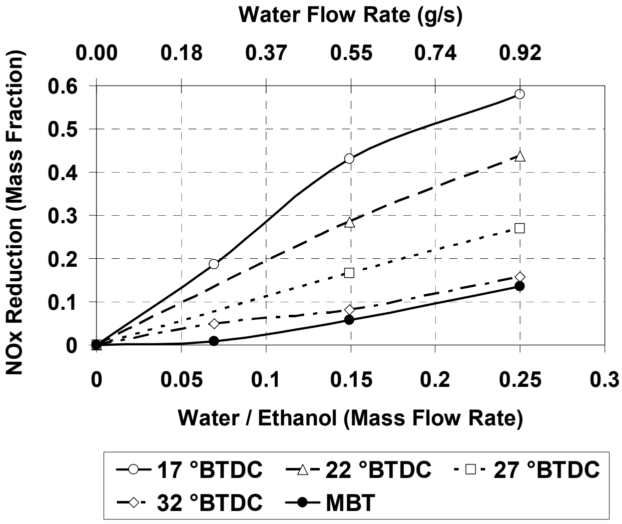

Figure 7 – NOx Response at 2000 rpm & 100kPa MAP.

Figure 7 illustrates reduction of NOx emissions in response to increasing water content, at a range of ignition timing conditions. Reduction in NOx appears directly related to the quantity of water added, and is higher at more retarded ignition timing. The reduction observed at MBT is somewhat less, due to the increase of NOx as MBT advances with increasing water content.

NOx reduction may be attributed to the decrease in peak combustion temperatures resulting from the specific heat capacity of the added diluent, and to some extent the additional charge cooling resulting from the diluent transitioning to vapour phase. The trends observed are consistent with the literature [19-23] and occur within the same scale of change, indicating that the mechanism of NOx reduction is similar between ethanol and non-oxygenated fuels. Certain researchers [22, 23] have noted a different response according to the method of water introduction to the engine, and as no previous study has examined the simultaneous direct injection of water and fuel it may be expected that the value of further comparison is limited.

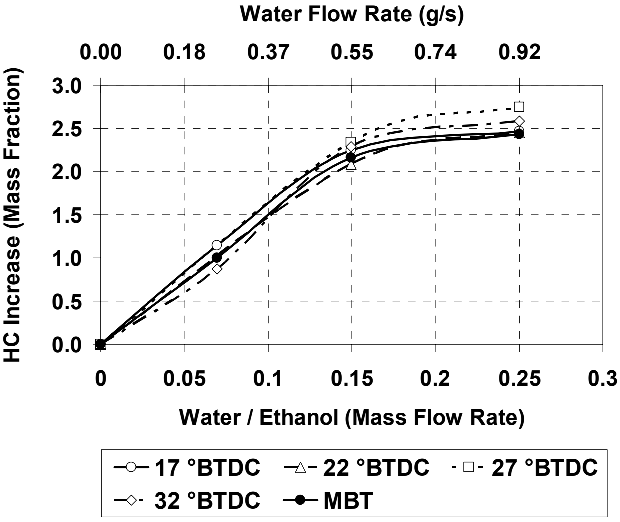

Figure 8 – HC Response at 2000 rpm & 100kPa MAP.

Figure 8 illustrates the increase of HC emissions in response to increasing water content, at a range of ignition timing conditions. HC emissions increase linearly with water content from E100 to E87h, and then stabilize to some degree. The response of HC emissions to water content is unaffected by ignition timing. The trends reported by other researchers are varied, ranging from minimal effect [23] to moderate and high [20-22], although in no instance was a reduction reported. A direct comparison with these results is believed to have limited value due to the absence of a clear trend, and the relative sensitivity of HC formation to the different engine and fuel system configurations employed.

Hydrocarbon emissions in SI engines with premixed fuel and air are considered to arise from four mechanisms [15]: (i) flame quenching, (ii) crevice filling, (iii) absorption to and desorption from the oil layer, and (iv) incomplete combustion. In this instance the complexity is increased with the use of direct injection, which adds the potential for direct wetting of the chamber surfaces and some degree of in-homogeneity. Whilst a detailed analysis is therefore beyond the scope of this study some limited comment may be made.

