

# the relationship between automotive diesel fuel characteristics and engine performance

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1. SUMMARY AND CONCLUSIONS

The oil industry and motor manufacturers, are currently facing a number of issues associated with diesel engines for road transport:

- (a) The growth in the use of diesel vehicles/fuels is raising concern about the level of particulate emissions and the possible effects of diesel exhaust on public health.
- (b) Further, the trend towards conversion in oil refineries and changing crude diets will alter the nature of the components used to blend diesel fuels. These changes in composition/characteristics could affect gaseous, particulate and noise emissions from diesel engined vehicles.
- (c) The automotive manufacturing industry is confronted with increasingly severe restrictions on exhaust emissions and engine noise.

This report discusses the influence of diesel fuel characteristics on emissions and performance, with particular emphasis on the effect of changing fuel properties on exhaust emissions. However, owing to the known highly complex nature of the interaction between engines and diesel fuel and the fact that more than one fuel property can be linked to emissions, the report discusses individual emissions species in conjunction with the range of fuel properties known to influence those emissions.

In order to compile this report, CONCAWE studied over 40 published papers, from various sources, and also included data from as yet unpublished oil industry research. The results recorded here have been obtained from current production engines and it is possible that engines designed to meet more stringent emissions legislation will exhibit much lower variability in their emissions performance. Nevertheless, existing differences in engine design and operating conditions have a significant influence so that, whilst emissions trends can be identified, it is not possible to quantify fuel effects absolutely.

Some examples of the beneficial/adverse impact of key fuel properties on engine performance are given below:

- Reduced cetane number can increase  $\text{NO}_x$ , gaseous HC, HC smoke and noise emissions; however, increased cetane number might have an adverse effect on particulates, CO and black smoke emissions due to the reduced air/fuel mixing prior to ignition.
- Increased viscosity can result in higher  $\text{NO}_x$  emissions but will decrease particulates.

- Lowered general volatility levels will increase NO<sub>x</sub> emissions. A more volatile front end will increase gaseous HC emissions. However, increasing mid boiling points will increase HC smoke.
- Increased density (e.g. via higher aromatic contents) will effect particulates and exhaust smoke unfavourably unless the maximum fuel delivery is correctly set. However, at the same time, increased density levels will lower the volumetric fuel consumption in a diesel engine.
- Increased n-paraffin content results in higher cetane number and improved ignition, but decreases low temperature operability. During cold starting of a diesel vehicle, cranking and white smoke tend to increase with reduced cetane number. Cold starting tests show that engine design is an important factor.
- Satisfactory low-temperature operability can be provided by the fuel CFPP specified for individual marketing areas. Proper vehicles' fuel system design providing good low-temperature operability can bridge untypical occurrences of abnormally cold weather conditions and thus avoid overly restrictive fuel low-temperature flow limits.
- Excessive pintle nozzle fouling has occasionally been reported. The precise mechanism for this phenomena is unclear but it is thought to possibly result from unfavourable injector design or environment and/or extremely unstable fuels. Fouling causes increase noise, hydrocarbon emissions and/or white smoke.

These examples demonstrate that the relationship between fuel properties and engine performance necessarily result in conflicting requirements, depending upon individual engine design. This explains why changes in the properties of diesel fuels have only a small influence on exhaust emissions, particularly when compared with the influence of current engine design and operating conditions. In practice the differences in emissions obtained with commercial fuels will be even smaller than those indicated in this paper since other performance requirements have to be satisfied.

The Task Force concluded that there is considerable literature available which describes and explains most aspects of the effect of engine design and fuel on the emissions and performance on diesel engines. This wealth of information is sufficient to establish an adequate technical position on this subject at this time, although growing interest in particulate emissions may stimulate the need for further investigations.

2.

OBJECTIVE

- (a) To prepare an internal briefing paper summarising current knowledge of the relationship between diesel fuel characteristics and emissions.
- (b) To identify any further work necessary to establish an adequate technical position on the subject.

3. NITROGEN OXIDES

Nitrogen oxides are formed in combustion systems by the reaction of nitrogen and oxygen at high temperatures. Clearly, since the maximum emission from diesel engines is significantly lower than the equilibrium concentration expected from combustion temperatures of 1500-2000°C, equilibrium is, in fact, never attained. Indeed, there is evidence that there is insufficient time available for equilibrium to be reached in a reciprocating internal combustion engine. In addition, there may be little oxygen available in the reaction zone. There is also evidence that the rate of quenching due to expansion by piston motion is sufficient to freeze the reaction at whatever value of nitrogen oxides has been attained. In view of the mechanism of formation of nitrogen oxides we would not expect fuel properties to have much influence on their formation. On the other hand, it would be expected that engine design could greatly affect the formation of nitrogen oxides, mainly through the fuel-air mixing process.

The influence of fuel properties on nitrogen oxide emission was examined (3) in engine tests using nine fuels having varying cetane numbers and volatility, including ignition-improved fuels. In these tests fuel volatility was found to have no effect on nitrogen oxides emission, but decreasing the cetane number from 54 to 44 gave up to 20% increase in nitrogen oxides emission. In engines equipped with distributor-type pumps an effect of viscosity was observed, a low-viscosity fuel giving substantially reduced emissions compared with a high-viscosity fuel. This was shown to be due to the effect of viscosity on fuel injection timing (i.e. retarded injection timing reduces nitrogen oxides emission)

Similar trends are reported in References (1 and 4). In this latter work, a shift of ten cetane numbers (52 to 42) increased the percentage of nitrogen oxides emissions by an amount similar to that established in (3) above. The influence of ignition quality on nitrogen oxides emissions has also been observed in Reference (5) but not in Reference (6). Bench engine tests (2) suggest that variations in  $\text{NO}_x$  are linked more closely to aromatic content than the cetane number (ignition delay), since using ignition improvers had no effect.

