ENGINE OPERATION ON LIGHT DIESEL FUELS

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Johannesburg, 1989.

I declare that this dissertation in my own, unaided work, except where specific reference is made. It is being submitted for the Degree of Master of Science in Engineering in the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination in any other University.

Robert S Falk.

9 day of January 1989

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ABSTRACT

Circumstances may require that blends of diesel refined from crude oil be blended with lighter hydrocarbons to extend the supply of diesel.

An ADE 236 diesel engine failed to complete a durability test using a 'worst case' blend of this 'light diesel' because of erosion of the piston crown, subsequently found to have been caused by the severe combustion characteristics of the fuel. Satisfactory durability performance can be achieved when using the worst case fuel by retarding the injection timing, or by retaining standard injection timing and using either an 'ignition-improved worst case' fuel or using a blend of diesel and heavy naphtha.

Two other engines fuelled with the worst case light diesel were tested. A standard ADE 314 successfully completed the durability test, but a Deutz F6L 413F failed due to piston crown and cylinder head erosion.

In general, rated power was reduced slightly and, depending on operating conditions, overall fuel consumption is expected to be higher. Wear of certain components of the fuel injection equipment was generally increased, particularly when using the blend containing heavy naphtha, thus restricting this fuel to emergency use only.

ACKNOWLEDGEMENTS

This dissertation describes tests which were carried out on diesel engines using the facilities of CSIR, Pretoria. Of the six tests carried out, the first was carried out under the supervision of another member of the staff before the author joined CSIR, the second was carried out as a joint project and the last four were carried out solely under the supervision of the author. All the tribological investigations were carried out by trained tribologists employed by CSIR.

The author wishes to acknowledge the assistance given by CSIR, Pretoria, for providing the facilities to carry out the work, the financial support from the National Energy Council incorporating the Foundation for Research Development, the support from Atlantis Diesel Engines (Pty) Limited, British Petroleum Southern Africa (Pty) Limited, Deutz Diesel Power (Pty) Limited, and the support from those who wish to remain anonymous, including a truck fleet-operator, without whom this investigation would not have been possible.

The author also wishes to thank the National Energy Council for permission to publish this dissertation.

CHANGE IN THE STRUCTURE OF CSIR

The Council for Scientific and Industrial Research, Pretoria, of which the National Mechanical Engineering Research Institute (NMERI) was part, was restructured in April 1988. The Institute's Heat Mechanics Division was incorporated into the Division of Production Technology (DPT), the institute's Tribology Division was incorporated into the Division of Materials Science and Technology (DMST), and the name 'Council for Scientific and Industrial Research' was changed to 'CSIR'.

The tests which were carried out before CSIR was restructured are refered to in the text as having been carried out by the National Mechanical Engineering Research Institute, whilst those carried out after restructuring are refered to as having been carried out by Division of Production Technology and the Division of Materials Science and Technology.

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GLOSSARY

ADE	Atlantis Diesel Engines (Pty) Limited
BP	British Petroleum Southern Africa (Pty) Limited
CEC	Coordinating European Council
CSIRO	Commonwealth Scientific and Industrial Research Organisation
DBAG	Daimler-Benz AG
DDP	Deutz Diesel Power (Pty) Limited
DIESANOL	Trade name for an ignition improved methanol
DMEA	Department of Mineral and Energy Affairs
DMST	Division of Materials Science and Technology, CSIR
DPT	Division of Production Technology, CSIR
EMA	Engine Manufacturers' Association
FRD	Foundation for Research Development, CSIR
IILD	Ignition improved light diesel - tops light diesel to which ignition improver has been added
KHD	Kloeckner Humboldt Deutz AG
NLD	Naphtha light diesel - 75 % diesel and 25 % heavy naptha by volume
NMERI	National Mechanical Engineering Research Institute, CSIR
SABS	South African Bureau of Standards
SATS	South African Transport Services
TLD	Tops light diesel - 75 % diesel and 25 % Tops by volume
Tops	Hydro-treated straight run tops

CHAPTER 1

BACKGROUND

1.1 Fuel Crisis

In the late 1970's the demand for diesel was growing more rapidly than that for petrol, as shown in Fig 1.1, and the possibility ex[illegible]sted that the refineries would not be able to produce sufficient diese[illegible] derived from crude oil.

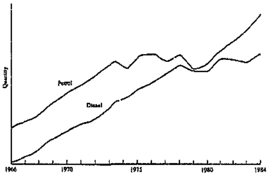

FIGURE 1.1 South African petrol and diesel demand, 1966 to 1984

Source: DMEA (1985)

The graph shows that the demands for petrol and diesel diverge after approximately 1980 giving the impression that the crisis had passed by then.

However, should a shortage of fuel occur for whatever reason, strategic transport and agriculture would have to be kept mobile. These sectors are generally powered by diesel-fuelled engines and therefore a need still exists for a strategy to maximise the yield of diesel from each barrel of crude oil.

To this end, exploratory tests were carried out by NMERI to determine what steps would be required to achieve this objective.

1.2 Exploratory Tests

These exploratory tests were carried out to evaluate methods of extending the quantity of diesel fuel produced from crude oil, but only those fuels and blends which could be used in a vehicle's existing fuel system were considered for evaluation because of the importance of being able to switch from one fuel to another quickly.

The products which were investigated as substitutes for, or extenders to, diesel fuel included petrol and heavy naphtha derived from crude oil, sun-flower oil, and the alcohols including methanol, ethanol, and 'propanol-plus' [CSIR (1978), CSIR (1979), CSIR (1980 a), CSIR (1980 b)]. Propanol plus is a mixture of propanol and higher alcohols and is a by-product from the Sasol-oil-from-coal process.

Laboratory tests were successful with blends of diesel-petrol and diesel-heavy naphtha [CSIR (1980 a)], and this led to a limited field trial being carried out using diesel powered buses belonging to the Pretoria City Council [Myburgh (1981)].

1.3 Involvement of Government and Oil Industry

The results obtained from the exploratory tests were submitted to the Department of Mineral and Energy Affairs (DMEA) which, by 1980, had established a 'Light Diesel sub-committee' which prepared a specification for a 'light diesel' which could be produced from crude oil in an emergency. At the same time discussions were held between CSIR and British Petroleum Southern Africa (Pty) Limited (BP) which led to the preparation of a specification for what BP considered to be a 'worst case light diesel' that could be produced from crude oil using products available from existing refinery streams. The physical properties of the DMEA's 'emergency light diesel' and BP's worst case light diesel were similar.

1.4 Laboratory Tests

The effects of using the worst case light diesel were investigated using an ADE 236 diesel engine which was selected because it was known to be sensitive to 'off-specification' fuels [Myburgh (1983)]. The engine was subjected to a durability test, and the first signs of mechanical stress were detected after only 100 hours. The test was stopped after 340 hours, because of severe erosion of the piston crown, an example of which is shown in Fig 1.2.

FIGURE 1.2 Severe erosion of piston crown

Source: Falk and Myburgh (1987)

This erosion was found to have been caused by the poor combustion characteristics of the fuel.

1.5 Aim of the Investigations

The failure of the ADE 236 fuelled with the worst case light diesel led to laboratory tests being carried out to determine what steps would be required to ensure that the engine could be operated satisfactorily when fuelled with light diesel fuels.

When this had been determined, similar tests were carried out on an ADE 314 diesel engine and on a Deutz F6L 413F diesel engine, both of which were selected because of their importance in transport.

The results of the first ADE 236 test carried out by Myburgh [Myburgh (1983)] are discussed in Chapter 7 together with the results of the subsequent work. Note should be taken that the experimental work described in this dissertation relates only to diesel and extenders derived from crude oil.

CHAPTER 2

ALTERNATIVES AVAILABLE

This investigation concentrated on the effects on engine performance and durability of operating on light diesel fuels which could be introduced as a short-term expedient in an emergency, and what steps, if any, would be necessary to ensure the engines' survival. Alternatives exist in the longer term for increasing the yield of diesel which may be used in engines requiring little or no modification, and for using 'new' fuels in modified engines. Some of the alternatives described below indicate how this investigation fits into the broader developments in the fields of fuels and engines.

Sections 2.1 to 2.3 deal with fuels which may be used in engines which are essentially unmodified, and Sections 2.4 and 2.5 deal with fuels for which modifications have to be made to the engines.

2.1 Diesel

Fuels derived from crude oil are expected to predominate until approximately 2010 by which time fuel from alternate sources will start to make an impact (Heywood 1981). Options exist for enabling vehicles to travel further either by reducing the quantity of diesel used by making vehicles more efficient, or alternatively, by increasing the total quantity of diesel fuel available. The latter option may be accomplished by changes in refining processes, or by adding water, other hydrocarbons not currently used, or by dual-fuelling engines with petrol, alcohol, or gas.

The manufacture of diesel from non-crude oil raw materials is dealt with below in Section 2.2, and mixtures of diesel with alcohol are dealt with in Section 2.5 below.

2.1.1 Changes in refining routes

The cetane number of diesel fuel in various industrialised countries varies between 40 and 50, although diesel of 32 cetane number is being used in Western Canada [Anon (1984 a), EMA (1979)]. For comparison, the minimum cetane number in South Africa is set by the South African Bureau of Standards (SABS) [SABS (1969)] at 45, although the industry norm is 48.

Changes in the quality of diesel fuel will result from changes in the refining processes if more cracked components are added to diesel to increase the yield [Van Paassen (1986)]. The net effect of adding these cracked components is that the cetane number will drop but the density will rise as shown in Table 2.1.

TABLE 2.1 Cetane number and density of diesel derived from different refining processes

		Distilled	Thermally Cracked	Catalytically Cracked
Cetane number		30 to 60	30 to 50	0 to 20
Density	kg/m^3	820 to 860	820 to 880	920 to 980

Source: Van Paassen (1986)

TABLE 2.2 Brief specification of possible future diesel fuels compared with SABS 342:1969

		SABS 342:1969	RF-70-A-84	RF-72-A-84
Cetane number		45 min.	43 to 46	40 to 42
Density	kg/m^3 @ 15 'C	not specified	870 to 885	815 to 860
Viscosity	cSt @ 40 'C	1,6 to 5,3	1,5 to 4,5	1,5 to 4,5
Flash point	'C	55 min.	56 min.	to be recorded

Notes:

RF-70-A-84 = Diesel fuel, European, Japanese, Australian, and New Zealand type, expected future quality (circa 1990 ·orst case)
RF-72-A-84 = Diesel fuel, South American, expected future quality (circa 1990)

Source: SABS (1969), Pearson and Hawkins (1986)

The quality of fuel which is expected in 1990 is reflected in the specifications prepared by the Coordinating European Council (CEC), as shown in Table 2.2.

Table 2.2 shows that the fuel RF-70-A-84 would appear to be similar to the existing [illegible] specification for diesel with the exception of density. The fuel RF-72-A-84 appears to be similar to the light diesel fuels which could be introduced in South Africa in an emergency. However, the minimum cetane number of the fuel in South Africa would most probably be 45 as specified by SABS and not 40 which was cited in a specification of an 'emergency light diesel' [FRD (1987)].

The quantity of diesel fuel may be extended by altering the specification to broaden the boiling range compared with current fuels [Titchener (1981)], or by adding other components which are usually lighter fractions. These fuels are termed 'broad cut' fuels. Benefits include an increase in the volume of distillate of between 3 and 5 % for an increase of 20 °C in the final boiling point (FBP) [Lanik and Ecker (1984)]. The effects of altering the boiling range of diesel, for example increasing the temperatures for 10 and 90 % recovery, includes an increase in exhaust smoke [Englin, et al (1981), Johnson (1986)], and a rise in gravimetric specific fuel consumption but the effects stabilise at 10 % recovery temperatures of 270 to 300 °C and 90 % recovery temperatures of 360 to 370 °C [Lanik and Ecker (1984)].

Broad cut fuels may be used successfully in compression ignition (CI) engines; for example, experiments carried out in Canada with 7 broad cut fuels indicated that whilst one diesel engine may operate satisfactorily on a diesel of 31 cetane number, it may be too low for others. Thus, a cetane number of 35 for diesel which is expected in Canada in 1990 may be too low. [Currie and Whyte (1981)].

As a means of alleviating a possible shortage of diesel fuel, an alternative suggested in the USA [Anon (1983 a)] is to divide the market into three sectors and to supply sufficient diesel of the appropriate quality to each, thus reducing the quantity which needs to be processed into the highest grade. The proposal called for a cetane number of 45 for city buses, light trucks and cars, a cetane number of 40 for trucks and tractors, and a third grade with a cetane number of 32 for railways, stationary engines and marine use. However, a distribution network would be needed, and the cost would probably preclude this solution.

2.1.2 Diesel water emulsions

Diesel-water emulsions have been developed as fire-resistant fuels [Weatherford et al (1979)], but may lead to reduced performance which can be restored by adjusting the fuel injection pumps. Two fuels were cited, diesel mixed with 10 % by volume water and 6 % surfactant, and diesel mixed with 5 % water, 3 % surfactant and 0,2 % anti-rust agent. Other experiments have shown [Anon (1987)] that an emulsion of diesel and water in the ratio of 10:1 with a 'selected organic compound' could be used successfully without any adjustment to the engine and produce higher torque up to mid-speed, but lower torque thereafter.

Benefits of adding water include reduced specific fuel consumption for which the quantity of water can be optimised, usually between 10 and 15 % [Crookes et al (1980)], reduced oxides of nitrogen emissions [Anon (1987)], inhibition of the formation of soot, promotion of more complete combustion thereby lowering carbon monoxide and unburnt hydrocarbons [Crookes et al (1980)].

Problems include the possibility that the surfactant would degrade when the fuel is recirculated from the engine to the fuel tank to prevent the water from coming out of solution [Weatherford et al (1979), Crookes et al (1980)], and pure diesel had to be used just before shutting down the engine to reduce the problem of corrosion in the fuel lines [Anon (1987)].

Adding water to diesel reduces the cetane number by 13 numbers from approximately 45 to 32 for an increase in the water content from 0 to 20 %, but this deficiency can be corrected by adding an ignition improver such as amyl nitrate at a concentration of 2 % to improve the cetane number by 10 to 15 numbers [Tsenev (1983)].

2.1.3 Extended diesel

Diesel may be blended with specific products, for example, petrol, heavy or light naphtha, or propanol-plus to produce an 'extended' diesel.

Considerable experience has been gained by the South African Transport Services (SATS) in using extended diesel fuels in Class 35-200 series railway locomotives [Tarboton (1980), Venter (1983), Falk (1986 a)]. Blends containing between 15 and 30 % petrol have been tested, but power was reduced by 3 % when operating on the blend containing 25 % petrol. Blends of diesel containing 15 to 40 % heavy naphtha in the boiling range 125 to 185 °C eventually led to the controls on the GM engine used becoming unstable when using the blend containing 40 % heavy naphtha. Although the tests were carried out with heavy naphtha, sufficient quantities were not expected to be available to make this a viable solution. Blends of diesel containing 20 to 30 % light naphtha in the boiling range 45 to 115 °C showed that governor hunting was a problem when operating on the blends containing 25 and 30 % light naphtha, and therefore, the 20 % blend would seem to be the most likely one to test.

Blending diesel with other products results in a larger quantity of diesel becoming available, but this benefit may be eroded by changes in fuel consumption of engines operating on the extended diesel. Tests carried out by Ricardo Consulting Engineers Limited on an indirect-injection engine suggested that when compared with operation on diesel, there would be an increase in fuel consumption of approximately 0 to 6 % when operating on diesel-naphtha blends [Needham and Cooper (1982)].

Other tests have shown that no improvement in efficiency could be obtained from using a diesel - petrol blend [Clark and Heim (1979)]. However, when the diesel was injected and petrol inhaled through the air-intake manifold the total fuel consumption could be reduced by 15 to 20 % with best results having been obtained with a diesel to petrol ratio of 85:15.

Adding petrol to diesel reduces the cetane number from, say, 50 to 32 when the blend contains 50 % petrol [Holmer et al (1980)]. Engines may still be able to operate on fuels of such low cetane number, either without modification or by using 'staged injection' whereby 0 to 20 % of the total fuel volume is injected early, the exact quantity and injection timing being determined experimentally [Anon (1983 b)].

The methods of extending diesel fuel described above may alleviate the shortage of diesel in countries which have crude oil or dollars with which to purchase crude oil. Countries lacking these resources but which have either plenty of coal or land and sunshine may rely on these alternative resources to produce synthetic diesel or alcohol [Eden (1979)].

2.2 Synthetic Diesel

Of the alternative sources of liquid fuels, the estimated reserves of distillate from shale are 3 times those of the recoverable reserves of crude oil in the Middle East [Dwyer (1984)] and two thirds of these are located in the USA [EMA (1979)]. Fuel from shale is low in sulphur, but the catalyst used in the process may be poisoned due to high nitrogen and metal contents [Lanik and Ecker (1984)].

Coal can be converted into distillate for which there are more than 150 patented methods. The Fischer-Tropsch method has been in commercial use in South Africa at Sasol One since 1955 [EMA (1979), Dry (1982)]. The NCB-LSE process has been in pilot operation in the United Kingdom, converting 2,5 t of coal per day into distillate with the hope of converting up to 10 Mt per year [Davies and Thurlow (1984), Anon (1986 a)].

In the Fischer-Tropsch fluidised bed 'Synthol' process 77 % of the product is liquid of which 52 % is diesel, whilst in the fixed bed process 87 % is liquid of which 75 % is diesel [Dry (undated)]. Diesel from the high temperature (325 °C) Synthol process has a cetane number of 55 whilst diesel with a cetane number of 75 can be produced by the low-temperature (220 °C) fixed bed process [Dry (1982)].

Diesel produced by Sasol generally has lower viscosity than that derived from crude oil, and experiments have shown that, when compared with operation on diesel derived from crude oil, there is a loss of still be able to operate on fuels of such low cetane number, either without modification or by using 'staged injection' whereby 0 to 20 % of the total fuel volume is injected early, the exact quantity and injection timing being determined experimentally [Anon (1983 b)].

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Diesel produced by Sasol generally has lower viscosity than that derived from crude oil, and experiments have shown that, when compared with operation on diesel derived from crude oil, there is a loss of power at rated speed of between 8,4 and 15,5 % when operating on diesel fuels derived from coal which have viscosities in the range 1,8 to 1,4 cSt [Hansen and Meiring (1982)].

There is also renewed interest in exploiting torbanite for producing distillate, but the project is still at the feasibility stage. Torbanite is a coal formed from algae and is found in layers sandwiched between conventional coal in the Eastern Transvaal.

Liquid fuels are also produced from coal in the USA with the trade names such as Exxon Donor Solvent (EDS), H-coal and SRC-II but these have to be blended with diesel to result in a fuel of acceptable cetane number. Whilst an engine has been operated on a blend of 75 % No 1D (US automotive) diesel and 25 % SRC-II by volume which had a cetane number of 37, the plungers on all injection pumps were coated with a black deposit, the removal of which revealed pitting of the plungers [Hoffman (1982)]. The deposit appeared to include particles of unburnt fuel and coal dust. Other problems included incompatibility of the fuel and seal materials.

2.3 Vegetable Oils

The use of vegetable oils is not new, for example, a generator driven by a diesel engine fuelled by soybean oil was exhibited at the 1932-33 Chicago World Fair [Baldwin (1983)].

