119. ROESSELER W.U.

Diesel engine research and development status. Highway vehicle systems contractors co-ordination meeting.

October 23, 24, 25, 1979. Dearborn, Michigan. Conf. 791082 p. 399.

120. PRIVATE COMMUNICATION WITH D.o.E. - May 1980.

121. ANON.

Texaco's engine developments.

National Academy of Sciences committee on motor vehicle emissions report, April 24 1974, Washington D.C.

122. MEURER J.S. AND URLAUB A.C.

Development and operational results of the MAN 'FM' combustion system.

SAE 690255, 1969.

123. Brief documentation on development status of MAN 'FM' combustion system.

MAN Report, Nürnberg, February 1975.

124. URLAUB A.G. and CHMELA F.G.

High speed, multifuel engine; L9204 FMV.

SAE 740122, 1974.

125. RICARDO DOCUMENT Z

126. ANONYMOUS.

Extract from "On the trail of the novel fuels: alternative fuels for motor vehicles."

Ricardo translation No. 360 from MTZ.

127. HARDY R.

Das verhalten eines Krupp-Zweitakt-Dieselmotors beim Betrieb mit verschiedenen Kraftstoffen.

MTZ December 1961 p. 458.

128. HAVEMANN H.A. ET AL

The effect of light diesel oil on high speed diesel engines.

Journal of the Indian Institute of Science. Vol. XXXVI No. 2, 1954.

129. ELLIOTT M.A.

Combustion of diesel fuel.

SAE Quarterly Transactions. July 1949. Vol. 3 No. 3 p. 490.

130. LYN W-T and VALDMANIS E.

The effects of physical factors on ignition delay.

Proc. I. Mech. E. 1966-67. Vol. 181 Pt.2A No. 1.

131. HARDENBERG H.

Investigations into the auto-ignition, detonation and smoke characteristics of fuels.

Ricardo Translation No. 368 from MTZ January 1975 (p.10).

132. PALUSKA R.T. ET AL

Design and development of a very high output (VHO) multifuel engine.

SAE 670520, 1967.

133. WILLIAMS C.G.

Creosote as a fuel for C.I. engines.

The Institution of Automobile Engineers No. 9186B, 1940.

134. JONES B.E. AND MACKWORTH J.D.

Induction swirl in spark ignition engines.

I. Mech. E. 19 February 1975 C85/75.

135. MUHLBERG E.

On the development of the MAN M-engine as a petrol engine with very high fuel utilization.

Translation from MTZ October 1957.

136. McCONNELL G. ET AL

Influence of fuel properties and effect of antismoke additives on diesel exhaust smoke.

SAE Journal March 1968. Vol. 76 No. 3 p. 36.

137. HILL S.H. and DODD J.L.

A low NOx lightweight car diesel engine.

SAE 770430, 1977.

138. SHELTON E.M.

Diesel fuel oils, 1977.

Bartlesville Energy Research Center,
Oklahoma. November 1977. BERC/PPS-77/5.

139. RICARDO DOCUMENT A1.

140. RICARDO DOCUMENT B1.

141. RICARDO DOCUMENT C1.

142. RICARDO DOCUMENT D1.

143. RICARDO DOCUMENT E1.

144. MITCHELL E. and ALPERSTEIN M.

Texaco Controlled - Combustion System-multifuel,
efficient, clean and practical.

Combustion Science and Technology Vol. 8
p. 39-49. 1973.

145. SUMMERSON W.A. and MITCHELL E.

Small military engine, stratified charge
combustion yields multifuel capability.

Reprint from Automotive Industries.

146. BARBER E.M. ET AL.

The elimination of combustion knock-Texaco
combustion process.

SAE Summer Meeting June 5-9 1950.

147. JAIN B.C. ET AL

A performance model for the Texaco controlled combustion stratified charge engine.

SAE 760116, 1976.

148. ANONYMOUS

Energy conservation + purer air with the TCCS.

Texaco publication (no date).

149. GOMPF H.L.

Evaluation of the Texaco stratified charge (TCP) M-151 army vehicle.

EPA, August 1972.

150. RICARDO DOCUMENT F1.

151. ANONYMOUS

MAN multifuel development.

Gas and Oil Power. January-February 1966.

152. MEURER J.S. and URLAUB A.

MAN modified M combustion system burns any grade of gasoline efficiently and without knock.

SAE Journal Vol. 77, 1969.

153. NEITZ A.

The MAN M-combustion system in the passenger car diesel engine.

SAE 770434, 1977.

154. RICARDO AND COMPANY ENGINEERS (1927) LIMITED

A study of stratified charge for light duty power plants.

EPA - 460/3-74-011A October 1975.

***

Other references of potential interest but located too late to include in the study :

155. HOLMER E. ET AL

The utilization of alternative fuels in a diesel engine using different methods.

SAE 800544, 1980.

156. THE SOUTHWEST RESEARCH INSTITUTE.

Emergency fuels utilization guidebook.

DoE/CS/54269-01. August 1980.

157. COURTNEY R.L. and NEWHALL H.K.

Automotive fuels for the 1980's.

SAE SP-446, 1979, p.31.

158. HULTKRANS H.R.

Role of fuel properties in diesel fleet economy.

SAE 650429, 1965.

159. SCHMIDT R.C.

Diesel engine operation in cold weather.

SAE 660011, 1966.

160. NICHOLS R.J.

Automobile engine fuel requirements.
Paper presented at the 5th International
Automotive Propulsion Systems Symposium
April 14-18, 1980.

161. RANSOME J.L.

Mixing petrol with DERV. The full story of
the Midland Red experiments.

Bus and Coach, April 1957, p.120.

162. LANDEN E.W. and DOYLE W.L.H.

Problems associated with use of diesel fuels.

ASME Paper No. 47-4-124.

163. WIMER W.W., J.A. RUSSELL & W.D. GILL

Emergency fuels composition and impact.

Phase 1: Emergency fuels constituents
identification.

Southwest Research Institute, Dec. 1978,
MED Report No. 109.

164. BAILEY B.K. and RUSSELL J.A.

Emergency fuels composition and impact.
Phase II: Formulation and screening -
diesel emergency fuels.

MED Report No. 101, presented at the DoE
Highway Vehicle Systems Meeting, Oct. 1979.

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APPENDIX 1

THE INFLUENCE OF FUEL IGNITION QUALITY - CETANE NUMBER

1. THE INFLUENCE OF CETANE NUMBER ON COLD STARTABILITY

Results from some recent work (3) are plotted in Figure 1. These results, obtained from a production multi-cylinder DI engine, demonstrate the time to starting for fuels of 33-50 cetane number at different ambient temperatures. The degradation in starting ability with reducing cetane is clearly shown, the effect becoming progressively more pronounced at low ambient temperatures and with the lowest cetane fuel. At an ambient temperature of 40^{o}F, reducing the cetane number from 50 to 37 resulted in the required cranking time being increased approximately twofold to 24 seconds. A further reduction in cetane to 33 resulted in a cranking time of over 60 seconds. This engine would therefore appear to have a critical cetane requirement for starting at low temperatures.

Derry and Evans (4) recorded the minimum temperature at which a start could be made utilising fuels of broadly similar distillation characteristics but a wide range of cetane. Utilising a DI engine, their results are shown in Figure 2, and clearly show the dependence upon high ignition quality for good cold start performance. It should be noted that the compression ratio of the test engine was very low at 13:1. For a higher and more representative compression ratio of 16-17:1, it is anticipated that the sensitivity of the response would be somewhat attenuated. In addition, the apparent absence of a critical cetane value may also be related to the low compression ratio.

The same authors extended their studies to commercially available multi-cylinder engines incorporating both DI and IDI combustion systems. Again fuels of broadly similar distillation range were utilised with a wide cetane range. These tests were carried out under simulated service conditions by cranking the engine with cold soaked batteries, at full rack, until starting was obtained. In the case of the IDI engines, the auxiliary air heating devices fitted were activated prior to cranking.

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Starting performance at a fixed ambient temperature was recorded as the cranking time required to sustain running without assistance from the starter. A synopsis of the data obtained is shown in Figure 3.

From Figure 3, it can be seen that the time required to start and the sensitivity to cetane number are influenced by overall engine design, it not being possible to categorise starting performance simply by the type of combustion system. The differences in cold start performance are attributed to various engine variables according to the authors. The relative insensitivity to cetane number of the IDI engines, D and E, and DI engine F, is attributed to adequate auxiliary intake air heater capacity and high cranking speed respectively. In the latter case, time to starting was also improved with respect to the other two DI engines, A and B.

The bad cold starting performance of the IDI engine C was attributed to inadequate heater plug capacity and rapid excess starting fuel pull-off by the governor, with regard to cetane sensitivity and long cranking periods respectively. Furthermore, the quicker time to starting of engine E relative to engine D has been attributed to the lower compression ratio of the latter. In this context it should be noted that the compression ratios of the IDI engines are low by current standards for light duty diesel engines. The starting performance may therefore not be representative.

Despite these differences, the adverse response to a reduction in cetane number is clearly evident. Furthermore, for most of the engines, the sensitivity to cetane during cold starting appears more marked in the range below 45-40. This trend is significant.

Other workers in the field (5, 6, 7, 8, 9) report similar findings with respect to the strong dependence of diesel engines on high ignition quality fuels for good cold starting.

2. THE INFLUENCE OF CETANE NUMBER ON EXHAUST SMOKE

2.1 The Influence of Cetane Number on Exhaust Smoke Following Cold Starting

Many researchers have explored the influence of cetane number on cold smoke emissions. Tests have been carried out to record the time to clear white smoke with the engine idling after cold starting. Results from such tests with multi-cylinder DI engines are shown in Figure 4 (3, 6, 10). It will be noted that agreement of results is excellent. All of the results demonstrate that, for a given ambient temperature, reducing cetane number increases the time to clear. The sensitivity of white smoke to cetane number is especially pronounced at low ambient temperatures. At high ambient temperatures above approximately 60°F, white smoke problems are relatively limited, even with low cetane fuels.

In tests carried out by Shell on their DI engine (6), a reduction in ambient temperature from 60°F to 23°F approximately doubled the smoking period. Results from a smaller DI engine returned an increase in smoking period by a factor of eight. In similar tests carried out utilising an IDI engine equipped with heater plugs, the same reduction in ambient temperature only increased the smoking time by 50%. This effect was in part attributed to the use of heater plugs for reliable starting. Even during these short smoking periods, sensitivity to cetane number is reported although no data are presented.

2.2 The Influence of Cetane Number on Exhaust Smoke at Normal Engine Operating Temperatures

Many researchers over the years have attempted to quantify the influence of cetane number on smoke emissions when operating under normal engine operating temperature conditions. These studies have been carried out on both direct and indirect injection engines. Much of this work has been limited to defining the relationship between cetane and smoke at, or near to, full load operation in order to observe the impact on smoke limited power.

In order to overcome the interaction of fuel parameters, several of the workers in the field have attempted to isolate the effects of ignition quality upon smoke by using fuel sets having variable cetane but with other leading fuel parameters essentially rationalised. This has frequently been achieved by utilising low cetane base fuels, plus commercially available ignition improvers, or fuels of naturally varying cetane but controlled distillation, generally either by range or mid point.

From the data extracted for this study, it is clear that not all workers in the field are in agreement as regards the influence of cetane number on smoke output. The findings are therefore presented in some detail.

Results from an extensive set of studies conducted by Shell (5, 6, 11) are summarised in Figure 5.

Results from the 85CID, single cylinder DI engine operated at 95 psi bmep, revealed a marked sensitivity with significant smoke reductions being recorded as cetane was reduced from approximately 45 to 30. The same results were obtained when the cetane was varied either naturally or by using ignition improving additives. All fuels had essentially similar mid-points.

Figure 5 also shows similar, but less sensitive trends obtained from a multi-cylinder DI engine operated at 1250 rpm and 100 psi bmep. The smoke sensitivity to reducing cetane below approximately 55-60 is remarkably similar for two fuels of radically different mid-points.

Results from additional studies carried out to investigate the influence of cetane, volatility and chemical composition demonstrated that,for fuels of the same chemical composition and mid-point, reducing cetane from approximately 45-50 to a minimum of 30 improved smoke output. At higher cetane ratings, the response was either slightly reversed or the sensitivity significantly attenuated, implying a cetane range to which the test engine, operated at constant load, was sensitive. This trend is thought to be particularly important. The results are shown in Figure 5.

In further studies with a multi-cylinder DI engine operated at 2400 rpm, the use of a 54 cetane ignition improved fuel was shown to influence adversely smoke compared with the 38 cetane base fuel over the entire upper 30% of the load range. Data was not presented for lower load factors.

The reduction in smoke observed with lower ignition quality fuels is attributed by the authors to the lower cetane fuels increasing the delay period resulting in a greater proportion of the fuel being burnt in a pre-mixed state. It is, however, stated that not all engines tested demonstrated similar tendencies. Several other, mainly DI engines, proved relatively insensitive to cetane change within the range 44-57 and ASTM 50% points of 437-536°F.

Studies carried out by Duval and Lys (12) utilising engines incorporating both swirl chamber IDI and DI "M" system combustion chambers, revealed that when operated to the Federal 13 mode test procedure, both engines returned higher smoke output when using a 60 cetane fuel compared with a 52 cetane fuel. Fuel distillation characteristics for each fuel were very similar and aromatic content was also rationalised. The results are,however, reported as trends rather than highly significant. The same speculation to account for the results as postulated by the Shell authors is given.

Utilising a single cylinder DI engine of low compression ratio (13), results have been obtained to compare exhaust smoke over the load range when utilising narrow boiling range fuels of naturally varying cetane between 40 and 90. Despite the low compression ratio of 12:1 these results are shown in Figure 6.

With the lighter fuels, having a distillation range of 400-500^{o}F, increased ignition quality returned higher smoke at all but the lightest loads. The largest influence generally appears to be when changing from a 40-55 cetane fuel, with smaller increases observed when changing from 55 to 70 and 70-90 cetane.

Similar tests conducted with a fuel set of 600-700^{o}F distillation range returned opposite trends, with the smoke output being reduced by the higher ignition quality fuels, especially at light load.

In the same test series, similar results were obtained when raising the ignition quality of the lowest 40 cetane fuels using additives for both distillation ranges.

The cetane smoke relationship reported for the lighter fuels is attributed by the author to the increased paraffinic content of the higher cetane fuels, resulting in the precipitation of more carbon early in the combustion phase due to pyrolytic action upon the more thermally unstable paraffin components. This increased carbon output is subsequently not fully consumed in the cylinders resulting in higher smoke emission. Regarding the converse trends recorded with the less volatile fuels, the author states that a blue smoke was emitted rather than black with unburnt fuel being present within the exhaust. It is speculated that the higher ignition quality fuels within these tests produced lower levels of smoke owing to the more readily oxidized paraffin fraction.

For the results with the ignition improved lighter fuels, the author concludes that the higher smoke levels with improved ignition quality were due to a cetane effect, whilst with the heavier fuels, the smoke reduction observed with 20% improver added is attributed to the low boiling improver having a favourable influence upon distillation characteristics and not to cetane effects.

In a paper presented by Ricardo (4) based upon the results shown in Figure 6, reduced smoke with lower cetane for the more volatile fuels was attributed to increased proportions of premixed combustion as a direct results of longer delay periods. The converse trends, noted for the heavier fuels, was thought to be due to the more rapid ignition of low volatility fuels in the remote parts of the combustion chamber.

The tendency for elevated ignition quality to influence smoke adversely at equal load has also been demonstrated by Derry et al (15). In this work, an increase in smoke was observed in the upper 25% of the load range of a DI engine running at 1800 rpm. Fuels of 40-70 cetane number were utilised with a rationalised mid-point. The author states that this is a general trend for DI engines but,provided the cetane is maintained above a certain minimal level, the differences are not large.

Young et al (16) comment on the influence of ignition quality on smoke on both DI and IDI combustion systems. In this paper, it is stated that higher smoke levels are a result of high ignition quality fuels due to short delay periods restricting the full development of the fuel spray and hence mixing prior to ignition. Fuel volatility, however, is also stated to be a prime factor. In tests with a swirl chamber IDI combustion system, minimum smoke density is reported with fuels of the relatively low 35-40 cetane range, independent of fuel volatility. In addition, fuels of very low ignition quality resulted in very late combustion and inadequate time to consume soot released within the cylinders, resulting in elevated smoke emissions.

Not all results obtained demonstrate smoke reduction with lower ignition quality fuels. McConnel and Howells (10) tested ignition improved diesel fuels and kerosenes in both DI and IDI engines. These results are shown in Figure 7. Increasing cetane number from 56 to 69 had no influence on smoke over the full load speed range up to 4000 rpm of a small IDI engine. In the same engine running at 1000 rpm, fuels of 47 and 52 cetane returned equivalent smoke over the upper 60% of the load range tested. Similar results were obtained at 4000 rpm except at full load, where the high cetane fuel returned lower smoke. In the same test series with a DI engine, smoke limited bmep remained unchanged in the tested speed range of 800-1800 rpm, when comparing 44 cetane kerosene and the same fuel ignition improved to 56 cetane.

The author's reason is that it is the combustion of the last few drops of fuel to enter the combustion chamber that play the greatest part in smoke production rather than the initial combustion experienced at the end of the delay period. In this manner, it is argued that, assuming changing delay with fuels of different cetane number, an influence of cetane number upon smoke would not be expected.

Figure 7 also shows data published by Howells and Walker (17). These results again demonstrate no influence of cetane number upon smoke for a DI engine operating at maximum fuelling for 1000 and 1600 rpm.

The influence of cetane on smoke over the load range for several engines has also been reported by Burk et al (8). In their studies, three fuels were selected covering the cetane range 37.5-60 whilst closely controlling viscosity and volatility. Their results for four engines are plotted in Figures 8, 9, 10 and 11.

The Fairbanks Morse IDI engine, running at a rated speed of 1200 rpm, showed little change at part load for each of the test fuels. At full load, smoke increased with the lowest cetane fuel but this can probably be related to API gravity. For the Hercules IDI engine with a rated speed of 2000 rpm, data are limited but reveal no change in smoke at any load when reducing cetane from 60 to 49. At 20% load, however, one data point with the 37.5 cetane fuel indicates a significant elevation in smoke output. The Lanova combustion system Mack engine, with a rated speed of 2200 rpm, showed little smoke sensitivity to cetane at part load, but significant sensitivity at full load. The marked reduction in smoke at this condition when reducing cetane from 49 to 37.5 is converse to changes in API gravity (35.9-30.7 respectively) and may therefore be significant. The GM DI engine with a rated speed of 2000 rpm showed a tendency for smoke to improve with reducing cetane especially at 25% load.

Utilising a 3.75" bore x 5" stroke IDI engine running at 1200 rpm 65 psi bmep. Landen (18) collected carbon solids from the exhaust. For a range of fuels, good correlation with the amount of carbon exhausted and cetane number defined by the range 29-84 was obtained. In this instance, the emission rate of carbon progressively rose by approximately 100% as cetane was reduced from 84 to around 45. At lower cetane values, carbon output rose markedly. At a cetane value of 29, carbon output was some 8-4 times higher than the emission rates for the 84-45 cetane range respectively. Ignition delay also followed similar trends.

The higher carbon output with the lower cetane fuels is attributed to longer ignition delays, allowing no formation of very rich mixtures in the prechamber prior to combustion. The carbon so formed is subsequently not consumed within the cylinders owing to the shorter burning interval.

El Nesr et al (19) utilised a single cylinder Wittle IDI diesel engine running at full load, 720 rpm, to investigate the influence of cetane with a number of fuels having a cetane number range of 22.7 to 71. Optimum smoke occurred at a cetane of 65. At cetane ratings below this, smoke increased rapidly. Acceptable smoke levels were exceeded at 47 cetane. The same trends were recorded when using ignition improving additives. Similar patterns are also reported as being observed over the entire load range.

The authors relate increased smoke levels with low cetane fuels to increased pre-mixed burning as a function of long ignition delay. Rather than being beneficial, it is speculated that the rapid combustion of the pre-mixed fuel lowers the general oxygen content and reduces the combustion rate of the rest of the fuel droplets. This, combined with the reduced burning interval due to the longer delay period, causes unburnt carbon to be released to the exhaust, where its oxidation is quenched, resulting in black smoke. The authors do however concede that the sensitivity of this particular engine may be due to such variables as low compression ratio and very retarded injection timing.

Utilising a small high speed (4000 rpm) swirl chamber IDI engine, Ricardo (20) evaluated smoke limited bmep for fuels of constant volatility (10%, 50%, 90%: 410^{o}F, 530^{o}F, 630^{o}F respectively) and cetane ranging from 50 upwards. Smoke limit was set at 30 Hartridge units. In the upper half of the speed range, smoke limited performance was not influenced by cetane. At the lower speeds smoke limited power was progressively improved with the higher ignition quality fuels.

Reference 7 provides a cetane number smoke correlation as deduced by Sinclair Refining Company. Unfortunately, details of test fuels and engines are not given although it would appear that 5 commercial engines were utilised in developing the correlation. Engine operating conditions are also not specified.