Although References (3 and 6) did not find any influence of volatility (i.e. higher 85% point) on nitrogen oxides emissions, such trends are reported in (1) in which less volatile fuels gave lower nitrogen oxides. It is believed that this is because the more involatile fuels restrict the effectiveness of pre-mixing during the delay period. However, this 'advantage' is in direct conflict with the influence of volatility on the production of white smoke following cold start (see Section 7).

Differences in specific rates of NO<sub>x</sub> emissions in excess of 140% were observed (7) between two direct injection turbocharged engines operating on the same fuel in the US 13 mode test. More recent work (8) showed threefold differences in NO<sub>x</sub> emissions (measured in g/kw.h) between six naturally aspirated engines operating on one fuel in both the US Federal and European 13 mode test (see Appendix). Further, NO<sub>x</sub> emissions from DI engines are more sensitive than from IDI engines over a greater part of the load/speed map.

The evidence provided by all references leads the Task Force to conclude that changes in fuel properties have only a minor influence on the emission of nitrogen oxides when compared with the influence of engine design and operating conditions.



4. CARBON MONOXIDE

Carbon monoxide is formed during the combustion of hydrocarbons when insufficient oxygen is available for complete combustion to carbon dioxide. In an automotive diesel engine, where there is no throttling of the aspirated air and, where even at full load there is approximately 20% excess air available, the diesel engine produces only low concentrations of carbon monoxide. The carbon monoxide that is generated is a measure of the incompleteness of mixing of the fuel and air during combustion.

In view of this, it might be expected that the fuel properties which influence the mixing and combustion of the fuel, e.g. volatility and cetane number, might also influence the emission of carbon monoxide. The results of tests (3) on a direct injection engine indicated that, at higher speeds, there was no significant difference in carbon monoxide emissions between two fuels having cetane numbers and mid-boiling points of 54/268°C and 43/190°C respectively. At idling speeds however, the lower cetane, more volatile fuel gave a mean reduction in emissions of about 15%. In another direct-injection engine, increasing the cetane number of a fuel from 42 to 60 by an ignition improver additive gave similar increases in the carbon monoxide emission at low loads but gave reduced emissions at high load. These results indicate that fuel ignition quality can influence carbon monoxide emission and that in some cases, the higher the cetane number (that is, the shorter the ignition delay), the higher the carbon monoxide emission. This is in line with later comments on black-smoke emission. It is to be expected that carbon monoxide and black smoke emissions from the diesel engine are related because of their dependence on the completeness of mixing and combustion.

However, engine design and operating conditions have a major influence on carbon monoxide emission, and in tests conducted (3) with one fuel on six different engines the emissions were found to range from 0.014-0.05% volume at low load to 0.09-4.5% volume at maximum load. Moreover, it would appear that engine design and operating conditions also influence the response to fuel properties in respect of carbon monoxide emissions. Driving an indirect injection passenger car on a cycle similar to ECE 15 indicated that carbon monoxide emissions were not influenced by fuel volatility, as expressed by the mid-boiling point. Such emissions were, however, influenced to a minor extent by cetane number. In this case the fuels of lower ignition quality (45 versus 58 cetane number) gave the higher emissions. This trend was also noted in References (4 and 8).

These results are in agreement with other findings. The Task Force concludes that the inherently very low levels of carbon monoxide emissions are only marginally affected by fuel properties when compared with the influence of engine design and operating conditions. (For example see Appendix).

5. HYDROCARBONS

Some important hydrocarbon emissions - white smoke, blue misfire smoke and particulates - will be dealt with in Sections 7, 8 and 9. In the following paragraphs we will only discuss gaseous emissions. There are indications that both cetane number and fuel volatility influence hydrocarbon emissions. References (1, 2, 3 and 5) suggest that lowering cetane number increases hydrocarbon emissions. The effect is non-linear so that lowering cetane number has only a marginal effect on hydrocarbon emissions until a threshold value is reached. This value is engine dependent but is estimated to lie in the low 40's for most current production engines in Europe (for example reference 1).

References (3 and 4) indicate that fuels with a more volatile front end (i.e. approaching U.S. levels) may lead to increased hydrocarbon emissions particularly at light load. Reference (4) also suggests that higher final boiling point fuels increase hydrocarbon emissions. This trend with less volatile fuels has been confirmed for direct injection engines by some recent (unpublished) work conducted by one member of the Task Force but, in the same study, no correlation could be found between volatility and hydrocarbon emissions for indirect injection engines. However, other work (2) has indicated similar volatility effects for both DI & IDI engines, particularly under light load operations. In Reference (6) higher 85% point fuels showed increasing or decreasing hydrocarbon emission levels depending upon the tested engine. References (1 and 6) emphasise that engine design and load/speed conditions have a significant influence on hydrocarbon emissions. The apparently conflicting (and complex) trends outlined above are discussed as follows:

The heterogeneous nature of diesel combustion leads to wide local variations in equivalence ratio and mixture temperature in the combustion chamber. The formation of hydrocarbon emissions is therefore a localised process and is highly dependent on load/speed conditions. Four modes of hydrocarbon formation can be readily envisaged:

- (a) Localised pockets of fuel rich mixture;
- (b) Localised pockets of fuel pre-mixed to a condition leaner than the lean limit of combustion;
- (c) Fuel entering the chamber too late to fully participate in combustion;
- (d) Fuel unavailable for combustion due to wall wetting and quench zones.

The engine designer sets out to avoid over-rich mixtures, wall wetting and late injection by optimizing the injection and air fuel mixing processes. However fuels with high end points will tend to exacerbate the effect of any over-rich zones by further impairing mixing. Additionally, heavier fuels will tend to encourage over penetration of the spray causing wall wetting and consequent poor mixing.