The trend is unrelated to changes in combustion stability and efficiency (Fig.1), indicating that incomplete combustion is not a major contributor. The higher injector flow rate associated with increase of water content is also not considered to be an issue, as increase of fuel flow for higher load does not produce the same response (Fig.12). It is also important to note that exhaust gas temperature is relatively unchanged (Fig.5) implying a similar level of post-combustion oxidation, although increased dilution may offset this. It is considered that the trend relates to an increase of quenching, and also the effect of fuel water content on preparation within the cylinder.

Summary of Ignition Timing Effect – In this sub-section the effect of ignition timing and ethanol water content has been observed on combustion behaviour, exhaust gas temperature and exhaust emissions at a speed of 2000 rpm and a MAP of 100 kPa. It has been found that increasing water content reduces burn rate, output and efficiency, and advances the MBT ignition timing. At MBT the output and efficiency are virtually recovered. Increasing water content at fixed ignition timing reduces emissions of NOx whilst increasing those of HC, and produces a nominal increase in exhaust gas temperature. At MBT the reduction in NOx is less, the increase in HC basically unaffected, and exhaust gas temperature may exhibit a nominal reduction. In the following sub-sections all data are presented at an ignition timing of MBT.

EFFECT OF LOAD – To illustrate the typical effects of load, data are presented at an engine speed of 2000 rpm, lambda 1, MBT ignition timing, and intake manifold pressures of 100, 140 and 170 kPa. Anhydrous ethanol and two different compositions of hydrated ethanol are examined.

Combustion Characteristics – Figure 9 illustrates the effect of engine load on engine operation with three levels of ethanol hydration. At 100 and 140 kPa engine operation is essentially consistent. At 170 kPa, a reduction of output in the order of 2% is observed from E100 to E93h, and a further reduction of 2% from E93h to E87h. The reduction for E93h may be attributed to reduced airflow, suggested to arise from a displacement of air during induction by increased water content. Lestz et al. [21] identified a reduction in output at MBT associated with direct injection of water on the intake stroke that was not observed with compression stroke injection. The further reduction in output for E87h may also be attributed to reduced efficiency (Fig.10).

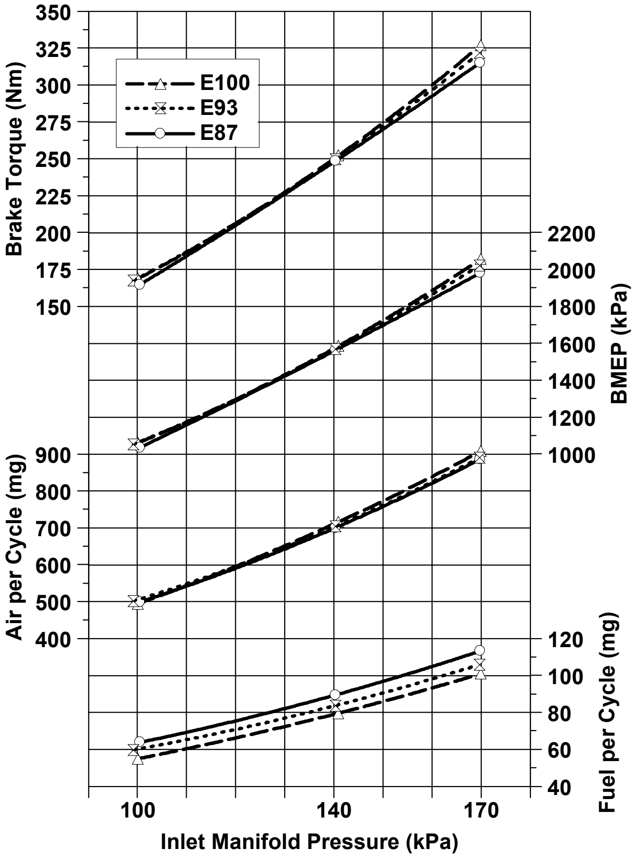

Figure 9 – Engine Output, Airflow and Fuel Flow at 2000 rpm & Various MAP.