An air cooled indirect injection Deutz diesel engine fuelled with degummed sunflower oil ran satisfactorily for the equivalent of 8 000 hours of farming duties [Fuls (1983)], and a Caterpillar indirect injection engine ran successfully on a blend of 30 % soybean oil and diesel [Dwyer (1984), Suda (1984)]. The heat plugs fitted to the pistons of the Caterpillar engines were modified to protect the aluminium pistons from concentrated thermal loads thus ensuring uniform wear of the piston rings and liners (Fig 2.1).

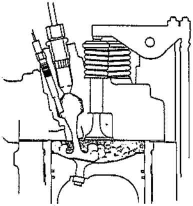


FIGURE 2.1 Pre-chamber Caterpillar engine with heat plug fitted to pistons

Source: Suda (1984)

More problems occur with direct injection engines than with indirect injection engines, including injector nozzle coking on the direct injection engines and filter clogging on both types of engine (Onion and Bodo (1982)]. The worst oils are crude soybean oil and crude peanut oil [Mumke and Barsic (1981)]. The nozzle-coking problems on direct injection engines may be overcome by using trans-este :fied sunflower oil where trans-esterification is a process carried out with the aid of a catalyst whereby the glycerol and fatty acids of the sunflower oil are converted to glycerol and the ester [Fuls (1983)]. Piston ring sticking may be overcome by reducing the clearance between the piston and the bore to 1/1000th of the piston diameter, reducing the height of the top land and aiming the fuel into the centre of the piston to avoid residues of fuel from collecting on the cylinder walls (Ziejewski and Kaufman (1982), Elsbett et al (1983)].

Other problems include carbon build-up in the inlet ports, and incompatability between the fuel and fuel system materials, for example, where the fuel acts as a paint stripper (Fuls et al (1984), Ziejewski and Kaufman (1982)].

2.4 Coal

The reserves of coal are estimated to be 4,5 times those of crude oil when compared on the basis of energy [Walker (1984)], and this energy could be maximised by using coal without first converting it into liquid fuel.

The original patent by Rudolph Diesel in 1892 related to operating an engine on solid and liquid fuels [Robben (1983)] Coal dust was used in Germany between 1928 and 1944 during which time engine were run at speeds of up to 1 600 r/min and developed up to 600 hp (450 kW). More recently engines have been operated by injecting stabilised coal-water slurries using 10 to 20 μ sized dust at concentrations of 50 to 60 % by mass [Robben (1983)], using only powdered coal [Kamo et al (1986)], or using micronised coal or carbon black as a component of a thixogel in which the liquid was diesel [Zatko et al (1982)]. A thixogel is a mixture of solids in a liquid which acts as a gel until it is pumped whereupon it acts as a liquid.

Problems do exist, for example, the engine which was operated on powdered coal as the sole source of energy had to be warmed up first using diesel [Kamo et al (1986)]. It was adiabatic, that is, it had no water cooling system, and the thermal efficiency was similar for operation on diesel or powdered coal. To reduce wear the inside of the engine had been sprayed with a ceramic layer to a thickness of 1,0 mm except for the cylinder head where the layer was 1,25 mm thick. However, wear of the piston rings remained a problem. The other engine was operated successfully using a thixogel containing up to 30 % solids in oil. (Zatko et al (1982)).

Irrespective of the system used, the coal has to be powdered to 5 to 20 μ by ball milling, micro-pulverisation or ultrasonically, after which the sulphur and ash have to be removed. In the overall costing of, for example, operating diesel engines on coal-oil mixtures, the monetary cost of grinding the coal may be of prime importance [CSIRO (1985)], whilst an indication of the energy cost involved is that micronising 8,3 t of coal to 40 μ for use in industrial boilers requires approximately 190 kWh [Smith (1986)].

An indication of the relative overall efficiencies for converting coal into motive power is given in Table 2.3:

TABLE 2.3 Overall efficiencies for coal energy conversion (raw material to motive power)

	%
Coal-solid fuel	59
Coal liquifaction	51
Methanol from coal	38
Electricity from coal	25

Source: EMA (1979)

2.5 Alcohols

The use of alcohol as a fuel for internal combustion engines can be traced back to at least 1903 when an engine of 17 l cubic capacity and developing 200 ch (french hp) powered a vehicle at 177,5 km/h [Agache (undated)].

Alcohols may be derived from renewable resources such as sugar cane, sago and nipa palms, cassava, grain sorghum, or maize, or as a by-product of the Sasol process, or from natural gas such as has been found at Mossel Bay [Ricardo (1982), Wilkinson (1983)]. Ethanol is used in Brazil in CI and spark-ignition (SI) engines to reduce Brazil's dependence on crude oil to the extent that 95 % of all new cars sold there run on 100 % alcohol [van Niekerk (1987)].

One disadvantage of using alcohols as a substitute for diesel is that the energy content on a volumetric basis is much lower than that of diesel, and the volume of ethanol and methanol need to be 169 % and 228 % respectively those of diesel for the same energy inputs [EMA (1982)]. Tests confirming these findings showed that the ratio was 160 % in the laboratory and 175 % on the road for ethanol with ignition improver [Acioli (1982)]. Since the volumes of fuel are so much greater, engine performance is limited by the volume of fuel which can be injected using currently-available fuel injection equipment, effectively restricting conversions to naturally aspirated engines [Weiss & Hardenberg (1986)].

Ten options were identified for enabling engines to operate on methanol (Kidd and Kreeb (1984)], but the comments are equally valid for other alcohol fuels. Some of these options are:

. convert the engine to spark-ignition
. pure alcohols (compression-ignition)
. fumigation (vapourised fuel or gas mixed with the intake-air)
. dual injection of diesel and alcohol
. emulsions and diesel-alcohol blends

2.5.1 Pure alcohol (SI engines)

Blends of petrol and methanol have been tested in the USA in cars fitted with SI engines where fuel economy improved provided the blend contained less than 12 % methanol [Anon (1984 b)]. In Canada cars have been operated successfully on methanol although stainless steel fuel tanks and nickel plated fuel pump components were fitted, and cold-start problems were overcome by adding 10 % petrol or 5 % dimethyl ester to the methanol [Anon (undated a)]. Improvements in efficiency may result from, for example, using ethanol in SI engines where a change in efficiency from 28 % for petrol to 38 % for ethanol may be realised provided that the engine settings are optimised for each fuel [Acioli (1982)].

Diesel engines may also be converted to operate on ethanol with spark assistance, thereby operating in a similar manner to SI engines. An engine with a 12:1 compression ratio (CR) has been developed from a fumigated alcohol-diesel and multi-fuel SI engine to promote fuels which would not ignite in CI engines [Agache (undated)]. The fuel is injected directly into the cylinders during the induction stroke and is ignited using a spark. Tests carried out by NMERI [Myburgh (1986 a)] showed that compared with the Perkins 4.236 diesel engine on which the conversion was based, the ethanol-fuelled engine developed 27.6 % more power at rated speed than its diesel equivalent.

The conversion from a standard diesel engine entailed machining the pistons to lower the compression ratio, installing an injection pump capable of injecting the necessary volume of fuel and fitting a spark-ignition system. Not only is this expensive, but the engine cannot be converted back for operation on diesel without replacing many of the components altered for the original conversion. Notwithstanding these comments, this type of engine could be used in sugar plantations such as in Natal where ethanol could be distilled from sugar cane and where the fuel distribution network would be confined to a small geographical area.

Trials are underway in several countries using alcohol-fuelled buses. The Golden Gate Bridge Highway and Transportation District in California is using two methanol-fuelled buses of which one is fitted with an MAN FM-system engine which incorporates stratified charge injection and spark ignition [Anon (undated b)] (Fig 2.2). The trials were scheduled to continue for at least 160 000 km and showed that the performance of the buses when compared with diesel-powered buses was the same, the exhaust emissions were lower, but the durability was not as good [Anon (1986 b)]. Coincident with hosting an international conference on the use of alcohol in transportation in New Zealand in 1982, the Auckland Regional Authority was operating two methanol-fuelled buses, one fitted with the MAN FM system and the other fitted with a Mercedes-Benz SI engine in which the methanol was vapourised into the intake manifold using a heater [Ricardo (1982)].

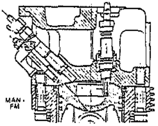


FIGURE 2.2 Schematic diagram of MAN FM combustion chamber

Source: Duggal et al (1984)

2.5.2 Pure alcohols (CI engines)

Pure alcohol without ignition improver and comprising a blend of 90 % alcohol and 10 % castor oil has been tried in Brazil in a diesel engine which had a compression ratio of 21:1, but did not self-ignite [EMA (1982)]. For self-ignition the engine would need a compression ratio of 25,3:1 if neat methanol were to be used [Schaefer et al (1987)].

Alcohols can be used in CI engines by adding ignition improvers which are mainly nitrate compounds such as amyl nitrate, hexyl nitrate or cyclohexyl nitrate at concentrations of approximately 15 to 16 % to give a cetane number of 40 [EMA (1982)]. In Brazil tri-ethylene glycol dinitrate (TEGDN) was developed using locally-available tri-ethylene glycol as a raw material because of the Brazilian government's desire to restrict imports of raw materials [van Niekerk (1987)].

Ignition-improved ethanol comprising 94,5 % azeotropic ethanol, 4,5 % TEGDN ignition improver, 1 % castor oil for lubricity, and 0,02 % morpholine for inhibiting corrosion has been used successfully in Brazil for several years [Hardenberg and Schaefer (1987)]. Changes to the fuel system included changing the pumping elements in earlier designs to incorporate pressure lubrication of the plungers [Anon (1977)], and the injector nozzles had to be re-set to open at a higher pressure and the hole size increased. Since ethanol burns almost soot-free, there is no lubrication of the valves and therefore the valves and seats had to be replaced by more wear resistant ones.

An advantage of an ethanol-fuelled CI engine is that the performance can be up to 15 % higher than that of the equivalent diesel-fuelled engine because the diesel-fuelled engine would require 30 % excess air for supressing smoke compared with only 10 % excess air for the ethanol fuelled engine [Hardenberg and Schaefer (1987)].

However, if ethanol is blended with other fuels such as diesel, petrol or vegetable oils there is nearly always a loss of power output (of up to 35 %), although when blended with babacu oil there is an increase in power of 1,6 %. Fuel consumption may rise by up to 58 % in the case of a mixture of 33 % ethanol, 33 % castor oil and 33 % diesel [Acioli (1982)].

Methanol may be used in a CI engine using two different technologies. If methanol without ignition improver is to be used some form of ignition aid must be provided, for example, a spark plug as described in 2.5.1 above, or surface-assisted ignition such as a glow plug as proposed for developing a 2-stroke CI engine [Kidd and Kreeb (1984)].

As an altern.tive to neat methanol, ignition-improved methanol has been used in an engine where the modifications were limited to alterations to the fuel injection system including fitting constant pressure valves in the injection pump to maintain a pressure of 100 bar (10 MPa) in the high pressure (HP) fuel pipes. The ignition-improved methanol contained 4,0 % TEGDN, 1 % castor oil and 0,02 % morpholine. The quantity of TEGDN was that which resulted in the ignition-improved methanol having the same ignition delay as that for diesel (Heinrich et al (1986)). Ignition delay was used as the comparator because correlation is difficult using cetane number when rating alcohol fuels with ignition improvers (Schaefer and Hardenberg (1981)).

An ignition-improved methanol known as 'DIESANOL' which is patented and manufactured in South Africa by Chemical Resources (Pty) Limited, an AECI Group Company, has been used successfully in a truck for approximately 60 000 km [Dick (1983)]. Claims include better energy efficiency, 'excellent' engine lubricating oil life and reduced engine wear. Laboratory tests carried out by NMERI on a DIESANOL-fuelled diesel engine in collaboration with Daimler-Benz AG (DBAG) indicated that fuel-related problems such as valve seat recession and cavitation erosion of the HP fuel pipes may be overcome by 'appropriate' technology [Myburgh (1985), Weiss & Hardenberg (1986)]. Tests are continuing at DPT to optimise a lubricating oil for use in DIESANOL-fuelled engines.

Tests to evaluate the use of propanol-plus as a diesel substitute indicated that up to 12 % by volume ignition improver would be required to reduce the ignition delay of propanol-plus to that of diesel [Myburgh (1986 b)]. However, the probability of propanol-plus being marketed is low because of the low volumes produced.

2.5.3 Fumigation

Fuels may be fumigated, that is, vapourised and passed into the intake-air of an engine, in a dual-fuel system where diesel is injected to initiate combustion. A typical installation is shown in Fig 2.3.

In a dual-fuel engine where the secondary fuel was ethanol, tests have shown that the limiting proportion of ethanol was set by knock caused by the ethanol igniting earlier than the diesel at 3/4 and full rack, and the energy substitution of diesel by ethanol was limited to 15 to 30 % because of roughness or knock [Broukhiyan and Lestz (1981)].

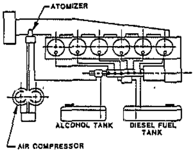


FIGURE 2.3 System for manifold injection of alcohol

Source: EMA (1982)

Tests in which methanol was fumigated into an engine using diesel as the pilot charge indicated that satisfactory performance on the road was possible with three different makes of engine, but piston crown erosion was seen on one engine, possibly caused by detonation of the end gasses, that is, knock [Naeser and Bennett (1980)].

Similar results were obc ned in tests carried out by the Fuel Research Institute of South Africa using petrol, ethanol, and methanol with diesel [Heim and Clark (1977), Clark and Heim (1978), Clark and Heim (1979)].

Thus, knock seems to be a problem encountered with all three fuel combinations cited.

2.5.4 Dual injection

Dual injection permits the use of two fuels in an engine, and may be accomplished by using one injector per fuel or one injector which injects both fuels. In the 'IDIS' system the primary fuel is diesel and is injected in the conventional manner [Fujisawa and Yokota (1981), Kishishita et al (1984)] (Fig 2.4).

The secondary fuel is introduced directly into the HP fuel line through a solenoid valve and a check valve when the pressure in the HP fuel line drops to a partial vacuum caused by the delivery valve in the injection pump retracting. Thus, a second fuel may be used with a minimum of modifications, and experiments have established that up to 20 to 40 % alcohol could be introduced as this secondary fuel.

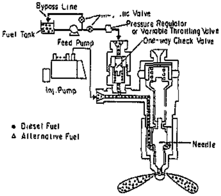


FIGURE 2.4 Schematic diagram of IDIS

Source: Fujisawa and Yokota (1981)

In another system both fuels are injected simultaneously through two separate pathways in the injector [Anon (1985)]. The volume of the secondary fuel varied between 60 % of the primary fuel at low brake mean effective pressures (BMEP) and 25 % at high BMEP. An advantage of the system is claimed to be reduced exhaust smoke.

In a dual-fuel engine using two separate injectors, smoke-free operation could be obtained if the quantity of the secondary fuel was limited. The limit was set at 75 % energy substitution of diesel by, in this case methanol, even though between 82 and 90 % of the energy input from diesel could be substituted [Holmer et al (1980), Pischinger et al (1979)].

2.5.5 Emulsions and diesel-alcohol blends

Alternatives which have been investigated by Letcher for extending diesel include emulsions of diesel with ethanol, methanol and water, dual fuelling diesel and ethanol/methanol, and blends of diesel with a wider diesel fraction, and diesel with ethanol plus a cosolvent (Letcher (1982)). Of these, a blend of 60 % diesel, 28 % ethanol, 8 % ethyl acetate and 4 % octyl nitrate, was considered the most suitable for a field trial using a VW car [Letcher (1983)]. The octyl nitrate was used to boost the cetane number of the blend and did not add to the energy output of the fuel. It enabled the engine to run more smoothly, a phenomenon also seen by Kamel [Kamel (1984)], and although 2 % would have been sufficient 4 % was used to improve cold-starting performance. The vehicle ran well, but material compatability was a problem with PVC items in the fuel system which had to be changed every 10 000 km because they became hard. A similar problem has been observed at NMERI with bowls of the fuel filters manufactured by Racor which have hardened and cracked, and which have now been replaced with alcohol-resitant bowls.

Blends of ethanol and diesel have been tested by the University of Natal in tractors, the effects of which are reductions of power at rated speed of 5 % when using a blend containing 15 % ethanol and 11 % when using a blend containing 30 % ethanol. Power could be restored by altering the fuelling level of the injection pump [Anon (1981)].

Propanol-plus can be added to diesels derived from crude oil or from coal without a blending agent, but the blends are susceptible to water contamination which leads to separation if the contamination is too great [Myburgh (1986 b)]. The effects of fuelling engines with blends of diesel and propanol-plus vary with engine and blend composition. In blends containing up to 40 % propanol-plus the changes in rated power of two engines tested differed, that of an ADE 236 was almost unaltered whilst there was a slight decrease in the rated power of a Deutz F5L 912W. The ADE 236 was subjected to a 300-hour durability test by NMERI using a blend of 80 % coal-derived diesel and 20 % propanol-plus which it successfully completed although there was higher than normal wear in the fuel injection pump which would need further investigation [Myburgh (1986 b)].

SATS has also investigated the use of propanol-plus as a diesel extender [Venter (1983)] in blends containing 10, 20 and 30 % propanol-plus. However, the injection pump needed to be reset to 106 % to restore a loss of power of 6,7 % when using a blend of 80 % diesel and 20 % propanol-plus [Tarboton (1980)].

If a straight blend of diesel and an alcohol such as ethanol is to be used, the blend's stability should be checked, for example, a blend of 185 to 200 proof alcohol and diesel will not blend and remain stable without an emulsifier, and the quantity of emulsifier required is directly proportional to the proof of the alcohol [Alcomotive (undated)]. If the quantity of alcohol exceeds approximately 20 % of the blend, oil must be added for lubricity, for example corn or peanut oil at concentrations up to 6 to 7 % are acceptable, but linseed oil is not suitable.

A solution to the problem of stability may be to mix the diesel with the alcohol just before injection such as in the locomotive which is currently being operated by SATS on a mixture of 75 % diesel and 25 % methanol [Falk (1986 a)]. Many modifications were carried out to the locomotive, including separate fuel tanks with on-board mixing using two fuel metering systems to ensure the correct quantities of methanol and diesel are received by the injection pumps in the correct proportion, and a diesel-only cold start facility.

As with dual-fuelling on diesel and alcohols using two injectors described in 2.5.4 above, the volume of methanol in the blend is limited to 40 % by volume by the onset of knock (Pischinger et al (1979)).

TABLE 2.4 Impact of alternate fuels on diesel engines

Coal synfuel	high aromatics, low cetane (35 to 38 CI)* cold smoke, noisy startability problems very low sulphur minor modifications to engine
Oil shale synfuel	high aromatics low cetane (40 to 45 CI) very low sulphur
Methanol	requires ignition aid difficult to inject larger fuel tank major engine change
Ethanol	as for methanol
Solid fuel	requries totally new fuel system high wear larger tank safety concern in handling

Notes: * CI = Cetane Index

Source: EMA (1979)

Alternatives to traditionally derived diesel fuels exist of which some have been described above, and their impact is summarised in Table 2.4.

2.6 Safety

Blends of diesel, derived from crude oil or coal, mixed with lighter hydrocarbon fractions or alcohols, or pure alcohols, may produce potentially explosive vapours. Flashpoints for different fuels are shown in Table 2.5:

TABLE 2.5 Flashpoints for different flammable liquids

	°C
Diesel containing 20 % propanol-plus	34
Diesel containing 25 % hydro-treated straight run tops	-34 to -6
Proposal for a wide cut fuel	35
Methanol	14
Ethanol	13
Diesel	approx. 70
SABS specification 342:1969	55 min.