The correlation obtained shows a linear increase in smoke with reducing cetane. The sensitivity of the response is defined by a 20% smoke increase for a drop of 19 cetane number from a reference point of 55.

Wetmiller and Endsley(21) performed an extensive study on the influence of fuel properties, including ignition quality, upon smoke under both steady state and accelerating conditions. Two popular automotive, but unspecified, diesel engines were utilised. Manufacturers recommended engine settings were adhered to.

Initially 13 commercial fuels with wide ranging gravities, distillation characteristics and cetane numbers were utilised. For these fuels, cetane varied between 20 and 72. The fuels were tested in one engine at steady state conditions, 1400 rpm, ranging from full load to motoring with fuel injection, the latter condition derived to simulate governor maladjustment. Particular emphasis was placed on low load operation. It was noted that as load was reduced from high load conditions smoke was attenuated until a light load condition was reached where a sudden smoke increase occurred. Continued load reduction from this point resulted in further smoke increase culminating in a maximum level after which smoke rapidly reduced as combustion ceased. Typical results for three fuels are shown in Figure 12A. The rapid increase in light load smoke was attributed to lean mixtures and inferior fuel injection patterns, the latter being deduced from bench tests which showed injector dribbling to be prevalent at very low fuelling levels. In addition, the smoke recorded at the higher loads prior to the light load rise was predominantly black, comprised of carbon particles, whilst under the heavy smoke, light load conditions smoke was comprised of unburnt fuel droplets and hence of the white type. Furthermore, when white smoke was observed, liquid fuel was observed in the exhaust system.

The fuelling level at which the large light load smoke increase and the maximum peak occurred correlated with cetane number as shown in Figure 12B.

The point of maximum light load smoke corresponds to the point of complete misfiring since motoring power was equivalent to friction horsepower.

From Figure 12B it can be observed that improved ignition quality of the fuel enables lighter loads to be achieved without white smoke penalties. The authors attribute this phenomenon to some compensation of the large delay periods, and hence late combustion, induced by lean mixtures and poor fuel spray form prevailing at light load.

Whilst these extreme light load conditions do not as a rule occur in service, the results are thought by the authors to be directly applicable to field service. It is argued that if light load conditions of white smoking are allowed to occur, fuel deposited in the exhaust system will be expelled by subsequent high load, high exhaust temperature operation with a resultant dense smoke cloud. This potential problem would be alleviated therefore by high ignition quality fuels. These foregoing deductions were validated by operating under conditions of total misfire, followed by full load operation and by idling followed by three-quarter load operation. From this study, it was therefore concluded that high cetane number is commensurate with low exhaust smoke in service owing to its ability to support combustion under adverse light load operating conditions.

In the same paper, the authors present data relating exhaust smoke to cetane number for three engine operating conditions, namely idle, full rack acceleration after idle to 1400 rpm and equilibrium running at 1400 rpm full load. The same engine as for the previous studies was utilised. The fuels were specially blended to isolate the effects of fuel composition, ignition quality and volatility. This resulted in an ignition quality range of 20-70 cetane number for various volatilities defined by ASTM 50% points of 420, 450, 495 and 570°F. Results demonstrating the influence of cetane number are shown in Figure 12C. Only one fuel type defined by chemical composition is shown. The remaining fuel types returned the same smoke/ cetane relationship, only the magnitude of the response varied.

From Figure 12C it can be observed that high ignition quality favours low smoke at idle. There is also a strong volatility influence although this will be discussed separately in Appendix 2.

At full load, fuel gravity is reported as the controlling factor on smoke. A cetane influence was however observed on the maximum smoke output recorded during the full rack acceleration following idle, as shown in Figure 12C. It can be observed that lower cetane increases smoke. This is in agreement with the earlier results presented by the authors.

Other workers in the field have reported little influence or no correlation of exhaust smoke with cetane number.

Broering and Holtman (3) tested six fuels in three 4 stroke DI engines over the Federal smoke test procedure. Two engines were turbocharged. The fuels differed significantly in both volatility, gravity, chemical composition and cetane. Cetane varied between 33-47. No correlation with any fuel variable for the small difference observed was obtained.

Utilising a small 4 cylinder, 4 stroke IDI engine, Cray (22) reported negligible influence upon smoke over the load range at 1500 rpm, when using ignition improving additives. No cetane data were given.

Whilst investigating the effects of fuel percent hydrogen content on smoke, Voorhies et al (23) found no correlation with smoke and cetane number. Ignition quality varied from 37.5 - 55.5. A Waukesha CFR engine with compression ratio set to 23:1 and fixed injection timing was used and tests were made at 800 and 1000 rpm.

Schweitzer (24) provides data which indicate that in an unspecified engine with adequate compression ratio for successful combustion, cetane number within the range 44-60 has no effect upon smoke over the load range. These results were independent of volatility.

A 426 CID, 2 stroke, naturally aspirated DI engine and a turbocharged, 672 CID, 4 stroke DI engine were tested by Gross and Murphy (25). Both engines were rated at 2100 rpm and had respective compression ratios of 18.7 and 15.0. The former engine was evaluated for smoke over the full load speed range. The latter engine was tested over the Federal smoke test cycle. A range of fuels was evaluated which differed in volatility, gravity, cetane and aromaticity. Cetane number varied between 29 and 48. Following a number of statistical correlations of various fuel variables and smoke, all were rejected, including cetane, except the volume of fuel distilling above 640°F.

3. THE INFLUENCE OF CETANE NUMBER UPON ENGINE PERFORMANCE

3.1 Ignition Delay

Data extracted from several sources are compiled in Figure 13. These data cover both DI and IDI combustion systems. By definition, reducing cetane number increases ignition delay and this is evident from Figure 13.

According to Shell data (26), the delay period measured in an IDI engine at high load, 3000 rpm, increases more markedly for cetane numbers below approximately 55. This trend was also demonstrated down the load range although the absolute value of ignition delay was marginally reduced. The results reported by Landen (13), Fiat (27), and Tsao et al (29) also indicate a tendency for increased sensitivity of the delay period at the lower cetane numbers. Ricardo (14) observed similar trends in the measured delay at idle. Reducing the cetane rating from 56 to 20 trebled the delay period whilst delay period only varied by 12% for cetane ratings between 52 and 100.

The Ricardo IDI data (20) shown in Figure 13 indicated a mixed response in the relatively high cetane range of approximately 50-70. For fuels of relatively low front end volatility, the response is generally flat. For fuels of relatively high front end volatility the sensitivity appears to be increased. In testing a 220 CID, 6 cylinder, swirl chamber IDI engine, Ricardo (30) observed no measurable difference in ignition delay when using fuels covering the cetane number range 40-56. Other fuel characteristics such as aromatic content and distillation characteristics were however different. These results were obtained at various part load operating conditions at two injection timings.

In the data supplied by Olson et al (28) the ignition delay response to reducing cetane number was more marked at low speed and load. Furthermore, it is also reported that of the three engines used to compile their data, the IDI engine proved the least sensitive to cetane number from the standpoint of ignition delay.

Sczomak and Henein (31) examined fuels of varying ignition quality in a pre-chamber engine and deduced that cyclic variations, defined by variations in the ignition delay, increased sharply for cetane numbers below 20.

3.2 Rate of Pressure Rise and Peak Pressure

Commensurate with increased ignition delay, rates of pressure rise and peak pressures are usually increased with reduced cetane number. The relationship between average rate of pressure rise and cetane number as extracted from two sources is shown in Figure 14.

The data presented by Olson et al (28) Figure 14, indicates that for a three engine average, reducing cetane number from 50 to 22 increases mean rate of pressure rise by 33 and 47% at low speed, low load and high speed, high load respectively. Contrary to the trends reported for ignition delay, the IDI engine in this case proved more sensitive to changes in cetane number than either of the DI engines. Although the authors correlated both ignition delay and rate of pressure rise with cetane number, peak cylinder pressure correlated poorly with all fuel properties and was dependent upon engine design and operating conditions. However, it is concluded that at maximum load, peak combustion pressures and average rates of pressure rise increase with increasing delay and hence reducing cetane. At minimum load conditions, the rate of pressure rise still increased with increasing delay periods, and hence reducing cetane, although peak pressures decreased owing to retarded combustion.

In the data supplied by Landen (13) Figure 14, it can be observed that as a general trend reducing cetane rating increases rate of pressure rise, the response being very pronounced with cetane numbers below 50. This general trend is in accord with the ignition delay data, Figure 13. It will also be observed that compression ratio has a secondary influence compared with injection timing.

Peak cylinder pressures are reported as higher with the lower cetane fuels, the advanced injection timings and the higher compression ratio.

Shell (26) furnished peak pressure data for their IDI engine load range tests at 3000 rpm, carried out using three fuels of different cetane number but similar distillation characteristics. The data reveal, for example, at a load of 80 psi bmep, that reducing cetane rating from 70 to 55 marginally increased peak pressure. This is in agreement with the small change in ignition delay reported in Figure 13. A further reduction in cetane to 40 resulted in the lowest peak pressure of all the three fuels. This would indicate retarded combustion owing to the significant elevation in delay period (as shown in Figure 13). Similar trends were recorded over the load range.

In tests carried out by Ricardo (32), little influence upon peak cylinder pressure was recorded over the full load speed range when comparing a 52 cetane fuel and the same fuel ignition improved to 62. A small high speed, swirl chamber IDI engine was utilised.

3.3 Fuel Consumption

Barrett and Freeston (26) recorded fuel consumption over the load range in both DI and IDI engines when using fuels of varying cetane numbers but similar distillation characteristics. These data are shown in Figure 15A. In the case of the DI engine, the bsfc is marginally worse over the load range for the lowest 40 cetane fuel. Similar results are evident for the IDI engine. These comments are only valid if the individual fuels have identical heat content on a mass basis. Other studies by the same authors appear to demonstrate that the volumetric fuel consumption of a diesel car was not affected by cetane number in the test range of 37-70. These results are shown in Figure 15B. The fuel consumption was corrected for gravity differences.

Landen (13) recorded bsfc for 12 fuels with various properties including a cetane range of 40-92, when operating a single cylinder DI engine at 1800 rpm and 95 psi bmep. Two compression ratios, 12 and 15:1 were employed and injection timing was adjusted accordingly. The data indicate that the bsfc was optimum for a cetane number of 55. Increasing cetane rating to 90 worsened fuel consumption whilst negligible change occurred when lowering ignition quality to 40. By inference, gravimetric heating value of the fuels was similar.

At equivalent injection timing, Shahed et al (33) observed worse bsfc when reducing cetane number from 56 to 52 to 40. Tests were made at constant speed and load in a 4 stroke DI engine. Again, trends are only valid for fuels of the same gravimetric heating value. In the case of the 52 and 40 cetane fuels, this may be a valid assumption since small quantities of ignition improver were used. These data are shown in Figure 21.

Wetmiller and Endsley (21) attributed increased idling fuel consumption on a volumetric basis to fuels of lower cetane ratings. Results are shown in Figure 15C. The trend was very marked below a cetane number of 40. The results are attributed to a degradation in light load combustion efficiency. Gravity was noted to be the controlling fuel parameter on full load economy.

Duval and Lys (12) present data which indicate very marginally better bsfc when reducing ignition quality from 60-52 in both a high speed swirl chamber IDI engine and a DI 'M' system engine. Other leading fuel parameters were constant and fuel consumption data were corrected to a reference gravimetric heating value. Engines were operated to the manufacturer's recommended settings.

Other works in the field (8, 30 , 32) record little or no significant influence of cetane number upon fuel consumption.

3.4 High Speed Misfire

High ignition quality assists in the avoidance of high speed, light load misfire, especially in high speed IDI engines. Results confirming these trends are reported in various references (5, 11, 32, 34).

Data according to Shell illustrating this point are shown in Figure 16. From Figure 16, it can be observed that the misfiring tendency is higher over the entire load range with the low cetane fuel, but is especially marked at low load. Shell argue that under these conditions misfire results as a function of the formation of weak mixtures which are not readily combustible.

Ricardo data (34) demonstrate that more advanced injection timings are required to clear high speed, light load misfire with low ignition quality fuels. These data are shown in Figure 24. The results can be attributed to the necessity to compensate longer ignition delays with low cetane fuels. Ricardo (32) also report the ability of higher ignition quality fuels to improve significantly light load misfiring tendency in two swirl chamber

IDI engines. Fuels of 52 and 62 cetane number were evaluated.

4. THE INFLUENCE OF CETANE NUMBER UPON HYDROCARBON (HC) EMISSIONS

Various recent studies have been carried out to investigate the influence of fuel ignition quality upon HC emissions. Generally, all of the data are in accord indicating that HC emissions are increased as fuel ignition quality is reduced. The sensitivity observed is, however, significantly different.

In four quiescent combustion chamber DI engines, Broering and Holtman (3) correlated an increase in HC emissions with reducing cetane over the Federal 13 mode test cycle. Engines were both turbocharged and naturally aspirated. Data from their studies are shown in Figure 17A. Whilst HC emissions always increased with reducing cetane number, as defined by the range 50-33, not all engines showed the same sensitivity. Nor could sensitivity be categorised by whether the engine was turbocharged or naturally aspirated. Significantly, however, the retarded injection, low NOx prototype engine was particularly sensitive to reducing cetane. In this engine, reducing cetane number from 48 to 33 increased HC levels sixfold. The influence of cetane upon HC emissions was demonstrated to be prevalent at the light load conditions as shown in Figure 17B.

These effects are attributed by the authors to retarded combustion following the increased delay period with the lower cetane fuels. However, in two turbocharged swirling combustion chamber DI engines, sensitivity of HC emissions to reducing cetane was virtually non-existent. These data are shown in Figure 17C.

Utilising a turbocharged 4 stroke and a naturally aspirated 2 stroke, Gross and Murphy (25) correlated increasing HC emissions with reducing ignition quality within the test range of 55-30 cetane. Both engines were tested over the 13 mode cycle and had a rated speed of 2100 rpm. The 4 stroke engine was significantly more sensitive to cetane reduction although both engines proved more sensitive for ignition quality ranging below 40-45 cetane. The authors report that for both engines the observed HC increases were predominantly at light load. Data extracted from this study are shown in Figure 18.

Bertodo (35) reports similar findings. In a 236 CID DI engine, reducing cetane number below 48 trebled light load HC emissions.

Marshall and Fleming (36) evaluated three DI engines encompassing naturally aspirated 4 stroke and an air scavenged 2 stroke. From their fuel set a correlation was made for 13 mode cycle HC emissions with volatility and cetane index. The correlations are reported as not very significant however. The sensitivity to reducing cetane index was more marked below approximately 42 for all three engines.

Mobil Oil (37) evaluated a range of heavy duty DI, US production engines. A statistical fuel design was utilised with major properties varying considerably. Cetane number ranged between 37 and 67. The 13 mode cycle HC emissions were noted to increase with reducing cetane number in three out of the four test engines, although the results are reported as not statistically significant over the range tested.

Shell (11) demonstrated increased HC emissions for a vehicle equipped with an IDI diesel engine operated on the chassis dynamometer to both Californian and European test procedures. Four fuels were evaluated with cetane numbers of 45 and 58, arranged in pairs of equal mid boiling point.

The results indicated that a low cetane, more volatile fuel (similar to US No. 1) increased HC emissions in both test cycles by 70-50%. Cold start performance was not responsible since the European procedure was driven from a hot start.

Duval and Lys (12) recorded elevated 13 mode cycle HC emissions when reducing cetane number from 60-52 for fuels of equal distillation characteristics, viscosity and aromatic content. High speed swirl chamber IDI and 'M' system DI engines were evaluated. Engine settings with respect to injection timing were left in accordance with the manufacturer's recommendations. HC emissions increased 11% and 30% for the IDI and DI engines respectively. Particular operating modes were not held responsible for these trends.

The trends reported by Duval and Lys are in accord with work carried out by BP (17). This paper reports that IDI engines were less sensitive to cetane changes as regards HC emissions than DI engines.

In testing a low speed, IDI diesel engine with very retarded injection, El Nesr et al (19) discovered that in reducing cetane number from 71 to 23, formaldehyde emissions at no load increased by a factor of nearly 3. The sensitivity was particularly pronounced in the cetane number range below approximately 45.

G.M. (38) examined 46 fuels in a light duty swirl chamber IDI diesel engine. In these studies, low HC emissions were noted to correlate with higher ignition quality.

5. THE INFLUENCE OF CETANE NUMBER UPON CARBON MONOXIDE (CO) EMISSIONS

Gross and Murphy (25) obtained a relationship between CO, cetane number and fuel 90% point in their experiments with a 2 stroke naturally aspirated engine and a 4 stroke turbocharged engine. Both engines were direct injection. Although the turbocharged engine produced lower 13 mode cycle CO emissions, the sensitivity of CO increase with cetane number reduction in the range of 52 to 28, for fuels of equal 90% point, was very similar in both engines i.e. a CO increase of approximately 2.0 - 2.5 g/bhp-hr. Such increases would not be problematical in meeting heavy duty emissions legislation. Data from these studies are presented in Figure 19A.

For quiescent chamber DI engines, Broering and Holtman (3) report similar findings. Their results are shown in Figure 19B. From these results, it can be observed that the turbocharged engines are less responsive to cetane number compared with the naturally aspirated engines, at approximately 0.7 and 2 g/bhp h CO increase for the cetane range 48-33 respectively. Similar results were also recorded in two turbocharged swirling chamber DI engines.

Hills and Schleyerbach (37) observed a statistically significant increase in 13 mode cycle CO emissions for cetane reduction within the range 71-38 for only one engine out of four. This engine was a heavy duty naturally aspirated DI unit. In another similar engine, however, reducing cetane number attenuated CO emissions.

According to Duval and Lys (12), reducing cetane number from 60 to 52 whilst maintaining other leading fuel parameters constant, reduced 13 mode cycle CO emissions by 15% in a small high speed swirl chamber IDI engine. The same fuels returned an increase of 17% when reducing ignition quality in an 'M' system DI engine. Both engines were operated to the manufacturer's recommended settings.

Utilising a low speed, retarded injection IDI engine, El Nesr et al (19) correlated increasing full load CO emissions to reducing cetane number as defined by the range 71-23. Overall, CO emissions increased by approximately 250%, although the sensitivity was more marked in the lower cetane range. Tests were made with both naturally and additive varying ignition quality.

With ignition quality improved by additives from 42 to 60, Shell (11) recorded increased CO emissions at light loads but the converse at high loads. These results were obtained from a DI engine. In cyclic tests with a car equipped with an IDI engine, the results indicated that CO emissions were influenced by cetane number, the lower ignition quality fuel returning a 19% increase in CO emissions.

6. THE INFLUENCE OF CETANE NUMBER UPON THE EMISSIONS OF NITROGEN OXIDES (NOx)

Broering and Holtman (3) found higher NOx with fuels of low cetane number. Engines were both naturally aspirated and turbocharged encompassing quiescent and turbulent chamber DI combustion systems. Extracted data are shown in Figure 20A. From this data it can be observed that the increase in NOx is not linear with reducing cetane number, the sensitivity generally being more marked at the lower cetane values. Reducing ignition quality from 48 to 33 results in a 13 mode cycle NOx increase of between 8 and 34% (mean 21%) according to these data. The sensitivity of NOx to ignition quality does not appear to be related to combustion chamber type, method of aspiration or base engine NOx level. The report indicates that the increased NOx output with fuels of lower cetane number can be attributed to the 13 mode cycle operating modes above 25% load factor as shown in Figure 20B.

Sensitivities in accord with this are reported by Bertodo (35). In a DI engine, NOx levels increased by 25% when reducing cetane number from 50 to 35. Ignition delay increased by 30%.

Similar observations are reported by Burt and Troth

(11) who recorded up to 20% increase in NOx emissions when reducing cetane number from 54 to 44. Engine type or test procedure were not specified.

Shahed et al (33) obtained the NOx-bsfc trade-off curves shown in Figure 21 with fuels of cetane number 40, 52 and 56 for wide variations in injection timing at equilibrium operating conditions. The 52 cetane fuel was obtained from the 40 cetane fuel by ignition improvers. The remaining fuel had a naturally high cetane number. A single cylinder 4 stroke DI engine was utilised. The results reveal that at either constant injection timing or bsfc the lower ignition quality fuels increase the NOx output, particularly at the more retarded timings. The effect is also generally more pronounced when comparing the 40 and 52 cetane fuels. Furthermore, at low and intermediate NOx levels, varying the injection timing for equivalent NOx emissions results in worse fuel consumption owing to the necessity for retard with the lower ignition quality fuels.

El Nesr et al (19) tested a variety of fuels ranging from 71 to 23 cetane number in a single cylinder, retarded injection, low speed IDI engine. Running at 75% of full load and full speed, NOx concentration was found to be a minimum for a cetane rating of approximately 60. Reducing ignition quality below this value increased NOx output. In the range 45 to 20 cetane, NOx increased at the rate of approximately 5ppm per reduction of cetane number. When using certain ignition improving additives containing nitrogen, NOx levels were higher at a given cetane compared with natural fuels. This was attributed to the nitrogen content of the additive.