Figure 10 illustrates a reduction in exhaust gas temperature with increasing water content, as described in the previous sub-section. Also illustrated is combustion stability, typically falling within 1% COV although showing a slight tendency to increase with water content at highest load. The response of efficiency to water content is somewhat inconsistent as load varies although shows a trend to increase with increasing load, indicative of the reducing impact of parasitic losses at higher indicated output.

Figure 11 illustrates an increase in cylinder peak pressure and rise rate associated with increase of cylinder charge mass. Other trends in rise rate and peak pressure appear primarily associated with combustion phasing rather than water content. With angle of 50% MFB between TDC and 7°ATDC it is possible to achieve MBT at all conditions tested, and it is evident that no knock constraint applies at any level of hydration.

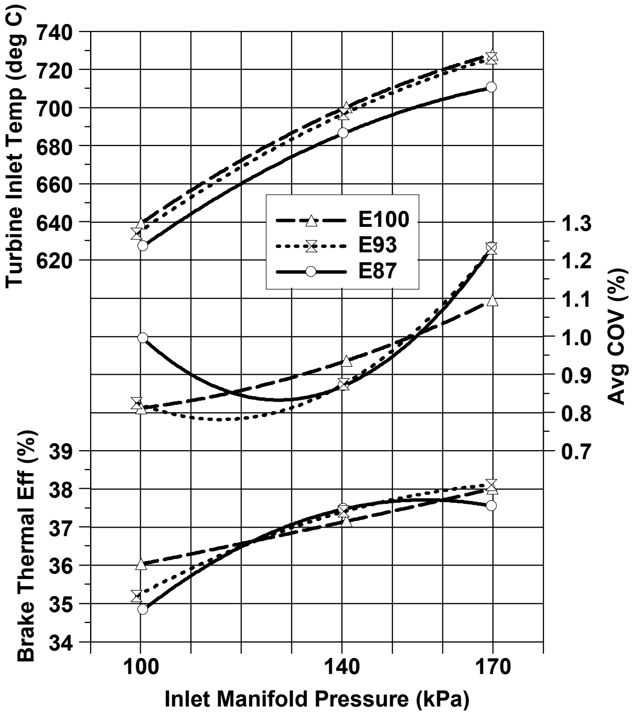

Figure 10 – Exhaust Temperature, Combustion Stability and Efficiency at 2000 rpm & Various MAP.

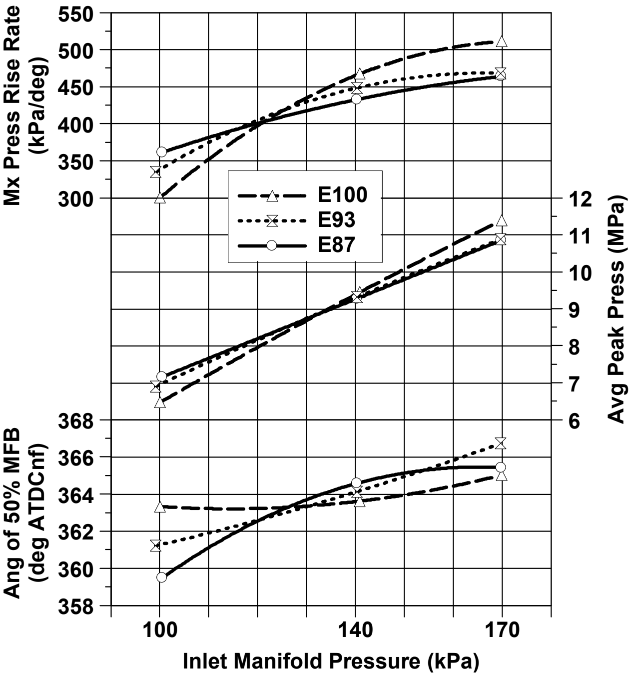

Figure 11 – Cylinder Pressure Metrics and Combustion Phase at 2000 rpm & Various MAP.

Some studies, however, have demonstrated knocking with ethanol at higher compression ratios [1, 33], and it is proposed that the knock suppression observed with water addition [22, 23, 26] would enable the use of higher compression ratio and boost pressure than for anhydrous ethanol. In addition it was noted at higher loads that increasing water content was an effective suppressor of pre-ignition, a load limiting phenomenon observed for anhydrous ethanol in a previous study [8].