Source: Tarboton (1980), Myburgh (1986 b), Lanik et al (1984), SABS (1969)

The risk of a tank exploding may be reduced by filling it with a patented material [Devden (undated)], but the vapours around the filler and vent would still remain hazardous. Some blends may pose handling problems because the vapours become explosive after the lighter fractions have evaporated. Safety precautions taken by SATS include pressurised tanks on which the vents are fitted with flame traps marketed by Link-Hampson, and dry-break fuel connections between the bulk supply and the locomotives' tanks for both the fuel and the vapours so that the vapours can be vented in a remote and safe location.

Some of the alternative fuels and the blends of diesel which have been tested by SATS and NMERI have flashpoints which put them in the same handling classification as petrol. This means that road tankers and railway tank-cars designed for carrying petrol rather than diesel must be used, and bulk storage above ground is prohibited.

Fire detection may pose a problem, for example, methanol burns with a clear flame which makes it very difficult to detect with the naked eye, and hence rai › the alarm that there is a fire [Mueller (1982)]. Even when the fire .s detected, fighting it is difficult because most alcohols absorb water. However, if sufficient water fogging is applied, the alcohol comes out of solution and can then be tackled with foams.

Methanol, propanol and butanol are toxic, whilst ethanol prepared as a fuel should have a denaturant added, for example, one percent petrol to make it toxic and to prevent its use to prepare alcoholic beverages

[Kirik (1984 a), Mueller (1982)]. Methanol is a cummulative toxin irrespective of whether the contamination is by contact on the skin, by inhalation or by swallowing [EMA (1982)]. The effect is damage to the optic nerve which may lead to temporary or permanent blindness (Mueller (1982)].

Care should also be taken when operating engines on, and storing, these alternative fuels since some degrade the properties of seals, tubing, or remove rust from steel containers leading to clogged fuel filters. Research on material compatability has been carried out by the Energy Research Institute, Cape Town [Leng (1980)], and components which are resistant to these alternative fuels may be obtained from the original equipment manufacturer and should be fitted to the fuel systems [Anon (1981)].

2.7 Viability of Alternative Fuels

The following tables indicates how much distillate can be produced from alternative sources and at what overall conversion efficiency. Note should be taken that only a certain proportion may be used as fuel in engines.

Production from non-renewable and renewable resources is given in Table 2.6, the reserves are given in Table 2.7 and the efficiencies of converting raw materials into liquid fuel are given in Table 2.8.

TABLE 2.6 Production of distillate from non-renewable and renewable resources

Production from non-renewable resources:		
Distillate from shale	barrel/t	1,4
coal	barrel/t	2,0
Production from renewable resources:		
Distillate from wood	l/t	500
Ethanol from sugar	l/hectare	3500
Sunflower oil	l/hectare	600

Source: EMA (1979), Kirik (1984 b), Emsley (1987), Bruwer et al (1980).

TABLE 2.7 Reserves and production of distillate

Reserves:		
Shale	billion barrels	900
Crude oil (Middle East)	billion barrels	300
Production:		
Sasol distillate	million barrels per year	52 (estimate)
Wood alcohol (USA)	million barrels per year	2.4 (potential)

Source: Dwyer (1984), Emsley (1987), DMEA (1983)

TABLE 2.8 Efficiency of obtaining alternative fuels

	%
Ethanol from sugar cane	34
wood methanol	27
natural gas	72 to 76
bituminous coal	55 to 65
wood	45 to 57
coal	45
Petrol and distillate from crude oil	87
SASOL 1 (all products)	56
SASOL 2 (all products)	38
NCB-LSE	63

Source: Ricardo (1982), Davies and Thurlow (1984), Holmer et al (1980).

The viability of introducing alternative fuels may be measured by, for example, specific energy consumption (SEC) of diesel compared with the alternatives, or the saving of foreign exchange spent on importing crude oil, or the need to become less dependent on outside influences. Some of the alternatives indicated above may not be economically viable if considered on the basis of SEC in MJ/kWh.

When comparing extended diesel fuels with diesel in terms of energy-efficiency, examples of improvement over diesel include a blend of 85 % diesel and 15 % ethanol which gave 3 to 5 % less power but improved SEC, and a blend of 90 % diesel with 10 % naphtha which gave 4 %

increase in SEC [Hill (undated)]. Even though SEC may be lower when operating on the alternative fuels than when operating on diesel, the volumetric fuel consumption will depend on the volumetric heat of combustion of the alternative fuel and the engine's thermal efficiency which may be different when operating on different fuels.

When comparing ethanol-fuelled SI and CI engines, the energy input for an SI engine is 25 % higher than that of the CI engine fuelled with ignition-improved ethanol. If the increase in the cost of preparing the ignition-improved fuel is less than this 25 %, preference should be given to using CI engines [Hill (undated)].

If ethanol is to be used, the quantity of land required for growing sugar cane for conversion into ethanol must be known. For example, in Brazil where production of distillate is expected to reach 15 billion litres in 1988, only 2 % of all arable land would be needed for the total substitution of crude oil imports [Kirik (1984 b), Rosillo-Calle and Hall (1988)]. A similar figure cannot be obtained for South Africa because the publication of statistics concerning the consumption of liquid fuels is prohibited.

Substituting one refinery process by another to yield more diesel may not substantially alter the efficiency of converting crude oil into fuels, but it would give an opportunity of satisfying a market demand.

Whatever course of action is decided upon and for whatever reason, in the short term fuels must be developed to suit the engines currently available. If any engine modifications are required the cost of these must be kept to a minimum and conversion back to standard must be possible at little or no extra cost. Investigations have already been carried out into different combustion systems to determine what the best compromises are between the development of fuels and engines to achieve the most energy efficient solution to the problem of fuel shortage. The conclusions are conflicting, indirect injection engines are favoured because they achieve lower exhaust emissions [Needham et al (1983)], and direct injection engines are favoured because of better fuel economy [Suda (1984)].

2.8 Evolution of the Test Programme

The failure of the first ADE 236 was attributed to the longer ignition delay and higher volatility of the worst case light diesel which resulted in more intense combustion and consequently higher thermal loads on the piston [Myburgh (1983)].

Therefore this investigation was undertaken to determine ways of reducing the stress so that the engine could operate satisfactorily on light diesel fuels. An indication of the thermal and physical stresses on engine components such as the pistons, piston rings, and bearings, can be obtained from the peak rate of pressure rise within the cylinder and the peak combustion pressure which may be calculated from data collected during combustion. Tests to obtain these data are termed 'combustion analyses' and were used to assist in assessing the results of the durability tests.

Combustion analyses were carried out to compare the peak rate of pressure rise and peak combustion pressure when using light diesel fuels with those of diesel. A test was also carried out to investigate the effect of changes in injection timing on peak rate of pressure rise and peak combustion pressure. This test led to the determination of an injection timing setting where neither the peak rate of pressure rise nor the peak combustion pressure exceeded the levels found when operating on diesel. A durability test was then carried out with the injection timing set at this new setting, and the engine survived [Myburgh and Falk (1985)].

In the following test, the worst case light diesel was used again, but an ignition improver was added to restore the cetane number to 48 which is regarded as the norm within the industry. The standard injection timing was retained, and the engine survived the durability test [Falk (1986 b)]. In practice the quantity of ignition improver which would have to be added to the worst case light diesel depends on its cetane number, and therefore a nomogram would have to be established to correlate the physical properties of light diesel with cetane number. This did not form part of the investigation.

As an alternative to adding ignition improver to the worst case light diesel, a blend of diesel containing less lighter hydrocarbons and which comprised 75 % diesel and 25 % heavy naphtha was used. The engine survived the durability test [Falk (1987)].

Having ascertained that satisfactory durability performance could be achieved on the ADE 236, the performance of two other engines was investigated. These engines were the ADE 314 built by ADE under licence from DBAG, and the Deutz F6L 413F built by Deutz Diesel Power (Pty) Limited (DDP) under licence from Kloeckner Humboldt Deutz AG, West Germany (KHD).

The ADE 314 operated satisfactorily on the worst case light diesel with the injection timing set to standard, and therefore no further tests were undertaken [Falk (1988 a)].

The Deutz F6L 413F with the injection timing set to the standard setting failed after only 97 hours of durablity testing due to piston crown and cylinder head erosion and further tests will be required to determine what steps are necessary to ensure the engine's survival.

The programme described above evolved through the need to obtain a solution which could be implemented at short notice to permit the continued use of engines in an emergency using fuel derived from crude oil. The only consideration was the mechanical survival of engines, and other considerations such as optimising the engines' settings for the fuels used and the effect on exhaust emissions did not form part of the investigation. However, in the longer term other possibilities exist for coping with a shortage of diesel fuel and some of these have been described above.

CHAPTER 3

FUELS USED

A general description is given here of the fuels which were used for the tests. Pertinent details of individual fuels are given in the Chapter 7 of this dissertation which deals with individual tests. Distillation curves of the fuels are shown in Fig 3.1, and a brief list of physical properties of the fuels is given in Table 3.1, whilst more details are given in Appendix A.

3.1 Diesel

The diesel which was used for all the tests was refined from crude oil and generally met the requirements of SABS specification for automotive diesel fuel SABS 342-1969 [SABS (1969)]. It was supplied by BP, and served as base stock for blending with the lighter hydrocarbons and also as a reference fuel for the combustion analyses and performance tests.

3.2 Light hydrocarbons and blends with diesel

3.2.1 Hydro-treated straight run tops (Tops)

Hydro-treated straight run tops (Tops) is a refinery product which is normally processed into solvents or petrol and was supplied by BP. The hydro-treatment is carried out mainly to remove sulphur and not, as in the USA, to saturate aromatics with hydrogen. The very light components in the Tops were included to satisfy fuel storage safety considerations by increasing the Reid Vapour Pressure (RVP) to a level which is above the upper flammability limit [SABS (1976)] when the Tops is blended with diesel.

3.2.2 Tops light diesel (TLD)

Tops light diesel (TLD) was a blend which comprised 75 % diesel and 25 % Topr by volume. The storage and transport requirements of TLD are similar to those for petrol because of the high RVP and low flash point when compared with diesel.

3.2.3 Ignition improved light diesel (IILD)

Ignition improved light diesel (IILD) was TLD to which Hicet 3 ignition improver had been addnd to ensure that the cetane number of the blend is 48,0, which is regarded as an industry norm in South Africa for diesel fuel. The Hicet 3, an iso-octyl nitrate, was manufactured locally by Chemical Resources (Pty) Limited, but has since been superceeded by Hicet 3a which is essentially 2-ethylhexyl nitrate and which is claimed to have similar properties.

3.2.4 Heavy Naphtha

The heavy naphtha was derived from crude oil by National Petroleum Refiners (Pty) Limited (Natref) and supplied by Sasol Fuels Marketing (Pty) Limited.

3.2.5 Naphtha light diesel (NLD)

Naphtha light diesel (NLD) was a blend of 75 % diesel and 25 % heavy naphtha by volume. The quantity of heavy naphtha which could be blended with diesel was limited by refinery production.

TABLE 3.1 Brief list of physical properties of diesel, tops light diesel and naphtha light diesel

	SABS 342:1969	Typical crude oil derived diesel	Typical tops light diesel	Typical naphtha light diesel
Cetane number	45 min.	48	42,1 to 46,0*	40,0 to 45,4
Density @ 20 °C kg/m^3	not specified	849,0	805,0 to 821,2	821,5 to 823,0
Viscosity @ 40 °C cSt	1,6 to 5,3	3,2	1,8 to 2,5	1,9 to 2,4

Notes: * With ignition improver 48,0

Source: SABS (1969), Falk and Myburgh (1987), Falk (1987)

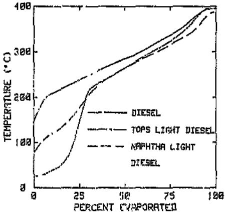


FIGURE 3.1 Distillation curves for diesel, tops light diesel and naphtha light diesel

Source: Falk (1986 b), Falk (1987)

CHAPTER 4

ENGINES

For brevity, the engines will be refered to by their model numbers.

The engines used for the tests were an ADE 236, an ADE 314, a Daimler-Benz (DB) OM 352, and a Deutz F6L 413F manufactured by DDP. Brief specification data for the engines are given in Table 4.1, and full specification data are given in Appendix B. Figures 4.1 to 4.3 show the engines mounted on dynamometers.

TABLE 4.1 Brief technical specifications of the engines tested

	ADE		Daimler-Benz	Deutz
	ADE 236	ADE 314	OM 352	F6L 413F
Swept volume l	3,86	3,784	5,675	9,572
No of cylinders	4	4	6	6
Arrangement	in line	in line	in line	Vee
Cooling	water	water	water	air
Injection pump make/type	Lucas-CAV distributive	Bosch in line	Bosch in line	Bosch in line

Note: all the engines are direct injection

Source: Falk (1987), Falk (1988 a), Falk (1988 b)

4.1 ADE 236

The tests using the ADE 236 were started in 1982 using the 'pre-update' version of the engine because of the large population of this model type, even though the newer design was already in production. Subsequently, all the tests were carried out on the pre-update engines to enable results from successive tests to be compared more easily. The differences in design were mainly in the cylinder head, in which the old design incorporated a cylinder head with 'non-venturi' ports whilst the new design incorporated 'venturi' ports to assist swirl, accomplished by machining the ports after casting. The fuel injection systems were also different: in the older design the fuel supply to the injector was on the side whilst in the later design the fuel supply was from the top.

For the performance and durability tests, the injection was set dynamically at full load and rated speed using reference diesel because one of the characteristics of the Lucas-CAV distributive injection pump is that the injection timing varies according to load, speed and fuel properties.

The cylinder heads were modified by ADE to accept a pressure transducer for measuring the pressure in cylinder no. 4.

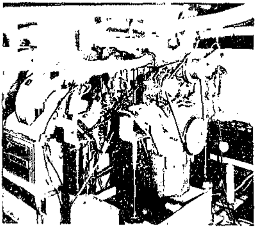


FIGURE 4.1 ADE 236 diesel engine mounted on a Schenck dynamometer

In the original durability test [Myburgh (1983)], two engines were run at the same time on two dynamometers, one fuelled with diesel to serve as reference and the other fuelled with TLD. In subsequent tests only one engine was used for each test except when testing NLD where four engines were used because the first two failed before reaching 70 hours of durability testing using NLD, and the third failed whilst undergoing pre-delivery performance tests at ADE using diesel. The pistons which failed were inspected by the NMERI's Tribology Division, and the conclusion drawn was that the failures resulted from piston scuffing over approximately 1 cm of circumference of the bore, and were not related to operation on NLD. The fourth engine completed the performance and durability tests.

During one of the durability tests the standard aluminium oil filter bowl housing developed hairline cracks leading to a loss of oil, probably caused by the extra mass of the oil cooler which was located between the housing and the filter. No further problems were experienced when the aluminium housings were replaced by cast iron housings.

4.2 ADE 314 and OM 352

The ADE 314 is the four cylinder version of the more popular six cylinder ADE 352. The ADE 314 used for these tests had been used previously for work not connected with this project, and therefore the cylinder head was removed and the valves lapped-in before starting these tests.

A pressure transducer could not be fitted to the ADE 314 to monitor pressure within any of the combustion chambers, and NMERI's own DB OM 352 was used for the combustion analyses. This engine is the same as the ADE 352 and the cylinder head was specially cast by DBAG to accept an adaptor to accomodate a pressure transducer to monitor pressure within No 6 cylinder.

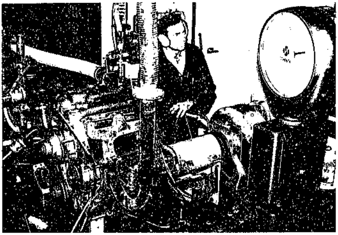


FIGURE 4.2 ADE 314 diesel engine mounted on a Schenck dynamometer

4.3 Deutz F6L 413F

The Deutz F6L 413F is a Vee-six engine in the FL 413F range which includes an in-line 6 cylinder and Vee engines with 6 to 12 cylinders. The rated speed of the Deutz F6L 413F is 2500 r/min compared with 2800 r/min for the ADE 236 and ADE 314 engines. Since the Deutz F6L 413F is air-cooled, a test cell was used which incorporated forced ventilation and ambient temperature control using evaporative cooling.

The engine had been used previously by a transport fleet operator and had been reconditioned, but not used since. To ensure that critical components in the fuel system were new at the start of the test the precaution was taken of fitting new injector nozzles, HP fuel pipes, injection pump elements and injection pump delivery valves. The injector nozzles were fitted in their holders by NMERI and adjusted using the Institute's Hartridge injector tester, whilst the pump was calibrated by the engine rebuilder, but was subsequently adjusted as described in 4.4.3 below.

This engine features individual cylinder heads, and a reconditioned cylinder head was modified to accept a pressure transducer. The drawing for the modification was supplied by DDP and amended to suit an adaptor which was already being used on the DB OM 352. The modified cylinder head was fitted to no 1 cylinder for the combustion analyses. The original head was refitted for the performance tests, but a new head was fitted to the engine for the durability test because signs of erosion were evident on both the original and modified heads, the origin of which had not been positively identified at the beginning of the durability test.

FIGURE 4.3 Deutz F6L 413F diesel engine mounted on a Schenck dynamometer

4.4 General Comments

4.4.1 Fuel temperature

In all instances the fuel returned from the injection pump to the filter was passed througl. a heat exchanger so that stable temperatures in the fuel injection pumps could be maintained. To aid cooling the fuel, the fuei filters on the ADE 314 and Deutz F6L 413F were mounted remote from the engines to reduce the effect of radiated heat.

4.4.2 Injection pump governors

When tests are to be undertaken which involve operation at part throttle and steady speed, such as in this inves[illegible]gation, the injection pumps fitted to the engines must be fitted with variable-speed governors so that [illegible] speed selected will be maintained irrespective of load. All th[illegible]ted complied with this requirement except the Deutz F6L 4[illegible]Q' governor had to be changed for an 'RQV' governor.

4.4.3 Derating for altitude

All the engines were derated for operation on the Highveld of which the altitude ranges from approximately 1220 to 1730 m. Once the injection pumps were set using reference diesel, their settings were not altered for operation on light diesel fuels except as indicated on the Deutz F6L 413F. Power output at rated speed of the Deutz F6L 413F was lower than that recommended by DDP [DDP (1988)], and the injection pump was reset whilst maintaining exhaust gas temperatures at rated speed within the limits set by DDP. When the full load performance test throughout the speed range was carried out, this limit was exceeded at an intermediate speed, and the injection pump was reset to the original setting.

4.4.4 Lubricating oil

BP *Vanellus* C3 SAE 30 lubricating oil was used throughout the investigation. Each engine except the ADE 314 was fitted with an integral oil cooler, with the result that the oil temperatures recorded during the ADE 314 test were similar to those recorded during previous performance tests (Falk (1986 c)), but higher than those subsequently recommended by ADE for continuous operation [ADE (1987)].