Murayama and Tsukahara (39) also observed similar trends using ignition improved diesel, kerosene and gasoline fuels in a single cylinder, pre-chamber engine operated over the load range at 2000 rpm with fixed injection timing. For the 55 cetane diesel fuel, raising ignition quality to 68 cetane resulted in somewhat higher NOx emissions over the majority of the load range. For kerosene, undoped gasoline and leaded gasoline (43, 35 and 16 cetane number respectively), raising ignition quality by approximately 10-15 cetane numbers resulted in lower NOx output in the mid load range. This trend was most pronounced with the low cetane, highly volatile gasoline fuels. These results are attributed to lower rates of heat release with the higher cetane fuels as a result of shorter ignition delays and less pre-mixed combustion. In the case of the diesel fuel having adequate cetane number, the reversed trend is presumably due to over-advanced combustion.

In studying seven fuels in three heavy duty, naturally aspirated DI engines, encompassing both 2 and 4 stroke cycles, Marshall and Fleming (36) recorded that NOx was strongly correlated with aromatics content in one engine, but only weakly in the other two engines. Examination of their data shows that NOx correlates reasonably well with cetane index for two of the three engines and may therefore represent the best correlation with NOx for this study. In this manner, the data suggest that NOx emissions increase with reducing cetane index within the test band of 37-50.

B.P. (10,40) have shown that for two fuels of 35 and 59 cetane number, adjusting the injection timing to give comparable start of combustion in a 4 stroke DI engine increased NOx emissions for the fuel of lower ignition quality. This is related to the advanced injection requirement of the low cetane fuel to compensate the larger delay period, resulting in more pre-mixed, impulsive combustion with attendant higher cycle temperatures. With fixed injection timing however, NOx differences between the two fuels were virtually eliminated over the load range. This is attributed to retarded combustion with the low cetane fuel offsetting impulsive burning. Similar results were also obtained in an IDI engine. These data are shown in Figure 22.

Hills and Schleyerbach (37) found no fuel factor to be correlated with NOx emissions whilst Gross and Murphy (25) related NOx output to aromatic content. In the latter tests however, a cetane improver was utilised to raise the ignition quality of a highly aromatic fuel from 29 to 34 cetane number which resulted in a 7% 13 mode cycle NOx reduction. This result is reported as possibly significant although the influence of cetane number on NOx emissions appears to be prevalent only with very aromatic fuels, similar cetane differences in low aromatic fuels resulting in no change in NOx levels.

Duval and Lys (12) appear to have demonstrated some influence of cetane number on 13 mode cycle NOx emissions although the results have not been commented upon. Two fuels of equal distillation characteristics, aromatic content and viscosity were utilised with cetane numbers of 52 and 60. In a high speed, swirl chamber IDI engine, reducing ignition quality reduced NOx emissions by 10% whilst the exact converse was recorded for an 'M' system DI engine.

7. THE INFLUENCE OF CETANE NUMBER UPON ENGINE NOISE

Longer ignition delay, resulting in higher rates of pressure rise with fuels of low ignition quality, culminating in higher noise levels, has been demonstrated by Broering and Holtman (3) in tests on naturally aspirated and turbocharged quiescent chamber DI engines. The turbocharged engines were observed to be more resilient to reduction in ignition quality as regards noise output, this being especially true with a retarded, low NOx prototype model. Figure 23A shows the noise levels recorded for a cetane number range of 33-48.

Troth (5) reported similar data as shown in Figure 23B. These relationships show that, in this instance, noise increases more markedly as cetane number falls below 45. Other studies carried out by the same author indicate that for a small high speed IDI engine increasing cetane number above 55 has little effect upon reducing noise levels.

Corroborative results have been obtained by Burk et al (8). Fuels of equal viscosity and volatility, with cetane number varying from 37.5 to 60, were tested in a Caterpillar D3400 IDI engine at rated speed, from full load to no load. Reduced ignition quality increased engine noise with sensitivity of the relationship apparently being more pronounced below approximately 50 cetane number. Aural observations made with a General Motors 2 stroke DI engine whilst accelerating under load also returned increased noise output for low ignition quality fuels.

Work carried out by Sinclair Refining Company (7) with five commercial engines revealed increased levels of noise when reducing cetane number from 55 to 30-35. Four engines behaved in a similar fashion, with relative noise levels increased by a factor of 2-2.5, whilst one engine proved relatively insensitive. Details of engine types, test procedures or test fuels are not, however, presented.

Fiat Research Centre (27) tested a small, high speed swirl chamber IDI engine on various fuels of different chemical composition and cetane numbers of 47-64. Noise measured at 2500 rpm, two-thirds maximum load showed a non-linear increase with reducing ignition quality, the increase being most pronounced between 60 and 50 cetane number. This trend was not matched by the measured delay period which followed a linear increase from 15-22 degrees crank angle for a cetane reduction from 64 to 47. At the same test conditions, another fuel set of different chemical composition but covering the same cetane range was evaluated. In this instance, combustion noise changed little while the delay period varied in a similar fashion as previously. The authors therefore concluded that chemical composition influenced noise and that a definite relationship between noise and delay period (hence ignition quality) does not exist. These views were verified by using fuel blends to give a range of chemical compositions at fixed cetane number.

According to Ricardo (32), increasing cetane number from 52 to 60 with an ignition improver resulted in no observed difference in noise levels, at fixed injection timing with two light duty, high speed swirl chamber IDI engines. A small reduction in noise was, however, possible with the high cetane fuel owing to the ability to be able to retard the injection timing slightly without incurring high speed misfire problems. The need to operate high speed swirl chamber IDI engines with relatively advanced injection timings for low cetane fuels, to avoid high speed misfire, and the implications on approaching the full load noise threshold is shown in Figure 24 (34). Low cetane fuels can therefore indirectly have an adverse affect upon noise levels because of requirements in other areas.

The data shown in Figure 24 reveals that cetane within the range 44-64 had no influence upon full load noise, despite the increased delay periods reported. Other studies by Ricardo (20), employing a high speed swirl chamber IDI engine, showed no change in noise level over the speed range at full load when varying cetane number between 45 and 75. Similar results were obtained at part load except for a slight tendency for noise attenuation with higher ignition quality fuels at speeds below 2000 rpm.

8. THE INFLUENCE OF CETANE NUMBER UPON EXHAUST ODOUR

Recording exhaust odour as aldehyde content, Wetmiller and Endsley (21) recorded maximum aldehyde output at light load with an unspecified diesel engine operating at steady state conditions, 1400 rpm. For a given fuel, reducing load from full load attenuated aldehyde emissions to a minimum at part load. Further reduction in fuelling resulted in a marked rise in aldehyde output culminating at a peak in the proximity of the lean limit of combustion. Typical results are shown in Figure 25A. These results are analogous with the smoke data recorded by the same authors and reported in section 2.2 of this Appendix. As with those data, very low fuelling levels were achieved by motoring with fuel injection.

From Figure 25A it can be observed that increased fuel ignition quality allows lighter loads to be achieved before the sharp rise in aldehyde output occurs and, in addition, allows leaner operation before the aldehyde peak, signified by the lean limit, is reached. Furthermore, at the lean limit, higher cetane fuels reduce the magnitude of the aldehyde peak. The observed fuelling level for the aldehyde peak and the corresponding misfire limit are shown in Figure 25B for fuels covering the cetane range 20 to 72.

With several fuel blends designed to isolate the individual effects of fuel composition, ignition quality and volatility, aldehyde emissions were measured at idle. The results were found to correlate with ignition quality and a composite plot for all the fuels tested is shown in Figure 25C. From these data it can be observed that reducing cetane number results in elevated levels of aldehyde emissions at idle, the effect being more pronounced for cetane numbers below approximately 50. In this instance, correlation of odour as measured by aldehyde content and a human panel was obtained.

A fuel treated with iso-propyl nitrate to vary ignition quality was tested by BP (10) in a multi-cylinder DI engine to observe the possible relationship at 1200 rpm. The results obtained are shown in Figure 26 and suggest that raising cetane number from 57 to 61 or 64 improves exhaust odour at light load.

Ainsley et al (7) report a marked increase in exhaust odour when reducing cetane number from 55 through to 30-35. The correlation was apparently established from tests with five commercial engines, although details of engine, test procedures and fuels are unfortunately not presented.

Burk et al (8) evaluated a 6 cylinder engine incorporating a Lanova type combustion chamber for exhaust odour with a range of fuels covering several prime variables, including a cetane number range of 37.5-60. From these tests it was concluded that within the limits of the experiment, no single fuel produced sufficient aldehyde emissions to result in objectionable odour output. Comparisons of individual fuels are however not presented.

Other workers in the field(36, 37) did not relate exhaust odour to fuel ignition quality. Hills and Schleyerbach (37) choose fuel distillation characteristics and aromatic content to correlate with exhaust odour and not ignition quality, although this parameter varied considerably. Marshall and Fleming (36) selected fuel sulphur content to correlate with odour in their tests with 2 and 4 stroke DI engines. Perhaps significantly however, cetane index appears to correlate reasonably well with odour for the two 4 stroke engines, in such a manner that odour increased with reducing cetane index when examined over the 13 mode cycle.

9. THE INFLUENCE OF CETANE NUMBER ON ENGINE DEPOSITS AND WEAR

Both references 7 and 8 indicate that reducing cetane number increases engine deposits. Burk et al (8) comment upon reduced combustion chamber deposits and piston lacquer when utilising fuels of higher cetane number defined by the test range of 37.5-60. Fuel viscosity and distillation characteristics were rationalised. A General Motors 2 stroke DI engine operating to a cyclic, predominantly full load, test procedure was employed for these studies. Similar trends are presented in reference 7 with combustion chamber deposits being shown to increase with cetane number reduced from 55.

Scott (14) comments upon incidences when running on diesel fuels of detonation damage to piston crowns, believed in some cases to be associated with engine operation under conditions of unusually long ignition delay. It is, therefore, speculated that reduced ignition quality of the fuel could incur such problems in certain instances. This has in fact been observed by Bertodo (35) in a DI engine.

APPENDIX 2

THE INFLUENCE OF FUEL VOLATILITY

1. THE INFLUENCE OF FUEL VOLATILITY ON COLD STARTABILITY

Derry and Evans (4) examined the cold starting performance of several different diesel fuels in a range of commercially available diesel engines encompassing both DI and IDI combustion systems. It was concluded that volatility influenced cold startability along with other parameters, of which ignition quality was generally the most important. Increased volatility generally improved starting and became more important for low cetane fuels. Analysis revealed that for volatile fuels (final boiling point below 572^{o}F), minimum starting temperature was improved by 21^{o}F for fuels of 30 cetane number but only by 9^{o}F at 60 cetane number. Data illustrating the beneficial influence of volatile fuels are given in Figure 27. In this case, the importance of volatility at lower cetane ratings is not very marked and this may be attributable to the relatively low compression ratio of the test engine.

Similarly, studies by Shell (5, 6, 11) concluded that starting performance was improved with fuels of lower mid-boiling point. Withers (41) also acknowledges easier starting with more volatile fuels.

Conversely, Burk et al (8) found that starting performance of a multi-cylinder engine equipped with Lanova cell combustion chambers only correlated with cetane number in the expected fashion. No trends for other physical fuel properties including distillation characteristics which varied between 394-608^{o}F mid point were found. Starting was not evaluated below temperatures of 30^{o}F.

2. THE INFLUENCE OF FUEL VOLATILITY ON EXHAUST SMOKE

2.1 The Influence of Fuel Volatility on Exhaust Smoke Following Cold Starting

Shell (5, 6, 11) studied extensively the relationship between fuel properties and smoking tendency after cold starting. It was concluded that both high ignition quality and high volatility improved cold smoking performance. In tests with DI engines, it was deduced that an increase of one cetane number had the same beneficial action as a reduction of mid-boiling temperature of 36-43^{o}F. In studies using an IDI engine equipped with heater plugs, a change of plus one cetane number had the same effect as a reduction in mid-boiling point of 41^{o}F. In general terms, smoking time was halved by reducing mid-boiling point by approximately 60^{o}F.

Other published results (17) indicate the beneficial action of higher volatility fuels on curtailing white smoke emission following cold starting.

2.2 <u>The Influence of Fuel Volatility on Exhaust Smoke at Normal Engine Operating Temperatures</u>

Not all references consulted in this area agree although the majority indicate that improved fuel volatility produces lower levels of exhaust smoke. These findings are presented initially.

Figures 28, 29 and 30 illustrate that for a range of engines, improving the volatility of the fuel can effect a reduction in exhaust smoke.

Data submitted by Muller (42) are shown in Figure 28. Regarding front-end volatility, smoke increases with raised initial boiling point in all engines. Tests were conducted at constant speed with fixed high load factors. The DI engine, however, is barely sensitive in this respect. According to the author, AFR remained constant throughout the tests. Such trends are related by the author to less favourable fuel preparation with the air charge for fuels of lower volatility, coupled with the higher oxygen demand of individual, higher molecular weight fuel molecules. The sensitivity of the engines was also more pronounced for the engines running with the richest air fuel ratios. The IDI therefore proved the most sensitive (equivalence ratio 0.98) and the DI least sensitive (equivalence ratio 1.8). The 'M' system engine had intermediate sensitivity (equivalence ratio 1.4).

With variable final boiling point, Muller observed no change in exhaust smoke for the DI engine provided that initial boiling point was sufficiently low to encourage rapid vaporization. In the 'M' system engine, reducing final boiling point below approximately 572^{o}F improved smoke output. Air fuel ratio was observed to remain constant and the smoke reduction is attributed to improved vaporization of the fuel deposited on the wall of the combustion chamber. In the case of the IDI combustion system, smoke progressively fell with final boiling point lowered from the upper test limit of approximately 716^{o}F. In this case, improved heat liberation was observed with fuels of lower final boiling point resulting in lower smoke emissions. Presumably this implies reduced fuelling levels required to maintain test conditions.

Gross and Murphy (25) reported the smoke/volatility correlation shown in Figure 29A. In this case, lower smoke emissions were observed in two heavy duty DI engines operated at full load conditions when fuels having lower volumes distilling above 640^{o}F were utilised. Burk et al (8), however, observed that fuel 90% point and end point could be varied widely without appreciable influence on smoking at both full and part load conditions. This trend was only valid for fuels of fixed cetane number and viscosity. Lowering the mid-boiling point was however conducive to reduced smoke output as shown in Figure 29B. These results represent the mean of a four engine study with smoke readings averaged for various speeds at each load.

Schweitzer (24) also reports that fuel 50% point is a better criterion for judging smoking tendency than 90% point. Results depicting the observed smoke attenuation with fuels of lower mid-point are shown in Figure 29C. It will be seen that the trend is predominant at both high and light load, but is more marked at the higher loads. In addition, the sensitivity of increasing smoke at a given air fuel ratio at the higher load factors to higher boiling fuels is more pronounced for 50% points above 495^{o}F. Schweitzer also presents data indicating the marked ability for the CFR engine to accept richer air fuel ratios for constant smoke when reducing mid-boiling point from 550^{o}F to 150^{o}F.

Data according to Landen (13) are shown in Figure 6 and again illustrate the tendency for lower boiling fuels to emit less smoke at equivalent cetane rating. In this case the smoke emitted by the higher boiling fuels was of a bluish cast compared with black for the more volatile fuels.

Duval and Lys (12) obtained higher smoke output with fuels of extended distillation characteristics and higher viscosity. Smoke was improved when blending lighter products to enhance front-end volatility. Observations were made over the 13 mode cycle utilising engines having DI 'M'.system and swirl chamber IDI combustion systems. The DI engine appears to be the most sensitive system and the IDI the least.

In a report by Shell (6) data are submitted indicating lower smoke levels at equivalent high load factor in a DI engine for more volatile fuels (lower mid-point) of equivalent cetane number. According to Shell a study of the influence of volatility is complex owing to accompanying variations in the chemical composition of the fuel. In this respect, Shell (11) derived the data shown in Figure 5. Results were obtained from a DI engine operating at constant load. From Figure 5, it will be observed that for entirely paraffinic fuels, smoke responded to cetane number and was insensitive to volatility changes characterised by mid-boiling points of 383°F and 419°F. For paraffinic/aromatic blends, smoking tendency was higher than for entirely paraffinic fuels at equivalent ignition quality. For such blends, the higher boiling (491°F mid-point) fuel produced higher smoke output than the more volatile blend (383°F mid-point). According to Shell, this is not due to volatility per se, but to the higher content of diaromatic hydrocarbons.

Wetmiller and Endsley (21) concluded that lower mid-boiling point was advantageous to limiting maximum white smoke. Data illustrating this point recorded at idling are shown in Figure 12C. Wetmiller and Endsley also argue that more volatile fuels are not condensed to such a degree in the exhaust system, thus lowering the tendency to produce blue smoke plumes by expelling collected fuel when operating at high load factor, immediately following prolonged periods of operation under marginal combustion conditions i.e. idle. Data supporting this view are presented and shown in Figure 30, where it can be noted that fuels having mid-boiling points above 500°F are critical in this respect. Withers (41) also favours more volatile fuels to reduce blue smoking tendency produced in this manner.

Other published data consulted (16, 43, 44, 45) support the view that lower smoke emissions are favoured by more volatile fuels.

Not all workers in the field would agree on this point. Results published by Derry et al (15) are shown in Figure 31. Here it will be seen that at equivalent high load factor and cetane number, smoke output of the DI engine tested is elevated with more volatile fuels characterised by lower mid-boiling points.

Other published data (3, 23) report that no correlation of exhaust smoke data and fuel volatility could be determined whilst other papers (17, 37, 46) acknowledge little influence of fuel volatility upon smoking tendency.

3. THE INFLUENCE OF FUEL VOLATILITY ON ENGINE PERFORMANCE

Little data indicating an influence of fuel volatility on features of engine performance have been located in this search.

Regarding ignition delay, Landen (13) observed that, in a DI engine at high load factor, less volatile fuels had an average ignition delay some 4% greater than the more volatile fuels. Distillation ranges were very narrow however at 600-700°F and 400-500°F. Fuel volatility had very little effect upon either rate of pressure rise or peak cylinder pressures. Furthermore, fuel volatility had no influence upon combustion duration. With respect to specific fuel consumption, Landen reports that the higher distillation range fuels had an average 4% worse bsfc than the more volatile fuels. Although tests were made at constant load, it is not possible to deduce the significance of this result in the absence of other details.

Qualitative agreement is however furnished by Duval and Lys (12). In their tests with three engines encompassing DI, 'M' system and IDI combustion chambers, 13 mode cycle fuel consumption was between 1 and 5% worse when utilising fuels of extended distillation range. The DI engine proved the most sensitive in this respect and the IDI the least. Values were corrected to a reference fuel gravimetric energy level. The opposite trends are reported when fuel volatility was improved.

Other literature consulted (8, 21, 25, 37, 45, 46) reports no influence of fuel volatility upon either fuel economy or power.

4. THE INFLUENCE OF FUEL VOLATILITY UPON HYDROCARBON (HC) EMISSIONS

Many investigations (11, 12, 17, 36, 37, 38, 42, 46, 47, 48) have demonstrated that fuel volatility can influence HC emissions.

Data according to Hills and Schleyerbach (37) and Duval and Lys (12) are shown in Figure 32. Both sets of data demonstrate reduced 13 mode cycle HC emissions for fuels of higher 50% point. These data, which cover both DI (N/A and TC), 'M' system and swirl chamber IDI combustion systems, show remarkably similar sensitivities. Hills and Schleyerbach report that these statistically significant trends were observed in three out of four engines. There was no change in the fourth engine. Similar correlations were obtained for the 10% point. Fuel 95% point was not a statistically significant factor in affecting HC emissions from any of the engines.

Duval and Lys (12) also comment on the relationship between reduced HC emissions for fuels of higher 10% point and that fuel 90% point had no influence. Similar observations are made by Barry et al (46) with respect to the insensitivity of HC emissions to fuel 90% point. This paper also acknowledges higher HC emissions for fuels with low boiling components. G.M. (38) however examined 46 fuels in a light duty swirl chamber IDI engine and found fuel 90% point to be an important factor. High 90% point correlated with lower HC emissions. The data according to Ford et al (47) are shown in Figure 33 and relate decreased HC emissions to fuel of higher 10% point.

B.P. (17) report that for a given ignition quality, there is a tendency for the more volatile fuels to emit slightly higher HC emissions. Compared with DI engines, IDI engines are noted as being relatively insensitive in this respect. Shell (11) however demonstrated reduced cyclic HC emissions from a car fitted with a pre-chamber diesel engine when using fuels with higher 50% points. Fuels had constant cetane numbers. Shamah and Wagner (48) also observed incidences of more volatile fuels of similar cetane number resulting in significantly higher HC emissions.