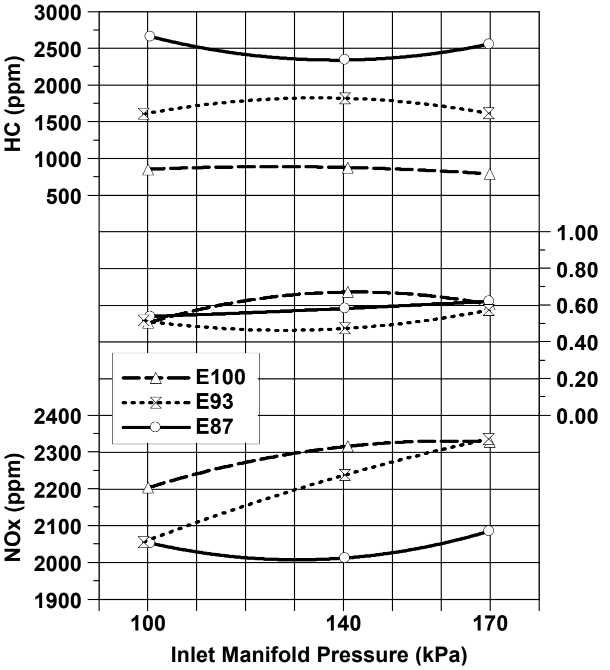

Figure 12 – Exhaust Emissions at 2000 rpm & Various MAP.

Figure 12 illustrates the response of exhaust emissions to variation in load and fuel water content. CO increases slightly with load but is unaffected by water content as noted in the previous sub-section. HC is affected by water content similarly at higher loads, and is somewhat independent of load. NOx at MBT decreases marginally with water content as previously noted, and the trend is somewhat independent of load.

Summary of Load Effect – In this sub-section the effect of engine load and ethanol water content has been observed on combustion behaviour, exhaust gas temperature and exhaust emissions at a speed of 2000 rpm. It has been found that an increase of load has little effect on the response due to water content of the fuel, excepting a slight reduction of 2% in output for E93h and 4% for E87h at the highest load of 170 kPa MAP. Whilst performance is not limited at this compression ratio and output, it is proposed that the suppression of knock and pre-ignition offered by hydration may present the greatest opportunity for extension of the engine operating regime. In the following sub-section data are presented at a fixed output.

EFFECT OF SPEED – To illustrate the typical effects of speed, data are presented at a target torque of 300Nm, lambda 1, MBT ignition timing, and engine speed between 2000 and 5000 rpm. Anhydrous ethanol and one composition of hydrated ethanol are examined.

Combustion Characteristics – Figure 13 illustrates engine output and efficiency with variation in speed. Brake torque is constant at 300Nm, equating to a BMEP of approximately 1900 kPa. Brake thermal efficiency declines for both fuels as engine speed increases. At 2000 and 3000 rpm there exists no clear trend between the fuels, whilst at 4000 and 5000 rpm it is observed that E100 is at least 1% BTE greater than E93h. An explanation may be presented in terms of burn rate, as below.