CHAPTER 5

INSTRUMENTATION

5.1 Dynamometer Installations

Schenck W 130 and W 150 eddy current and D 360 and D 400 hydraulic dynamometers were used for testing the engines. Electronic throttle position controllers were fitted to the engines for the combustion analyses and performance tests but during durability testing the engines were fitted with pneumatic throttle position controllers which enabled either full throttle or idle to be selected. In the event of an emergency-stop or a failure of the electric power supply or air pressure the throttle lever would return to 'idle' and the stop lever on the ADE 236 to the 'stop' position, both by spring tension. The ADE 314 and Deutz F6L 413F engines were fitted with Bosch fuel injection pumps which did not incorporate separate stop controls. The throttle was controlled as before, and a 24 V solenoid valve was fitted to the fuel supply line ahead of the injection pump to act as a fail-safe emergency stop. By the time the Deutz F6L 413F was tested a system was devised whereby the idle speed was set by a second pneumatic piston which blocked the throttle from returning to the 'stop' position other than on shut-down or in an emergency.

Fuel flow was measured using AVL type 730 gravimetric fuel flow meters manufactured by AVL List GmbH, Austria (AVL).

Exhaust smoke opacity was measured using a Hartridge smoke meter when testing the ADE 236 engines, and a Bosch smoke meter when testing the ADE 314 and Deutz F6L 413F engines.

A general description of the facilities and instrumentation and their operation is given in CSIR Report ME 1751 (Myburgh (1982)).

5.2 Pressure Measurement

Pressure in the rearmost cylinder of each engine was measured using a Kistler 6121 piezo electric pressure transducer mounted in the cylinder head of each engine. The 'dead volume' between the combustion chamber and the face of the transducer was minimised to reduce resonance. The signal obtained from the transducer was amplified using a Kistler 5001 charge amplifier fitted with a 10 kHz filter and with the gain set to 10.

5.3 Injector Needle-Lift Sensor

Injectors modified by NMERI were fitted to the cylinder in which the pressure measurements were made. Each incorporated a linear variable differential transformer inside the injector nozzle so that injector needle displacement could be monitored to determine the start of injection.

The start of injection was determined using methods of calculation termed 'Perkins-Old' and 'Mercedes' [Myburgh (1986 c)]. In the Perkins-Old method the start of injection is the point at which the tangent to the rising curve of needle-lift intersects the baseline and the method was used when testing the ADE 236 engines. In the Mercedes method the start of injection is the point where 13 % of total needle-lift has been reached and the method was used when testing the DB OM 352 and Deutz F6L 413F engines. The methods are shown graphically in Fig 5.1.

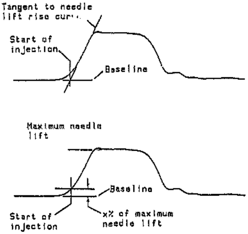


FIGURE 5.1 Methods of determining start of injection, showing Perkins-Old method (upper) and Mercedes method (lower)

Source: Myburgh (1986 c)

5.4 Crank Angle Measurement

Two methods were used to display crank angle accurately, an optical encoder developed by NMERI and one manufactured by AVL.

5.4.1 NMERI optical crank angle encoder

A crank angle encoder using infra-red light passing through holes and serrations in a ring designed by NMERI was fitted to the engine flywheel adjacent to the cylinder in which the measurements were being made, thus minimising the effects of torsional oscillations of the crankshaft.

The sweep on the oscilloscope was triggered from a signal at 35 degrees before top dead centre (°BTDC), whilst signals were obtained to give spikes on the oscilloscope screen every 10 crankshaft degrees (°CA), plus 5 °BTDC, top dead centre (TDC) and 5 degrees after top dead centre (°ATDC). The serrated edge of the encoder ring produced a signal with a rectangular wave which gave half-degree resolution.

5.4.2 AVL optical crank angle encoder

An AVL model 360C/600 optical crank angle encoder was fitted to the crankshaft pulley, and gave the same display of crank angle on the oscilloscope as did the NMERI encoder. Since the encoder was fitted at the end of the crankshaft remote from the cylinder in which the combustion data were taken, checks were carried out to determine if there was an error due to torsional vibration. The error was measured by comparing the display on the oscilloscope of TDC on the flywheel with TDC on the encoder. On the Deutz F6L 413F, which was the first engine exclusively using the AVL encoder, the error was 0,2 °CA at 2500 r/min irrespective of load.

5.4.3 Determination of top dead centre

Top dead centre was determined accurately by turning the crankshaft by hand either side of TDC on the firing cycle until the rearmost piston touched a valve which was blocked open using a thin spacer. The flywheel was lightly marked with a centre punch through the hole in the mounting bracket for the TDC sensor, and the mid-point between the two marks made on the flywheel was taken as TDC and marked heavily with a centre-punch.

Two methods were used to display TDC on the oscilloscope and to align the TDC marker on the crank-angle display to it. In the early tests, a magnetic sensor detected the indentation in the flywheel to give a sinusoidal reference signal for TDC on the oscilloscope, where zero output at the transition from positive to negative output indicated TDC. The position of the crank-angle display was then adjusted by moving the infra-red transmitter/receiver in its mounting until its TDC marker was coincident with the zero output from the magnetic TDC sensor whilst the engine was running at test-speed.

In later tests, the magnetic sensor and amplifier were replaced by a cheaper optical unit built by NMERI and which used infra-red light. Instead of using the centre-punch mark on the flywheel, TDC was indicated using a pin set in the flywheel which passed through the light beam thus producing a square wave output. With the engine stationary at TDC the sensor was moved in its mounting past the pin to produce the square wave 'manually' after which the sensor was clamped. The body of the AVL crankangle encoder could be rotated about its axis, and was clamped when the fall of the square wave output from the AVL encoder at TDC was coincident with the rise of the square wave output from the TDC marker on the flywheel.

5.5 Fast Data Capture System for Combustion Analyses

The signals from the transducers were fed to a Nicolet 4094 four-channel digital storage oscilloscope which could acquire up to 3968 data points per channel at a sampling rate equivalent to a minimum of 0,5 µs per point. Cycle to cycle variation in data gathered was minimised by real time averaging over a number of cycles. When the data had been stored by the oscilloscope it could be transfered to either of two 130 mm floppy discs.

The oscilloscope was controlled by a Hewlett Packard HP 9836 computer which was programmed to perform the necessary calculations after retrieving data stored on disc. The program used for capturing and processing the data was developed by Hodgson as a BIng(Meg) final year project at Pretoria University under the guidance of Myburgh and it has been revised periodically by Myburgh during the period over which the tests were carried out. By the time the last combustion analysis test was carried out, the program had been developed to the stage where the time per point of data gathered was automatically set by the computer to give a minimum screen-width covering from 30 °BTDC to at least 30 °ATDC. Calculations performed on the data gathered included break mean effective pressure (BMEP), accurate engine speed measurement over the period during which the data were gathered, peak rate of pressure rise and the crank-angle at which it occured, peak combustion pressure and the crank-angle at which it occured, start of injection, end of injection, point of combustion and ignition delay.

Ignition delay may be calculated using several methods, for example, in a simulated combustion chamber two methods were used, pressure delay and luminous delay [Siebers (1985)]. Pressure delay was defined as the time taken from injector opening until the pressure in the chamber reached 0,25 atm (25 kPa) above the pressure that would have existed if no fuel had been injected. Luminous delay was defined as the time taken from injector opening until the first luminosity is sensed by a photodiode located outside a window fitted at one end of the cylinder.

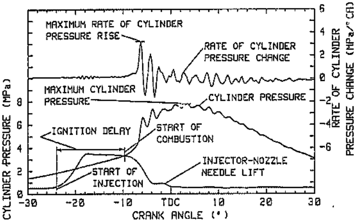


FIGURE 5.2 Explanatory example of combustion parameters calculated by computer

Source: Myburgh (1986 c)

The definition of ignition delay used in this investigation using the program developed by Myburgh was taken as the time from injector opening calculated as indicated in 5.3 above, to the point of ignition. On the ADE 236 and DB OM 352 the point of ignition was calculated as the point of the first strong positive increase in the rate of pressure rise, and is shown in Fig 5.2. However, on the Deutz F6L 413F the transition between compression and ignition was so smooth that the program was amended by Myburgh to redefine the point of ignition as follows. A theoretical pressure-time curve was calculated based on the pressure and volume at two points in the compression cycle, and the point of ignition was taken where the actual curve exceeded the theoretical curve by more than 5 %.

5.6 Computer Facilities

The results of the combustion analysis tests were printed on a Hewlett Packard HP 82905 printer and plotted on an HP 7470A plotter, both of which were coupled to the HP 9836 computer.

An HP 216 computer coupled to an HP 9133 20 Megabyte Winchester / 89 mm disc drive was used for program development, and for calculating the results of performance tests, which were printed on an HP 2934A printer and plotted on an HP 7475A plotter.

By the time the Deutz F6L 413F was tested, a computer program had been written to control the dynamometer and record data during the durability test using a second HP 216 computer coupled to an HP 3054A data acquisition unit.

CHAPTER 6

EXPERIMENTAL PROCEDURE AND DATA PROCESSING

The engines were installed and operated according to the operating conditions which were specified by ADE and DDP, as shown in Table 6.1.

TABLE 6.1 Limits on engine operating conditions

		ADE		Deutz
		ADE 236	ADE 314	F6L 413F #
Intake depression	mm H2O	300	200	750 max.
Exhaust back pressure	mm H2O	1050	600	750 max.
Coolant outlet temperature	'C	93 ± 1	85 ± 5	90 max.
Fuel temperature*	'C	54	30 to 35	no limit
Oil temperature **	'C	As found	115 max.	130
Exhaust gas temperature	'C	700	700	700

Notes:

\# F6L 413F intake depression left as found - 65 mm H2O
exhaust back pressure set to 600 mm H2O
fuel temperature controlled to 30 'C
injection pump reset to give 700 'C exhaust gas temperature
coolant outlet = air outlet on sides of engine

* Temperature of fuel in cambox on ADE 236, injection pump fuel entry on ADE 314 and Deutz F6L 413F

** On ADE 236 oil temperature to be controlled to 104 ± 2 'C, but 'as found' if integral oil cooler fitted as in this case
On ADE 314 this temperature was exceeded - see section 7.5.3

Source: Myburgh (1983), Falk (1988 a), DDP (1988)

6.1 Combustion Analysis

As described in Chapter 2 above, an indication of the stresses on engine components may be obtained from the peak rate of pressure rise and the peak combustion pressure, which in turn may be calculated from data collected during combustion including the pressure within the cylinder, the injection pressure and the lift of the injector needle.

Alterations in injection timing were believed to affect peak rate of pressure rise and peak combustion pressure [Perkins (1984)], and therefore the ADE 236 was tested with the injection timing set to 14 °BTDC, 19 °BTDC, 24 °BTDC (standard) and 27 °BTDC.

The DB OM 352 was tested with the injection timing set to the standard setting of 16 °BTDC static, and the Deutz F6L 413F with the injection timing set to the standard setting of 22 °BTDC static.

The engines were run at their respective rated speeds using various loads between full and no loads.

The combustion data used for calculating peak combustion pressure and peak rate of pressure rise were the averages taken from data collected during 40 consecutive combustion cycles. The number of cycles had been established by Myburgh as the minimum number to minimise the effects of cyclic dispersion, although on SI-engine research 250 cycles captured at random over a 15 minute period [Lyon (1987)] and 300 consecutive firing cycles [Dye (1985)] have been cited.

A transcription of a typical analysis is shown in Table 6.2.

TABLE 6.2 Typical example of calculated combustion data

Engine		Daimler Benz OM 352
Date		30/07/88
Fuel type		tops light diesel
Test number		CA 002
BMEP	kPa	660,06
Peak rate of pressure rise	MPa/°CA	1,639
Occuring t	°CA	0,9
Peak pressure	MPa	7,164
Occuring at	°CA	5,8
Combustion at	°CA	-3,176 *
Injection beginning at	°CA	-15,85
Ignition delay	°CA	12,64
Injection ending	°CA	0,81

Note: * - = °BTDC

Source: Falk (1988 a)

6.2 Perfo mance Tests

Tests were carried out at full and part loads so that comparisons could be made of, for example, power, injection pump fuel delivery, exhaust smoke, volumetric specific fuel consumption, brake thermal efficiency, and exhaust gas temperature for operation on diesel and the light diesel fuels.

Fuel temperature was controlled using a heat exchanger in the fuel return-line from the injection pump to the filter, engine coolant temperature was thermostatically controlled in the case of the watercooled engines, and oil and exhaust gas temperatures were also monitored to ensure that the manufacturers' recommended values were not exceeded. This is important because the tests were carried out at an altitude of 1365 m above sea level and exhaust gas temperatures may be up to 100 °C higher than those recorded at sea level (Falk (1986 c)), and consequently may be above the safe operating limits set by the manufacturer.

For the full load tests, data were gathered starting at rated speed and reducing the speed in 200 r/min steps to 1 000 r/min. However, in the case of the Deutz F6L 413F the first step was from 2 500 r/min (rated speed) to 2 400 r/min, and because the engine was air-cooled the lowest speed was 1200 r/min. At part load, engine speed was held constant at the same speeds that were used for the full load tests, starting with the highest speed, and gathering data from the highest torque at a given speed to the lowest torque in steps of 20 Nm in the case of the ADE 236 and ADE 314, and in steps of 50 Nm in the case of the Deutz F6L 413F.

The reason for starting with the highest speed and highest torque was that temperatures stabilise quicker when the engine speed and torque are reduced, thereby reducing the time taken for carrying out the tests and minimising the quantity of fuel used. However, the time taken for the temperatures to stabilise after each change in load and/or speed on the air-cooled Deutz F6L 413F was approximately 10 min compared with approximately 5 min for the water-cooled engines.

Power and specific fuel consumption were corrected for variations in ambient conditions at altitude using the correction factor derived from the formula [SABS (1982)]:

$$kd = \left(\frac{87,0}{p}\right)^{0,65} \left(\frac{t+273}{298}\right)^{0,5}$$

where kd = correction factor
p = ambient pressure in kPa
t = intake air temperature in °C

Where a comparison of theoretical power is desired the percentage change may be calculated as the change in energy input, which may be expressed as:

$$Dp = \left(\frac{Vf(\text{comparison fuel}) \times Hv(\text{comparison fuel})}{Vf(\text{base fuel}) \times Hv(\text{base fuel})} - 1\right) \times 100$$

where Dp = theoretical change in power output, assuming the same combustion efficiency, %
Vf = volume of fuel consumed in unit time, l/h
Hv = heat of combustion calculated on a volumetric basis, kJ/l

6.2.1 Data processing

Two computer programs were prepared in HP BASIC for processing the data recorded manually during performance tests. The first program was developed by Myburgh to key-in data manually from the test record sheets, perform the necessary calculations and produce a printed output for one test at a time. It has been rewritten during this light diesel investigation to be 'user-friendly' so that technicians with virtually no computer experience could key-in data and obtain printed results from tests, and permits data to be stored on diskette, recalled from diskette, edited if required and re-stored. Only data recorded during the tests and not processed data are stored to reduce the memory space required. The program has been updated periodically to cater for different engine types which demand different tabular presentations, and to automatically select the appropriate correction factor for tests carried out at sea-level or altitude based on the atmospheric pressure keyed-in. Key questions and their implications are shown in Appendix C.

Other programs which retrieve and manipulate data stored on the discs include one for plotting the results of the performance tests, and a program for tabulating the 'average' differences between tests. A general plotting program developed by Hurlin of NMERI was used for presenting the data from the combustion analysis tests.

6.3 Durability Tests

6.3.1 Durability cycle

Several cycles are available for carrying out durability tests, for example, a cycle suggested by the Engine Manufacturers Association (EMA) in the USA for a 200-hour evaluation of alternative fuels [Anon (1982)], a 500-hour test for testing blends of sunflower oil and diesel (Ziejewski and Kaufman (1982)), a cycle recommended by DBAG [DBAG (1986)], and the durability cycle which was recommended by ADE [Rogers (1981)]. The cycle recommended by the Perkins Divisison of ADE was the one used, and is shown in Table 6.3, whilst the other test procedures are shown in Appendix D for comparison. Note should be taken that the 200-hour evaluation test is the only one which specifies a condition for failure, namely, an uncorrectable reduction in power of 5 %.

TABLE 6.3 Durability cycle recommended by ADE

Time per condition	Accumulated time *	Load %	Speed
5 min	5 min	0	low idle
5 min	10 min	50	1600 r/min
3 h#	3 h 10 min	30	as governor curve
5 min	3 h 15 min	0	low idle
4 h#	7 h 15 min	85	as governor curve
5 min	7 h 20 min	0	high idle
4 h	11 h 20 min	100	rated
5 min	11 h 25 min	0	low idle
4 h	15 h 25 min	85	as governor curve
5 min	15 h 30 min	0	high idle
4 h	19 h 30 min	100	rated
5 min	19 h 35 min	0	high idle
4 h	23 h 35 min	100	rated
5 min	23 h 40 min	0	low idle
20 min	24 h		service shutdown

Notes:
* Cycle duration 24 hours
# During the 3 and 4 hours periods, the engine to be cycled for 5.1/2 minutes at the specified load condition followed by 1/2 minute at low idle.

Source: Rogers (1981)

6.3.2 Performance checks

Engine performance was monitored by frequent random checks of torque developed at rated speed, and of piston blowby using a gas flow meter connected to the breather on the crankcase to monitor piston ring and cylinder bore conditions. These random checks enabled a quick assessment to be made of the condition of the engine without stopping the test. In addition to these random checks, full performance tests were carried out at the start and end of the durability test, and also at intervals of 100 hours in the case of the tests carried out on the ADE 236 engines.

6.3.3 50-hourly inspections

Every 50 hours the engine was stopped to carry out a visual examination of the bores and piston crowns using a borescope. At the same time, compression pressures were checked and the conditions of the injector nozzles were checked using a Hartridge injector tester.

In the tests carried out on the ADE 236 engines which were fitted with Lucas-CAV fuel injection equipment, the injectors were checked for opening pressure, 'leak-back time', 'seat leakage' and spray pattern. The test for leak-back time determines the needle-to-bore clearance by measuring the time taken for the pressure to drop from 16,2 to 10,1 MPa. The test for seat leakage was carried out by holding the injector tip against a piece of blotting paper and observing the growth of the stain produced by the fuel whilst maintaining a pressure just below opening pressure. The specifications were opening pressure of 21,0 MPa, leak back time of 6 to 45 s and seat leakage stain of 4,8 mm (maximum) in 1 min.

The performance of the injectors fitted to the ADE 314 and Deutz F6L 413F engines which were fitted with Bosch fuel injection equipment was measured in terms of opening pressure, seat leakage and spray pattern. The method of determining seat leakage was to maintain a pressure of 17,0 MPa and measure the time taken for a drop of fuel to form on the injector nozzle. The specifications set by ADE were 20,0 to 21,0 MPa for the opening pressure of new injectors, 18, [illegible] 1 injectors and the time for a drop to form was to exceed [illegible] specifications set by DDP were 18,0 to 18,8 MPa for the opening pressure of new injectors and 17,5 to 18,3 MPa for used injectors. The specification for seat leakage given by ADE for the ADE 314 was used for the injectors fitted to the Deutz F6L 413F.

A summary of the checks which were carried out and their frequency is shown in Ta [illegible] 6.4.