It has been demonstrated that the residual volume of fuel in the sac of multi-hole injectors is blown out into the combustion chamber later in the expansion stroke and that this fuel has a significant influence on total hydrocarbon emissions (1). The injection of this fuel may be propelled by high volatility components which "boil off" so that high volatility gas oils could increase hydrocarbon emissions from this source (9). Modern injector designs feature minimised sac volumes but complete elimination of the sac poses structural and geometric problems in nozzle design. Pintle nozzles fitted to IDI engines do not incorporate a sac, so that this phenomenon does not feature as an emissions problem for IDI power units. However, both DI and IDI engines can suffer from secondary injection (i.e. a momentary reopening of the nozzle after the designed fuel delivery is complete). This source of hydrocarbon emissions applies particularly to engines equipped with poorly tuned injection systems operating at high engine speed.

It can be argued that the generation of hydrocarbon emissions from pockets of over-lean mixture is related to reductions in cetane number. Longer ignition delays allow more pre-mixing (albeit with more fuel in the chamber) so that pockets of over-lean mixture can be formed. This is supported by evidence that the use of ignition improvers reduces emissions. Reference (1) quotes work by Lucas - CAV which concludes "in indirect injection engines the 'lean limit' source of hydrocarbons is a major contributor". This is in agreement with Reference (3) which points out that, at high speed, the rate of air motion in the indirect injection engine is very rapid. Under these conditions it is possible that fuel-air mixing proceeds so quickly and so thoroughly that 'some of the mixture is too weak for combustion'. Reference (4) suggests that more volatile front end fuels produce the same 'lean limit' effect by promoting 'improved' air fuel mixing.

Although some fuel characteristics can be correlated with hydrocarbon emissions their influence needs to be put in perspective. In Reference (1) regression analysis of emission tests conducted on 13 fuels in three engines indicated that distillation characteristics had no significant effect. The same reference quotes an investigation of cetane number influence on four engines. A reduction in cetane number from 60 to 40 increased hydrocarbon emissions by approximately 35% for two engines, 28% for a third engine, whilst the remaining power unit showed a marginal reduction in hydrocarbon emissions. Of much greater significance, however, was the variation in hydrocarbons emitted by the four engines. At 50 cetane number two of the engines were producing over three times the hydrocarbons emitted by the 'best' power unit. Other work quoted in the same reference and in References (6, 8 and Appendix) confirms that engine design influences hydrocarbon emissions to a greater extent than the changes in fuel characteristics envisaged in Europe.

6. SULPHUR DIOXIDE

The sulphur present in automotive gas oil will, on combustion with air, form sulphur oxides and most of these gases will be emitted from the exhaust as sulphur dioxide. The emission of sulphur oxides is directly proportional to the fuel sulphur content and is a maximum at engine full load conditions. Fuel sulphur content will also influence particulate sulphate content (See Section 9).

7.

WHITE AND BLUE SMOKES

The distinction between white and blue smokes is mainly one of droplet size, the droplets in white smoke being generally greater than  $1\mu$ , and those in blue smoke mainly less than  $0.4\mu$ , in diameter, the latter appearing blue owing to greater light scattering at shorter wavelengths. These smokes are mainly the result of the condensation of fuel vapour with cooling of the exhaust, though it is possible that white smoke can include fuel droplets which have not evaporated. Tests with a cold smoking engine have shown that both the residence time and the temperature in the exhaust system can influence the appearance of the smoke. Increasing time or decreasing temperature can change the colour from blue to white. Analysis has shown that the smokes also contain products of partial combustion, such as aldehydes, which give these aerosols a pungent odour and lachrymatory effect, and that blue smoke produced from fully warmed-up engines at part load can contain polynuclear aromatic hydrocarbons in concentrations similar to those in black smoke (3).

White and blue smokes are evidence of absence of combustion or, at best, only partial combustion of the injected fuel. The emission of white and blue smoke is very dependent on engine design. The prolonged emission of white and, subsequently, blue smoke following starting is generally associated with direct injection engines and applies particularly to those power units with relatively low compression ratios. The fuel properties which influence the emission of such aerosols are those which affect the vaporisation and ignition of the fuel. White smoke is therefore decreased by decreasing mid-boiling point or by increasing cetane number of the fuel. In fact, an empirical simple linear equation containing these terms can describe the performance of a fuel in this respect but not necessarily any specific engine/fuel combination (3).

Unfortunately, this concept is complicated by the conflicting fuel characteristics it presents. The ease of ignition of a fuel generally increases with increasing n-paraffin molecular weight. As a consequence the combination of high cetane number and low mid-boiling point is, to some extent, incompatible. Even the use of ignition improvers is only a partial solution since the effectiveness of these additives is considerably reduced under cold engine conditions. This means that ignition improved cetane number is not equivalent in cold performance to natural cetane number. Furthermore, those fuels which manage to combine high cetane number and low mid-boiling point have a low volumetric energy content. The selection of a 'best' fuel from both engine performance and emissions viewpoints is therefore a matter of compromise.

Fuels can influence the blue smoke resulting from misfire in high speed engines by changing the extent of the misfire. Misfire can be particularly acute in certain types of small, high speed, separate-chamber engines but it can also occur in small, high speed, direct-injection engines. One reason why this particular

problem is more severe in separate-chamber engines may well be because of the marked degree of cyclic dispersion of combustion exhibited by this type of engine under marginal combustion conditions. Because of this, combustion may take place on one cycle but not on the next. The direct-injection engine, on the other hand, is much more regular in its operation and, consequently, if conditions do not result in combustion on one cycle it is unlikely that combustion will take place under the same conditions on subsequent cycles. The exact cause of misfire is not fully understood but there is evidence that it may be related to poor flame spread in weak fuel-air mixtures. The separate-chamber engine, at high speeds, has a very rapid air movement in the combustion chamber, and when only small quantities of fuel are injected it is possible that fuel-air mixing is so rapid, and so thorough, that some of the mixture formed is too weak for combustion (3).

8.