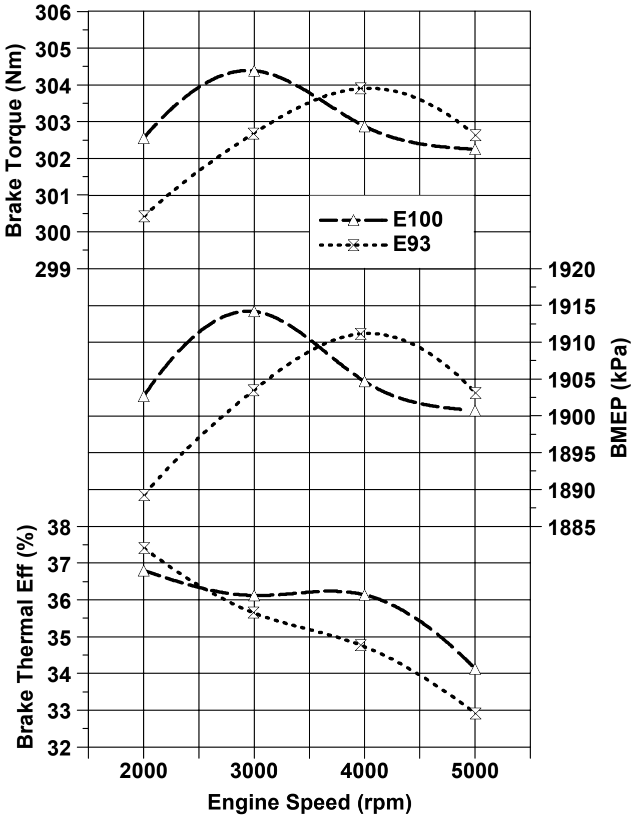

Figure 13 – Engine Output vs. Speed

Figure 14 illustrates the variation in MBT ignition timing with engine speed. For both fuels, advance is required with increase of speed, although the advance for E93h is greater by ~5°CA at 2000 rpm and increases to ~7°CA at 5000 rpm. Lambda is maintained at 1 throughout the test, and to achieve the target torque at reduced efficiency it is necessary to increase airflow for E93h at 4000 and 5000 rpm. The exhaust temperatures follow trends observed previously up to 4000 rpm, whilst at 5000 rpm the temperature of E93h exceeds that of E100, indicating the relative loss in efficiency.

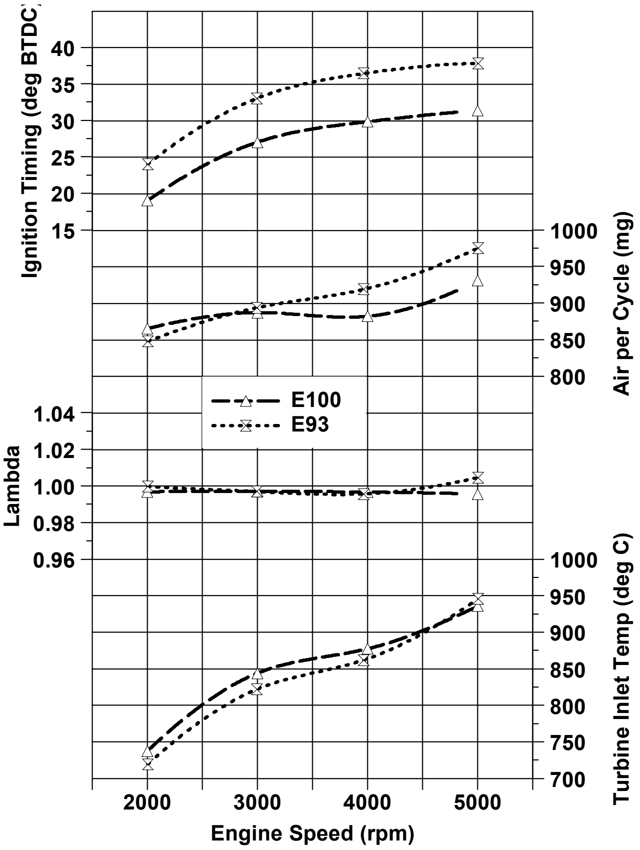

Figure 14 – Ignition Timing, Airflow, Lambda and Exhaust Temperature vs. Speed

Figure 15 presents burn data for the two fuels. Peak pressure is consistent between the fuels at 3000 rpm and above. As anticipated, ignition delay and burn duration increase for both fuels with increase in engine speed. Relatively, E93h exhibits both greater ignition delay and burn duration than E100. At 2000 rpm E93h exhibits an ignition delay ~2°CA higher than E100, and at 5000 rpm ~7°CA higher. Burn duration is similar at 2000rpm but is greater for E93h at 4000-5000 rpm by ~3°CA. In consequence the overall burning angle for E93h is greater by ~10°CA at 4000-5000 rpm. Noting that combustion phase is near optimal for E93h and E100 at these speeds, the longer duration of E93h combustion is held primarily responsible for the reduction in efficiency. An analysis of this mechanism would form the basis of a further study.