Muller (42) carried out extensive studies with three engines incorporating DI, 'M' system and IDI combustion chambers and utilising fuels of fixed cetane number and aromatic content and either variable initial or final boiling points. Results obtained at constant speed and constant high load factor for normal engine operating temperatures are shown in Figure 34. Tests were carried out at fixed injection timings.

When varying initial boiling point with other fuel parameters fixed, both the DI and 'M' system engines revealed no change in HC emissions until a temperature of approximately 392°F was reached. For initial boiling points below this value, HC increased at a similar rate for both engines. These trends are attributed to early break up of the fuel spray with the more volatile fuels. This restricts penetration and mixing in the case of the DI engine and prevents complete wall impingement as designed, in the 'M' system engine. Fuel is also lost to areas such as valve pockets, etc. from which successful combustion is difficult. In the case of the IDI engine, HC rose progressively with increase of initial boiling point. This result is related to the partially burnt charge leaving the pre-chamber and being quenched, to some extent, in the relatively cool main chamber.

As regards variable final boiling point, no influence upon HC emissions was apparent in the DI and 'M' system engines until a temperature of 464°F was reached. Reduction in final boiling point below this value resulted in a similar marked rise in HC emissions for both engines. These trends are again attributed to reduced fuel penetration and dispersal of fuel to remote parts of the combustion chamber owing to the more volatile nature of the fuel. In the IDI chamber, HC emissions progressively fell as final boiling point was reduced. This trend is related to improved fuel vapourisation in the pre-chamber, resulting in less un-vaporised fuel reaching the cooler main chamber.

Similar trends are recorded for fuels of higher aromatic content although data were not presented.

Under cold idle conditions with fuels of equal cetane rating, decreased initial boiling point was favourable to attenuated HC emissions in both IDI and 'M' system engines (42). Data are not presented for the DI. The sensitivity in this respect was more marked for the highly aromatic fuel. According to the author, reduced fuel front-end temperature results in more favourable fuel preparation under cold cylinder conditions. For fuels of low aromaticity (15%), reduced final boiling point had little influence upon HC output from the DI and 'M' system engines but improved HC emissions were observed for the IDI chamber. For the highly aromatic fuels (45%), reduced final boiling point significantly increased HC emissions in both the IDI and 'M' system engines. No data were presented for the DI engine in this case.

In addition to the speculations made by Muller to explain the results observed in his studies, other researchers in the field (12, 37, 47, 48) argue that HC emissions are lower with less volatile fuels due to a reduction of fuel discharged from the uncontrolled volume below the injector needle. Reduced HC emissions with injector designs incorporating smaller uncontrolled volumes below the needle have been demonstrated (47).

Gross and Murphy (25) correlated 13 mode cycle HC emissions from two heavy duty engines to viscosity and cetane number. HC emissions reduced with more viscous fuels for a given ignition quality. For the fuel utilised, viscosity and mid-boiling point were highly correlated; therefore, by implication, HC decreases for increased 50% point. Limited tests using polymer thickened fuels at equivalent mid-point resulted in lower HC emissions so the authors currently favour correlation with fuel viscosity.

5. THE INFLUENCE OF FUEL VOLATILITY UPON CARBON MONOXIDE (CO) EMISSIONS

In the studies of Gross and Murphy (25), 13 mode cycle CO emissions from two heavy duty DI engines were most satisfactorily correlated with fuel 90% point and cetane number. These data are shown in Figure 19A. For a fixed cetane rating, CO emissions are decreased by lowering the fuel 90% point within the test range of 700-550°F. This trend was, however, not very significant in the turbocharged engine returning low CO emissions, whilst for the air scavenged two cycle engine, reported as running at AFRs near to stoichiometric at full load, the influence of fuel 90% point was pronounced.

Similarly, Duval and Lys (12) report that CO emissions over the 13 mode cycle were adversely affected by elevated distillation range. Tests were conducted at fixed injection timings and equal power settings using engines incorporating DI, 'M' system and swirl chamber IDI combustion chambers. These trends appear to be least evident in the IDI engine and maximum sensitivity was recorded for the DI combustion system.

Other workers in the field (11, 36, 37, 46) report little influence of volatility on CO emissions in tests covering both heavy duty DI and light duty IDI engines.

6. THE INFLUENCE OF FUEL VOLATILITY UPON THE EMISSIONS OF NITROGEN OXIDES (NOx)

Some research into this aspect has shown a fuel volatility/NOx relationship whilst other studies have indicated no effect.

Data according to Broering and Holtman (3) are shown in Figure 35A. For fuels of equivalent cetane number and 50% point, lowering initial boiling point by 54°F effected a mean 6% 13 mode cycle NOx increase in four engines. Results are attributed to the development of higher peak pressures and temperatures following the more homogeneous mixtures produced during the delay period. Perhaps significantly, the production turbocharged engine demonstrated the maximum per cent NOx increase, possibly as a function of increased inlet air temperatures encouraging fuel vaporization during the ignition delay. This was not the case for the retarded turbocharged engine possibly indicating a compensating effect due to late combustion.

Figure 35B presents data according to Gross and Murphy (25). In this study, NOx was proportional to aromatic content. Because aromatic content was highly correlated with the more readily available specific gravity and 50% point, the correlations were accordingly re-modelled resulting in the data shown in Figure 35B. These data show that, for a fixed gravity, NOx emissions are increased by reducing the fuel 50% point. These trends were evident in two radically different heavy duty DI engines and are in accord with the findings of Broering and Holtman.

According to Murayama and Tsukahara (39) however, more volatile fuels when compared on an equivalent cetane basis produce lower NOx emissions compared with heavier fuels in a pre-chamber engine. These results were obtained by comparing ignition improved gasoline with kerosine and ignition improved kerosine with diesel fuel and are shown in Figure 54. More rapid combustion with the volatile fuels owing to improved vaporization, resulting in reduced residence time for NOx formation is speculated as being responsible.

In a light duty swirl chamber IDI engine, G.M. (38) recorded similar trends in that fuels with low 90% point appeared related to low NOx output.

Other studies (11) report no influence of fuel volatility upon NOx emissions. In the report published by Muller (42), neither variable initial nor final boiling point for fuels of fixed aromatic content and cetane rating affected NOx emissions. Tests were conducted at high load steady state conditions employing engines representing DI, 'M' system and IDI combustion chambers. Hills and Schleyerbach (37) report no correlation of fuel variables with 13 mode cycle NOx emissions from four DI engines despite wide ranging fuel 10% and 95% points. Barry et al (46) comment upon NOx emissions being relatively unaffected by fuels of differing 90% points in a range of tests with both heavy duty DI and light duty IDI diesel engines. In studies carried out by the Amoco Oil Company (48), NOx emissions were also not influenced by fuel volatility at full load, part load and idle in a three engine study covering both DI and IDI combustion systems. Narrow cut fuels of similar ignition quality were employed.

7. THE INFLUENCE OF FUEL VOLATILITY UPON ENGINE NOISE

Whilst Fiat (27) note an influence upon combustion noise in high speed IDI engines of both cetane number and chemical composition, fuel viscosity and volatility are reported as having a negligible influence based upon results of unreported tests.

8. THE INFLUENCE OF FUEL VOLATILITY UPON EXHAUST ODOUR

Exhaust odour has been demonstrated to be affected by fuel volatility in studies carried out by Hills and Schleyerbach (37). Engine design had major effects and fuel 10% and 95% points had secondary influences.

At idle and part load, increased 10% point resulted in decreased odour. This effect was however only pronounced in one of three engines. Raising fuel 95% point decreased odour in all three engines with the effect being most noticeable at part load. These results are shown in Figure 36.

Ford et al (47) revealed that HC emissions and odour were attenuated by minimising the uncontrolled volume beneath the injector needle. In addition, for a given volume, HC emissions were decreased when raising the 10% point of the fuel by reducing the amount of fuel discharged from the uncontrolled volume due to expansion and evaporation. By implication, increased 10% point of the fuel should therefore reduce exhaust odour. This is in accord with the data reported by Hills and Schleyerbach (37).

According to Wetmiller and Endsley (21), cetane number was the only controlling fuel variable on exhaust odour at idle. Engine type was not specified. Marshall and Fleming (36) did not include distillation characteristics to correlate with exhaust odour but chose fuel sulphur content.

9. THE INFLUENCE OF FUEL VOLATILITY ON ENGINE DEPOSITS AND WEAR

In a test programme carried out by Amoco (49), raising fuel volatility incurred slightly lower levels of combustion chamber deposits and top ring wear. Fuels tested had fixed sulphur levels and aromatic content. Distillation range (10% - 90%) was varied between 380/520°F - 460/590°F. Results were obtained from four production diesel engines run over a 250 hour mixed part load/full load test cycle.

B.P. (41) have observed that fuels having less than 90% distilling to 662°F aggravate injector nozzle deposits as the volatility is further reduced. More volatile fuels therefore improve deposits in this respect.

Burk et al (8) recorded combustion chamber deposits and piston lacquering in a heavy duty two stroke DI engine using fuels of comparable properties except for boiling range. Three fuels were evaluated having boiling ranges of 484-558°F, 371-654°F and 333-700°F. The latter two fuels had little influence upon deposits and piston lacquering, but the narrow cut fuel, with significantly lower end point, produced noticeably lower levels of both deposits and piston lacquer.

APPENDIX 3

THE INFLUENCE OF FUEL CHEMICAL COMPOSITION

1. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION ON EXHAUST SMOKE AT NORMAL ENGINE OPERATING TEMPERATURES

Figure 37 presents a compilation of the influence of aromatic content upon exhaust smoke according to several sources.

In the data submitted by Hills and Schleyerbach (37), two similar engines returned opposite trends. One engine gave increased smoke for the more aromatic fuels although the results are only stated as significant for aromatic content above 28%. Despite these data the paper concludes that smoke was relatively unaffected by fuel changes within the range tested.

With the exception of the DI 'M' system engine, all of the other data shown in Figure 37 indicates increasing exhaust smoke with the higher aromatic fuels.

The results according to Lindeman et al (49) were obtained from fuels of equal distillation characteristics and gravity effects were precluded by testing at constant power. Ignition quality data were not provided. B.P. (17) also adjusted for equal power but no fuel data other than gravity are given.

Muller (42) obtained a smoke/aromatic correlation in three engines utilising fuels of equal initial and final boiling points (302 and 662°F respectively). Cetane number was also constant at 45. Each engine was tested at an identified load and speed which was kept constant for each fuel. Injection timing was fixed. Muller extensively explains the reasons for the trends observed as detailed below.

In the DI engine, the ignition delay was observed to increase with higher aromaticity despite constant cetane number. This resulted in a loss in bmep and a resultant increase in fuelling culminating in slighty higher smoke output. For the IDI engine a similar situation was prevalent. The very high smoke levels even with the low aromatic fuels are however attributed to the AFR reaching stoichiometric. With the DI 'M' system engine, increasing aromatic content affected ignition delay in such a manner as to require slightly increased fuelling levels for constant load. In this instance smoke reduced slightly with increased aromatic content suggesting an association of wall wetting and the increased resistance of aromatics to cracking.

Bertodo (35) varied aromatic content between 15 and 30% in a DI engine. As the aromatic content was progressively raised to the upper limit, black smoke became unacceptable. Longer ignition delay was held responsible although no ignition quality data are given.

Other workers in the field (11, 12) report a tendency for increasing aromatics content to increase smoke in both DI and IDI combustion systems. The data according to Shell (11) are shown in Figure 5. In the studies by Gross and Murphy (25), increasing smoke was correlated in two engines with the volume of fuel distilling above 640°F. Whilst this may imply increasing aromatics content, the authors concluded that no aromatic effects were present.

Landen (13) recorded no appreciable influence of increasing aromatics content upon exhaust smoke. Tests were made by blending benzene and alpha methyl naphthalene with a normal paraffin fuel (cetane). The blends had a cetane rating of 50 compared with 100 for straight cetane.

Ricardo (30) also observed no significant influence of fuel specification upon smoke at several part load conditions when testing a light duty swirl chamber IDI engine at both advanced and retarded injection settings. Fuels varied in all respects including aromatics which ranged 19-45%. Similarly, Marshall and Hurn (50) concluded neglible effect of fuel properties on smoke emissions in a heavy duty DI engine. Aromatics varied between 14 and 44%.

Standard Oil Company (44) observed increased smoke output in heavy duty engines when comparing catalytically cracked fuels with straight run fuels of similar distillation characteristics.

Six diesel fuels with hydrogen content ranging from 12.4 - 14.4% were evaluated by Voorhies et al (23) in a Waukesha CFR, IDI engine at low speed with fixed injection timing. At a given air fuel ratio, smoke decreased with increasing per cent hydrogen in the fuel. Neither fuel ignition quality nor volatility provided a satisfactory correlation. The trends are attributed to faster rates of carbon combustion as a result of the increased hydrogen to carbon ratio. By inference, these results are commensurate with higher smoke emissions from more aromatic fuels.

2. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION UPON ENGINE PERFORMANCE

Isolated references to the influence of fuel chemical composition upon engine performance, other than the direct implications of chemical composition upon gravity and hence power at fixed rack, have been located, as follows:

Olson et al (28) recorded increased ignition delay at light load in three engines when blending increasing quantities of mixed aromatics with JP-4 base fuel. Such trends however are explainable by ignition quality differences according to the authors. Similar results were observed when blending mixed olefins with the base JP-4 fuel.

Muller (42), however, recorded that increased aromatic content raised ignition delay despite constant cetane number in three engines encompassing both DI and IDI combustion systems. This resulted in increased fuelling levels being required to maintain constant, relatively high loads with fixed injection timings. Increased fuelling resulted in higher smoke output with the more aromatic fuels in two out of three engines. By inference, fuel consumption was therefore worse for all engines.

According to Duval and Lys (12), raising aromatics content within the range 20-40% with other leading fuel parameters held essentially constant, increased fuel consumption by 1.5-3% in 13 mode cycle tests on both DI and high speed IDI engines. These small increases appear significant in view of the standard deviations quoted. It would seem that the DI engine tested, as opposed to the DI 'M' system and IDI engines, was more sensitive to increased aromatics content in respect of fuel consumption. Fuel consumption data were corrected to a reference energy/mass value.

According to Hills and Schleyerbach (37), specific fuel consumption, power and peak torque did not change when testing various heavy duty engines, regardless of fuel properties. In their fuel set aromatics varied between 14 and 48%. Ricardo (30) also noted no significant influence of various fuels with wide ranging properties including aromatic content of 19-45%. Tests were conducted at part load with two injection settings in a high speed swirl chamber IDI engine.

3. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION UPON HYDROCARBON (HC) EMISSIONS

For DI, including 'M' system, and IDI combustion systems, Muller (42) recorded a linear increase in HC emissions for fuels of greater aromatic content when idling with controlled cylinder head temperatures of 23^{o}F. Fuels had equal boiling ranges and a constant cetane number of 45. HC emissions were increased by a factor of 2-8 dependent upon engine combustion system for an aromatic range of 10-50%. The sensitivity of the IDI and DI engines was very similar but the 'M' system diesel was significantly less sensitive. The author attributed the results to the higher temperatures required for spontaneous ignition of aromatics not being reached in all parts of the combustion chamber. Results obtained are shown in Figure 38.

Under equal conditions of high load with normal operating temperatures, however, HC increased very little for higher aromatic fuels of constant boiling range and cetane number, except in the IDI engine, where a marked increase was observed. These results are also shown in Figure 38. This latter trend was attributed, however, to the very rich air fuel ratios under which the IDI engine was operated at the chosen test condition. Reducing load factor and raising the excess air ratio revealed only a slight influence of aromatics upon HC levels according to the author, the sensitivity then being very similar to the two DI engines.

In Reference 12, data are presented which demonstrate an adverse influence of higher aromatic fuels on 13 mode cycle HC emissions. Increases of 20-60% were observed in tests on three engines incorporating swirl chamber IDI, DI and 'M' system combustion chambers. Fuels ranged in aromatics content from 20 to 40% whilst other parameters were held essentially constant. The 'M' system engine appears to be the most sensitive in this respect and the IDI the least. Despite other fuel parameters being held essentially constant while aromatic content varied, the authors speculate that the generally slightly lower cetane rating accompanying increased aromatic content, coupled with other minor variations such as viscosity, may be concealing the true effect of the aromatic content.

G.M. (38) observed increased HC emissions from a light duty, swirl chamber IDI engine with fuel sets blended to increase aromatic content. In analysing 46 other fuels however, aromaticity was not included in the HC correlation developed.

Other workers in the field (37, 50) have reported little or no influence of chemical composition of the fuel upon HC emissions. Other studies (25, 30) have chosen other fuel parameters to best correlate their observations. The work covered by these studies included both DI heavy duty engines (25) and a light duty, swirl chamber IDI engine (30).

4. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION UPON CARBON MONOXIDE (CO) EMISSIONS

In the tests conducted by Mobil (37), a small but statistically significant increase in 13 mode cycle

CO emissions with higher aromatic fuels was observed in two out of four DI engines. Duval and Lys (12) noted CO emissions increasing in three engines representing swirl chamber IDI, DI and 'M' system combustion chambers for fuels of higher aromatic content. In the aromatic range 20-40%, CO was seen to increase by 5-17%. The 'M' system diesel was the most sensitive engine whilst both the DI and IDI were marginally affected.

B.P. (17) report a 7% increase in CO emissions when raising aromatic content of the fuel from 3 to 21% (by weight). Except for gravity, no other fuel data are documented. Tests were carried out on a multi-cylinder DI engine operating at 1200 rpm and a constant high load factor.

Other studies (25), utilising two heavy duty DI engines, revealed relatively low correlation coefficients for 13 mode cycle CO emissions and aromatic content compared with other fuel parameters. Ricardo (30) concluded that cetane number differences best explained CO variability in light load tests upon a swirl chamber IDI engine, despite aromatic content varying between 19 and 45%.

5. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION UPON THE EMISSIONS OF NITROGEN OXIDES (NOx)

Several researchers have recorded an adverse influence of higher aromatic fuels upon NOx output (25, 36, 42). Data are shown in Figure 39. Muller (42) recorded linear NOx increases within the range 15-30% in three engines with different combustion systems when varying aromatic content from 15-50% whilst holding fuel distillation range and cetane rating constant. Despite the constant cetane number of 45, the higher aromatic fuels increased the delay period and fuelling had to be increased to maintain power. The resultant increase in peak pressure and temperature resulted in the higher NOx output according to the author.

Gross and Murphy (25) and Marshall and Fleming (36) selected higher fuel aromatic content to correlate with their observations of NOx increase when testing heavy duty DI engines. In the latter paper, the correlation was only strong in one out of the three engines.

G.M. (38) also observed higher NOx emissions from a light duty, swirl chamber IDI engine with fuels of increasing aromaticity.

Other studies (37 50) revealed that fuel variables did not influence NOx emissions despite wide ranging aromatic content. Ricardo (30) related higher light load NOx emissions from a light duty, swirl chamber IDI engine to reduced ignition quality although aromatic content of the test fuels was significantly different.

6. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION UPON ENGINE NOISE

Fiat (27) related chemical composition to engine noise. In their tests, a swirl chamber IDI engine was run at two-thirds full load, 2500 rpm, on fuels having radically different chemical characters and constant cetane number. Aromatic and cyclo-paraffinic fuels produced higher noise emissions than both iso- and n-paraffins. Viscosity and volatility differences were rejected as being responsible.

7. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION UPON EXHAUST ODOUR

In a test programme conducted by the U.S. Bureau of Mines (50), fuels of different chemical characteristics were evaluated for odour formation in a two stroke engine. Little influence of fuel specification was observed at several test conditions including steady state and transient operation. Differences that were observed could not be reliably related to any single fuel factor or combination of factors. Additional studies with a four stroke DI engine also failed to establish reliable correlations between odour intensity and fuel composition.

According to Hills and Schleyerbach (37), a slight trend towards increased odour existed with fuels of higher aromatic content. These trends, obtained from three heavy duty DI engines evaluated at idle and part load, are regarded however by the authors as not statistically significant.

8. THE INFLUENCE OF FUEL CHEMICAL COMPOSITION ON ENGINE DEPOSITS AND WEAR

Four production diesel engines were evaluated for deposit and wear levels in 250 hour mixed load tests by Lindeman et al (49), using fuels of 15 and 50% aromatic content. Sulphur level was constant at 0.3% for both fuels. Top ring gap and combustion chamber deposits slightly increased following operation with the highly aromatic fuel.

Union Oil Company (51) report that, despite essentially no difference in the instantaneous smoking tendencies of four different No. 2 diesel fuels, injector fouling levels differed markedly. A fuel which was hydrogen treated by the unifining process increased injector life by two- to threefold in comparison with straight run fuels when evaluated in truck fleet tests.

APPENDIX 4

THE INFLUENCE OF FUEL VISCOSITY

1. THE INFLUENCE OF FUEL VISCOSITY ON COLD STARTABILITY

Derry and Evans (4) comment that in some cases cold starting improves with increased viscosity. This occurs in engines where pump internal leakage increases significantly with reduced viscosity and where there may be a deficiency in fuelling levels delivered during starting.