BLACK SMOKE

The greater part of the fuel in a diesel engine, following ignition, is burnt in a high-pressure turbulent diffusion flame. The formation of soot (or soot precursors) is characteristic of diffusion flame combustion and films of diesel engine combustion clearly show much white or yellow radiation, characteristic of incandescent particles, at all diesel engine operating conditions. At low fuel inputs, with smoke-free exhaust, the radiation disappears early in the engine cycle when temperatures are still high. At high fuel inputs, with smoking exhausts, the radiation persists, changing to orange and red as cooling takes place during expansion. This expansion, due to piston movement, leads to quenching of the combustion products and thus the soot formed at high loads is emitted in the exhaust products. The soot in black diesel smoke is similar in structure and composition to soot emitted from diffusion flames in other combustion systems, for example, industrial burners and laboratory diffusion flames. Black-smoke emission from all diesel engines increases with increasing fuel input, and thus the maximum power output of an engine is ultimately limited by the intensity of the smoke emission.

At high engine power the chemical and physical properties of fuels might be expected to influence smoke formation through the effects of hydrocarbon composition and mixing on carbon formation in flames. Experimental work has shown that fuels of different molecular structure differ markedly both in the extent to which they produce smoke in diffusion flames and in the amount of pre-mixed air that they require for the suppression of carbon formation. However, because diesel fuels are complex mixtures of different hydrocarbon molecules containing from 10 to 25 carbon atoms, it is not practicable to describe the composition in chemical terms. Instead, fuels are specified by certain measured properties; for example, cetane number and volatility. While these properties are important to other aspects of diesel engine performance they are not necessarily directly related to smoking tendency. Figure 1, (3), is an illustration of how the cetane number and composition of a fuel can influence smoke emission at constant load in a direct-injection engine. This shows that with entirely paraffinic fuels, i.e. fuels of inherently low and similar smoking tendencies, decreasing cetane number reduces smoking. This is due to the effect of ignition delay on the mixing of fuel and air in the combustion chamber. The lower the cetane number of the fuel, the longer the ignition delay. Thus with fuels of low cetane number the extent of pre-mixing before combustion is increased, thereby increasing the quantity of fuel burnt as a pre-mixed flame and smoke is only formed in pre-mixed flames with grossly rich mixtures.

Black smoke emissions for fuels having the same cetane number but varying composition follow the pattern of soot propagation from

laboratory diffusion flames in that aromatic hydrocarbons (particularly di-nuclear aromatics) give more smoke than paraffins. The increased use of conversion products in the automotive gas oil pool can affect both composition and cetane number. The influence of this trend on soot formation is therefore complex - lowered cetane numbers will reduce black smoke emissions (i.e. decrease diffusion combustion) whilst increased aromatic contents will produce more soot in the shorter diffusion combustion phase.

Variations of fuel volatility, either mid boiling point or back end volatility, appear in general to have little influence on black smoke emissions, though in some specific studies correlation of black smoke with volatility has been observed (2). In other instances the apparent effect of fuel volatility has been due to changes in hydrocarbon composition (3).

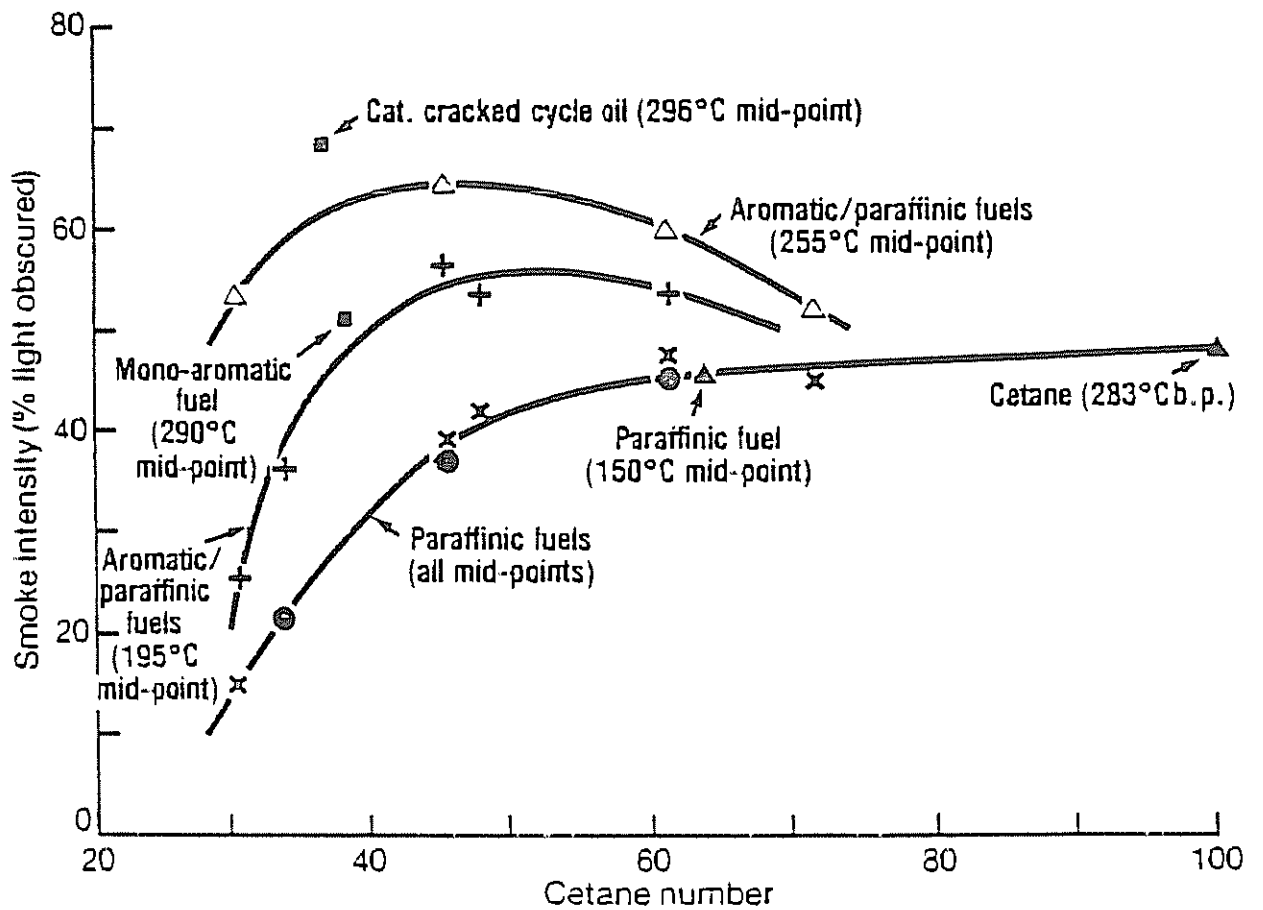
In practice the scope for reduction of black smoke emissions through fuel changes is limited by the often conflicting requirements of other aspects of performance and the nature of components available from conversion processes. For example, aromatic hydrocarbons in an automotive gas oil confer increased volumetric calorific value whilst a fuel containing a large proportion of straight chain paraffins would exhibit poor cold temperature operability and a lower volumetric calorific value.