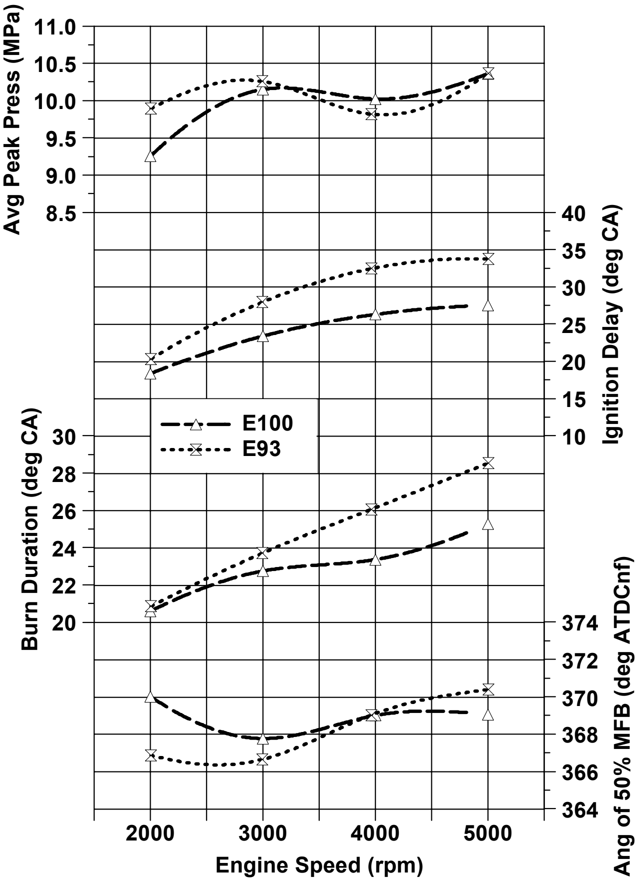

Figure 15 – Burn Data vs. Speed

Summary of Speed Effect – In this sub-section the effect of engine speed and ethanol water content has been observed on combustion behaviour at a BMEP of 1900 kPa. It has been found that an increase of speed has some effect on the response due to water content of the fuel, such that at 4000-5000 rpm E93h demonstrates a reduction of 1% BTE relative to E100.

SUMMARY

Comparative testing of anhydrous (E100) and hydrated (E93h, E87h, E80h) ethanol fuels has been completed on a direct injection multi-cylinder turbocharged engine at a compression ratio of 10.4:1. Evaluation was conducted at high load, and initially at 2000 rpm and manifold pressure of 100 kPa to assess variation in ignition timing. Subsequent testing at 2000 rpm evaluated increase in manifold pressure to 140 and 170 kPa. Finally the effect of engine speed was assessed at a BMEP of 1900 kPa.

The key findings of the work are typically in agreement with the literature [15-23], and may be summarized as follows.

1. Ignition delay and burn duration are increased with increasing water content at fixed ignition timing, as a consequence of charge dilution. Engine output, efficiency and combustion stability are decreased and MBT ignition timing is advanced.

2. Engine output, efficiency and combustion stability are typically recovered at MBT ignition timing. Some reduction remains at engine speeds of 4000-5000 rpm, and load of 1900 kPa BMEP and above.

3. Emissions of CO are unaffected by fuel water content.

4. Emissions of NOx decrease linearly with increasing fuel water content at fixed ignition timing, as a function of diluent specific heat and consequent reduction of peak combustion temperatures. At MBT ignition timing the reduction is typically less than 10%.

5. Emissions of HC increase linearly with increasing fuel water content for E93h and E87h, the trend being largely independent of ignition timing. The mechanism is proposed to be an increase of flame quenching, and also the effect of water content on fuel preparation within the cylinder.

6. Exhaust gas temperature increases slightly with increasing fuel water content at fixed ignition timing, as a consequence of later combustion. The increase is in the order of 20°C. At MBT, increasing water content may reduce EGT in the order of 10°C. This is attributed to reduced combustion temperatures arising from increased heat capacity of the charge, and also the latent heat of vaporisation.

7. MBT ignition timing was achieved at all conditions tested and with all levels of fuel hydration. Further increases in boost pressure and compression ratio are therefore feasible, and it is proposed that the suppression of knock and pre-ignition offered by hydration may present the greatest opportunity for extension of the engine operating regime.