TABLE 6.4 Summary of checks carried out and frequency

Check	Frequency
Maximum power at rated speed	daily
Exhaust gas temperature at full load and rated speed	daily
Piston blowby at full load and rate speed	daily
Oil consumption rate	daily
Injector nozzle condition	every 50 hours
Lubricating oil analysis, samples drawn	every 50 hours
Compression pressure	every 50 hours
Visual examination of bores and piston crowns	every 50 hours
Injection pump delivery	every 100 hours

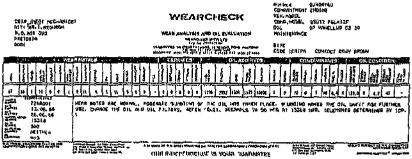
Source: Myburgh (1983)

6.3.4 Oil analyses

Samples of oil were drawn from the engine sump every 50 hours for analysis by Wearcheck, Pinetown, Natal, who specialise in spectrometric oil analysis, and the Institute's Tribology Division. The spectrometric analyses were carried out using a computer controlled Rotrode Emission Spectrometer for the ADE engines, and by the recently introduced Inductively Coupled Plasma (ICP) for the Deutz F6L 413F. Analysis by ICP is claimed by Wearcheck to be more accurate, but the results from the two methods are different, and therefore the trends rather than absolute values should be reviewed to determine the condition of an engine. A typical analysis report is shown in Table 6.5, and gives the accumulation of the metal contaminants in the oil in parts per million (ppm) by mass of iron, chromium, nickel, molybdenum, aluminium, copper, lead, tin, silver, magnesium, calcium, zinc, phosphorus and barium. It also includes the other contaminants silicon (dust), sodium, and boron in ppm and water, fuel and sludge in per cent, together with a description of the oil.

Oil samples taken in-house were analysed using a Duplex Ferrograph and Scanning Electron Microscope (SEM), whilst detailed metal examination was carried out using Energy Dispersive X-ray Analysis (EDP). The amount of iron debris was measured using a particle quantifier.

TABLE 6.5 Example of a Wearcheck oil analysis report



Source: Falk (1988 b)

6.3.5 End-of-test strip-down

At the end of the durability test each engine was stripped and the components inspected. The two ADE 236 engines which were used in the original test [Myburgh (1983)] were stripped and inspected with the assistance of ADE and the fuel injection equipment was sent to Lucas-CAV Limited in England for strip-down and inspection. The fuel injection equipment of subsequent engines were stripped and inspected with the assistance of the NMERI's Tribology Division and DMST who then prepared a report on their findings covering the condi ion not only of the fuel injection equipment, but also the engine components.

6.4 Reporting Results of Tests

An official CSIR report incorporating the tribological report as an appendix was prepared on the completion of each test and submitted to the Foundation for Research Development, now incorporated into the National Energy Council, who sponsored the investigation.

CHAPTER 7

RESULTS

The original test carried out by Myburgh [Myburgh (1983)] using the standard ADE 236 fuelled with TLD did not form part of the current investigation, but was the reason for it, and the results are included here so that they may be refered to more easily. The tests and their results are given in chronological order because the outcome of each test influenced the way in which the investigation evolved.

The results are commented on below only where they deviate from what would normally have been expected. Descriptions are given of the fuels used and the results from the combustion analysis and durability tests. Results from the performance tests are grouped together in 7.6 below, and summarised in Table 7.1. Tables showing the physical properties of the fuels are given in Appendix A, and the results from full load performance tests in Appendix E.

7.1 ADE 236 with Standard Injection Timing and using TLD (the original test) [Myburgh (1983)]

7.1.1 Fuels used

Four batches of diesel were used for the tests of which the cetane numbers of the first two were both 52,0 and the second both had a cetane number of 48,0.

Only one batch of Tops was used for the test and when blended with the diesel produced batches of TLD of which the first batch had a cetane number of 46,0 and the other three had a cetane number of 44,0.

7.1.2 Combustion analysis

At the time this first test was carried out, NMERI did not possess equipment for fast-data-gathering and therefore Polaroid photographs were taken of oscilloscope displays to determine peak combustion pressure and peak rate of pressure rise. The results were analysed according to the method specified by Perkins [Myburgh (1983)], and indicated that the peak combustion pressure and peak rate of pressure rise were 5,5 and 20,0 % higher respectively for operation on TLD than diesel. Subsequent tests were carried out using the fast-data-capturing equipment and the results are shown in Fig 7.1.

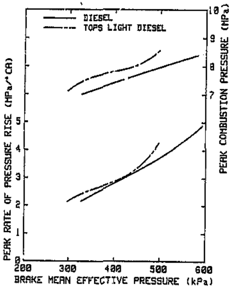


FIGURE 7.1 Peak rate of pressure rise and peak combustion pressure (ADE 236, diesel and tops light diesel)

Source: Falk and Myburgh (1987)

The results shown in Fig 7.1 indicate that at the maximum load common to operation on diesel and TLD, the peak combustion pressure when operating on TLD was 8,6 MPa compared with 8,0 MPa for diesel and the peak rate of pressure rise was 4,2 MPa/°CA for TLD compared with 3,9 MPa/°CA for diesel [Falk and Myburgh (1987)].

7.1.3 Durability

Throughout the test there was no appreciable loss of power or deterioration in cylinder compression pressure of either the engine fuelled with diesel which served as a reference or the engine fuelled with TLD. However, in comparison with the start of the test, the blowby on the TLD-fuelled engine had doubled by 315 hours and doubled again by 340 hours when the test was stopped.

The first signs of what may have been erosion of the pistons were seen as early as 100 hours by the appearance of a matt silvery deposit, thought to have been aluminium, on the cylinder wall above top ring reversal. By 175 hours, the first signs of erosions were seen, and at 340 hours severe erosion was evident on Nos 2 and 3 pistons as shown in Fig 1.2.

The strip-down inspection revealed that the increase in blowby was caused by sticking rings on No 2 piston and the condition of the bores was worse than that on the diesel-fuelled engine.

The pistons of the engine fuelled with diesel showed signs of cracking around the lip of the combustion bowl believed to have been caused by the omission of the chamfer when the pistons were machined. Subsequent engines were fitted with pistons which had a 1 mm deep chamfer machined at 18,5 °, and no further problems were encountered in the tests.

The fuel injection equipment was sent to Lucas-CAV Limited in England for inspection. The conditions of the fuel injection equipment on both engines were similar; there was no breakdown of lubrication although the pump operated on TLD appeared to have suffered more wear. A deposit of sulphur and copper was found on the injector needles, and the presence of sulphur was surprising because of the low sulphur content of the fuels. The origin of the copper may have been the HP fuel pipes which were spirally wound copper-steel [Myburgh (1983)].

7.2 ADE 236 with Retarded Injection Timing and using TLD [Myburgh and Falk (1985)]

7.2.1 Fuels used

Two batches of diesel were delivered. The cetane number of the first was 48,0, and that of the second was 45,7 which is marginally higher than the minimum specified by SABS.

Two batches of TLD were prepared, of which one was used for the combustion analyses and the other for the performance and durability tests. The cetane numbers of the batches were respectively 44,8 and 42,1.

7.2.2 Combustion analysis

The effect of changes in injection timing was investigated to determine if reductions in the peak combustion pressure and peak rate of pressure rise could be obtained as suggested by Perkins [Perkins (1984)]. The results are shown in Fig 7.2 and indicate that when the engine was operated at rated speed and at a load equivalent to a BMEP of 500 kPa, reductions in the peak combustion pressure and peak rate of pressure rise could be achieved when the injection timing was altered between 27 and 14 'BTDC,. The injection timing had to be retarded to 22,8 'BTDC for the peak combustion pressure not to exceed that when operating on diesel, and to 19,6 'BTDC for the peak rate of pressure rise not to exceed that when operating on diesel [Myburgh and Falk (1985)].

These results confirmed the recommendation made by Perkins that the injection timing should be set to 19 'BTDC, and led to the decision to carry out a test with the injection timing set to 19 'BTDC [Perkins (1984)].

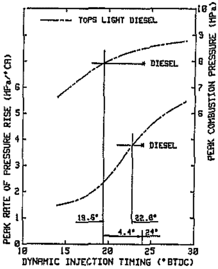


FIGURE 7.2 Variation in peak rate of pressure rise and peak combustion pressure with change in injection timing (ADE 236, tops light diesel)

Source: Falk (1988 c)

7.2.3 Durability test

The test was terminated after 300 hours because no serious deterioration appeared to have occured in the condition of the engine, although the power at rated speed was 4,3 % lower at the end of the test compared with the beginning.

The engine was found to be in relatively good condition when it was stripped down at the end of the test. The piston rings were all free in their grooves and were free from scores.

Inspection of the fuel injection equipment revealed that although the condition of the cam ring of the fuel injection pump was considered to have been satisfactory, there was slight scuffing of the pump plunger-shoe conjunction which indicated that a break-down of lubrication had occured (Fig 7.3). The injector needles again had a deposit which comprised copper, sulphur and zinc.

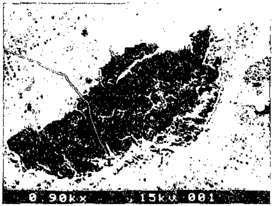


FIGURE 7.3 Plunger shoe conjunction
(ADE 236, retarded injection timing, TLD)

Source: Myburgh and Falk (1985)

7.3 ADE 236 with Standard Injection Timing and using IILD [Falk (1986 b)]

7.3.1 Fuels used

The first batch of diesel used as base stock was the same as the second batch used in the test described in 7.2 above and had a cetane number of 45,7, whilst the second batch had a cetane number of 48,0, which is considered to be the 'Industry Norm'.

The decision to use fuel with a cetane number of 48 led to the preparation of IILD of which two batches were prepared, the first of which required 0,23% by volume Hicet 3 ignition improver and the second 0,16%. The method used to determine the quantity of ignition improver required is shown in Fig 7.4

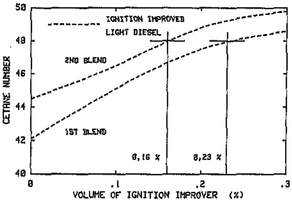


FIGURE 7.4 Cetane number vs volume of ignition improver added

Source: Falk (1986 b)

Cetane number was used as the comparator instead of ignition-delay as recommended by Heinrich et al [Heinrich et al (1986)] because should the need arise to introduce this fuel, cetane number can be checked far more easily than ignition delay. At the time that these tests were carried out the cost of treatment should not have added more the 1 % to the retail price of the fuel.

7.3.2 Durability

The test was stopped after 250 hours because no fuel-related deterioration had occured. There was no evidence of the piston crown erosion that was a feature of the original test.

When the engine was stripped for inspection, the piston rings were free in their grooves.

Inspection of the fuel injection equipment revealed that slight fatigue of the cam ring of the fuel injection pump had occured. The deposit was present on the injector needles again which could have eventually led to the blocking of the injector nozzle holes. Impact fatigue of the line contact seat between the injector body and the needle was more severe than in previous tests, and is shown in Fig 7.5.

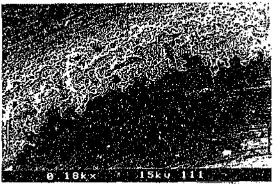


FIGURE 7.5 Impact fatigue we.. on ho . line seat of injector (ADE 236, IILD)

Source: Falk (1986 b)

7.4 ADE 236 with Standard Injection Timing and using NLD [Falk (1987)]

7.4.1 Fuels used

Two batches of diesel were used, of which the batch used for the combustion analyses had a cetane number of 45,7 which is just above the minimum specified in SABS 324-1969 and the batch used for the performance and durability tests had a cetane number of 47,1, but the temperature for 90% by volume recovery was 383 'C, which is above the limit of 362 'C set by SABS. A high temperature for 90% by volume recovery is usually associated with a high concentration of heavier fractions, which consequently shows up as a higher carbon residue, in this instance 0,[illegible] which is above the 0,2% specified. For good combustion all fuel components must be in a gaseous or vapour phase during combustion. Since the temperatures are lower during start-up or at idle, these heavier fractions might not be burnt, resulting in increased exhaust emissions. However, since this test comprised virtually no start-ups and only a very small amount of time spent at idle, the inclusion of these heavier fractions should not have had any deleterious effect on the results of the test.

Two batches of NLD were prepared, and the blend used for the combustion analysis had a cetane number of 40,0, whilst the blend used for the performance and durability tests had a cetane number of 45,4. The difference between the first and second batches of diesel was responsible for the difference in properties of the blend since only one batch of heavy naphtha was used.

7.4.2 Combustion analysis

The results of the combustion analysis test are shown in Fig 7.6. They show that when operating at the maximum common load, the peak combustion pressure was 8,4 MPa for NLD compared with 8,1 MPa when operating on diesel. However, throughout the load range the peak rate of pressure rise was similar for the two fuels, except at the maximum common load where the peak rate of pressure rise for operation on NLD was 3,3 MPa/°CA compared with 3,6 MPa/°CA for diesel. In view of the shapes of the curves, this difference may be considered to be small.

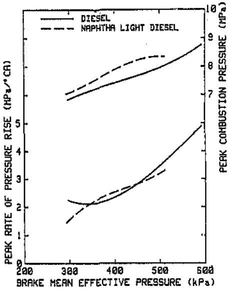


FIGURE 7.6 Peak rate of pressure rise and peak combustion pressure (ADE 236, diesel and naphtha light diesel)

Source: Falk (1987)

7.4.3 Durability

The test was stopped after 250 hours because no fuel-related deterioration had occured. Power at rated speed increased by 1,4 % between the beginning and the end of the test.

The oil consumption rate was steady up to 200 hours, but then increased slightly.

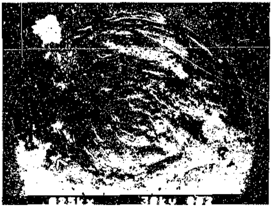


FIGURE 7.7 Adhesive wear on one of the plunger/shoe conjunctions (ADE 236, NLD)

Source: Falk (1987)

The strip-down inspection at the end of the test indicated that the engine had been operating satisfactorily. The deposit which was seen on the injector tips in previous tests was present again, but it would not have affected the operation of the injectors. Wear on one of the plunger/shoe conjunctions in the injection pump was excessive, shown in Fig 7.7 above, the deterioration in the injectors due to impact fatigue, shown in Fig 7.8, and cavitation erosion are cause for concern.

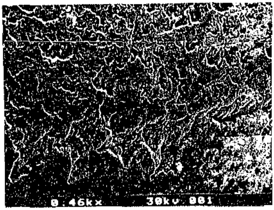


FIGURE 7.8 Impact fatigue of injector needle *(ADE 236, NLD)*

Source: Falk (1987)

The deterioration of the components of the fuel injection equipment preclude *the use of this fuel except in an emergency where* the rates of wear may have to be accepted until the problem has been overcome.

7.5 ADE 314 with Standard Injection Timing and using TLD (Falk (1988 a))

7.5.1 Fuels used

Two batches of diesel were used of which the first batch had a cetane number of 48,0, generally complied with SABS 342-1969, and was used for the combustion analyses. The second batch had a cetane number of 48,5, and was used for performance tests and as base stock for use in the durability tests. The distillation temperature for 90% volume recovery of the second batch of diesel was higher than that specified [SABS (1969)], but despite this, the gross heats of combustion, densities and cetane numbers of the two batches of diesel were almost identical.

Two batches of TLD were prepared in which two batches of diesel and two batches of Tops were used. The cetane numbers were 44,8 and 42,6 respectively.

7.5.2 Combustion analysis

The results are shown in Fig 7.9.

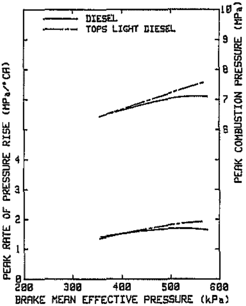


FIGURE 7.9 Peak rate of pressure rise and peak combustion pressure (DB OM 352, diesel and tops light diesel)

Source: Falk (1988 a)

The highest common load for operation on diesel and TLD was equivalent to a BMEP of 570 kPa. At this load, the peak combustion pressure

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The results are shown in Fig 7.9.

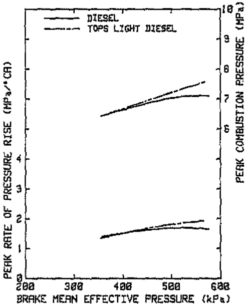

FIGURE 7.9 Peak rate of pressure rise and peak combustion pressure (DB CM 352, diesel and tops light diesel)

Source: Falk (1988 a)

The highest common load for operation on diesel and TLD was equivalent to a BMEP of 570 kPa. At this load, the peak combustion pressure whilst operating on TLD was 7,5 MPa compared with 7,0 MPa when operating on diesel, and the peak rate of pressure rise was 1,9 MPa/°CA compared with 1,6 MPa/°CA. The results are shown in Fig 7.9, and reveal that the peak rates of pressure rise were substantially lower than those seen on the ADE 236 engines (compare Fig 7.1 with Fig 7.9).

7.5.3 Durability

The test was stopped after 300 hours because no fuel-related deterioration had occured. However, the rate of oil consumption was 43,5 % higher than that seen on the ADE 236 engines.

Inspection of the engine at the end of the test revealed that the crankshaft had a yellow colour, and the big end bearings had suffered from cavitation erosion. Both NMERI's Tribology Division and DBAG (DBAG (1987)) concluded that the probable cause was too high an oil temperature which ranged from 122 to 134 °C. In subsequent discussions with ADE the recommendation was made that the oil temperature should not have exceeded 115 °C for continuous engine operation [ADE (1987)].

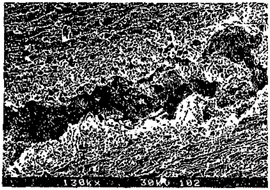

FIGURE 7.10 Cavitation erosion of the HP fuel pipes (ADE 314, TLD)

Source: Falk (1988 a)

Cavitation erosion shown in Fig 7.10 was found to have occured on the inside of the HP fuel pipes, but there was no deposit on the injector needles.

The lack of deposit may have been due to the engine being fitted with solid steel HP pipes compared with spirally wound copper-steel HP pipes fitted to the ADE 236, or due to changing the fuel pipes connecting the outside bulk storage drums to the day-run-tank from copper to flexible alcohol resistant hose.

7.6 Deutz F6L 413F with Standard Injection Timing and using TLD (Falk (1988 b))

7.6.1 Fuels used

One batch of diesel which had a cetane number of 47,1, and one batch of Tops were used for the test. The TLD blended had a cetane number of 42,9.

7.6.2 Combustion analysis

The peak combustion pressure and peak rate of pressure rise for operation on diesel and TLD were lower than those for the ADE 236 and DB OM 352 (compare Figs 7.1 and 7.9 with Fig 7.11). At the highest common load of 520 kPa, the peak combustion pressure was 6,2 MPa for TLD compared with 6,6 MPa for diesel, and the peak rate of pressure rise was 0,9 MPa/°CA compared with 0,75 MPa/°CA for diesel. However, the shapes of the curves of peak combustion pressure and peak rate of pressure rise both show maxima at a BMEP of 400 kPa compared with diesel which falls slightly as the load is reduced.

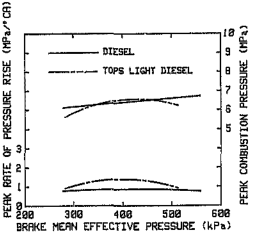

FIGURE 7.11 Peak rate of pressure rise and peak combustion pressure (Deutz F6L 413F, diesel and tops light diesel)

Source: Falk (1988 b)

7.6.3 Durability

The first signs of erosion were seen on the piston crowns after only 50 hours of durability testing, and the test was stopped after 97 hours.