Burk et al (8) report that a definite trend existed between the ease of cold starting and cetane number and note that no trend was found with other physical fuel parameters, including viscosity.

2. THE INFLUENCE OF FUEL VISCOSITY ON EXHAUST SMOKE AT NORMAL ENGINE OPERATING TEMPERATURES

Duval and Lys (12) recorded increased smoke emissions over the 13 mode cycle for fuels of higher viscosity in three engines covering DI, 'M' system and IDI combustion chambers. The DI engine appeared the most sensitive. These results are, however, closely allied with changes in the fuel distillation characteristics which accompany the viscosity difference, i.e. increased viscosity commensurate with extended distillation range.

Similar results were recorded by Burk et al (8). Their data showed that, for four engines, smoke generally increased with higher viscosity fuels at various loads. Cetane rating of fuels was similar between 44 and 49. Since for fuels of constant cetane number the viscosity closely follows the mid-boiling point, similar smoke results were evident for the latter fuel factor. Fuel 90% point and end point, however, could be varied widely without significant effect upon exhaust smoke provided that cetane and viscosity were held constant.

Hills and Schleyerbach (37) demonstrated little influence of 95% point and viscosity on smoke whilst Gross and Murphy (25) correlated their smoke data with the quantity of fuel distilling above 640^{o}F and did not include viscosity terms.

3. THE INFLUENCE OF FUEL VISCOSITY UPON ENGINE PERFORMANCE

At equal rack setting, performance of diesel engines can be influenced by viscosity effects altering the quantity of fuel delivered, owing to higher viscosity reducing the degree of fuel leakage past the pumping element. Such a phenomenon is supported by data presented in Reference 21. These results were however of secondary importance compared with gravity effects upon mass fuel flow. Lindeman et al (49) and Shell (11) also studied the effect of fuel viscosity on fuel delivery characteristics. Shell also mentioned the influence of viscosity on fuel injection timing that can occur with certain types of injection equipment.

Duval and Lys (12) demonstrated that when extending the distillation range of fuels by the addition of heavier fractions, together with an accompanying increase in viscosity, an adverse influence upon fuel consumption in the 13 mode test procedure was observed. Variations were small but significant and ranged between 1% and 5% for IDI and DI engines respectively. Viscosity varied between 3 and 7 centistokes. Fuel consumption was corrected to a reference gravimetric heating value and tests were performed at constant power settings.

The tests conducted by Mobil (37) revealed no detectable fuel related variations in either power or fuel economy within the limitations of test repeatability, despite significant differences in fuel variables. Gross and Murphy (25) did not include a viscosity term in their correlations of power and fuel economy with fuel variables.

4. THE INFLUENCE OF FUEL VISCOSITY UPON HYDROCARBON (HC) EMISSIONS

According to Gross and Murphy (25), 13 mode cycle HC emissions from two heavy duty DI engines were lowest for fuels with high viscosity and cetane number. At equivalent cetane number, therefore, increasing fuel viscosity reduced HC emissions. The sensitivity to viscosity was more marked in the naturally aspirated two stroke engine compared with the turbocharged four cycle unit. These data are shown in Figure 18. Viscosity was also demonstrated to represent the preferred fuel variable in lieu of mid-boiling point despite high inter-correlation of the two fuel factors.

The correlation coefficients evolved by Marshall and Fleming (36), from a study of three heavy duty DI engines, reveal 13 mode cycle HC emissions to correlate reasonably strongly with viscosity. Correlation coefficients were negative indicating an HC reduction for increasing viscosity.

Duval and Lys (12) also comment upon reduced HC emissions for fuels of higher viscosity in tests on both DI and IDI combustion systems, with the DI engine being the most sensitive. An 'M' system diesel appeared relatively insensitive in this respect.

Despite presenting the data to a viscosity base, the authors relate these trends to the accompanying elevation of distillation characteristics.

G.M. (38) observed lower HC emissions with higher viscosity when analysing 46 fuels in a light duty, swirl chamber IDI engine.

Hills and Schleyerbach (37) observed that fuel mid-point correlated best with their HC data, with increasing 50% point resulting in decreased HC output. No mention is made of any attendant viscosity effect although viscosity appears related to the mid-point from the fuel data presented.

5. THE INFLUENCE OF FUEL VISCOSITY UPON CARBON MONOXIDE (CO) EMISSIONS

Using developed fuels, Duval and Lys (12) recorded a slight increase in 13 mode cycle CO emissions with fuels of heavier distillation characteristics and high viscosities. Equal power settings were utilised for the three engines evaluated. The DI engine proved the most sensitive whilst a swirl chamber IDI unit appeared to be relatively unaffected. These results are supported by higher smoke output and worse fuel economy.

Extensive studies of the influence of fuel factors on engine performance and emissions (25, 36, 37) make no reference of viscosity correlating with 13 mode cycle CO emissions. Other fuel factors are chosen to explain the small effects recorded.

6. THE INFLUENCE OF FUEL VISCOSITY UPON THE EMISSIONS OF NITROGEN OXIDES (NOx)

Various test programmes (25, 36), using developed fuels to cover a wide range of fuel variables, report correlations of NOx emissions with other fuel factors and make no mention of viscosity, whilst in one instance (37) no fuel variables affected NOx output.

7. THE INFLUENCE OF FUEL VISCOSITY ON ENGINE DEPOSITS

Using a two cycle DI engine, Burk et al (8) correlated an increase in both combustion chamber deposits and piston lacquering to the use of more viscous fuels. Fuels had equivalent cetane ratings and,as would be expected, distillation characteristics were elevated for the higher viscosity fuels. Tests were conducted over a mixed but predominantly full load cycle. Reasons for these trends are not given but it may be significant that the higher viscosity fuels also have lower API gravity. Overall fuel input per cycle would therefore be affected in the absence of any stated fuelling correction for equal power settings.

APPENDIX 5

THE INFLUENCE OF FUEL DENSITY

Regarding exhaust smoke at full load with fixed injector rack position, many workers (e.g. 21, 48) have noted smoke to reduce with lower density (higher API gravity) fuels. Similarly, power output also varies in the same manner (8, 21, 25). The tendency for lower density fuels to reduce full load smoke at fixed rack is purely a function of leaner AFR in direct proportion to the change in density, since the pump meters the fuel volumetrically. The reduction in power observed is directly related to the lower volumetric energy content of the lighter fuels, neglecting changes in pumping efficiency due to viscosity.

In a similar fashion to smoke, CO emissions at full load are also affected by fuel density at fixed rack position (17, 36, 48).

These relationships have no direct bearing upon this study as future engines would have adapted fuelling levels, and/or injection rates, to compensate for density differences in order to attain an acceptable smoke limited performance.

Lower density fuels have higher energy content on a gravimetric basis. Assuming that fuels do not affect combustion efficiency, brake specific fuel consumption on a mass basis will improve when using lighter fuels (3, 8, 25). Such improvements will however be small since the energy content of petroleum derived fuels only varies marginally on a mass basis.

Volumetrically, fuel consumption is worse for fuels of lower density owing to the lower energy content on a volume basis. Therefore, to obtain a fixed load or drive a given cycle, volume fuel flow must be greater to provide the same energy input. Illustrations of this point are given in the literature at steady state conditions (3, 8, 21) and under transient fixed cycle conditions (46, 52). This is probably the most important consequence of changing fuel density since the adoption of lighter fuels has a direct deleterious effect upon miles per gallon, which to the operator may seem unacceptable although engine efficiency may not have changed.

APPENDIX 6

THE INFLUENCE OF FUEL IMPURITIES

1. THE INFLUENCE OF FUEL IMPURITIES UPON THE EMISSIONS OF NITROGEN OXIDES (NOx)

Fuel bound nitrogen can be converted to NOx within the combustion process. Potentially this may pose a problem with future products as shale derived fuels are expected to contain significantly more fuel bound nitrogen than current petroleum derived fuels.

Only one reference was located providing data in this field. In this paper, Tuteja and Clark (53) studied the influence of fuel bound nitrogen upon NOx emissions in a DI two cycle engine. Four fuels were utilised covering No. 1 / No. 2 diesel fuels and JP-4. The latter fuel was blended with cetane to obtain equal ignition quality between fuels. These fuels were chosen to cover a wide range of both aromatics and volatility since turbine burner studies suggest that such variables influence nitrogen conversion. To each of these fuels, pyridine (C_5H_5N) was added to adjust fuel bound nitrogen to approximately 0.25, 0.5 and 1.0% by weight.

The results obtained from the 1% nitrogen doped fuels indicated that no significant fuel bound nitrogen to NOx conversion was apparent. The authors speculate that these results were due to NO attaining near equilibrium concentrations in the major NO producing zones, this being induced to some extent by running the test engine relatively advanced with high NOx output. It is therefore thought that more representative retarded injection timings will result in equilibrium not being attained, with a possible tendency for fuel bound nitrogen to be converted to NOx.

2. THE INFLUENCE OF FUEL IMPURITIES UPON PARTICULATE EMISSIONS

It has been well documented that fuel sulphur is converted within the cylinders predominantly to gaseous sulphur dioxide (SO_2), whilst a smaller fraction is emitted as sulphate (SO_4). This latter quantity is emitted in the form of particulate. Various workers (54, 55) have demonstrated that 1-6% of the fuel sulphur input to the engine is converted to SO_4. Ricardo data are in accord with this. The level of SO_4 emission therefore varies directly with the level of fuel sulphur.

For non-catalyst equipped light duty diesel vehicles, driven over the 1975 FTP cycle, Ricardo measured approximately 2-5% of the total particulate as sulphate for typical diesel fuels containing 0.2-0.3% by weight of sulphur. This result is in close agreement with EPA (54). The impact of fuel sulphur on particulate emissions is not therefore of great concern.

However, if catalysts are fitted to diesel engines for particulate control, then fuel sulphur becomes a problem. Under conditions of high load with exhaust temperatures above approximately 570-670°F, Ricardo (56) have demonstrated a considerable influence upon particulate emissions. At or above these temperatures, the catalyst efficiently converts gaseous SO_2 to sulphate. This results in particulate emissions being significantly elevated above the levels normally observed with the standard exhaust system, despite the fact that the normal major particulate fractions of HC and carbon are being suppressed by the catalyst. At lower exhaust temperatures, the catalyst appears to act less efficiently in this respect and particulate control can be restored. With very low sulphur fuel, particulate control can be retained over the entire load range (56).

In the light of Ricardo experience, catalysts for particulate control do not significantly influence sulphate output in the 1975 FTP test because mean exhaust temperatures are very low. The situation is however reversed in the higher load factor highway cycle, where it has been observed that catalysts which provide 25-40% particulate control in the FTP test provide little or no overall control due to sulphate generation.

3. THE INFLUENCE OF FUEL IMPURITIES UPON EXHAUST ODOUR

Marshall and Fleming (36) correlated exhaust odour with fuel sulphur content in their studies with heavy duty DI engines operated over the 13 mode cycle. The relationship evolved indicated that exhaust odour was reduced when fuels of lower sulphur content were utilised. The correlations obtained are however reported as not highly significant for all three engines. G.M. (57) also report finding that a very high sulphur fuel gave rise to a more odorous and irritant exhaust. Other studies however (50) have not been able to establish a correlation between fuel sulphur and exhaust odour.

The impact of fuel sulphur upon exhaust odour may be aggravated by the sulphate generation of catalysts if such units are adopted for future particulate control.

4. THE INFLUENCE OF FUEL IMPURITIES ON ENGINE DEPOSITS AND WEAR

Lindeman et al (49) conducted wear and deposit tests in four engines to investigate the influence of fuel sulphur level. Fuel sulphur level was varied from 0.3-1.2% whilst volatility and aromatic content were held constant. Tests of 250 hour duration were utilised and covered both part and full load operation. Raising sulphur level from 0.3 to 1.2% increased combustion chamber deposits slightly at part load but had little effect at full load. Top ring wear was generally increased at both test conditions. Tests were not made to investigate fuels of lower sulphur content. Other workers (41, 58, 59) have also observed increased levels of engine deposits and wear with fuels of higher sulphur content. Blanc (58) also reports that fuel bound nitrogen did not influence engine deposits or wear although nitrogen content was only varied in the narrow range of 0.07-0.08%.

5. THE INFLUENCE OF FUEL IMPURITIES UPON FUEL INJECTION EQUIPMENT

Ricardo (43) have observed that the very low sulphur levels encountered with kerosine type fuels are directly related to observed damage of the cam ring of rotary type fuel injection pumps. This can be related to the fact that the sulphur within the fuel contributes directly to its extreme pressure (EP) lubricity.

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APPENDIX 7

FUEL LOW TEMPERATURE PERFORMANCE

Fuels such as diesel precipitate wax crystals as they are cooled. Cloud point represents the temperature at which a haze of these wax crystals appears as the fuel is cooled. The precipitation of these crystals can restrict flow and can therefore adversely affect cold starting (4, 5, 49).

In the United States, low temperature performance of diesel fuels has been judged by the cloud point. Cloud point is stated as a "local requirement" by ASTM and is specified according to geographical areas for the winter period. Satisfactory operation should be achieved in most cases if the cloud point is specified at 10^{o}F above the tenth percentile minimum ambient temperature for the area and season under consideration (60).

For future fuels having a wax content, low temperature flow performance should be comparable to current fuels. The relative importance of such parameters as cloud, pour and cold filter plugging points, with respect to low temperature flow, will therefore need to be determined.

APPENDIX 8

ALTERNATIVE FUELS

1. THE INFLUENCE OF ALTERNATIVE FUELS ON COLD STARTABILITY

Earlier sections of these Appendices have made it quite clear that the startability of diesel engines depends primarily upon the ignition quality of the fuel. In this respect gasoline, with high octane number and hence low ignition quality, poses the ultimate test regarding the startability of compression ignition engines. Data in this area are therefore dealt with initially in this section.

1.1 Gasolines

Firstly, comments relating to the startability of essentially unmodified diesel engines will be dealt with.

Ricardo (61, 62, 63, 64, 65) have gained some past experience on the starting characteristics of both swirl chamber IDI and DI heavy duty engines with gasoline at test shop ambient temperatures. Engines were not specially modified for multifuel capability and additional starting aids such as inlet charge heaters were not employed. Single cylinder IDI engines (61, 62) are reported as starting satisfactorily on 70 octane gasoline with Pintaux nozzles. Difficulty was experienced with a similar multi-cylinder engine however (63). In this engine, starting was also not possible with 86 octane gasoline blended with 10% lubricating oil to improve ignition quality (64). Pintaux nozzles were again utilised. In a DI engine however (65), a slight rise in compression ratio to 19:1 ensured good starting with 86 octane gasoline.

International Harvester (66) have noted starting difficulties with a standard pre-chamber IDI diesel engine with 91 octane gasoline. Starting in this case was improved when adding 10% lubricating oil. International Harvester also recorded that longer ignition delays were prevalent in IDI engines having larger pre-combustion chamber surface to volume ratio. Such engines were also more difficult to start with gasoline as fuel.

Loeffler (67) reports acceptable starting of a standard DI engine when using a 75%/25% blend of 97 octane gasoline and diesel fuel respectively. No data were given for straight gasoline. In multi-fuel studies carried out by Thornycroft Limited (68), a standard heavy duty swirling DI engine has been demonstrated to start acceptably with 80 and 82 octane gasolines at 32°F. At lower temperatures, intake preheaters were required to ensure positive starting. This particular engine had a favourable long stroke and hence low combustion chamber surface to volume ratio.

An insight into the engine developments required to obtain reliable cold starting characteristics of diesel engines on gasoline, especially at low ambient temperatures, is offered by the experiences of the multi-fuel engine developer (e.g. 69, 70, 71, 72). Examination of this literature reveals that, without exception, compression ratios have been raised by comparison with standard diesel practice. In addition, supplementary devices are generally utilised for very low ambient temperatures. A summary of these developments together with any reported starting performance are tabulated at the end of this section.

Other published data (73, 74) specify the need for relatively high compression ratios as one of the requirements to ensure reliable starting with gasoline type fuels.

These data therefore demonstrate that diesel engines can be adapted to start even at low ambient temperatures with fuels of low ignition quality.

1.2 Jet and Other Lower Distillate Fuels

The situation regarding the startability of diesel engines on the less volatile, higher ignition quality products such as Jet Fuels is not so well documented as for gasolines. Arguably, demonstrated starting ability on gasoline implies better starting characteristics on such fuels.

This philosophy is upheld by the data published by Bailey (72). In this instance, the multi-fuel IDI combustion system developed by Caterpillar for operation on high octane gasoline has a three-fold superior starting ability at -25°F when comparing CIE (35 cetane number) and gasoline (18 cetane number) fuels. Such judgements are made upon cranking times. Whether or not the glow plugs required for gasoline were also required for CIE fuel is not stated. The Lycoming multi-fuel concept (70) provides nearly equivalent startability at -25°F on both CIE and gasoline fuels but can tolerate lower glow-plug voltages with CIE fuel.

Other workers (66, 67) observed no starting difficulties with JP-4 and kerosene fuels in standard diesel engines.

1.3 Broadcut Fuels

EPA (52) observed harder starting of a light duty IDI diesel vehicle when using a "100°-600°F" broadcut fuel having a cetane number of 35. Ricardo (35) found that a "100°-700°F" broadcut fuel offered no starting difficulties provided vapour locking was prevented. In this case, the broadcut fuel had an ignition quality comparable with diesel fuel.

MULTI-FUEL DIESEL ENGINES - STARTING CHARACTERISTICS

REFERENCE	ENGINE/ COMBUSTION SYSTEM	COMPRESSION RATIO	STARTING AIDS	DEMONSTRATED ACCEPTABLE STARTING ABILITY
69	Continental Hypercycle - DI 'M' System	20:1	Intake manifold flame heater	91 octane gasoline - to 30°F without heater (0°F for diesel fuel) 91 octane gasoline - to 40°F with flame heater
70	Lycoming S & H - Quiescent DI	18.7:1	Glow plugs installed in combustion chamber	91 octane gasoline - to -25°F with 30 sec glow plug preheat and cold batteries 91 octane gasoline - up to -65°F with longer preheat and warm batteries or motor-generator set.
71	Detroit Diesel 6V-53 - Quiescent DI	23:1	Flame primer in air box	91 octane gasoline - to 40°F without primer 91 octane gasoline - to -25°F with primer
72	Caterpillar LVDS 1100 & LDS 750 - IDI	19.5:1	Glow plugs installed in the pre-chamber	91 octane gasoline - to -25°F with glow plugs

2. THE INFLUENCE OF ALTERNATIVE FUELS ON EXHAUST SMOKE AT NORMAL ENGINE OPERATING TEMPERATURES

At fixed injector rack position, fuels such as gasoline or kerosene give lower levels of exhaust smoke. Many examples of this are available in the literature (10, 11, 17, 48, 67). Lower smoke output with such lighter alternative fuels can be attributed predominantly to reduced fuel delivery owing to a combination of lower density and viscosity, the latter resulting in increased fuel leakage past the pumping element. These trends are therefore not a measure of the smoking tendency of the fuel but simply reflect engine derating. Whilst this could undoubtably concern the true multi-fuel operator, in the context of this report, fuelling levels or injection rates would be raised to give equivalent power on such lighter fuels. Smoke data on various fuels have only therefore been extracted from the literature to enable comparisons at equal power output.

2.1 Gasoline, Jet Fuels, Kerosenes and Diesel Fuels

Data compiled from several Ricardo sources (56, 63, 65, 76, 77) are shown in Figure 40. Additional data are given in Figures 41 and 42.

Regarding swirl chamber IDI engines, Figures 40 and 41, these data generally indicate little influence upon smoke limited bmep when using fuels ranging from gasoline, through diesel fuel to heavier distillates. For swirling direct injection engines, however, Figure 40, with lower rates of mixing, the use of volatile fuels such as gasoline appears to result in a marked improvement in exhaust smoke limited bmep. In Figure 42, it can be noted that smoke was generally improved over the load range with more volatile kerosene as fuel. At equivalent high load factor however, with rich AFR's and high smoke levels, kerosene in fact generally made smoke worse. Such a result is no doubt due to the extended injection periods utilised to obtain the power output with the lighter kerosene.

BP (10) also observed similar results with kerosene as fuel as shown in Figure 43. At full load fixed rack, kerosene as expected returns lower smoke in a DI engine compared with diesel fuel. Reducing the diesel fuel delivery for equal power resulted in nearly equivalent smoke levels between both fuels. Increasing the fuel delivery on kerosene to give the diesel torque curve, however, increased smoke output. This latter trend was also observed in another DI engine by BP and similar results may be traced in the data published by Loeffler (67).

Shell (6, 11) have recorded lower smoke at equivalent power output over the load range when comparing kerosene with diesel fuel. Other workers in the field (16, 68) have also observed lower smoke with kerosene type fuels compared with diesel fuel. Shamah and Wagner (48) noted little influence of fuel type when evaluating JP-4, kerosene and various diesel fuels at equal power to that dictated by the lightest fuel in both IDI and DI engines.