CONCLUSIONS

A direct-injected turbocharged multi-cylinder engine may be operated at high load with the same output and efficiency, with either anhydrous or hydrated ethanol. The key differences arising from fuel water content are reduced burn rate requiring advance in ignition timing, increased fuel mass flow rate, a decrease in engine emissions of NOx and increase of HC, and higher potential for increase of compression ratio and output.

When reviewing powertrain applications for anhydrous vs. hydrated ethanol fuels, key areas of difference may include fuel preparation, catalyst specification and control system calibration. Items not addressed within this study but also requiring consideration include compatibility and durability, lubrication, cold start capability, and fuel system capacity.

When considering the tank to wheel emissions inventory, anhydrous and hydrated ethanol will yield similar emissions of CO_2 as indicated by efficiency. At the tailpipe and when warm, emissions of CO and NOx may be relatively unaffected by water content, whilst emissions of HC may increase.

Further work in support of this area is on-going, and includes development of low temperature starting capability, and turbo-charger application development for transient performance and high specific output.

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ACRONYMS & ABBREVIATIONS

AFR: Air Fuel Ratio

Ang: Angle

ATDC$_{f}$: After Top Dead Centre (firing)

ATDC$_{nf}$: After Top Dead Centre (non-firing)

Avg: Average

BMEP: Brake Mean Effective Pressure

BTDC$_{f}$: Before Top Dead Centre (firing)

BTDC$_{nf}$: Before Top Dead Centre (non-firing)

BTE: Brake Thermal Efficiency

CA: Crank Angle

CFR: Cooperative Fuels Research

CO: Carbon Monoxide

CO$_2$: Carbon Dioxide

COV: Coefficient of Variation (of IMEP)