Measured torque fell by 10 % between the beginning and end of the test, engine blowby increased by 28 %, but the rate of oil consumption was steady throughout the test during which 16,38 l oil had been used.

Only the cylinder heads, barrels, pistons and fuel injection equipment were removed from the engine for inspection because of the short duration of the test. Erosion was seen on all the piston crowns, an example of which is shown in Fig 7.12, and also on all the cylinder heads (Fig 7.13). Some of the inserts between the inlet and exhaust valves had also been distorted.

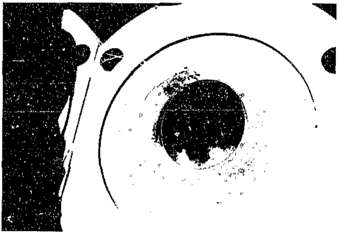

FIGURE 7.12 Erosion of piston crown No 6 after 97 hours (Deutz F6L 413F, TLD)

Source: Falk (1988 b)

FIGURE 7.13 Erosion of cylinder head No 6 after 97 hours (Deutz F6L 413F, TLD)

Source: Falk (1988 b)

The fuel injection equipment was in good condition which was expected because it had been subjected to less than 100 hours instead of the more 'normal' 250 to 300 hours of durability testing.

7.7 Performance

The performance of the engines on light diesel fuels compared with diesel may be summarised as shown in Table 7.1. This table was prepared from the data contained in the Tables and Figures in Appendix E.

TABLE 7.1 Summary of engine performance when operating on light diesel fuels compared with diesel

	Engine model					
	ADE 236				ADE 314	Deutz F6L 413F
	TLD	TLD retarded injection timing	IILD	NLD	TLD	TLD
Change in power at rated speed %	-4,8*	-6,7	-6,3	-2,1	-4,0	-7,0
Change in maximum exhaust temperature °C	-69	-62	-61	-31	-31	-60
Change in maximum smoke **	-34	-37	-30	-26	-0,6	-1,6
Change in thermal efficency *** %	-1,8 to 3,6	-1,6 to 4,0	-1,0 to 2,8	5,2 to 8,6	-0,9 to 3,4	0,4 to 3,4
Change in volumetric sfc *** %	1,8# to 3,8	-0,8 to 7,2	0,5 to 7,1	-2,2 to 1,1	3,3 to 8,5	1,2 to 4,6

Notes:

* - = lower value
** Exhaust smoke measured in Hartridge Smoke Units (HSU) for ADE 236 and Bosch Smoke Units (Bosch) for ADE 314 and Deutz F6L 413F
*** Figures are arithmetic averages of different loads at steady speeds
\# Part load investigated at 3 speeds only

The table shows that in all instances operation on light diesel fuels led to a reduction in rated power. There was also a reduction in full load power throughout the speed range which led to reduced exhaust gas temperatures and reduced smoke emission due to the engines operating with slightly greater excess air than when operating on diesel.

Changes in thermal efficiency and volumetric specific fuel consumption given in the table are the highest and lowest arithmetic averages at set speeds within the speed range over which part load performance was tested. This method of presenting data only gives an indication of the the changes attributable to the use of light diesel fuels. A better method of obtaining and presenting this type of data may be found in the report of field trials carried out by Natal University [Lyne (1986)]. In the report 3-D presentation has been used to determine the proportion of time an engine is operated at a given load and speed combination so that a more realistic assessment may be made of, for example, changes in overall fuel consumption.

The figures of engine performance at full load shown in Appendix E show that the quantities of fuel injected by both the Lucas-CAV distributive pump and Bosch in-line pumps were affected by the change in physical properties of the light diesel fuels compared with diesel. In all cases for the same throttle settings the volume of fuel delivered was lower when operating on light diesel fuels, leading to lower power output.

An example of the difference in performance between the use of diesel and light diesel fuels may be seen in the ADE 314 where full load power was consistently lower throughout the speed range when operating on TLD, with a sharp drop-off below 2000 r/min. This characteristic may have been caused by the increased compressibility of the D compared with diesel resulting in a reduction of the quantity of fuel injected, and hence reduced power output [ADE (1988 a)].

Several results were seen only on the Deutz F6L 413F and these are commented on here. Operation on TLD led to maximum power being developed at 2400 r/min instead of at the rated speed of 2500 r/min, although the change in power given in Table 7.1 is the percentage change at 2500 r/min. Exhaust gas temperatures and smoke measured on the left-bank were higher than those measured on the right-bank throughout the speed range except for 1400 and 1600 r/min when operating on diesel, and higher throughout the whole speed range when operating on TLD. This phenomenon has also been seen in similar tests carried out by a vehicle developer.

On several occasions difficulty was experienced in restarting the engine when operating on TLD. A brief test showed that when the engine had been operated at 2200 r/min and full load and then had been shut down, the temperature of the fuel at the injection pump inlet rose to 58 °C after as little as 15 minutes. Restarting was impossible due to vapour lock.

CHAPTER 8

DISCUSSION

This series of tests was prompted by the failure of an ADE 236 to operate satisfactorily on a worst case light diesel due to erosion of the piston crown. In the first test the pistons of the engine fuelled with diesel, which served as reference, also showed signs of cracking around the lip of the combustion bowl which was sharp-edged, possibly due to the omission of the chamfer when the pistons were machined.

Problems of cracking, or in advanced cases, erosion, of the piston crown in the region of the combustion bowl may be caused by a combination of thermal and mechanical stresses [Wacker and Coelingh (undated)]. Thermal fatigue cracks occur at the edge of the combustion bowl because of excessive temperature and temperature gradients such as the cyclic thermal stresses found during the combustion cycle.

Mechanical fatigue cracks occur as a result of high firing pressures and/or high rates of pressure rise. Rates of pressure rise are increased under transient accelerating conditions when maximum fuelling is introduced into a cool combustion chamber when full throttle is selected after idle, namely the conditions prevailing in the durability cycle used in these tests [Perkins (1984)].

Open combustion bowls with rounded edges are less susceptible to problems of this nature than re-entrant bowls with sharp edges. Unfortunately, sharp-edged combustion bowls aid combustion by promoting swirl leading to better fuel mixing [ADE (1986 b)]. The method used by ADE to resolve the problem on the ADE 236 was to machine a 1 mm deep chamfer on the lip of the combustion bowl at 18,5 ° and to provide a radius where the chamfer meets the bowl. Subsequent engines tested had pistons fitted with the chamfer, and no further piston cracking occured Other more expensive methods of overcoming thermal fatigue cracking include hard anodising the surface of the piston with a layer

40 to 70 μ thick which improves the situation by a factor 3 to 5, changing the combustion chamber to the Perkins 'Quadram' squared shapo which is incorporated in a new range of engines which Perkins claim reduces the i nition delay by 10 °CA and peak pressure by 10 %, changing to a combustion bowl which is shallower, or to introduce internal piston cooling [Wacker and Schoekle (1979), Scott (1986)].

In an emergency these costly and long term options are not available, since the aim is to provide an almost instantaneous way of continuing to operate diesel engines with a minimum of fuss.

The investigation has shown that the peak combustion pressures for TLD and NLD were similar, but there was a difference in the peak rates of pressure rise, especially when comparing the results from TLD and NLD as shown in Fig 8.1.

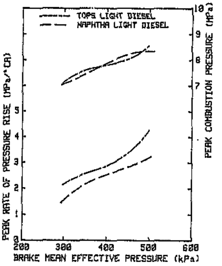

FIGURE 8.1 Peak rate of pressure rise and peak combustion pressure for TLD and NLD on the ADE 236

Source: Falk (1988 c)

The peak rate of pressure rise for TLD was higher than that for NLD and diesel, especially at the highest common load attainable on all fuels. The failure of the ADE 236 may therefore be more dependent on peak rates of pressure rise than on peak combustion pressures, and especially under transient accelerating conditions as suggested by Perkins (1984). Durability tests carried out on the ADE 236 established that the engine could survive between 250 and 300 hours of testing without the recurrence of the piston crown erosion that was a feature of the frst test. On this basis each test was classified as a 'pass', although increased wear of components in the injections pumps remain cause for concern. These components include the injector nozzle needles/seats (Fig 7.6), and adhesive wear on one of the plungers (Fig 7.8) when operating on NLD which precludes the use of this fuel except in an emergency. Tests to determine the load carrying capacity of the light diesel fuels have been carried out by NMERI's Tribology Division [Luszczewski (1986)], but the four-ball test method does not appear to take into account the operating temperature of the fuel, and therefore the value of the results is questioned.

The peak combustion pressure and peak rate of pressure rise for the ADE 314 as tested in the DB OM 352 were lower than those for the ADE 236 (compare Fig 7.10 with Fig 8.1). The design of the ADE 314 already incorporates a radiused combustion bowl lip, which together with the results from the combustion analysis are believed to be the reason for this engine's survival. However, whilst the fuel injection equipment was generally in better condition on this engine than on the ADE 236, cavitation erosion had occured in the HP fuel lines.

The damage to the pistons of the Deutz F6L 413F occured on the sharp edged section of the lip in the direction of swirl at a location between the injector and the exhaust valve (Figs 7.13 and 7.14). In the first ADE 236 test the failure was also on a portion of piston crown located close to the exhaust valve. The failure of the air-cooled Deutz F6L 413F might be ascribed to the longer time taken to reach stable temperatures compared with water-cooled engines thus extending the period of high transient rates of pressure rise. However, KHD are unwilling to ascribe a cause of failure because the engine was rebuilt and suffered from high oil consumption (Falk (1988 d)).

Power at full load throughout the speed range on all the engines was reduced as a result of operating on light diesel fuels (see Appendix E). The cause may be explained as follows. The lower viscosities of the light diesel fuels resulted in a reduction in transfer pressure in the Lucas-CAV pumps fitted to the ADE 236 engines thus leading to a reduction in the quantity of fuel delivered. The higher compressibility of the light diesel fuels also led to a reduction in the quantity of fuel delivered as indicated in Chapter 7.7. The quantity of input energy was further reduced because of the lower volumetric heat of combustion of the light diesel fuels compared with diesel.

The results of the durablity and performance tests may be summarised as shown in Table 8.1:

TABLE 8.1 Summary of durability test results, and performance compared with operation on diesel

Engine	Injection timing	Fuel	Result	Rated Power	Expected Volumetric Fuel Consumption
ADE 236	Standard	TLD	fail	lower	higher
ADE 236	Retarded	TLD	pass	lower	higher
ADE 236	Standard	IILD	pass	lower	higher
ADE 236	Standard	NLD	pass *	lower	higher
ADE 314	Standard	TLD	pass	lower	higher
Deutz F6L 413F	Standard	TLD	fail	lower	higher

Note: * Wear in injection equipment unacceptable except in emergency

These laboratory tests were not designed to evaluate problems that may only occur when operating engines in vehicles. According to Grigg et al (1986) hot re-start problems can be expected if the temperature of the fuel in the injection pump exceeds 55 °C when using 2,0 cSt fuels, such as the light diesel fuels used in these tests. The equivalent temperature for operation on diesel derived from crude oil which has a viscosity of approximately 3,0 cSt is 70 °C. Hot re-start problems have occured on the Highveld where diesel produced from coal is marketed and which has a lower viscosity than diesel produced from crude oil. The situation has been resolved by the fuel producer ensuring that the viscosity of the diesel is at least 2,2 cSt which is well above the SABS minimum [SABS (1969)].

During these tests hot re-start problems were only experienced on the air-cooled Deutz engine. The cause was not believed to have been related to viscosity but to vapour lock, established by monitoring the temperature of the fuel at the injection pump inlet on shut-down. The possibility exists that the Vee range of any engine whether water or air cooled and where the injection pump is located in the Vee may be susceptible to vapour lock because of heat-soak from the crankcase.

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CHAPTER 9

CONCLUSIONS

Following combustion analyses, performance, and durability tests which were carried out on the ADE 236, ADE 314 and Deutz F6L 413F diesel engines, the following conclusions may be drawn:

1. In standard form the ADE 236 did not survive a durability test when fuelled with a worst case light diesel fuel due to erosion of the piston crowns.

2. The ADE 236 can achieve satisfactory durability performance if:
   - the injection timing is retarded by 5 °CA and the worst case light diesel fuel is used,
   - or the standard injection timing is retained and the engine is fuelled either with the worst case light diesel to which ignition improver has been added or a blend containing fewer light hydrocarbons, namely the blend which contained 25 % heavy naphtha.

3. The ADE 314 survived the durability test in standard form when fuelled with the worst case light diesel.

4. The Deutz F6L 413F did not survive the durability test in standard form when fuelled with worst case light diesel due to erosion of the piston crowns and cylinder heads.

5. Wear in the fuel injection equipment of the ADE engines was more severe when operating on the light diesel fuels than on diesel and this precludes the use of the blend containing 25 % heavy naphtha except in an emergency. The test carried out on the Deutz F6L 413F was too short to comment on the effect of the fuel on the fuel injection equipment.

6. Operation on light diesel fuels led to a reduction of up to 7,0 % in full load power. Full load exhaust temperatures and smoke were lower as a result of the lower power outputs throughout the speed range. Volumetric fuel consumption is expected to increase by up to 8,5 %.

7. Hot re-start problems occured due to vapour lock on the Deutz F6L 413F Vee engine, and this problem may also occur on other Vee engines.

CHAPTER 10

RECOMMENDATIONS

1. Further work should be carried out to determine what steps are necessary to reduce the wear in the fuel injection equipment when operating on light diesel fuels.

2. Further tests on the Deutz F6L 413F are planned using a new engine instead of a rebuilt one. A repeat of the test using the worst case light diesel is proposed followed by a test using the worst case light diesel to which ignition improver is added to determine if the engine would survive the durability tests. A test with the injection timing retarded and using the worst case light diesel fuel is not recommended because of the conditions under which trucks fitted with Deutz F6L 413F and F10L 413F engines operate.

APPENDIX A PHYSICAL PROPERTIES OF THE FUELS TESTED COMPARED WITH THE REQUIREMENTS OF SABS 342:1969

	SABS 342:1969	Typical crude oil derived diesel	Typical tops light diesel	Typical naphtha light diesel
Flash point 'C	55 min.	57	-34 to -5	-4
Viscosity @ 40 'C cSt	1,6 to 5,3	3,2	1,8 to 2,5	1,9 to 2,4
Cetane number *	45 min.	48	41,0 to 46,0	40,0 to 45,4
Cold filter plugging point 'C	**	-8 to -1	-9 to -7	-7
Boiling range 'C	-	160 to 388	39 to 392	59 to >400
Distillation temp 'C for 90 % recovery	362 max.	353	364	356 to 366
Density @ 20 'C kg/m^3	-	849.0	805,0 to 821,2	821,5 to 823,0
Heat value (kJ/kg)	-	45400	45640 to 46050	43550 to 45660
(kJ/l)	-	38545	36760 to 37280	35840 to 38120
Ash content % mass	0,01 max.	0	0	0,002
Water content % vol	0,05 max.	0,05	0	0,05
Sulphur cont. % mass	0,55 max.	0,42	0,45	0,43
Sediment cont. % mass	0,01 max.	0,01	0	0,005
Carbon residue on 10 % (vol) distillation residue % mass	0,2 max.	0,06	0,11	0,13
Copper strip corrosion (3 h @ 100 'C)	1 max.	1	1	1

Notes: * Tops light diesel with ignition improver 48.0
** Cold filter plugging point -
Winter -4 'C
Transition 0 'C from 15 April to 14 May, 1 to 30 September
Summer 3 'C

Source: Myburgh (1986 c), Falk (1987)

APPENDIX B FULL TECHNICAL SPECIFICATIONS OF THE ENGINES TESTED

Make and model		ADE 236	ADE 314	Daimler-Benz OM 352	Deutz F6L 413F
Type of combustion chamber		open	open	open	open
Injection (direct/indirect)		direct	direct	direct	direct
Bore	mm	98,4	97,0	97,0	120,0
Stroke	mm	127,0	128,0	128,0	125,0
Swept volume	l	3,86	3,784	5,675	9,572
No of cylinders		4	4	6	6
Arrangement		in line	in line	in line	Vee
Compression ratio	:1	16	16	17	17
Cycle		4	4	4	4
Cooling		water	water	water	air
Aspiration		nat asp*	nat asp	nat asp	nat asp
Injection pump	make	Lucas-CAV	Bosch	Bosch	Bosch
	model	DPA	A	A	A
	type	distrib#	in line	in line	in line
	model no	3249F532	PES4A 90D410 RS2570	PES6A 90D410 RS2293	PES6A 95D410 LS2450
Injector nozzle	type	Perkins multihole	Bosch multihole	Bosch multihole	Bosch multihole
	number	2645655	DLLA 142 S 791	KDAL 7453/19	DLLA 28 S 656
Injection timing °BTDC		24 @ 2800 r/min	16 ± 1 static	16 static	22 static

Notes: * nat asp = naturally aspirated
distrib = distributive (ie rotary)

APPENDIX C 'ENGTEST' PROGRAM FOR CALCULATING AND TABULATING DATA FROM PERFORMANCE TESTS

Key questions and implications

Question	Answer	Implication
Diskette size	large / small	semi-automatic address selection for file of information irrespective of computer used
Input type	manual automatic from file mass print	manual input of data automatic input of data - not ready yet retrieve information from file mass print of up to 50 test results on one diskette
Engine type	compression ignition spark ignition	selection of questions and table format, correction factors, questions for data input
In-line or Vee		table format and questions for data input
Water / air cooled		table format and questions for data input
Naturally aspirated / turbocharged/ turbocharged intercooled		table format, correction factors (turbocharged = 1), questions for data input
2-stroke / 4-stroke		calculation of BMEP

Test title sheet input:
date, test number*, engine make/model, displacement*, number of cylinders*, fuel, density*, heat value (gross or nett)*, atmospheric pressure*, comments, (items marked * must be entered for program to continue)

Print style	normal / compressed	certain tables only compressed
Smokemeter	Bosch / Hartridge	table format
Fuel density	pump temperature / 20 'C	conversion from sfc mass to sfc vol based on fuel temperature at pump inlet or 20 'C. If pump temperature then decrease density by 0,68 kg/'C temperature rise above 20 'C

Question	Answer	Implication
Turbo boost pressure or manifold depression	mm Hg / mm H2O	table format, conversion to kPa
Print-out	yes / no	continue input with/without printout of most recently entered test
Sfc	mass / volume	table format, conversion mass / vol
On Mass Print soft key options:		
Full print out		data input, calculated input, calculated output
Results only		calculated input and calculated output
Data in only		data input
Sfc by mass		sfc by mass or volume - table format
Sfc by volume		
Print normal		print 12 characters per inch
Print narrow		print standard compressed
Density calculation		pump temperature or 20 'C

Automatic features:

Compressed print style for certain tables

Barometer input whether mm Hg or kPa will automatically be printed as kPa. The correction factor for altitude or sea level according to SABS 013 is based on the barometric pressure calculated in kPa. The comments section of table format states which correction factor has been applied, namely, SABS 013 Part I for tests at sea level or Part II for tests at altitude.

Correct selection of correction factor based on input of engine type (compression ignition/spark ignition, naturally aspirated/turbocharged)

Error trapping on manual input if data are not within 'reasonable limits'

TABLE D1 Proposed light diesel project using ADE OM 314 engine - Diesel fuel testing engine based on the OM 366 engine.