Fuel injection equipment is basically a volume metering system and thus differences in fuel density can influence mass fuel input, engine power output and smoke emissions. As a result an automotive gas oil of high density will tend to give more smoke and power in addition to any differences in its inherent smoking propensity. This trend is noted in all the references but, as reference (6) points out, if pump deliveries are adjusted to give equal power then the variation in smoke emissions between fuels of differing densities is sharply decreased.

Diesel engines equipped with distributor-type (rotary) injection pumps can show additional fuel effects on black-smoke emission. The metering of the fuel is influenced by fuel viscosity and in some designs viscosity can affect fuel injection timing. This is relevant as both fuel quantity and timing influence smoke emission (3,4 and 8). For example, measurements of fuel delivery of a distributor pump with fuels having a range of viscosities from 1.6 to 7.5 cSt at 40°C have shown that the volumetric delivery was 13% higher with the most viscous fuel. In addition to the volume increase, the high-viscosity fuel would also have a density about 9% higher, giving, in total, an increase of over 20% in mass delivery. This factor alone will markedly influence black-smoke emissions, irrespective of differences in the inherent smoking quality of the fuel. However, in this type of injection pump an opposite effect is also observed in that increased viscosity advances the start of injection. This timing change will, in general, reduce smoke emission. The timing can be advanced by some 6° crank angle when fuel viscosity is increased from 1.6 to 7.5 cSt (3).



**Figure 1** Relation between black smoke emission and ignition quality for fuels of different volatility and hydrocarbon composition in a DI engine at constant load.



9. PARTICULATES

The term particulates with respect to exhaust gases, as defined by the US EPA, embraces all filterable material obtained from an exhaust diluted with air down to a temperature below 52°C. Thus particulates are composed of carbonaceous material on which hydrocarbons are condensed and adsorbed. The aspects of fuel quality which influence particulate emissions are aromatic content and high back end volatility (12).

The proportion by weight of the soluble organic fraction of a particulate sample is highly dependent upon operating conditions and can be as high as 85% at low to intermediate loads and as low as 5% at full load (9, 10). The proportion of soluble hydrocarbons in a particulate sample is also strongly engine dependent. Reference (10) quotes proportions of 60% and 5% over a composite Federal 13 mode cycle for a 2- and 4-stroke diesel engine respectively. Total particulates tend to be high under low and high load conditions due to condensed hydrocarbons and carbon respectively and reach a minimum value at some intermediate condition.

The soluble fraction of particulates is of major importance due to the presence of potentially carcinogenic polycyclic aromatic compounds, PAH. However it has never been satisfactorily established whether this material can be extracted in an active form within the human body (22).

There is also considerable interest at present in the role played by fuel composition in determining the chemical composition, and specifically the bio-activity of diesel exhaust particulates. Current data (9) suggests that a small proportion of the PAH present in the fuel contributes directly to particulate PAH but that additional PAH are produced during the combustion process. The potential for PAH emissions if all fuel PAH remained unmodified during the combustion process is far higher than that actually measured. As a consequence, engine malfunctions (such as secondary injection) could greatly increase the PAH emission burden via the unburned fuel.

Fuel sulphur content influences particulates composition (4) in that increasing fuel sulphur content has a pro-rata effect on particulate sulphate. In this study between 1 to 5% of the fuel sulphur was converted to sulphate. According to Reference (4) this is "typical for diesel engines with no catalytic exhaust post treatment".

The Task Force concluded that, although there is a correlation between automotive gas oil composition and particulate emission, there is insufficient current data to compare any inherent fuel effect with the influence of engine design. Reference (8) subjected eight engines to the 13 mode test (both European and U.S. Federal modes) and clearly identified engine design as a major variable for particulate emissions levels. These emissions, measured in g/kW.h varied between 0.27 and 1.34 in the European mode and 0.33 and 1.58 under U.S. test conditions (see Appendix).

The current Ricardo 'Future Diesel Fuels' Project will be studying these effects (along with all other aspects of engine performance) and this programme is in part sponsored by a number of members of CONCAWE. This work, in conjunction with other published studies should provide relevant data.

10.

NOISE EMISSIONS

The combustion noise emissions of a diesel engine are essentially influenced by the cetane number of the fuel (1 and 11). Reference (11) also showed that there was no evidence to suggest that any other automotive gas oil property influenced combustion knock. Both references indicate that changes in injection timing influence the sensitivity of engines to changes in ignition quality (i.e. advancing injection timing increases sensitivity) and Reference (1) shows that engine design also influences this sensitivity. References (1 and 11) suggest that increasing cetane numbers above 50 provides only marginal benefit in terms of noise reduction and (11) indicates that reducing cetane numbers to 45 should not create unacceptable increases in noise emissions, especially 'drive-by' noise where the effect of changing cetane number is much less noticeable.

11. INJECTOR FOULING

In general, there will be little scope for improving performance in diesel engines which are operating to manufacturers' design parameters. However, deposits build up to a greater or lesser extent in all engines, and in fuel injectors in particular, and engine performance may then be affected. The level of deposit build up will depend on many parameters including vehicle operation, maintenance, engine design and fuel characteristics, and may manifest itself through changes in noise, emissions (including smoke), fuel economy and power.