deg: Degree

DI: Direct Injection

E100: Ethanol at 100% concentration

E80h / E80: 80% ethanol + 20% water by mass

E87h / E87: 87% ethanol + 13% water by mass

E93h / E93: 93% ethanol + 7% water by mass

EDI: Ethanol Direct Injection

Eff: Efficiency

EGT: Exhaust Gas Temperature

EGR: Exhaust Gas Recirculation

EVC: Exhaust Valve Closing

Exh: Exhaust

H$_2$0: Water

HC: Hydrocarbon

IMEP: Indicated Mean Effective Pressure

Inl: Inlet

IVO: Intake Valve Opening

LHV: Lower Heating Value

MAP: (Intake) Manifold Absolute Pressure

MBT: Minimum advance for Best Torque

MFB: Mass Fraction Burned

Mx: Maximum

NO: Nitric Oxide

NO$_2$: Nitrogen Dioxide

NOx: Nitrogen Oxides

Press: Pressure

rpm: revolutions per minute

SI: Spark Ignition

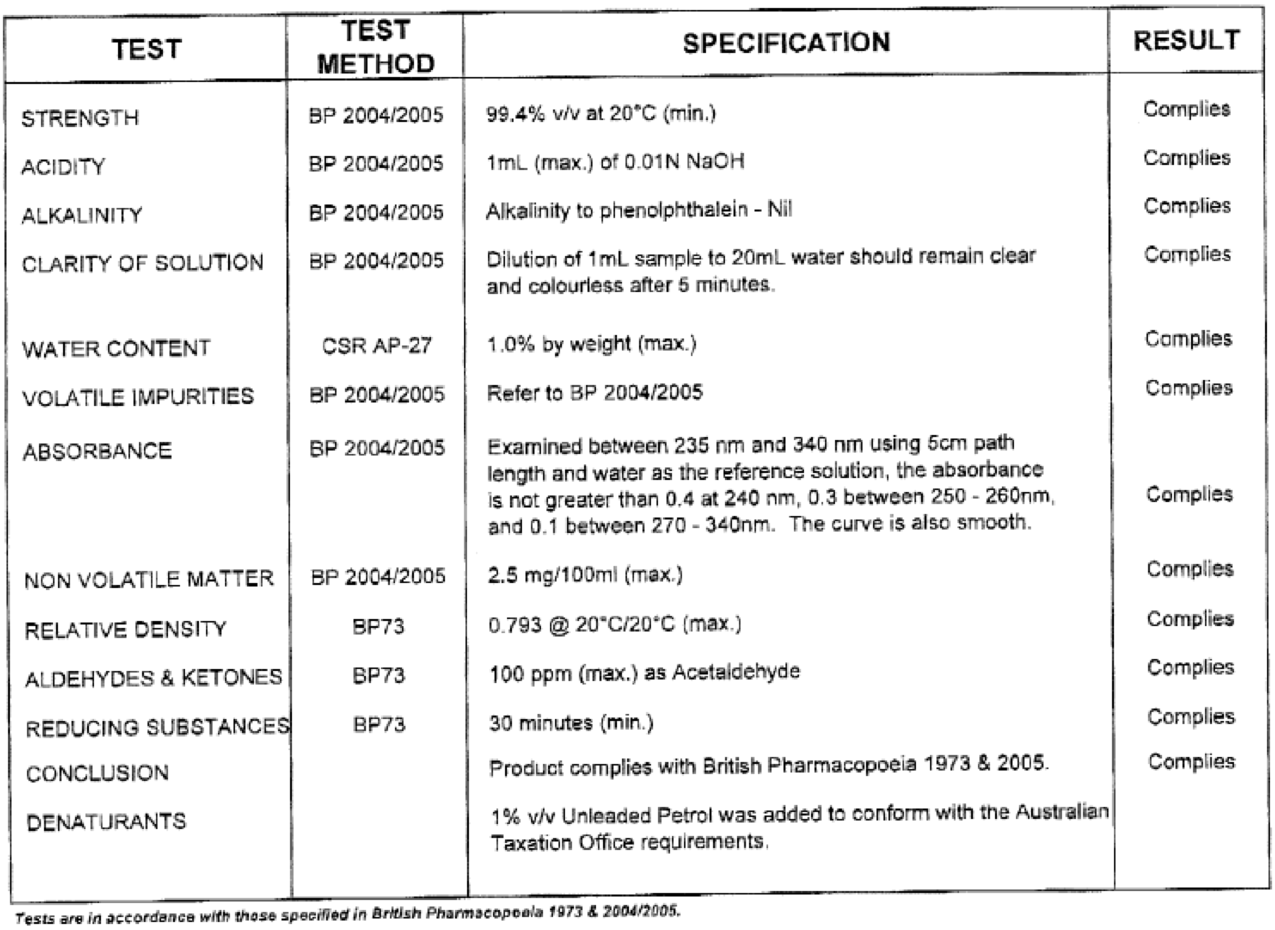
TDC: Top Dead Centre

Temp: Temperature

WFR: Water Fuel Ratio

WOT: Wide Open Throttle

APPENDIX: ETHANOL SPECIFICATION



TEST	TEST METHOD	SPECIFICATION	RESULT
STRENGTH	BP 2004/2005	99.4% v/v at 20°C (min.)	Complies
ACIDITY	BP 2004/2005	1mL (max.) of 0.01N NaOH	Complies
ALKALINITY	BP 2004/2005	Alkalinity to phenolphthalein - Nil	Complies
CLARITY OF SOLUTION	BP 2004/2005	Dilution of 1mL sample to 20mL water should remain clear and colourless after 5 minutes.	Complies
WATER CONTENT	CSR AP-27	1.0% by weight (max.)	Complies
VOLATILE IMPURITIES	BP 2004/2005	Refer to BP 2004/2005	Complies
ABSORBANCE	BP 2004/2005	Examined between 235 nm and 340 nm using 5cm path length and water as the reference solution, the absorbance is not greater than 0.4 at 240 nm, 0.3 between 250 - 260nm, and 0.1 between 270 - 340nm. The curve is also smooth.	Complies
NON VOLATILE MATTER	BP 2004/2005	2.5 mg/100ml (max.)	Complies
RELATIVE DENSITY	BP73	0.793 @ 20°C/20°C (max.)	Complies
ALDEHYDES & KETONES	BP73	100 ppm (max.) as Acetaldehyde	Complies
REDUCING SUBSTANCES	BP73	30 minutes (min.)	Complies
CONCLUSION		Product complies with British Pharmacopoeia 1973 & 2005.	Complies
DENATURANTS		1% v/v Unleaded Petrol was added to conform with the Australian Taxation Office requirements.	

Tests are in accordance with those specified in British Pharmacopoeia 1973 & 2004/2005.