Durability cycle 100 hours

Stage	Time: Per stage min	Time: Cummulative min	Engine speed r/min	Load % of full load
1	60	60	2800	100
2	30	90	1800	100
3	30	120	1000	100
4	30	150	max. 2950 (high idle)	0
Repeat stages 1 - 4 up to 50 h total time 3000				
5	3000	6000	2800	100

DBAG do not recommend removing injectors every 50 hours for inspections because of the danger of foreign matter getting into the bores.

Source: DBAG (1986)

TABEL D2 200 hour screening test for alternate fuels

Fail if power drops by 5 % and cannot be corrected

Stage	Speed %	Torque %	Power %	Duration min
1	rated	-	rated	60
2	85	max.	approx. 95	60
3	90	28	25	30
4	low idle	0	0	30
Total time per cycle				180

Repeat 5 cycles, then shut down for 9 hours.

R ver 200 hours.

Source: Anon (1982)

TABLE D3 Endurance test of a sunflower oil/diesel fuel blend

Stage	Speed	Load	Time min
1	high idle	0	3
2	peak torque	peak torque	10

Repeat continuously for 500 hours.

Source: Ziejewski and Kaufman (1982)

APPENDIX E FULL LOAD PERFORMANCE DATA

Full load performance curves and tabulated data from which the curves were derived for:

ADE 236, standard injection timing, tops light diesel
Source: Myburgh ([illegible]), Falk (1988 c)

ADE 236, retarded injection timing, tops light diesel
Source: Falk and Myburgh (1985), Falk (1988 c)

ADE 236, standard injection timing, ignition improved light diesel
Source: Falk (1986 b), Falk (1988 c)

ADE 236, standard injection timing, naphtha light diesel
Source: Falk (1987), Falk (1988 c)

ADE 314, standard injection timing, tops light diesel
Source: Falk (1988 a)

Deutz F6L 413F, standard injection timing, tops light diesel
Source: Falk (1988 b)

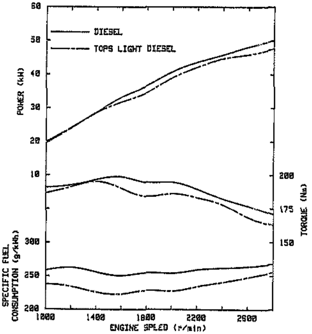

FIGURE E1 Full load power, torque and specific fuel consumption (ADE 236, standard injection timing and using TLD)

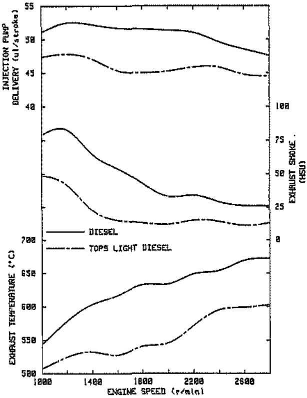

FIGURE E2 Full load injection pump delivery, exhaust smoke and exhaust temperature (ADE 236, standard injection timing and using TLD)

TABLE E1 Full load performance data (ADE 236 standard injection timing and using diesel)

DATE: 82/02/04

TEST NUMBER: PC359

ENGINE: ADE 236-2

DISPLACEMENT: 3.86 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

FUEL: 100% BP COASTAL DIESEL

DENSITY @ 20 'C: 841.0 kg/m3
HEAT VALUE (GROSS): 44260 kJ/kg

ATMOSPHERIC PRESSURE: 86.80 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD POWER DATA
INJECTION TIMING SET AT 24' BTDC
54'C INJECTION PUMP CAMBOX TEMPERATURE
WATER-IN TEMPERATURE NOT MEASURED

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	'C	'C	'C	'C	'C	'C	l/h	ul/str	
1000	93	0	98	39	51	544	6.13	51.08	1.025
1200	93	0	99	39	51	579	7.56	52.49	1.025
1400	93	0	101	39	51	604	8.70	52.24	1.025
1600	93	0	101	39	51	617	9.92	51.69	1.025
1800	93	0	103	39	52	633	11.13	51.53	1.025
2000	92	0	104	39	53	634	12.32	51.33	1.025
2200	93	0	105	39	55	649	13.47	51.01	1.025
2400	91	0	106	39	57	654	14.29	49.62	1.025
2600	93	0	106	38	57	669	15.14	48.53	1.023
2800	89	0	103	38	54	672	15.90	47.57	1.023

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	MEASURED POWER	MEASURED SFC	MEASURED BMEP	CORRECTED TORQUE	CORRECTED POWER	CORRECTED SFC	CORRECTED BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	BSU
1000	186.0	19.5	265	606	190.6	20.0	258	621	30.7	79
1200	188.0	23.6	269	612	192.7	24.2	263	627	30.2	83
1400	192.0	28.1	262	625	196.8	28.8	256	641	31.0	65
1600	194.0	32.5	257	632	198.8	33.3	251	647	31.7	54
1800	190.0	35.8	261	619	194.7	36.7	255	634	31.1	43
2000	190.0	39.8	260	619	194.7	40.8	254	634	31.2	33
2200	185.0	42.6	266	602	189.6	43.7	259	617	30.6	34
2400	178.0	44.7	269	580	182.4	45.8	262	594	30.3	29
2600	173.0	47.1	270	563	177.0	48.2	264	576	30.1	26
2800	167.0	49.0	274	544	170.9	50.1	268	556	29.6	26

TABLE E2 Full load performance data (ADE 236 standard injection timing and using TLD)

DATE: 82/02/12 TEST NUMBER: PC363

ENGINE: ADE 236-2

DISPLACEMENT: 3.86 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

FUEL: 75% BP COASTAL DIESEL
25% HYDROTREATED STRAIGHT-RUN TOPS

DENSITY @ 20 °C: 816.5 kg/m3
HEAT VALUE (GROSS): 45710 kJ/kg

ATMOSPHERIC PRESSURE: 87.20 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD POWER DATA
INJECTION TIMING SET AT 24° BTDC
54°C INJECTION PUMP CAMBOX TEMPERATURE
WATER-IN TEMPERATURE NOT MEASURED

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	°C	°C	°C	°C	°C	°C	l/h	ul/str	
1000	91	0	96	34	54	507	5.69	47.40	1.013
1200	92	0	98	33	54	525	6.89	47.06	1.012
1400	93	0	99	33	54	533	7.92	47.12	1.012
1600	95	0	101	33	54	528	8.71	45.37	1.012
1800	92	0	100	33	54	541	9.73	45.03	1.012
2000	94	0	103	33	54	546	10.88	45.27	1.012
2200	93	0	102	33	54	572	12.11	45.87	1.012
2400	93	0	105	33	54	595	13.21	45.86	1.012
2600	94	0	105	32	54	599	13.96	44.75	1.010
2800	94	0	103	32	54	603	14.95	44.51	1.010

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	HSU
1000	184.0	19.3	241	599	186.5	19.5	238	607	32.7	49
1200	189.0	23.8	237	615	191.2	24.0	234	623	33.2	42
1400	193.0	28.3	228	629	195.3	28.6	228	636	34.5	23
1600	188.0	31.5	226	612	190.2	31.9	223	619	34.9	15
1800	182.0	34.3	232	593	184.2	34.7	229	600	34.0	13
2000	184.0	38.5	230	599	186.2	39.0	227	606	34.2	12
2200	181.0	41.7	237	589	183.1	42.2	234	596	33.2	15
2400	176.0	44.2	244	573	178.1	44.8	241	580	32.3	14
2600	167.0	45.5	251	544	168.7	45.9	248	549	31.4	11
2800	161.0	47.2	259	524	162.6	47.7	256	530	30.4	13

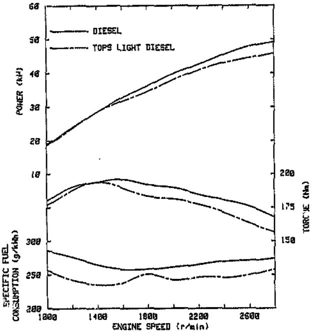

FIGURE E3 Full load power, torque and specific fuel consumption (ADE 236, retarded injection timing and using TLD)

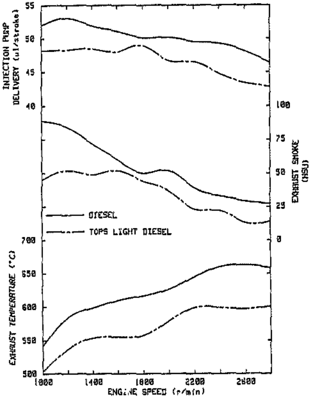

FIGURE E4 Full load injection pump delivery, exhaust smoke and exhaust temperature (ADE 236, *retarded* injection timing and using TLD)

TABLE E3 Full load performance data (ADE 236 standard injection timing and using diesel)

DATE: 85/03/12

ENGINE: ADE 236-1

FUEL: DIESEL

TEST NUMBER: PC 223

DISPLACEMENT: 3.86 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

DENSITY @ 20 °C: 865.0 kg/m3
HEAT VALUE (GROSS): 45010 kJ/kg

ATMOSPHERIC PRESSURE: 86.82 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD POWER DATA
INJECTION TIMING SET TO 24 °BTDC
NOTE: water-in temperatures were not measured

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	°C	°C	°C	°C	°C	°C	l/h	ml/str	
1000	92	0	96	31	53	541	6.24	51.07	1.011
1200	92	0	97	32	53	584	7.64	53.08	1.013
1400	92	0	98	31	53	599	8.72	51.90	1.011
1600	92	0	99	31	53	610	9.80	51.05	1.011
1800	92	0	101	31	54	617	10.81	50.06	1.011
2000	92	0	103	31	54	627	12.04	50.17	1.011
2200	93	0	105	30	55	645	13.07	49.50	1.010
2400	93	0	105	30	55	660	14.18	49.25	1.010
2600	93	[illegible]	105	30	55	563	15.01	48.11	1.010
2800	93	0	104	30	55	658	15.58	48.37	1.010

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	MEASURED POWER	MEASURED SFC	MEASURED BMEP	CORRECTED TORQUE	CORRECTED POWER	CORRECTED SFC	CORRECTED BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	HSU
1000	178.0	18.6	289	580	180.0	18.9	286	586	27.6	09
1200	188.0	23.6	280	612	190.5	23.9	276	620	28.6	04
1400	192.0	28.1	268	625	194.2	28.5	265	632	29.9	72
1600	194.0	32.5	261	632	196.2	[illegible]	258	639	30.7	60
1800	190.0	35.8	[illegible]	619	192.2	36.2	258	626	30.6	50
2000	188.0	39.4	[illegible]	612	190.1	39.9	262	619	30.2	52
2200	183.0	42.2	268	596	184.9	42.6	266	602	29.8	39
2400	179.0	45.0	273	583	180.7	45.4	270	589	29.3	33
2600	174.0	47.4	274	567	175.7	47.8	271	572	29.2	29
2800	166.0	48.7	277	541	167.6	49.1	274	546	28.9	27

TABLE E4 Full load performance data (ADE 236 retarded injection timing and using TLD)

DATE: 85/03/04

ENGINE: ADE 236-1

FUEL: 75% DIESEL
25% TOPS

TEST NUMBER: PC 201

DISPLACEMENT: 3.86 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

DENSITY @ 20 'C: 821.2 kg/m3
HEAT VALUE (GROSS): 45720 kJ/kg

ATMOSPHERIC PRESSURE: 86.03 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD POWER DATA
INJECTION TIMING SET TO 19 'BTDC
NOTE: water-in temperatures were not measured

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	'C	'C	'C	'C	'C	'C	l/h	ul/str	
1000	93	0	96	30	52	503	5.79	48.22	1.010
1200	94	0	98	30	53	537	6.96	48.30	1.010
1400	93	0	99	31	53	554	8.15	48.49	1.011
1600	93	0	100	30	53	555	9.21	47.99	1.010
1800	92	0	100	29	53	558	10.57	48.95	1.008
2000	92	0	102	29	54	579	11.21	46.70	1.008
2200	93	0	103	29	54	599	12.27	46.50	1.008
2400	91	0	104	29	54	599	12.90	44.78	1.008
2600	92	0	104	29	54	597	13.53	43.37	1.008
2800	92	0	104	29	55	601	14.39	42.82	1.008

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	MEASURED POWER	MEASURED SFC	MEASURED BMEP	CORRECTED TORQUE	CORRECTED POWER	CORRECTED SFC	CORRECTED BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	HSU
1000	175.0	18.3	259	570	176.7	18.5	257	575	30.4	45
1200	186.0	23.4	244	606	187.0	23.6	242	612	32.2	52
1400	192.0	28.1	236	625	194.2	28.5	235	632	33.1	49
1600	188.0	31.5	240	612	189.8	31.8	238	618	32.8	52
1800	182.0	34.3	253	593	183.4	34.6	251	597	31.1	44
2000	180.0	37.7	244	586	181.4	38.0	242	591	32.3	37
2200	176.0	40.5	249	573	177.4	40.9	247	578	31.7	23
2400	170.0	42.7	248	554	171.4	43.1	246	558	31.8	22
2600	162.0	44.1	257	528	163.3	44.5	250	532	31.3	13
2800	155.0	45.4	260	505	156.2	45.8	258	509	30.3	14

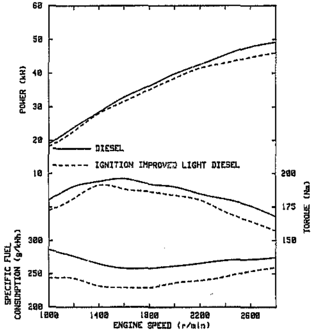

FIGURE E5 Full load power, torque and specific fuel consumption (ADE 236, standard injection timing and using IILD)

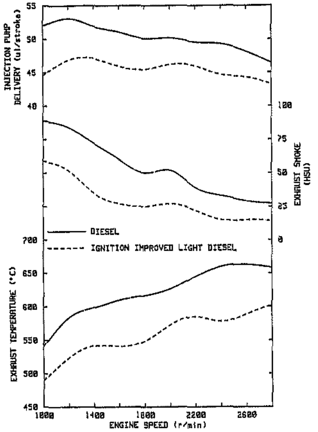

FIGURE E6 Full load injection pump delivery, exhaust smoke and exhaust temperature (ADE 236, standard injection timing and using IILD)

TABLE E5 Full load performance data (ADE 236 standard *injection timing and using diesel*)

DATE: 85/03/12

TEST NUMBER: PC 223

ENGINE: ADE 236-1

DISPLACEMENT: 3.86 1
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

FUEL:DIESEL

DENSITY @ 20 'C: 865.0 kg/m3
HEAT VALUE (GROSS): 45010 kJ/kg

ATMOSPHERIC PRESSURE: 86.02 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD POWER DATA
INJECTION TIMING SET TO 24 'BTDC
NOTE: water-in temperatures were not measured

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	'C	'C	'C	'C	'C	'C	l/h	ul/str	
1000	92	0	96	31	53	541	6.24	51.97	1.011
1200	92	0	97	32	53	584	7.64	53.08	1.013
1400	92	0	98	31	53	599	8.72	51.90	1.011
1600	92	0	[illegible]	31	53	610	9.80	51.05	1.011
1800	92	0	101	31	54	617	10.81	50.06	1.011
2000	92	0	103	31	54	677	12.04	50.17	1.011
2200	93	0	105	30	55	645	13.07	49.50	1.010
2400	93	0	105	30	55	660	14.10	49.25	1.010
2600	93	0	105	30	55	663	15.01	46.11	1.010
2800	93	0	104	30	55	658	15.50	46.37	1.010

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	HSU
1000	178.0	18.6	289	580	180.0	18.9	286	586	27.6	89
1200	188.0	23.6	280	612	190.5	23.9	276	620	28.6	84
1400	192.0	28.1	260	625	194.2	28.5	265	632	29.9	72
1600	194.0	32.5	261	632	196.2	32.9	258	639	30.7	60
1800	190.0	35.8	261	619	192.2	36.2	258	626	30.6	50
2000	188.0	39.4	265	612	190.1	39.8	262	619	30.2	52
2200	183.0	42.2	268	596	184.8	42.6	266	602	29.8	39
2400	179.0	45.0	273	583	180.7	45.4	270	589	29.3	33
2600	174.0	47.4	274	567	175.7	47.8	271	572	29.2	29
2800	166.0	48.7	277	541	167.6	49.1	274	546	28.9	27

TABLE E6 Full load performance data (ADE 236 standard injection timing and using IILD)

DATE: 85/06/03

TEST NUMBER: PC 246

ENGINE: ADE 236-1

DISPLACEMENT: 3.86 l
NO. OF CYLINDEES: 4
CYCLE: 4 -stroke

FUEL: 75% DIESEL
25% TOPS
0.23% HICET 3

DENSITY @ 20 'C: 821.2 kg/m3
HEAT VALUE (GROSS): 45720 kJ/kg

ATMOSPHERIC PRESSURE: 87.42 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA
INJECTION TIMING SET TO 24 'BTDC
NOTE: water-in temperatures were not measured

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	'C	'C	'C	'C	'C	'C	l/h	ml/str	
1000	93	0	95	25	55	490	5.35	44.57	.997
1200	94	0	96	25	53	524	6.75	46.08	.997
1400	94	0	98	25	53	542	[illegible]	[illegible]	.997
1600	93	0	98	25	53	541	[illegible]	45.90	.997
1800	92	0	99	25	53	549	9.81	45.[illegible]3	.997
2000	92	0	101	26	53	573	11.08	46.18	.999
2200	93	0	103	24	53	585	12.13	45.94	.995
2400	93	0	105	24	53	578	12.86	44.65	.995
2600	93	0	106	24	53	589	13.82	44.28	.995
2800	93	0	105	23	53	602	14.52	43.21	.994

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	HSU
1000	173.0	18.1	242	563	172.5	18.1	243	562	32.5	59
1200	182.0	22.9	242	593	181.4	22.8	243	591	32.5	52
1400	192.0	28.1	231	625	191.4	28.1	232	623	34.1	35
1600	189.0	31.7	229	615	188.4	31.6	229	614	34.5	27
1800	187.0	35.2	229	609	186.4	35.1	229	607	34.4	25
2000	184.0	38.5	236	599	183.7	38.5	237	598	33.3	27
2200	181.0	41.7	239	589	180.1	41.5	240	587	33.0	22
2400	172.0	43.2	244	560	171.2	43.0	245	557	32.2	15
2600	165.0	44.9	253	537	164.?	44.7	25?	535	31.2	15
2800	158.0	46.3	257	515	157.0	46.0	259	511	30.6	14

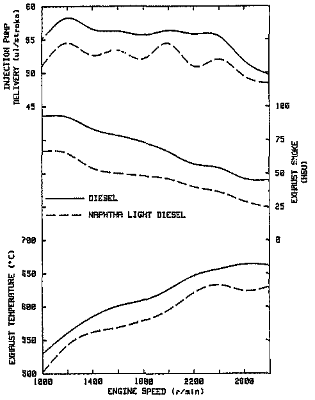

FIGURE E7 Full load power, torque and specific fuel consump ... DE 236 standard injection timing a ... NLD)

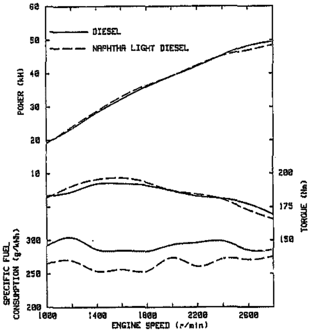

FIGURE E8 Full load injection pump delivery, exhaust smoke and exhaust temperature (ADE 236, standard injection timing and using NLD))