Henein and Bolt (78) examined gasoline, CITE and diesel fuels in a single-cylinder DI engine operating at a fixed AFR (32:1) at 2000 rpm. Smoke on diesel fuel was relatively high at 50 (Hartridge) units. CITE fuel reduced smoke to 40 units whilst gasoline significantly improved smoke to 5 units.

With gasoline as fuel, BP (10) obtained the results also shown in Figure 43. These data show gasoline to attenuate smoke over the load range in a "multi-fuel" DI engine at both high and low speed and at low speed in a standard DI engine. In this engine, gasoline at high speed significantly elevated smoke output. Other workers in the field (39, 72, 73, 79, 80) have also reported a tendency for gasoline fuels to return lower smoke levels over the load range compared with diesel fuel.

2.2 Diesel Fuel Derived from Syncrude

Tuteja and Clark (53) recorded smoke over the load range at three speeds up to rated with diesel fuels derived from tar sands and shale oil. The latter fuel was of a marine grade. All results were obtained from a DI 2-cycle engine at fixed injection timing and were compared with regular No. 2 diesel fuel. These data are shown in Figure 44. These results show that, at all conditions, the lower ignition quality tar sands, derived fuel gave the highest smoke and the relatively high cetane shale fuel generally returned the lowest smoke. The regular No. 2 diesel fuel had intermediate ignition quality and generally produced the intermediate smoke levels. According to the authors, these results do not indicate any relationship with higher distillation characteristics, viscosity or aromatics but do indicate some relationship with H/C ratio.

Bertodo reports on smoke limited power when comparing diesel fuel with blends of diesel fuel and shale/coal derived products. Ricardo also evaluated smoke output with coal derived fuels. The findings of these studies are reported in the alternative fuels section 4.2, dealing with gaseous emissions.

3. THE INFLUENCE OF ALTERNATIVE FUELS UPON ENGINE PERFORMANCE

In a similar manner as reported under the section dealing with alternative fuels and exhaust smoke, engine power output at fixed rack is reduced when using lighter fuels, such as gasoline and kerosenes, due to the derating effect arising predominantly from the combination of lower gravity and viscosity. Again many examples of this are available in the published literature (66, 67, 71, 72, 74, 81, 82, 83, 84) and typical examples to illustrate the point are shown in Figure 45. Adopting the same philosophy as before, comparisons have only been made under conditions of equivalent power output between fuels.

3.1 Gasoline, Jet Fuels, Kerosenes, Diesel Fuels and Heavier Distillates

3.1.1 Fuel Economy

Ricardo have gathered significant experience in this field in the past with both DI and swirl chamber IDI engines. The findings are presented as follows in sub-sections i), ii) and iii).

i) Tests with Heavier Distillates and Fuel Oils

Ricardo have observed that supercharging a single-cylinder swirl chamber IDI engine rendered the engine relatively insensitive to fuel quality, including cetane number. A series of tests (43) were therefore carried out to investigate the ability of the engine to digest a wide range of fuels ranging from light distillates to heavy fuel oils. Fuels had a similar rather low cetane number and gravimetric heat content. Tests were carried out at 500 and 1250 rpm at various inlet air temperatures with 20 ins. Hg. boost. Results were also obtained running naturally aspirated at 500 rpm. Data simulating a turbocharged condition are shown in Figure 41.

From these load range curves it can be noted that at low speed none of the fuels can match the automotive gas oil, with the heaviest fuel giving the worst performance. At the higher speed with boost, the lighter fuels match the gas oil but the heavier fuels are still somewhat worse. The inferior performance of the heavier fuels is a direct result of slower combustion as recorded from the pressure diagrams taken. The better overall performance of the fuels at the higher speed is thought to be due to higher mixing rates and the higher inlet air temperature. Results obtained at the higher speed, boosted condition, but with much reduced inlet air temperature (86°F) showed a marked deterioration in performance with the heaviest fuel.

Quite clearly, when considering these results, due regard should be given to the low speed range evaluated.

ii) Kerosene and Jet Fuels

Comparative performance of a small DI engine running on kerosene and diesel fuel is shown in Figure 42. At 1000 rpm, the economy improvement is quite marked with kerosene and probably indicates improved fuel injection/ air swirl matching with kerosene. At the two higher speeds fuel economy was virtually identical between fuels. These results are directly comparable on a thermal efficiency basis since both fuels had very small (1.5%) differences in gravimetric heat content.

Ricardo (85) also compared Avtag (similar to JP-4) and diesel fuel in a similar DI engine over the full load speed range up to 2700 rpm. Fuelling was increased to give equivalent power with the lighter Avtag and diesel fuel. The Avtag fuel had an appreciably lower ignition quality with respect to the diesel fuel (c.40 compared with c.50-55). No other engine re-optimisation was carried out and standard diesel compression ratio was utilised. At full load, in the lower speed range, Avtag marginally improved brake specific fuel consumption by 2.5% whilst above 1400 rpm consumption was on average 5% worse. Fuel consumption was not significantly affected by Avtag over the load range at 1600 and 2700 rpm.

Ricardo (76) also evaluated Avtag in a small, high speed, swirl chamber IDI engine over the load range at several speeds up to 2500 rpm. The engine had been increased in compression ratio to 21:1 for gasoline operation but injection timing was fixed for both fuels at the diesel setting. At 1200 and 2500 rpm in the upper half of the load range, brake specific fuel consumption was identifical for both fuels. At these load factors at lower speeds Avtag returned a very marginal, 2% worse consumption. At lower loads (20 psi bmep), Avtage worsened fuel consumption by an average 9% at all speeds.

iii) Gasolines

Data obtained by Ricardo (43, 65), comparing 82 and 86 octane gasolines with diesel fuel in a DI engine are shown in Figure 46. For these tests, compression ratio was increased from 17 to 19:1 to assist with gasoline operation. Injection timing was also advanced for gasoline. At the lower load factors, brake specific fuel consumption tends to worsen with gasoline this being particularly pronounced with the higher octane fuel at high speed. This trend is due to the onset of misfiring due to the longer ignition delays encountered. At high load factors, with the exception of the high octane fuel at rated speed, gasoline generally imparts somewhat improved fuel consumption compared with diesel fuel.

Additional Ricardo studies (63) utilising a boosted, single cylinder, swirl chamber IDI engine of heavy duty size demonstrated 5-10% worse fuel consumption over the load range with 86 octane gasoline compared with diesel fuel. These trends were most pronounced at high load rated speed. Injection parameters were optimised for each fuel but compression ratio was not raised.

In a light duty, naturally aspirated multi-cylinder engine incorporating a swirl chamber IDI combustion system (76), running with gasoline was not a practical proposition, despite attempts to optimise injection timing. Compression ratio was however not raised for gasoline operation.

Ricardo (77) also evaluated gasoline/diesel blends (66/33%) in a single cylinder, naturally aspirated swirl chamber IDI engine of heavy duty size. Without engine modifications, brake specific fuel consumption was comparable over the load and speed range (2000 rpm maximum) for the gasoline blend and straight diesel fuel.

Other workers in the field have evaluated kerosene and gasoline fuels in diesel engines.

Pre-chamber IDI engine data according to International Harvester (66) are shown in Figure 47. From this Figure, it can be observed that with lighter jet fuel of comparable ignition quality to diesel fuel, full load brake specific fuel consumption, with rack adjusted to give the same power as diesel fuel, was marginally worse, reaching 2.5% maximum deviation. Figure 47 also shows that for a similar engine, operation with 91 octane (84 MON) gasoline at advanced injection timing to combat long ignition delays returned appreciably worse fuel consumption within the range 4-15% compared with diesel fuel. The economy differential was most pronounced at light load and full load (15% and 11% respectively).

In another engine with higher compression ratio at fixed injection timing, a lower octane gasoline (70 MON) only increased fuel consumption compared with diesel by a maximum of 3-4% over the upper half of the full load speed range. Fuelling was adjusted for equal power as shown in Figure 47. The raised compression ratio reduced ignition delay on this fuel by 6 degrees crank.

The MWM, balanced pressure pre-chamber IDI design has been run on a wide variety of fuels covering gasolines, JP-4, diesel fuels and heavier distillates including lubricating oils (86). At full load 2000 rpm, where fuelling was adjusted for equal power on all fuels, brake specific fuel consumption varied by up to +6.5% for this range of fuels with respect to diesel. Fuel consumption penalties were most evident when using SAE 10 lubricating oil and 88 octane gasoline.

Murayama and Tsukahara (39) observed slightly higher specific consumption at low load with gasoline (35 cetane) compared with diesel fuel (55 cetane). The converse was recorded at high load. When using ignition improved gasoline (c.55 cetane), specific fuel consumption was improved over the majority of the load range. These results were obtained from a single cylinder, 46 CID pre-chamber engine running at 2000 rpm with fixed injection timing.

Thornycroft Limited have published data (68) providing fuel economy comparisons at equal full load conditions with three swirling DI engines operated on a range of fuels.

A 691 CID engine was fuelled to give equal power at 1900 rpm rated speed (standard diesel datum) on diesel fuel (53 cetane), Avtag (43 cetane) and 80-82 octane gasoline (cetane number less than 20). Brake specific fuel consumption and thermal efficiency were comparable for each fuel. These tests were conducted with the engine in standard diesel form. Similar results were obtained with a 600 CID engine having identical design philosophy. In this instance, the comparison was made at the slightly higher rated speed of 2000 rpm. Thermal efficiency and brake specific fuel consumption remained essentially constant for the various fuels at full load rated speed when utilising a turbo-charged version of the 691 CID engine.

Using gasoline as fuel in a small 252 CID engine with a higher rated speed of 2600 rpm however caused difficulties. The diesel torque curve could not be matched despite advanced injection timings. Where equal power did occur between diesel and gasoline, thermal efficiency and fuel consumption were worse with gasoline.

Other workers (67) have demonstrated no adverse influence upon brake specific fuel consumption when comparing kerosene and gasoline/lubricating oil blends at equivalent full load power settings, rated speed, in a DI engine.

Various other published literature (72, 73, 87, 88) demonstrates little change in thermal efficiency and/or brake specific fuel consumption when comparing gasoline/kerosene fuels with diesel fuel in multi-fuel engines designed for operation on such fuels. Typical data are shown in Figure 48. In developing these engines for successful multi-fuel operation with low cetane fuels, compression ratios have been raised to relatively high levels and to maintain combustion efficiency at light load various methods have been adopted: e.g. water heated intake manifold (Continental engine), intake throttling (Lycoming engine), water inter-cooler with turbocharger (Caterpillar engine).

When considering multi-fuel engines, mention must be made of the opposed piston 2-stroke developments. This type of engine is regarded by many as eminently suitable for operation on various fuels because the opposed piston design forms a hot combustion chamber, which aids the combustion of low ignition quality fuels. This is demonstrated by the published fuel consumption maps for the Rootes diesel engine (89). This engine represents a heavy duty automotive unit of 199 CID with a compression ratio of 16:1. These data, shown in Figure 49, demonstrate a favourable comparison of fuel consumption between diesel fuel and 80 octane gasoline, especially at high speed, light load where the low ignition quality of gasoline can be particularly problematical. In achieving this, the low compression ratio relative to other multi-fuel diesel engines should be noted.

This engine has also been demonstrated at higher ratings on both fuels, with comparable brake specific fuel consumption over the full load speed range and over the load range at rated speed.

In achieving these standards, this engine relies upon the merits of the opposed piston design, regarding the formation of the hot combustion chamber, has partially insulated piston crowns and benefits from a relatively hot inlet charge developed by the mechanical scavenge blower.

In cyclic tests with light duty IDI diesel cars, various workers (46, 52, 90, 91) have observed little difference in fuel economy when comparing the lighter No. 1 (Jet A) type fuels with No. 2 fuels, including high boiling No. 2 fuels. EPA (54) have however observed an instance for the lighter Jet A/No. 1 fuels to return improved fuel economy by up to 10% compared with the heavier No. 2 fuels. Fuels had equivalent ignition quality.

3.1.2 Ignition Delay, Rate of Pressure Rise and Peak Pressure

As might be anticipated, the major problem of running compression ignition engines on alternative fuels to diesel is one of ignition quality. In order to obtain successful operation with low ignition quality fuels, ignition delay must be reduced to avoid late combustion and the attendant problems of low load misfire, roughness and loss of thermal efficiency, and hence economy as already presented. The published literature provides many examples of combustion performance illustrating the problems of low ignition quality fuels such as gasoline (39, 43, 66, 73, 74, 80, 81, 82, 86, 92).

Typical data are shown in Figure 50. From Figure 50, the adverse influence of low ignition quality fuels on delay period and rate of pressure rise is clearly shown. Conversely, fuels like JP-4 with higher ignition quality than gasoline influence these parameters marginally and hence diesel engines generally have no problem in utilising such fuels.

Figure 50 also shows that increased compression ratio can offset the deleterious influence of low ignition quality and help restore more favourable ignition delay. Indeed, this is one of the features of the multi-fuel engine. Combustion chamber design can also help to maintain combustion performance with a range of fuels. Notable in this respect is the MWM pre-chamber engine (86) which displays remarkably constant rates of pressure rise for fuels ranging from gasoline to diesel fuel, despite increased delay periods. The M.A.N. 'M' system combustion chamber (80) also displays controlled rates of pressure rise with fuels of wide ranging properties.

Furthermore, it will be noted from Figure 50 that long ignition delay due to low ignition quality does not automatically give rise to higher peak pressures owing to retarded combustion.

3.2 Diesel Fuel Derived from Syncrude

3.2.1 Fuel Economy

Tuteja and Clark (53) evaluated diesel fuels derived from tar sands and oil shale and compared the results with regular No. 2 fuel. Fuel consumption was measured over the load range at three speeds up to rated in a 213 CID, 2 cycle DI engine with fixed injection timing. Results obtained are shown in Figure 51, together with basic fuel properties. These data generally indicate that the heavier marine grade shale derived fuel returned the highest fuel consumption, No. 2 the lowest and the tar sands fuel intermediate. Considering the stated reproducibility of results, the trends are only significant at higher load factors. The heating values of the fuels were essentially the same so the slight general increase in fuel consumption observed with the synfuels represents a degradation in thermal efficiency.

Bertodo reports on fuel economy when comparing diesel fuel with blends of diesel fuel and shale/ coal derived products. Ricardo also examined economy with coal derived fuels. The findings of these studies are reported in the alternative fuels section dealing with gaseous emissions.

3.2.2 Ignition Delay, Rate of Pressure Rise and Peak Pressure

In the aforementioned studies with synfuels carried out by Tuteja and Clark, ignition delay was at a maximum with the tar sands derived fuel and lowest with the shale derived fuel, with No. 2 fuel in between. Rate of pressure rise also generally followed these trends. The pattern of results is entirely predictable from the ignition qualities of the three fuels. No peak pressure information was supplied.

Bertodo examined diesel fuels diluted with shale and coal derived products. The shale fuel imparted a reduction in ignition quality and the coal products induced long ignition delays and adverse burning characteristics. These findings are fully reported in the alternative fuels section dealing with gaseous emissions.

3.3 Broadcut Fuels

3.3.1 Fuel Economy

Transient cycle fuel economy comparing broadcut and diesel fuels in light duty IDI vehicles has been obtained (46, 52, 75). With 100 - 600°F broadcut fuels, FTP fuel economy has been noted to decrease by 8-24% (46) and 3-6% (52) with respect to diesel fuel operation. Such trends are largely attributable to lower density and hence lower volumetric heats of combustion. Ricardo (75) however compared road consumption in two vehicles equipped with swirl chamber IDI engines when operating on a 100 - 700^{o}F broadcut fuel and diesel fuel. Both fuels had similar cetane numbers. In this instance, fuel consumption varied little between fuels. This was attributed to reduced performance of the vehicle when operating on the broadcut fuel, with it being speculated that rack adjustment to give comparable performance (and hence road driving) would worsen fuel consumption due to gravity differences.

3.4 Other Fuels

3.4.1 Fuel Economy

Bechtold and Lestz (93) utilising a single cylinder DI engine, recorded progressively decreasing thermal efficiency when raising the quantity of used lubricating oil blended with diesel fuel up to 15% by volume. Brake specific fuel consumption was accordingly penalised.

4. THE INFLUENCE OF ALTERNATIVE FUELS UPON HYDROCARBON (HC), CARBON MONOXIDE (CO) AND NITROGEN OXIDE (NOx) EMISSIONS

4.1 Kerosene, Jet Fuels and Diesel Fuels

Ricardo (85) compared 13 mode cycle emissions from a heavy duty DI engine when operated on Avtag (similar to JP-4) and DERV diesel fuels. Engine settings were identical for each fuel except for rack adjustments to give equal power. Compression ratio was left at the standard diesel value. With Avtag as fuel, HC and CO emissions increased by 76 and 39% respectively. NOx was however reduced by 21%. Based upon these results, it was speculated that retarded combustion owing to the lower ignition quality of Avtag (c. 40 compared with c.50-55) was responsible. The large increase in HC emissions may also have been aggravated by the more volatile characteristics of Avtag due to fuel evacuation from the nozzle sac volume. Individual operating modes were not responsible for these results.

BP (17) have demonstrated both higher aldehyde and total HC emissions from a range of 4 cycle DI engines when utilising more volatile kerosene in place of diesel fuel. Ignition quality was similar for both fuels. These results are shown in Figure 52. It will be noted that total HC emissions are increased over the load range while aldehydes are predominantly higher at lower load factors. IDI engines are reported as less sensitive in this respect. With regard to CO emissions, BP have also demonstrated the expected reduction of CO emissions with kerosene fuels compared with diesel fuels owing to the derating effect of the lighter kerosene at fixed rack position.

Shamah and Wagner (48) evaluated JP-4, kerosene No. 1 and a variety of No. 2 diesel fuels covering both straight run and catalytically cracked distillates, in three heavy duty engines incorporating IDI and DI combustion systems. Tests were carried out at full load, part load and idle. HC emissions levels between engines varied markedly and in two engines, levels were so low that fuel effects could not be resolved. In the 2-stroke DI engine studied HC output was significantly higher and a fuel volatility effect was recorded. In this instance, the most volatile JP-4 fuel returned two-fold higher HC levels. CO emissions were little affected by fuel type at equal power settings and only varied amongst fuels appreciably at equal full rack, when the lighter fuels produced lower levels due to derating. Fuel type had no significant influence upon NOx emissions.

EPA (91) utilised a light duty vehicle equipped with a Nissan IDI diesel engine to evaluate a No. 1 and No. 2 diesel fuel and a No. 2 smoke test fuel. In both cold and hot start 1975 FTP cycles and the Highway procedure, increasing HC emissions appeared to correlate best with decreasing fuel density and/or viscosity. HC increases of 70, 82 and 93% for the respective test cycles were noted with the lightest No. 1 fuel. CO emissions followed similar patterns as for HC whilst NOx was singularly unaffected.

Other studies by EPA (54) examined Jet A, No. 1 and various No. 2 diesel fuels, all fuels having similar ignition qualities, in a light duty vehicle fitted with a swirl chamber IDI engine. Regarding HC emissions, the two more volatile fuels (Jet A, No. 1) returned 34% lower HC emissions in the 1975 FTP test whilst levels were similar for all fuels in both the Highway and Sulphate test procedures. These tests are both higher load factor and do not employ a cold start. CO emissions were relatively unaffected by fuel type. NOx emissions were 27%

lower with the more volatile fuels in the FTP test and were little influenced in the Highway and Sulphate tests.

Mobil research (46) also compared more volatile No. 1 with the No. 2 diesel fuels in four light duty IDI diesel cars driven over the 1975 FTP cycle. Expressed as a mean for the four vehicles, HC emissions were 31% lower with the more volatile No. 1 fuel compared with No. 2 fuel. Both fuels had similar ignition qualities. CO and NOx emissions were generally little affected.

Additional programmes carried out by Southwest and/or EPA on heavy duty DI (94) and other light duty IDI engines (52, 90) concluded little significant difference of gaseous emissions between the more volatile No. 1 (Jet A, kerosene) type fuels and the heavier No. 2 fuels. In the latter report, however, (90) one of the vehicles returned higher HC emissions when utilising a "minimum" quality No. 2 fuel during operating schedules containing substantial idle time. This fuel had both low ignition quality and higher aromatic content.

Murayama and Tsukahara (39) observed slightly lower NO emissions at higher load factors when comparing kerosene with diesel fuel. These results are more fully presented in the section dealing with gasoline fuels.

In tests carried out by Shell (11) kerosene (43 cetane, 374^{o}F mid-point) and diesel fuel (54 cetane, 514^{o}F mid-point) were compared for CO emissions at two speeds over the load range in a DI engine. At the higher test speed of 1400 rpm, there was no significant difference between fuels but at the lower speed of 700 rpm, the kerosene fuel gave a mean reduction in CO emissions of about 25%. Limited tests conducted by McConnell and Howells (10), however, have demonstrated that a kerosene fuel increased CO emissions by approximately 30-60% in a small high speed, swirl chamber IDI engine tested at 3000 rpm up to 75 psi bmep.