There are two types of diesel engine which need to be considered. First, there is the indirect injection (IDI) diesel engine, so called because fuel is injected into a pre-chamber where ignition occurs and the combustion then spreads to the main chamber. These engines are mainly used for small high speed applications such as passenger cars and light vans. They are designed for fuel economy relative to the gasoline engine, but low noise and performance are also important. The second type of engine is the direct injection (DI) version, in which the fuel is sprayed directly into and ignited in the main combustion chamber. These engines are used for medium and large trucks and offer high power output and good fuel economy relative to the IDI engine but they are considerably noisier.

In IDI engines the type of fuel injection nozzle commonly employed is the "pintle" nozzle. This is designed to give low noise by controlling fuel flow during the early stages of needle lift (the so called 'pre-lift' or 'overlap' region). The size of the annular clearance in the overlap zone is critical to the combustion characteristics of the engine - too large a gap allows excessive premixed burning, high cylinder pressure rise and potential incomplete combustion of the main charge. Nozzle fouling influences this initial slow rate of injection as deposits tend to form in the prelift portion of the injector.

Injector manufacturers have recognised for many years that some degree of fouling is bound to occur. This tends to happen fairly early in the life of the nozzle and thereafter the condition stabilises. As a consequence the tendency has been to fit nozzles with slightly oversized prelift clearances so that the optimum flow is achieved once the deposit level has plateaued.

It should again be emphasised that injector fouling is not new. However, in recent years there have been a number of reports from the United States which indicate that, with some fuels, deposit formation does not stabilise but continues to increase until the prelift clearance is completely lost. Engine manufacturers are now suggesting that isolated instances of this condition have been encountered in Europe.

Performance problems do not appear to manifest themselves until the prelift clearance is completely, or nearly completely, blocked. This delays the start of injection and the resultant fuel line pressure rise increases the injection rate once fuel begins to flow. As a result the rate of cylinder pressure rise increases, with a consequent impact on engine noise. The retarded timing effect also has an impact and leads to an increase in hydrocarbon emissions and/or white smoke. Because this condition only influences fuel injection in the prelift region the effect is minimised under high speed, high load conditions.

The mechanism for pintle nozzle fouling is not fully understood. There are indications that the deposits are essentially products of combustion but why such deposits should continue to be laid down under certain circumstances is far from clear. Some investigations conducted in the United States suggested that fuel stability might influence the coke deposit mechanism (16). However, attempts to relate the phenomenon to fuel quality aspects, such as cetane number, aromatic content, volatility, stability and carbon residue have been generally unsuccessful. There are also indications that mechanical solutions to this in-service problem may exist. For example, the 'flatted' injector needle apparently not only avoids excessive deposit formation but also permits closer 'tuning' to the optimum flow rate as it is less sensitive to any initial deposit build-up.

In DI engines the simple hole-type nozzles can experience external deposits which periodically build-up and then break-off. Generally, they have little effect on performance unless the condition becomes so severe that the deposits interfere with the spray pattern. Lacquering and gum may also form on the needles and seat within the injector. Then the needles may stick or the nozzles may no longer close tightly. If so, fuel will leak through and cause excessive coking around and in the holes, which ultimately results in a poor spray pattern, inefficient combustion and a deterioration in fuel economy and performance (including smoking). Such severe degradation of nozzle condition is extremely rare and the major European injection equipment manufacturers do not consider DI nozzle fouling to be an operational problem.

12. THE INFLUENCE OF COLD START TEMPERATURES ON EMISSIONS AND PERFORMANCE

Ignition delay is arguably the most important single aspect of engine performance. It is particularly significant for cold start and warm up when conditions in the engine cylinders are least conducive to ignition and sustained combustion. The ignition characteristics of diesel fuels, involving both physical and chemical aspects, are defined by cetane number, which rank fuels in order according to ignition delay in a standard engine test. Cetane number, may therefore be expected to be the most important fuel property when addressing diesel engine startability and warm up. Starting aids, on the other hand, seem to be the most important feature of engine design where cold start performance is concerned.

The IDI engine is more difficult to start than a DI power unit and, as a consequence, glow-plugs are incorporated as a standard fitment in IDI engines. References (1 and 5) indicate that the starting performance at temperatures down to minus 10 to minus 12°C of a range of indirect injection engines was not influenced by ignition quality down to a cetane number of 40.

DI engines for temperate European operations are generally not equipped with starting aids and therefore tend to exhibit increased sensitivity to cold starting. The DI engine tested by (7) which was not provided with a starting aid shows how sensitive DI engines can be. At 0°C the start time for this particular power unit increased from 18 seconds to nearly 35 seconds when the fuel was changed from 50 cetane to 37. With a 47 cetane number fuel the cranking time had increased to 25 seconds. The increase in cranking time is also thought to be one of the factors responsible for the higher white smoke emissions observed as cetane number decreases. While the engine is cranking, injected fuel accumulates in the exhaust system. After the engine has fired this accumulated fuel will vaporise and add to the smoke burden generated within the combustion chambers. Consequently the same DI engine tested by (7) showed that the time required for smoke to reach an acceptable level after starting is affected by cetane number. Reference (7) found that, with a 50 cetane number fuel, the time required to clear white smoke increased from one minute to four minutes as the ambient temperature dropped from 21°C to 5°C. The same engine, under the same temperature conditions, saw an increase in clearance time from two minutes to ten minutes respectively when operating on a 38 cetane number fuel and from two minutes to seven minutes when running on a 47 cetane number automotive gas oil.

As indicated earlier, starting aids for DI engines tend only to be fitted when it is known that they will be expected to operate under severe low temperature conditions. In recent, as yet unpublished, work by one CONCAWE member it was found that a range of DI engines equipped with starting aids showed virtually no discrimination in starting times. These tests were conducted at minus 20°C with a matrix of twelve fuels of varying volatility and with cetane numbers in the range 34.5 to 45. The same tests conducted at minus 5°C without starting aid assistance demonstrated the same trends as noted by reference (7). It can therefore be concluded that the use of starting aids at somewhat higher ambient temperatures would reduce or eliminate any deterioration in cold start performance (cranking time and white smoke).

As yet unpublished data from work conducted by one CONCAWE member has shown that engine design has an impact on cold starting performance (cranking time and white smoke) beside the influence of the cetane quality. The data obtained using six vehicles powered by IDI and DI engines document that different engine designs exhibit varying sensitivity to cetane quality changes. E.g., with one DI engine fitted with a standard starting aid, no significant influence on cold starting performance was observed when decreasing the cetane number by 11 numbers.

Reference (3) studied the effect of fuel properties on cold misfire in a separate-chamber engine under cold, light load conditions. The general conclusions drawn from tests on nine fuels covering all combinations of three different cetane numbers and three different mid-boiling points were that, in this engine, fuel volatility had little effect on misfire, but to prevent misfire a fuel of reasonably high cetane number was required. During the tests coolant temperature was shown to have a much greater influence on misfire than air temperature; the lower the coolant temperature, the more the misfire. This work therefore suggests that appropriate design and control of the coolant system can do much to alleviate cold misfire.

None of the oil industry sources quoted in this report have examined the impact of the low ambient temperatures on exhaust emissions. However, the EPA (15) measured emissions from two diesel passenger cars as a function of ambient temperature (6°C - 28°C). The fuel employed was a conventional US 2-D automotive gas oil with a cetane number of 46, its other characteristics being representative of a good quality product meeting 2-D requirements. The tests were conducted over the urban dynamometer driving schedule of the Federal Test Procedure. It was concluded that low ambient temperature did not affect rates of emission of HC, CO or NO<sub>x</sub>. Total particulate matter increased with decreasing test temperature as did particulate organic emission rates (due primarily to absorption of unburnt diesel fuel). PAH emission and mutagenicity were not influenced by decreasing ambient temperature.



Ignition improver additives can offer a partial solution to white smoke emitted following a cold start. In addition the cold start times for DI diesels may be improved, but only 50-70% of the cetane boost provided by the additive is realised as a performance benefit. Smoke and HC emissions during cold idle and warm-up are reduced and the time to clear white smoke is improved for both engine types. Again, only about 50% of the cetane improvement is seen as a performance enhancement (13). Some improvement in cold idle quality (noise and misfire) and driveability immediately after starting can be experienced but engine design appears to be an important factor. Fully warmed-up engines show much less sensitivity to cetane number and the presence of ignition improvers. Effects on performance are variable and engine dependent, the exceptions being ignition delay and noise which appear to be reduced in line with expectations based on cetane enhancement (13).

The major drawbacks in employing ignition improvers are that they appear to offer only about half the performance improvement indicated by the cetane boost and that their effectiveness reduces with reducing base fuel cetane number. The reasons for this latter phenomena are not fully understood but it is thought that the radicals generated by the additive are 'soaked up' by aromatic structures in the fuel. A further reduction in additive response is reported, based on a survey of European diesel fuel quality (50 samples) over a three years period (14). On average, a 2.5 point reduction in the cetane number gains due to ignition improver additives was observed over the three years and this is ascribed to the adoption of these additives which reduces the efficiency of further treatment. It could also be argued that this reduction in cetane gain was due to changes in refinery processing i.e. increased conversion.

13. LOW-TEMPERATURE OPERABILITY

One of the major problems of diesel engines is starting and operating at low temperatures. After starting the low temperature operation of a diesel engine depends on the low temperature flow properties of the diesel fuel and the design of the fuel system. These two parameters have to be matched to avoid accumulation of fuel wax crystals in the lines or on filters thereby restricting fuel flow and - in the worst case - causing engine stall.

To provide satisfactory low-temperature quality of automotive diesel fuel, the Cold Filter Plugging Point (CFPP) is used to specify low-temperature performance in industry and national standards for European automotive diesel fuels. The CFPP provides a realistic indication of the temperature above which a diesel fuel (with or without flow improver) will perform satisfactorily in typical European fuel systems (17).

Refineries supply automotive diesel fuel according to low-temperature flow characteristics specified for individual marketing areas and expected low temperatures based on weather data extending back over several decades. Nevertheless, under abnormally cold weather conditions or with vehicles equipped with poorly designed fuel systems, operational problems can occur. More stringent low-temperature flow specifications might somewhat reduce these isolated difficulties; but it does not seem realistic to introduce overly restrictive specifications to satisfy a minority sector of the diesel vehicle population (17). Such a move would have consequences for diesel fuel yields or lead to the processing of more crude oil to meet demand. This approach would be contrary to the world wide efforts to use crude oil resources as effectively as possible.

An alternative approach to alleviate occasional low-temperature operability problems is to identify and improve sensitive vehicle fuel system designs. General criteria for an insensitive fuel system design are provided in References (17,18,19).

In References (20,21) it is shown that proper fuel system design is important to provide satisfactory low-temperature operability of a diesel passenger vehicle. Based on a survey and testing of European diesel passenger cars, it was concluded (20,21) that systems being sensitive and insensitive to low temperature operation are in service. Sensitive vehicles, failing at or above ambient temperatures corresponding to the fuel CFPP, were significantly improved by simple modifications of the standard fuel system. Larger filters, the removal of the tank screen, or the use of an electric fuel heater provided good low-temperature operability well below the fuel CFPP. Good operability was obtained down to about 11°C below the fuel CFPP in some cases.

The strategy of such modifications is - first - to improve the fuel flow to increase the amount of time the engine can be operated before the filter becomes plugged with wax and thus allow the engine and the fuel to warm-up to melt any accumulated wax in due time and/or - second - to raise the fuel temperature immediately after start-up by additional heating.

14.