4.2 Diesel Fuel Derived From Syncrude

Detroit Diesel Allison (53) evaluated the emissions characteristics of diesel fuels derived from oil shale and tar sands and compared the results with those obtained from regular No. 2 diesel fuel. The shale derived fuel was of a marine grade. A 2-cycle DI engine was employed and tests were conducted over the load range at three speeds up to rated speed.

Gaseous emissions results together with main fuel specifications are presented in Figure 53.

In general, NOx emissions were highest over the load range with tar sands fuel and lowest with the shale fuel. Taking into account reproducibility of results, there is a possible significance of the NOx differences between shale fuel and regular No. 2 whilst the comparisons between tar sands fuel and No. 2 are definitely significant. According to the authors, the trends are somewhat consistent with other studies in so much that the more aromatic, lower cetane tar sands fuel produces higher NOx whilst the lower aromatic, higher cetane shale fuel returns the lowest NOx output.

From Figure 53 it can be observed that HC emissions with tar sands fuel are about 30-40% lower than No. 2 fuel, whilst with shale fuel they are reduced by approximately 60-70%. The authors relate these trends to higher viscosity and mid-points with the synfuels and therefore favour the explanation of reduced fuel mass evacuated from the nozzle sac volume during the expansion stroke.

With respect to CO emissions, tar sands fuel generally showed the highest emissions followed by regular No. 2 and shale fuel. These patterns are related to cetane number according to the authors.

In conclusion, the authors state that the emission trends observed with these syncrude products can be explained by physiochemical fuel properties in a similar fashion to the various philosophies developed for standard petroleum products.

Bertodo (35) examined a blend of No. 2 diesel fuel and 20% shale oil in a 236 CID multi-cylinder swirling DI engine. This fuel had a cetane number of 39 and an aromatic content of 32%. Previous experience had suggested that these levels of aromatics and ignition quality would adversely affect ignition delay, emissions and light load performance. To compensate for this, the compression ratio was raised from 16 to 18:1 and the dynamic injection timing retarded from 22° BTDC to 14° BTDC. Comparisons of the smoke limited performance, 13 mode cycle emissions, particulates and noise were made between the modified engine with the shale oil blend and the standard engine running with UK specification diesel fuel (cetane number 50, aromatic content 20%).

Smoke limited rated power output was reduced by 2% with the shale oil blend whilst volumetric specific fuel consumption was not affected. NOx and CO emissions were increased by 9 and 15% respectively but HC emissions were reduced by 8%. Particulates increased by 21% with the shale oil blend. Noise was also raised by 2 dB. The data contained within the paper imply that the increased compression ratio and retarded injection timing would explain most of these trends.

Bertodo also examined the suitability of straight coal derived products and diesel fuel diluted with coal derived products.

Regarding straight coal derived products, success was largely dictated by C/H ratio with high C/H ratio degrading suitability. Coal tar fuel oil, with C/H ratios above 16, could not be employed directly in a diesel engine but could be rendered suitable by hydrogenation to provide satisfactory operation. The best results were obtained with paraffinic products obtained from the Fischer-Tropsch process although fuel consumption was 5% worse.

Blended fuels comprised of UK specification diesel fuel with 25 and 50% of tar oil and hydrogenated creosote oil were also evaluated. These dilution fuels were based upon the selection of high conversion efficiency products in the interest of maximised production efficiency and therefore do not necessarily reflect the requirements of the engine.

The engine results were influenced by the slow burning rates of the dilutents having low H/C ratios of 0.1 and 0.125 for tar oil and hydrogenated creosote oil respectively, compared with typically 1.9 for diesel fuel. Ignition delay was also increased due to the depressed ignition quality of the blends ranging between 36-41 (25% blends) and 24-33 (50% blends) compared with 47 for the diesel fuel. These factors resulted in operation above 25% dilution with tar oil and 50% with hydrogenated creosote oil being impossible.

With the same engine builds utilised for the shale oil study, smoke limited rated power was reduced by 11% with a 25% blend of creosote oil and volumetric specific fuel consumption increased by 15%. 13 mode cycle HC, CO and NOx emissions were all increased by 12, 15 and 24% respectively. These trends are thought to represent actual fuel differences based upon the reported effects of the different injection timing and compression ratio employed when comparing diesel fuel with the blends.

The use of the various diesel/coal product blends also resulted in various adverse degrees of nozzle fouling, lubricating oil sludging and consequent wear.

Ricardo (95, 96, 97, 98, 99) have also evaluated fuels derived from coal by the liquid solvent extraction process in a heavy duty swirling DI engine. Initially, straight run and distillates without hydrogenation were examined, but operation was not satisfactory.

A hydrogenated sample was examined having a cetane number comparable with "minimum quality" No. 2. Performance levels were comparable with diesel fuel over the entire load and speed range. Smoke was marginally reduced at full load and was attributed to the lower end point of the coal fuel.

In addition, a more commercially viable, hydrogenated coal derived fuel having higher distillation characteristics was evaluated. This fuel had a higher cetane number comparable with typical No. 2 fuel. Performance and gaseous emissions, except for an increase in HC levels, was competitive with diesel fuel. Exhaust smoke and particulate emissions were both lower with the coal derived fuel.

4.3 Gasolines

Murayama and Tsukahara (39) studied extensively the influence upon NOx emissions when using gasoline as fuel in a single-cylinder, pre-chamber engine. Tests were conducted over the load range at 2000 rpm with fixed injection timing. A diesel fuel (55 cetane) was compared with kerosene (43 cetane), undoped gasoline (35 cetane) and leaded gasoline with 5% heavy gas oil as a cetane improver (16 cetane). Results obtained are shown in Figure 54. From these results, it can be seen that there is a considerable difference of NOx emissions between fuels. Using diesel fuel as baseline, the lighter lower ignition quality kerosene had little influence upon NOx output at light load but returned lower levels in the upper two thirds of the load range. NOx output was approximately 10% lower with kerosene at peak NOx production occurring at three quarters load. The undoped 35 cetane gasoline, increased NOx emissions by approximately 5-13% up to about 60% load and at higher load factors returned a mean 15% lower NOx emission rate. The very low ignition quality leaded gasoline markedly attenuated NOx emissions by 24-42% at low load factor but had little influence at higher load factors.

From these results, it would appear that a definite trend of NOx emissions with cetane number or volatility exists. NO_2 emissions were little affected by fuel type.

Murayama and Tsukahara also observed that if gasoline is ignition improved to give the same cetane rating as kerosene, or kerosene is improved to equivalent ignition quality of diesel fuel, the more volatile fuels returned lower NOx emissions than the heavier fuels. These results are shown in the inset of Figure 54. According to the authors, this phenomenon is thought to be due to the rapid vaporisation of the more volatile fuels acting advantageously on the control of oxygen in the pre-chamber, presumably by increasing combustion rate and reducing residence time with oxygen supply for NOx formation.

BP (40) observed appreciably lower NOx emissions over the load range when running a 2-stroke DI engine on 80 octane gasoline in lieu of diesel fuel. Injection timing was fixed and results were attributed to late combustion owing to the long ignition delay incurred with the gasoline fuel.

4.4 Broadcut Fuels

EPA(52) evaluated a broadcut fuel in comparison with No. 1 and No. 2 diesel fuels over both 1975 FTP and Highway cycle test procedures utilising a light-duty vehicle equipped with an IDI diesel engine. The broadcut fuel had a distillation range of approximately 100° - 600°F compared with 325° - 538°F and 352° - 580°F for the No. 1 and No. 2 fuels respectively. Both diesel fuels had cetane numbers of 46 whilst the broadcut fuel had lower ignition quality at 35.

In both test cycles, gaseous emissions were almost identical for both diesel fuels. By comparison over the 1975 FTP test, the broadcut fuel increased HC emissions by 115% and CO emissions by 32%. NOx was unaffected. NOx was also not influenced during the Highway cycle but HC and CO emissions were again increased by 159% and 69% respectively. These results are attributed to the low ignition quality of the broadcut fuel. However, the exceptionally large HC increase may in part have been aggravated by the vehicle running somewhat retarded as judged by the relatively low NOx emissions with respect to vehicle inertia weight (1.5 g/mile 75 FTP NOx emissions - 3500 lbs. test inertia).

Mobil Research (46) also examined a broadcut fuel of similar distillation range to that evaluated by EPA but of higher ignition quality (44 cetane index). In testing three light-duty IDI diesel cars, 1975 FTP HC emissions were increased in all three vehicles with the broadcut compared with No.2 diesel fuel by 27%, 121% and 262%. CO emissions were also increased within the range 13-64% whilst NOx emissions were reduced by a mean 16%.

4.5 Other Fuels

Pennsylvania State University (93) evaluated blends of used lubricating oil and diesel fuel in a single cylinder DI engine. Oil blending was controlled from 2.5 - 15% by volume. These blends were evaluated over the load range at three speeds up to 3000 rpm. HC emissions were recorded at a minimum with 5% oil blends but the reduction compared with diesel was only slight. Generally speaking, oil blends did not appreciably influence HC emissions compared with diesel fuel with the exception of full load, 3000 rpm with 15% blends, when HC emissions were increased two-fold. CO emissions were unaffected at light loads but at full rack, CO increased for the higher blends. NOx emissions were marginally reduced with increasing quantity of blended oil.

5. THE INFLUENCE OF ALTERNATIVE FUELS UPON PARTICULATE EMISSIONS

5.1 Jet, No. 1 and No. 2 Diesel Fuels

Several workers in the field have recorded particulate emissions when running diesel engines on Jet, kerosene and various diesel fuels. These data have been obtained using dilution tunnel particulate collection techniques in accordance with legislative procedures.

EPA (54) evaluated Jet A, No. 1 and two No. 2 fuels in a light duty car incorporating a swirl chamber IDI diesel engine driven over the 1975 FTP, Highway and Sulphate test cycles. Particulate emissions varied similarly amongst fuels for each test cycle. Higher particulate emissions were correlated with increased fuel aromatic content, the correlation being particularly strong for the results from the 1975 FTP test. In this case, the least aromatic Jet A fuel (13% aromatic) returned 26% lower particulate output than the more aromatic(27%) No. 1/No. 2 fuels. The other No. 2 fuel with intermediate aromaticity also produced intermediate particulate levels.

In a test programme conducted by Southwest and EPA (90) utilising two light duty diesel cars with both swirl and pre-chamber combustion systems, particulate emissions were lowest when using a Jet A fuel and highest when using a "minimum" quality No. 2 fuel. Although the latter fuel was of lower ignition quality, aromatic content was nearly three-fold that of the Jet A fuel (13% compared with 35%).

No significant fuel effect upon cyclic particulate emissions was recorded by EPA (91) in other light duty diesel studies. Fuels ranged from No. 1, a No. 2 and a higher boiling No. 2. In this instance, ignition quality was similar for all fuels and aromatic content varied within the narrower range of 23-35%.

Heavy duty turbocharged DI engine studies conducted by Southwest and EPA (94) revealed fuel effects upon particulate emission rates recorded during the 13 mode cycle. For both engines,particulate emissions were generally lower with a No. 1 fuel and highest with a heavy No. 2 fuel. In this respect, the No. 1 fuel returned mean modal particulate reductions of 19 and 43%. The least sensitive engine was a 2-cycle engine compared with the more sensitive 4-cycle unit. The third fuel generally returned intermediate particulate levels and also had intermediate properties. All fuels had similar ignition qualities but different distillation characteristics and significantly variable aromatic content, which ranged from 9% for the No. 1 fuel, through 23% for the intermediate fuel to 35% for the No. 2 fuel.

Frisch et al (55) also recorded lower part load particulate emissions from a heavy duty DI engine when running with a more volatile, less aromatic No. 1 fuel.

G.M. (38,100) have recorded results which appear to be in general agreement with the aforementioned data. In the G.M. light duty diesel programme No. 1 type fuels having both lower upper distillation characteristics and lower aromaticity compared with No. 2 fuels, can reduce particulate emissions by approximately 25%. G.M. have also concluded that, regarding fuel chemical composition, aromatics tend to have the most dominant influence upon particulate output.

In studies by Ricardo (56), particulate emissions were reduced by some 50% over the load range at both intermediate and rated speeds when utilising a lighter fuel, similar to No. 1 specification compared with a heavier European specification diesel fuel. Major fuel differences included lower higher distillation characteristics and much lower aromatic content with the No. 1 type fuel. These results were obtained in a heavy duty DI engine with fixed injection timing. Analysis of collected particulate revealed that both organic and carbon fractions were suppressed with the lighter fuel.

Bertodo (35) examined fuels of 15-30% aromatic content in a DI engine and noted that as the aromatic content was raised progressively to the upper limit particulate emissions doubled. This was attributed to the longer ignition delays observed. No ignition quality data were presented however.

Kittelson et al (101) examined No. 1 and No. 2 diesel fuels, 40 and 50 cetane number secondary reference fuels in a pre-chamber IDI engine operated over the load range at 1800 rpm. The lower 40 cetane reference fuel returned decreased particulate emissions. According to the authors, this was due to cleaner pre-mixed burning induced by the longer ignition delay. G.M. (38) also observed a similar result. When a light fuel was ignition improved to 46 cetane number, a significant increase in particulates was observed, independent of injection timing. These results were obtained from a light duty swirl chamber IDI engine.

Other Ricardo programmes (30) however concluded insignificant fuel effects on particulate emissions when evaluating several light load conditions, at two injection timings with a light duty swirl chamber IDI engine. In this programme, four fuels were evaluated covering European and US specification diesel fuels. Cetane number, higher distillation characteristics and aromatic content varied markedly.

5.2 Diesel Fuel Derived From Syncrude

Ricardo (95, 96, 97, 98, 99) tested coal derived fuel in a heavy duty DI engine. A hydrogenated fuel with a cetane number comparable with typical No. 2 diesel fuel produced lower particulate emissions when compared with diesel fuel operation.

6. THE INFLUENCE OF ALTERNATIVE FUELS UPON ENGINE NOISE

6.1 Gasoline, Jet, Kerosene and Diesel Fuels

As would be expected, low ignition quality gasoline, by inducing long ignition delays and higher rates of pressure rise results in increased noise output compared with diesel fuel. This fact has been well documented (eg 32, 43, 61, 62, 63, 64, 65, 66 67, 73, 76).

By suitable development, such as higher compression ratio, (65) noise can be reduced and in the case of a fully developed multi-fuel engine (79, 80, 86) noise levels with gasoline as fuel can be made indistinguishable from diesel operation.

The less volatile, higher ignition quality Jet and kerosene fuels do not impose such severe noise penalties as gasoline and even in standard diesel engines such fuels can return acceptable noise levels (39, 66, 67, 68, 85).

6.2 Broadcut Fuels

EPA (52) report high engine noise levels in comparison with diesel fuel when operating a light duty IDI diesel engine on a "100^{o}-600^{o}F" broadcut fuel having a cetane number of 35. Ricardo (75) observed little difference in engine noise when comparing a "100^{o} - 700^{o}F" broadcut fuel with diesel fuel in two light duty swirl chamber IDI engines. In this instance, however, the broadcut fuel had similar ignition quality to the diesel fuel.

7. THE INFLUENCE OF ALTERNATIVE FUELS UPON EXHAUST ODOUR

EPA (54) measured 35% lower 1975 FTP aldehyde emissions from a light duty car equipped with a swirl chamber IDI diesel engine when using lighter Jet A/No. 1 fuels compared with No. 2 fuels. No correlation with exhaust odour was made. Other light duty IDI studies conducted by Southwest and EPA (90) utilising Jet A and various No. 2 fuels concluded little clear trends between aldehyde emissions and fuels.

G.M. (57) recorded odour intensity with two heavy duty DI engines using three fuels. Two fuels were similar to No. 1 and No. 2 diesel fuels, whilst the third had distillation characteristics closely resembling No. 2 fuel, but a substantially lower ignition quality. In one engine, fuels did not influence odour measured at idle and full load, high speed. In the other engine, tested over the load range at intermediate speed, the exhaust odour was slightly less when comparing the lighter No. 1 fuel with the No. 2 fuel. Both fuels had equivalent, relatively high ignition qualities.

BP (10) noted that when comparing kerosene with diesel fuel in a DI engine, there was a tendency for kerosene to impart more noticeable odour and irritation characteristics to the exhaust in the upper part load portion of the load range. At both lighter load factors and at full load, the converse was true. BP also compared exhaust odour with gasoline and diesel fuels in a turbocharged DI engine. In this case the exhaust was rated as more odorous when running on gasoline.

8. THE INFLUENCE OF ALTERNATIVE FUELS UPON THE POTENTIAL HEALTH HAZARDS OF DIESEL PARTICULATES

It is known that carcinogenic polynuclear aromatic hydrocarbons (PNA) are emitted from diesel exhaust in the form of particulate matter. Of these PNA emissions, the potentially most potent is thought to be benzo(a)pyrene or BaP for short.

EPA (54) measured BaP emissions from a light duty vehicle equipped with a swirl chamber IDI engine when driven over the 1975 FTP, Highway and Sulphate drive cycles. Jet A and No. 2 fuels were compared, both fuels having equivalent cetane number. Jet A fuel emitted greater or equal BaP levels compared with the No. 2 fuel despite having significantly lower aromatic content.

Southwest and EPA (90) also measured BaP emissions from two light duty IDI diesel cars, operated over various schedules and with Jet A and several No. 2 diesel fuels. Highest average BaP emissions were noted for both vehicles when a "minimum quality" No. 2 fuel was used having a cetane index of 42, whilst the lowest BaP output was recorded when using "premium" quality No. 2 fuel of 53 cetane index. Complementary studies (102) recorded that the mutagenicity of light duty diesel particulate samples is influenced by fuel type. Both the BaP content and the mutagenic activity in the Ames test were highest when the vehicles were run on a "minimum quality" No. 2 fuel. This fuel had both low ignition quality and high aromatic content.

Ricardo (103) also recorded similar results. In these studies, the potential health hazard of the particulate matter emitted during the 1975 FTP test was generally higher when using a typical No. 2 fuel compared with a European diesel fuel having higher ignition quality. The engine was optimised for the No. 2 fuel.

9. THE INFLUENCE OF ALTERNATIVE FUELS UPON FUEL INJECTION EQUIPMENT

Many of the references consulted for this study have highlighted the requirement for modified fuel systems for suppressing vapour formation with volatile fuels such as gasoline or broadcut (43, 62, 67, 71, 72, 73, 82, 104). Vapour suppression is generally achieved by increasing the fuel supply pressure to the injection pump and incorporating a much higher fuel flow rate through the pump than the engine requires.

Furthermore, fuels such as gasoline and kerosene do not have the adequate lubrication properties of diesel fuel. To avoid the seizure of pumping elements and damage to other components therefore, the fuel injection pump frequently requires modification by providing additional lubrication (43, 67, 71, 82, 104). In the case of rotary pumps which cannot readily be supplied with supplementary lubrication and rely on the fuel lubricity alone, material specifications and working clearances may require attention.

APPENDIX 9

THE SPARK IGNITED COMET COMBUSTION SYSTEM

1. HISTORY AND BRIEF DESCRIPTION OF THE SYSTEM DESIGN AND COMBUSTION PROCESS

The Ricardo Comet V combustion system has been used extensively in light duty diesel engines. The intense, organised air swirl induced in the pre-chamber and in the later stages of combustion, the piston cavities of the main chamber, results in fast rates of combustion with high air utilisation. This allows a wide speed range with favourably high smoke limited bmep enabling high specific power output to be achieved.

In the latter half of 1970, Ricardo explored the potential of this combustion system, with spark ignition, as the basis of an unthrottled stratified charge engine with multi-fuel capabiltiy. Utilising a single-cylinder engine of 3.35" bore and stroke, approximately 100 different engine builds were examined. The various parameters studied included injector position, spray type, spark plug position and size, main chamber/pre-chamber volume ratio, compression ratio, throat size, rate of injection, nozzle opening pressure and fuel type. These development programmes have been reported (105, 106, 107, 108) and the final arrangement evolved is shown in Figure 55.

The main parameters of the Ricardo Comet V design are retained for this engine but the compression ratio is reduced to 12:1 from typically 20-23:1 for diesel applications. The pre/main chamber ratio is held as close to 1:1 as possible. The major changes are that the injector is re-located to provide a horizontal spray axis and a sparking plug with extended electrodes has been fitted to the original injector position.

Combustion in the spark ignited Comet consists of three phases:-

a) Fuel/air preparation.
b) Combustion within the pre-chamber.
c) Completion of combustion within the main chamber.

The first and last of these phases are similar and

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not overly influenced by fuel specification. Fuel specification does however influence combustion within the pre-chamber.

Fuel is injected near to top dead centre in the rapidly swirling air generated by the throat leading to the pre-chamber. Fuel is evaporated and mixed with the air, creating an ignitable mixture at the sparking plug which is timed to spark at a suitable interval after the start of injection. The kernel of flame caused by the spark develops until there is an appreciable change in the cylinder pressure. With gasoline as fuel, this flame front expands throughout the pre-chamber until it is quenched by high local AFR or contact with the chamber walls. With diesel fuel, as the flame develops, the self ignition temperature of the fuel is reached and at a number of sites auto-ignition occurs. The combustion gases then leave the pre-chamber through the throat at high velocity and combustion is completed within the main chamber, mixing with air in the process due to the design of the cavities within the piston crown.

As with the IDI diesel engine, load factor is governed by the quantity of injected fuel and the torque curve is smoke limited.

Typical indicated performance and emissions data for the system operating on 91 octane gasoline and diesel fuel are shown in Figures 56 through 59 (108). These data were obtained with injection and ignition timings set for maximum economy. Figures 56 through 59 also show performance and emissions of the Comet diesel version of the same engine and a modern single cylinder gasoline engine. The diesel engine data were obtained with the injection timing optimised for maximum full load torque whilst the gasoline engine data were derived with the mixture strength set for best part load economy and weakest mixture for maximum power at full load. In addition, the ignition timing of the gasoline engine was adjusted in each case for maximum torque.

Since the spark ignited Comet was developed using a single cylinder research engine, designed for DI diesel loadings, the friction levels are entirely unrepresentative for the combustion system, as they should be between conventional IDI diesel and gasoline levels with a bias towards the latter. The choice of making indicated comparisons precludes this friction factor and places the engine comparisons in better perspective.

2. COMBUSTION CHARACTERISTICS

Under full load conditions, the shape of the pressure diagrams between gasoline and diesel fuels is very similar, as can be judged by the indistinguishable combustion noise between fuels reported later in this section.

Peak cylinder pressures at full load are of the order of 1000 psi compared with pressures up to 900 psi for gasoline engines and 1100 psi for Comet diesel engines.

Rates of pressure rise are typically higher than the gasoline engine (30 psi/$^\circ$crank) at 80-100 psi/$^\circ$crank. This value is in accord with typical Comet diesel engines.

3. PERFORMANCE AND FUEL ECONOMY

Comparative fuel consumption data are shown in Figure 56. Over the speed range, it can be observed that the indicated fuel consumption of the spark ignited Comet engine is very similar for either gasoline or diesel fuel. The part load indicated efficiency lies approximately midway between the gasoline and diesel engines. The tendency for the fuel consumption to turn up sharply at low load factor as displayed by the gasoline engine is not evident and the response with load is similar to the diesel characteristic.

From Figure 56 it can be noted that the smoke limited imep is higher with gasoline than diesel fuel, particularly at the lower speeds. The smoke limited imep is however lower than the diesel engine with both fuels.

The motoring loss of a multi-cylinder spark ignited Comet engine has been estimated using data available from conventional engines and a knowledge of the influence of the various parameters involved. The estimated motoring loss falls midway between conventional multi-cylinder gasoline and Comet diesel engines. In this way, the somewhat inferior indicated fuel economy and smoke limited imep with respect to the diesel engine is offset and the predicted brake smoke limited performance and light load fuel economy for a multi-cylinder spark ignited Comet engine more closely matches the diesel engine.

Ricardo have gained some experience with a multi-cylinder spark ignited Comet engine although development was very restricted. On a range of fuels covering gasoline to diesel fuel, the typical smoke limited torque curve associated with the diesel engine could not be matched. Fuel consumption was also generally worse, particularly at higher load factors, although at light load the upper limit of typical diesel fuel consumption was approached at some speeds with some fuels. During this work difficulty was experienced with reproducing the required injection characteristics defined from the single cylinder programme. The results obtained therefore should be regarded with caution and simply reflect the limited development effort expended.

4. GASEOUS EXHAUST EMISSIONS

HC emissions measured from the single cylinder programme are shown in Figure 57. Gasoline and diesel engine emissions are again shown for comparison. It will be observed that the HC response at light load is different between fuels in the spark ignited Comet, with gasoline returning notably higher emission levels than diesel fuel. At higher load factors, HC emissions are equivalent or lower with gasoline. The elevated light load HC emissions with gasoline are thought to be due to very weak mixture areas existing in some parts of the chamber and the quenching of combustion within areas of the pre-chamber due to high local AFR or contact with the chamber walls. With diesel fuel, the self ignition temperature of the fuel is reached within the pre-chamber as the flame becomes established thus assisting with the promotion of combustion and reducing the probability of zones of unburnt fuel being formulated. Fuels of good cetane quality are therefore advantageous in this respect.

Comparative CO emissions are shown in Figure 58. The spark ignited Comet engine generally tends to return somewhat lower CO emissions with gasoline. This is probably due to the higher volatility of gasoline promoting improved mixture preparation and the reduced likelihood of rich mixture pockets.

NOx emissions are shown in Figure 59. With gasoline as fuel, the spark ignited Comet returns similar NOx emissions to a gasoline engine at light load. The NOx peak is reached at a much lower load factor however and emissions reduce more closely to the diesel performance at higher load factors. Diesel fuel returns appreciably lower NOx emissions in the spark ignited Comet by comparison with gasoline. Tests with various other fuels have confirmed that the magnitude of NOx levels is closely linked with fuel volatility.

5. RESPONSE TO GASEOUS EXHAUST EMISSION CONTROLS

In order to combat emissions for meeting legislative standards, throttling, exhaust gas recirculation and combustion retard were explored. These data are summarised in Figure 60. Considerable NOx control can be achieved by either EGR or retard without seriously affecting fuel economy or HC emissions. CO emissions are, however, penalised. Throttling also induced similar NOx reductions but with considerable HC, CO and fuel economy penalties.

Potentially viable control technology to achieve low NOx levels would therefore be based upon either EGR or retard, or a combination of both. Throttling does however have the one advantage of eliminating extremely lean mixtures at light load which can result in occasional misfire.

6. NOISE

Comparisons of combustion noise are presented in Figure 61. Of particular interest is the insensitivity of the spark ignited Comet to change in fuel specification, the noise levels being indistinguishable on gasoline and diesel fuel. Noise levels are similar to the gasoline engine up to 600 Hz. Beyond this point, noise levels do not reveal the typical rapid drop associated with the gasoline engine (60 dB/decade). In the frequency range of 600 Hz to 3kHz, noise levels exceed that of the diesel engine.

7. COLD STARTING CHARACTERISTICS

Starting characteristics on both gasoline and diesel fuel have not been examined at low ambient temperatures. Under test cell conditions, however, starting is immediate on either fuel.

APPENDIX 10

THE TEXACO CONTROLLED COMBUSTION SYSTEM - TCCS

1. HISTORY AND BRIEF DESCRIPTION OF THE SYSTEM DESIGN AND COMBUSTION PROCESS

The TCCS stratified charge combustion process has been under development now for several years. The finally evolved form of the system is shown schematically in Figure 62. Engines typically have compression ratios of 12:1.

In the TCCS system, high intensity air swirl is generated within the cylinders by the inlet port and/or masked valve. This swirl is magnified at TDC by the bowl-in-piston arrangement. Fuel is injected, mixes with the swirling air and is carried over the sparking plug positioned adjacent and downstream of the injector. The initial portion of fuel admitted to the cylinder is ignited by the spark which in multi-fuel applications is timed coincident with the injector opening. The burning zone formulated between the spark plug and injector is then fed by fresh fuel from the injector, which controls the rate of combustion, and by air due to the induced swirl. The fuel injection rate and level of air swirl must be closely matched to ensure that the flame neither extinguishes nor burns over-rich resulting in carbon formation. The engine load is solely governed by the quantity of fuel injected which also affects the duration of combustion.

As for the diesel engine, the power output of the TCCS process is smoke limited.

This system has been applied to cylinder sizes of between 3.25" and 5.75" bore but most of the more recent development appears to have been restricted to the smaller sizes for light duty applications.

Relevant data applicable to this system as extracted from the literature are presented as follows:-

2. COMBUSTION CHARACTERISTICS

Full load cylinder pressure diagrams for the TCCS process (109) indicates little influence of fuels of wide ranging specifications. This is achieved with coincident injection and spark timing set constant relative to TDC for each fuel. Cylinder pressure diagrams are notably smooth and rates of pressure rise are similar to gasoline engines at 30 psi/crank degree. Maximum cylinder pressures at full load in naturally aspirated TCCS engines are typically 700 psi compared with up to 900 psi for the gasoline engine. This reduction is due to the smoke limited torque output but means that the TCCS engine can utilise gasoline components and avoid the high friction levels of the IDI diesel. Diagrams also remain stable with reducing load.

As will be seen later in this Appendix, the similarity of cylinder pressure diagrams between fuels with the TCCS process enables vehicle noise levels to be maintained when changing fuels.

3. PERFORMANCE

Full load torque data for naturally aspirated engines extracted from several sources (109, 110, 111, 112, 113, 114) are shown in Figure 63. Data from several IDI diesel engines are also presented for comparison. With one exception all of the TCCS data are for multi-fuel engines.

Three of the sets of data reveal that the multi-fuel TCCS system matches the smoke limited band ascribed to IDI diesel engines. These Texaco results have been obtained using predominantly gasoline but the data for the 144 HCC engine are reported as typical for gasoline, jet and diesel fuel. When judging these comparisons, caution must be exerted since smoke limits for the TCCS data were not quantified. The published data describe the torque curves as being limited by acceptable, or just visible,smoke. By Ricardo standards, this would represent approximately two units on the Bosch scale, this level having been mainly used to formulate the band of IDI diesel data. Where the Texaco just visible rating lies in relation to this is not clear.

Two sets of data derived by Bartlesville Energy Technology Centre (111, 112) were quantified by smoke limits as shown in Figure 63. The data obtained from the LIS-183 multi-fuel engine (112) show very low smoke limited output on fuels ranging from gasoline to diesel. The incomplete development of this engine should,however, be noted. Perhaps significantly, smoke limited bmep was higher for the more volatile gasoline and broadcut fuels compared with diesel fuel. In evaluating the L-163-S engine (111) Bartlesville set a smoke limit of 4 Bosch units with diesel as fuel. The data obtained clearly show that, at the lower speeds, this particular engine must smoke excessively judging by the low levels of bmep. Only at the higher speeds does the performance approach the acceptable.

When analysing these results, due regard must be given to the multi-fuel nature of the engine. In being multi-fuel, leading parameters such as injection timing, spark timing, injection rate, air swirl,etc must represent a developed compromise. The potential for single fuel optimisation therefore exists. In the limit, this is illustrated by the gasoline optimised 147 engine (110). In this case, advantage is taken of the octane number of the gasoline, the injection is advanced relative to the spark and by inducing a greater degree of pre-mixed burning smoke limited bmep is elevated by up to approximately 10% as shown in Figure 63.

The smoke limited power output of the TCCS system has been demonstrated to be extended by approximately 25% with turbocharge (115) whilst still retaining multi-fuel capability.

4. FUEL ECONOMY

4.1 Steady State Fuel Economy

Test bed fuel consumption data for several TCCS multi-cylinder, multi-fuel engines are shown in Figure 64 (109, 111, 112, 114). These data encompass naturally aspirated engines and fuels from gasoline to diesel.

By virtue of low compression ratio and open chamber design, the TCCS engine has favourable friction levels in comparison with the IDI diesel. Friction levels for the L-141 engine for example (114) are indistinguishable from the gasoline engine from which the multi-fuel engine was derived. The friction levels published are also typical of a European gasoline engine and are approximately 40% lower than a typical IDI diesel engine.

Lower friction enables the TCCS engine to compete favourably with the light duty IDI diesel engine as regards brake specific fuel consumption. Figure 64 shows that at part load the TCCS engine generally returns fuel economy within the lower levels of the IDI diesel band. Bartlesville (111) did experience exceptionally high fuel consumption at light load with the L-163-S engine on diesel fuel in comparison with gasoline, indicating a combustion problem. These data appear somewhat untypical and at higher load factors fuel consumption was similar for both fuels. These results are also shown in Figure 64. In a similar multi-fuel TCCS engine, Bartlesville report that broadcut fuels (50/50 diesel/gasoline) return fuel consumption levels intermediate to both base fuels.

In the upper third of the load range, there is a definite general tendency for the fuel consumption to rise significantly in comparison with the IDI diesel engine which produces a much flatter characteristic. This indicates combustion inefficiency at richer AFR and potentially will have an adverse effect upon smoke limited torque.

Turbocharging (115) enables a much flatter trend in high load specific fuel consumption to be achieved, particularly at the higher speeds. Turbocharging has little effect however upon light load fuel consumption, the results on a range of fuels being in close agreement with the naturally aspirated data.

When comparing these data, it should again be remembered that the engine represents a compromise multi-fuel development and that the potential for improvement by single fuel optimisation exists. In this respect however the gasoline optimised 147 engine (110) showed little difference in fuel economy when compared with typical multi-fuel results.

In the multi-fuel engine, injection and spark are coincident whilst in the gasoline optimised engine the injection is advanced relative to the spark. Bartlesville (112) examined the influence of spark retard at part load with the LIS-183 engine. With gasoline, retarding the spark timing reduced engine efficiency and a similar but smaller influence was also observed with broadcut fuel. Retarding the spark had no effect upon efficiency with diesel fuel. It was concluded that best efficiency or zero penalty was achieved by retaining the coincident multi-fuel philosophy with respect to spark and injection timing.

With coincident spark and injection timing at road load conditions, thermal efficiency was generally lowest with gasoline as fuel. Maximum efficiency was dependent upon test conditions and fuel. At 1500 rpm, 19 psi bmep, diesel fuel gave the best efficiency. At 2600 rpm, 37 psi bmep, broadcut fuel returned maximum efficiency. At 2000 rpm, 26 psi bmep, broadcut fuel gave the best efficiency with advanced settings whilst diesel fuel returned maximum efficiency with retarded settings. At full load smoke limited conditions, diesel fuel proved to have the worst efficiency, this being commensurate with the lowest smoke limited torque as previously recorded. Whilst these results are relevant, the scatter of thermal efficiencies were however small between fuels at approximately 2%.

At part load conditions, the setting of the coincident timings relative to TDC appeared to have little effect on thermal efficiency compared with the influence of fuel.

The combination of unthrottled intake and gasoline friction levels enables the TCCS engine to idle with low fuel consumption. TCCS engines typically require $5mm^3$ of fuel for each working cycle compared with $10mm^3$ for light duty IDI diesel and gasoline engines of similar size. This results in a considerable improvement in idle fuel consumption (113) and may be extremely beneficial in some operating circumstances.

In some instances, Texaco employ intake throttling at light load to elevate exhaust temperatures to enhance emissions control. Single-cylinder studies (116) based on the L-141 engine with throttle applied below load factors of 60 psi imep demonstrate an increase in indicated fuel consumption of between 3 and 10%.

Considering the low part load fuel economy of the TCCS engine, vehicle fuel consumption should also show favourable results.

Level road fuel consumption for the naturally aspirated L-141 engine fitted to a Jeep (116) demonstrates a 32% improvement over the standard gasoline engine at light load (25-30 mph) reducing to 25% at higher load factor (50 mph). These comparisons are made with gasoline fuel and the improvements are commensurate with the steady state test bed economies of typical TCCS and gasoline engines. As would be expected, volumetric fuel consumption is further improved at the same operating conditions to 52 and 50% respectively when using diesel fuel. Other published data provide supporting evidence (e.g. 113).

Following the trends of the boosted TCCS test bed data, turbocharge (116) improves level road fuel consumption by some 16% at higher speeds and loads in comparison with the naturally aspirated engine, but at lower speeds and loads this improvement was not apparent.

4.2 Transient Fuel Economy

In transient road operation (116) the Jeep equipped with the L-141 TCCS engine returned considerable economy gains when running on gasoline in comparison with the standard gasoline engined vehicle. These improvements ranged between 73 and 39% dependent upon cycle, the cycle with a high proportion of idle time returning the best improvement.

Published 1975 FTP economy comparisons (117) for the standard Jeep and the turbocharged L-141 TCCS equipped version reveals a 59% improvement in economy when using gasoline as fuel. More significantly in the context of this report, Texaco (117) evaluated gasoline, broadcut and diesel fuels in a Cricket vehicle fitted with the turbocharged L-141 TCCS engine operated over the 1975 FTP. In this case, fuel economy varied only by the expected degree owing to the different volumetric heat contents of the fuels and ranged from 28-30.2 mpg (see Table 2 Line 1 attached to this Appendix). Other data (118) also demonstrate little influence of fuel specification upon transient economy (see Table 2, Line 2 attached to this Appendix). These results were accomplished without changing engine settings and reflect the excellent multi-fuel character of the TCCS system.

In the aforementioned FTP tests, both TCCS equipped vehicles returned NOx levels within the range 1.5-2.0 g/mile. This provides a convenient means of comparison and Figure 65 shows these and other TCCS data compared with 1978/79 model year gasoline and IDI diesel certification vehicle results (119) (i.e. equivalent NOx). From Figure 65, it is apparent that the excellent part load test bed economy of the TCCS engine is not reflected in these data under the conditions imposed. In the case of the diesel fuelled TCCS data, the economy figures do not reach the IDI diesel band and only excel the gasoline engine data due to differences in volumetric heat content. Fuel efficiency of the gasoline fuelled TCCS data also does not, with the exception of the Gremlin result, even approach the IDI diesel gasoline equivalent volumetric consumption. It should be stated, however, that the results compare favourably with the gasoline engine, the gasoline fuelled TCCS results either being superior to or generally lying within the upper limits of the gasoline economy band.

These data present an anomaly for which there is no apparent immediate answer. One possible explanation is that test bed economy has been sacrificed in the interest of lower emissions. In this context, all of the TCCS vehicle data, except for the turbo-charged L-141 Jeep, were obtained with catalysts fitted for HC control to approximately 1.0 g/mile. In comparison with the diesel, which will not be fitted with such units, some fuel economy may have been sacrificed due to back pressure penalties. In addition, both sets of data relative to the L-163-S installations were also obtained with EGR for NOx control to 1.5-2.0 g/mile, EGR apparently not being required for the other applications at this NOx level. Relative to the diesel, some further economy penalties may be evident therefore due to the EGR in these instances. Whilst this assortment of emission controls makes direct comparisons with diesels more confused, the choice of comparing at equivalent NOx levels is entirely logical, since it is known that in all classes of engine NOx control can adversely affect fuel economy. In this respect, the IDI diesel has the general advantage of meeting the comparative NOx levels utilised with little control requirement.

Other factors may reflect upon the TCCS fuel consumption. These include, especially in the case of the Jeep, lower transmission efficiency relative to the certification vehicles or vehicle/engine/ transmission mis-matching. Furthermore the TCCS powered Cricket is equipped with automatic transmission and compared with 1978/79 vehicles of which the majority probably have manual transmissions is this inertia class. In addition, the lowest fuel consumption data for TCCS Cricket and Jeep vehicles shown in Figure 65 were obtained by gravimetric determination and not carbon balance. This imposes a further variable making direct comparison more difficult. In this respect, Ricardo experience suggests that carbon balance and gravimetrically determined fuel consumptions during the cold start

FTP test differ by up to 10% with gravimetric measurements returning higher fuel consumption. This difference is reduced to between 2 and 5% for hot start FTP tests. Fuel consumption with the TCCS engine in comparison with the diesel may also be adversely influenced by the TCCS system suffering from some form of transient combustion problem.

Recent Texaco data (110), however, demonstrate favourable fuel consumption with TCCS engines when operating to a simulated parcels delivery schedule involving many engine stops and starts (US Postal Service Programme). Under these conditions, delivery vans equipped with gasoline, TCCS and diesel engines of similar rating, returned c.40% worse fuel consumption with the gasoline engine. The TCCS and diesel engine equipped vehicles gave similar fuel consumptions with a bias in favour of the TCCS. The TCCS engine was run on diesel fuel. Data extracted from correspondence between Texaco and DoE, November 1980 submitted following Texaco's examination of the orginal draft of this report. In other parcels delivery test programmes (United Parcels Service Programme) the TCCS equipped vehicle returned a 40% consumption improvement with respect to the standard gasoline vehicle when operated on gasoline and 25% diesel/75% gasoline fuels. In this specific UPS programme, a diesel engined vehicle was not evaluated. A diesel engined vehicle was, however, tested during other programmes and showed a 21% economy improvement relative to the gasoline vehicle.

5. GASEOUS EXHAUST EMISSIONS

5.1 Steady State Gaseous Exhaust Emissions

The Texaco data uncovered for this study are predominantly restricted to vehicle FTP results with some limited test bed data covering the influence of emission control parameters with gasoline as fuel in single and multi-cylinder versions of the L-141 engine. These data are covered in Sections 5.2 and 6 respectively of this Appendix.

Bartlesville (111) examined the influence of both speed and load on gaseous emissions emitted by the L-163-S multi-fuel engine operated on both diesel and gasoline fuels. Results are shown in Figure 66.

Regarding NOx emissions, the trends for both fuels