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EMISSIONS PERFORMANCE OF A RANGE OF CURRENT PRODUCTION ENGINES

ENGINES EMPLOYED IN THE MULTI-CYLINDER ASSESSMENT  
 CONDUCTED BY REFERENCE (9)

Engine Type	Max. Power (kW)	Specific Power (kW/litres)	Smoke (b.s.u)	B.S.F.C. (g/kWh)
NA Quiescent DI	170.4	12.16	3.8	250.1
TC Swirling DI	259.5	21.32	0.9	222
TC Swirling DI	113.5	19.59	2.2	240
NA Swirling DI	170.4	14.00	2.4	239
NA Swirling DI	88.9	15.33	3.9	260
NA Squish Lip DI	62.6	15.39	2.5	239
NA 'M' System DI	100.7	17.71	4.1	248
NA IDI	67.6	10.84	1.3	257

Note: Figures quoted are those measured by Reference 8, not Makers specification.

FUEL EMPLOYED IN THE MULTI-CYLINDER ASSESSMENTS

Relative Density at 15.6°C/15.6°C		0.8730
Total Sulphur Content	% wt	0.23
Kinematic Viscosity at 40°C	cSt	4.22
Flash Point (PM closed)	°C	103
Cold Filter Plugging Point	°C	-7
Pour Point	°C	-18
Carbon Residue (Con) on 10% Residue	% wt	0.02
<u>Distillation IP 123</u>		
IBP	°C	231
10% Volume Recovered at	°C	257
30% Volume Recovered at	°C	272
50% Volume Recovered at	°C	286
80% Volume Recovered at	°C	314
90% Volume Recovered at	°C	330
95% Volume Recovered at	°C	342
FBP	°C	351
Distillate/Residue/Loss	% vol.	99/1/-
Aniline Point	°C	63.6
Determined Cetane Number		45
Calculated Cetane Index		44.5
Hydrogen Content (NMR) (Standard Tridecane)	% wt	12.80
Aromatics by HPLC	% wt	32

Fig. 1 Comparison of emission specific rates from a range of current production engines

13 Mode Test Results (European)

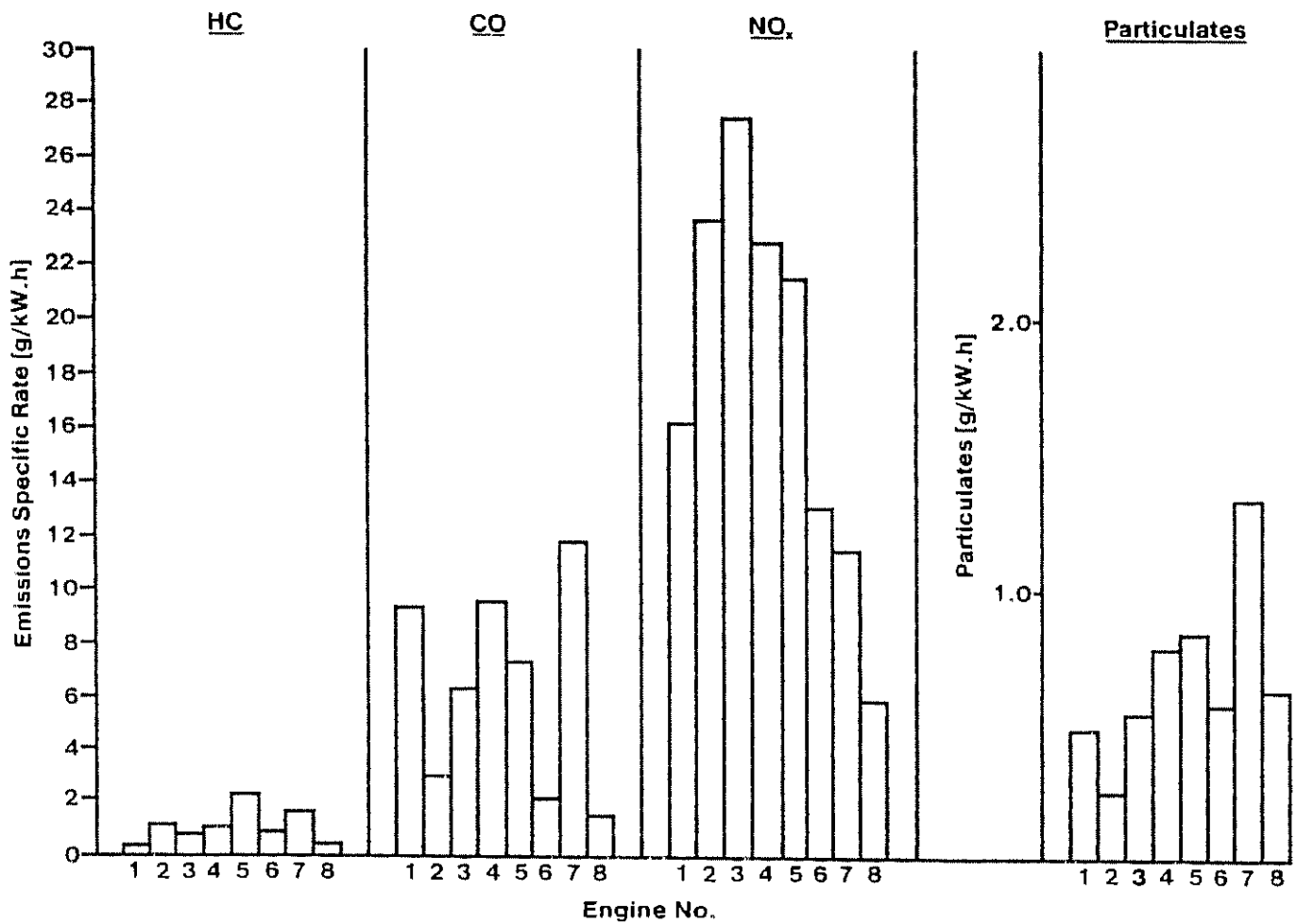




Fig. 2 Comparison of Emission Specific rates from a range of current production engines