TABLE E7 Full load performance data (ADE 236 standard injection timing and using diesel)

DATE: 86/01/15

ENGINE: ADE 236

FUEL: DIESEL

TEST NUMBER: PC 403

DISPLACEMENT: 3.86 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

DENSITY @ 20 'C: 849.5 kg/m3
HEAT VALUE (GROSS): 44770 kJ/kg

ATMOSPHERIC PRESSURE: 87.15 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMACE DATA
NOTE: water-in temperatures not measured

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	'C	'C	'C	'C	'C	'C	l/h	ul/str	
1000	94	0	95	27	53	530	6.80	55.03	1.002
1200	94	0	97	27	53	561	8.39	58.27	1.002
1400	93	0	99	27	54	585	9.50	56.55	1.302
1600	93	0	100	27	54	601	10.88	56.25	1.002
1800	93	0	101	26	54	610	12.02	55.69	1.001
2000	93	0	102	26	55	626	13.53	56.39	1.001
2200	93	0	103	26	55	647	14.73	55.81	1.001
2400	93	0	103	26	54	657	15.99	55.52	1.001
2600	93	0	104	26	54	664	16.17	51.84	1.001
2800	92	0	105	26	54	662	16.67	49.61	1.001

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	RSU
1000	183.0	19.2	293	596	183.4	19.2	292	597	27.5	93
1200	186.0	23.4	305	.06	186.4	23.4	304	607	26.4	92
1400	192.0	28.1	287	62[illegible]	192.4	28.2	286	627	28.0	83
1600	192.0	32.2	285	625	192.4	32.2	285	62[illegible]	28.2	78
1800	191.0	36.0	284	622	191.1	36.0	283	62[illegible]	28.3	73
2000	187.0	39.2	294	609	187.1	39.2	293	609	27.4	66
2200	183.0	42.2	297	596	183.1	42.2	297	596	27.1	57
2400	181.0	45.5	299	589	181.1	45.5	298	590	26.9	54
2600	177.0	48.2	285	576	177.1	48.2	285	577	28.2	46
2800	169.0	49.6	286	550	169.1	49.6	286	551	28.1	45

TABLE E8 Full load performance data (ADE 236 standard injection timing and using NLD)

DATE: 86/01/22

TEST NUMBER: PC 414

ENGINE: ADE 236

DISPLACEMENT: 3.86 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

FUEL: 25% NAPHTHA
75% DIESEL

DENSITY @ 20 °C: 823.0 kg/m3
HEAT VALUE (GROSS). 43550 KJ/kg

ATMOSPHERIC PRESSURE: 87.20 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	°C	°C	°C	°C	°C	°C	l/h	ul/str	
1000	93	0	96	25	53	502	6.12	51.03	.998
1200	93	0	97	25	53	543	7.85	54.53	.998
1400	93	0	98	25	53	562	8.84	52.64	.998
1600	93	0	99	25	53	569	10.26	53.42	.998
1800	93	0	101	24	54	579	11.25	52.08	.997
2000	93	0	103	24	54	595	13.07	54.47	.997
2200	93	0	103	24	54	621	13.47	51.01	.997
2400	93	0	104	24	54	633	14.99	52.05	.997
2600	93	0	104	24	53	624	15.44	49.49	.997
2800	93	0	104	24	53	630	16.26	48.39	.997

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	HSU
1000	182.0	19.1	264	593	181.7	19.0	265	592	31.3	67
1200	191.0	24.0	269	622	190.7	24.0	270	621	30.7	65
1400	196.0	28.7	253	638	195.7	28.7	254	637	32.6	54
1600	197.0	33.0	256	642	196.7	33.0	256	641	32.3	58
1800	194.0	36.6	253	632	193.4	36.5	254	630	32.7	48
2000	188.0	39.4	273	612	187.4	39.2	274	610	30.3	46
2200	185.0	42.6	260	602	184.4	42.5	261	601	31.8	40
2400	181.0	45.5	271	589	180.4	45.3	272	588	30.5	36
2600	173.0	47.1	270	563	172.4	47.0	271	562	30.6	29
2800	166.0	48.7	275	541	165.5	48.5	276	539	30.1	24

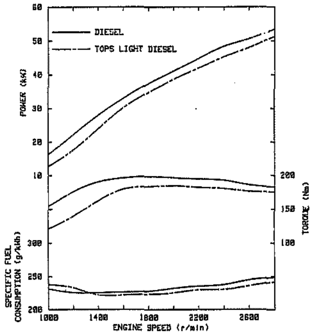

FIGURE E9 Full load power, torque and specific fuel consumption (ADE 314, standard injection timing and using TLD)

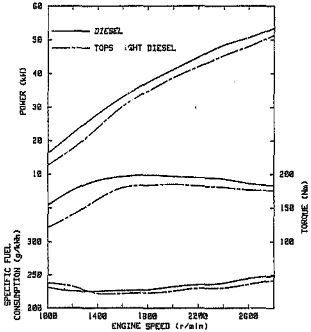

FIGURE E9 Full load power, torque and specific fuel consumption (ADE 314, standard injection timing and using TLD)

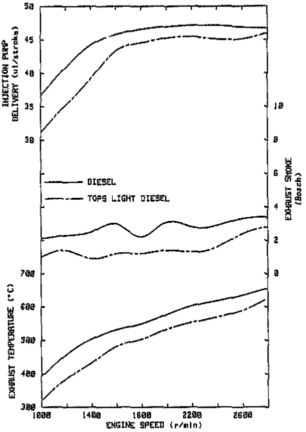

FIGURE E10 Full load injection pump delivery, exhaust smoke and exhaust temperature (ADE 236, standard injection timing and using TLD)

TABLE E9 Full load performance data (ADE 314 standard *injection timing and using diesel*)

DATE: 86/09/16

ENGINE: ADE 314

FUEL: DIESEL

TEST NUMBER: PC 566

DISPLACEMENT: 3.784 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

DENSITY @ 20 'C: 847,4 kg/m3
HEAT VALUE (GROSS): 45300 kJ/kg

ATMOSPHERIC PRESSURE: 87.10 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	'C	'C	'C	'C	'C	'C	l/h	ml/str	
1000	85	80	112	26	31	391	4,40	36,67	1,001
1200	86	80	115	26	31	458	5,88	40,86	1,001
1400	85	79	119	27	31	504	7,43	44,21	1,003
1600	86	80	120	27	31	532	8,79	45,80	1,003
1800	86	81	123	27	31	550	10,06	46,58	1,003
2000	86	81	126	27	31	578	11,27	46,97	1,003
2200	86	80	128	27	31	604	12,45	47,15	1,003
2400	85	80	128	27	31	619	13,57	47,13	1,003
2600	85	80	130	28	30	635	14,61	46,82	1,004
2800	85	80	130	27	30	655	15,60	46,60	1,003

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	[illegible]	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	Bosch
1000	154	16.1	231	512	154.1	16.1	231	512	34.4	2.1
1200	[illegible]	22.1	225	585	176.2	22.1	225	585	35.3	2.3
1400	[illegible]	27.9	226	631	190.5	27.9	225	633	35.2	2.5
1600	[illegible]	32.8	227	651	196.9	32.9	226	653	35.0	3.0
1800	198.0	37.3	228	658	198.5	37.4	228	659	34.8	2.2
2000	196.0	41.1	233	651	196.5	41.2	232	653	34.2	3.1
2200	194.0	44.7	236	644	194.5	44.8	235	646	33.7	2.8
2400	192.0	48.3	238	638	192.5	48.4	238	639	33.3	2.9
2600	185.0	50.4	246	615	185.8	50.6	245	617	32.3	3.3
2800	182.0	[illegible]	249	605	182.5	53.5	248	606	31.9	3.4

TABLE E10 Full load performance data (ADE 314 standard injection timing and using TLD)

DATE: 86/09/17

TEST NUMBER: PC 567

ENGINE: ADE 314

DISPLACEMENT: 3,784 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

FUEL: 75% DIESEL
25% TOPS

DENSITY @ 20 °C: 803.9 kg/m3
HEAT VALUE (GROSS): 46050 kJ/kg

ATMOSPHERIC PRESSURE: 87.51 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	°C	°C	°C	°C	°C	°C	l/h	ul/str	
1000	84	81	106	[illegible]	31	[illegible]	3.74	31.16	.998
1200	85	80	110	[illegible]	[illegible]	[illegible]	5.08	35.24	.998
1400	85	81	113	[illegible]	[illegible]	[illegible]	[illegible]	39.18	.998
1600	86	81	117	[illegible]	[illegible]	[illegible]	[illegible]	43.34	1.000
1800	85	80	120	27	[illegible]	[illegible]	9.62	44.54	1.000
2000	85	80	123	27	[illegible]	[illegible]	10.86	45.25	1.000
2200	85	80	126	28	31	554	12.00	45.46	1.001
2400	85	80	127	28	31	578	12.98	45.07	1.001
2600	84	79	127	29	31	590	14.08	45.12	1.001
2800	85	80	126	28	30	624	15.45	45.98	1.001

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	POWER	SFC	BMEP	CORRECTED TORQUE	POWER	SFC	BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	Bosch
1000	121.0	12.7	237	402	120.7	12.6	238	401	33.6	1.0
1200	140.0	17.6	[illegible]	465	139.7	17.6	232	464	33.7	1.4
1400	163.0	23.9	221	541	162.7	23.9	222	540	35.3	.9
1600	180.0	30.2	222	598	179.9	30.1	222	598	35.2	1.2
1800	184.0	34.7	223	611	183.9	34.7	223	611	35.1	1.2
2000	185.0	38.7	225	615	184.9	38.7	225	614	34.7	1.4
2200	182.0	41.9	230	605	182.2	42.0	230	605	34.0	1.3
2400	180.0	45.2	231	598	180.2	45.3	230	599	33.9	1.6
2600	176.0	47.9	236	585	176.2	48.0	236	585	33.1	2.4
2800	175.0	51.3	242	581	175.2	51.4	242	582	32.3	2.8

TABLE E10 Full load performance data (ADE 314 standard injection timing and using TLD)

DATE: 86/09/17

TEST NUMBER: PC 567

ENGINE: ADE 314

DISPLACEMENT: 3.784 l
NO. OF CYLINDERS: 4
CYCLE: 4 -stroke

FUEL: 75% DIESEL
25% TOPS

DENSITY @ 20 °C: 803.9 kg/m3
HEAT VALUE (GROSS): 46050 kJ/kg

ATMOSPHERIC PRESSURE: 87.51 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA

TEST CONDITIONS

SPEED	TEMPERATURES WATER OUT	WATER IN	OIL	INLET AIR	INJ. PUMP	EX-HAUST	FUEL FLOW	INJ. PUMP DELIVERY	CORREC-TION FACTOR
r/min	°C	°C	°C	°C	°C	°C	l/h	cl/str	
1000	84	81	106	26	31	319	3.74	31.16	.998
1200	85	80	110	26	30	386	5.08	35.24	.998
1400	85	81	113	26	31	433	6.58	39.18	.998
1600	86	81	117	27	31	482	8.32	43.34	1.000
1800	85	80	120	27	31	504	9.62	44.54	1.000
2000	85	80	123	27	30	534	10.86	45.25	1.000
2200	85	80	126	28	31	554	12.00	45.46	1.001
2400	85	80	127	28	31	570	12.98	45.07	1.001
2600	84	79	127	28	31	598	14.08	45.12	1.001
2800	85	80	126	28	30	624	15.45	45.98	1.001

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	MEASURED POWER	MEASURED SFC	MEASURED BMEP	CORRECTED TORQUE	CORRECTED POWER	CORRECTED SFC	CORRECTED BMEP	EFFIC-IENCY	EXH. SMOKE
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	Bosch
1000	121.0	12.7	237	402	120.7	12.6	238	401	33.0	1.0
1200	140.0	17.6	232	465	139.7	17.6	232	464	33.7	1.4
1400	163.0	23.9	221	541	162.7	23.8	222	540	35.3	.9
1600	180.0	30.2	222	598	179.9	30.1	222	598	35.2	1.2
1800	184.0	34.7	223	611	183.9	34.7	223	611	35.1	1.2
2000	185.0	38.7	225	615	184.9	38.7	225	614	34.7	1.4
2200	182.0	41.9	230	605	182.2	42.0	230	605	34.0	1.3
2400	180.0	45.2	231	598	180.2	45.3	230	599	33.9	1.6
2600	176.0	47.9	236	585	176.2	48.0	236	585	33.1	2.4
2800	175.0	51.3	242	581	175.2	51.4	242	582	32.3	2.0

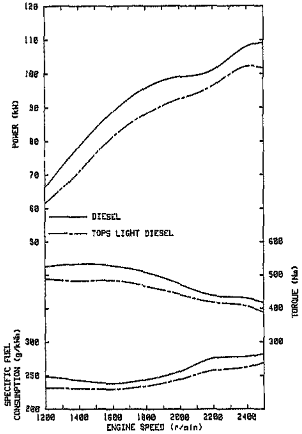

FIGURE E11 Full load power, torque and specific fuel consumption (Deutz F6L 413F, standard injection timing and using TLD)

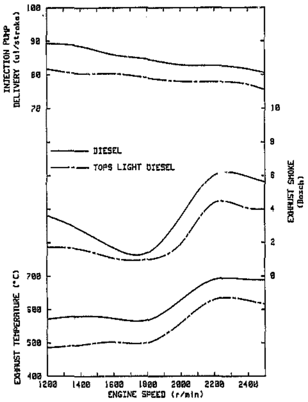

FIGURE E12 Full load injection pump delivery, exhaust smoke and exhaust temperature (Deutz F6L 4.3F, standard injection timing and using TLD)

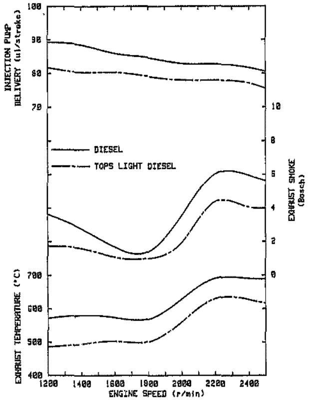

FIGURE E12 Full load injection pump delivery, exhaust smoke and exhaust temperature (Deutz F6L 413F, standard injection timing and using TLD)

TABLE E11 Full load performance data (Deutz F6L 413F standard injection timing and using diesel)

DATE: 88/05/02

TEST NUMBER: PC 658

ENGINE: DEUTZ F6L 413F

DISPLACEMENT: 9.572 l
NO. OF CYLINDERS: 6
CYCLE: 4 -stroke

FUEL: DIESEL

DENSITY @ 20 'C: 851.0 kg/m3
HEAT VALUE (GROSS): 45590 kJ/kg

ATMOSPHERIC PRESSURE: 88.06 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA

TEST CONDITIONS

SPEED	TEMPERATURES CYL HEAD LEFT	CYL HEAD RIGHT	OIL	INLET AIR LEFT	INLET AIR RIGHT	INJ. PUMP	EXHAUST LEFT	EXHAUST RIGHT	FUEL FLOW	INJ. PUMP DELIVERY	CORRECTION FACTOR
r/min	'C	'C	'C	'C	'C	'C	'C	'C	l/h	ul/str	
1200	160	145	93	17	17	28	581	561	19.27	89.21	.979
1400	159	143	94	17	17	28	574	586	22.29	88.47	.979
1600	158	143	95	18	18	28	567	579	24.74	85.90	.980
1800	152	140	94	19	19	28	583	554	27.41	84.61	.982
2000	144	135	93	22	22	29	663	595	29.83	82.86	.987
2200	142	132	95	20	19	29	780	680	32.74	82.67	.983
2400	145	132	99	18	17	31	693	687	35.27	81.64	.980
2500	145	133	99	18	18	31	694	682	36.21	80.47	.981

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	MEASURED POWER	MEASURED SFC	MEASURED BMEP	CORRECTED TORQUE	CORRECTED POWER	CORRECTED SFC	CORRECTED BMEP	EFFICIENCY	EXHAUST SMOKE LEFT	EXHAUST SMOKE RIGHT
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	Bosch	Bosch
1200	537.0	67.5	243	705	525.6	66.0	248	690	32.5	3.9	3.3
1400	545.0	79.9	237	716	533.4	78.2	243	700	33.3	3.0	2.6
1600	539.0	90.3	233	708	528.2	88.5	238	694	33.9	1.7	1.7
1800	517.0	97.5	239	679	507.8	95.7	244	667	33.0	2.2	.6
2000	479.0	100.3	253	629	472.9	99.0	256	621	31.2	5.1	2.0
2200	446.0	102.8	271	586	438.2	101.0	276	575	29.1	6.7	5.4
2400	438.0	110.1	273	575	429.1	107.8	278	563	29.0	6.1	5.8
2500	425.0	111.3	277	558	416.9	109.1	282	547	28.5	6.0	5.2

TABLE 512 Full load performance data (Deutz F6L 413F standard injection timing and using TLD)

DATE: 88/05/04

TEST NUMBER: PC 667

ENGINE: DEUTZ F6L 413F

DISPLACEMENT: 9.572 l
NO. OF CYLINDERS: 6
CYCLE: 4 -stroke

FUEL 75 % DIESEL
25 % TOPS

DENSITY @ 20 °C: 885.0 kg/m3
HEAT VALUE (GROSS): 45900 kJ/kg

ATMOSPHERIC PRESSURE: 87.60 kPa

COMMENTS: PERFORMANCE DATA CORRECTED TO SABS 013, 1982, PART II (ALTITUDE)
FULL LOAD PERFORMANCE DATA

TEST CONDITIONS

SPEED	TEMPERATURES CYL HEAD LEFT	CYL HEAD RIGHT	OIL	INLET AIR LEFT	INLET AIR RIGHT	INJ. PUMP	EXHAUST LEFT	EXHAUST RIGHT	FUEL FLOW	INJ. PUMP DELIVERY	CORRECTION FACTOR
r/min	°C	°C	°C	°C	°C	°C	°C	°C	l/h	ml/str	
1200	147	132	87	18	18	29	490	481	17.65	81.69	
1400	147	134	90	18	18	31	486	502	20.23	80.29	
1600	148	135	92	19	19	31	495	510	23.11	80.75	
1800	144	134	93	21	21	31	510	491	25.60	79.01	.98
2000	148	132	93	24	24	33	592	529	28.05	77.32	.994
2200	138	128	94	22	22	33	640	620	30.81	77.81	.990
2400	140	128	98	21	20	33	630	624	33.26	77.00	.988
2500	139	127	98	20	19	32	619	612	33.93	75.40	.986

PERFORMANCE RESULTS

SPEED	MEASURED TORQUE	MEASURED POWER	MEASURED SFC	MEASURED BMEP	CORRECTED TORQUE	CORRECTED POWER	CORRECTED SFC	CORRECTED BMEP	EFFICIENCY	EXHAUST SMOKE LEFT	EXHAUST SMOKE RIGHT
r/min	Nm	kW	g/kWh	kPa	Nm	kW	g/kWh	kPa	%	Bosch	Bosch
1200*	497.0	62.5	227	653	488.9	61.4	231	642	34.5	2.3	1.2
1400	490.0	71.8	227	643	482.0	70.7	230	633	34.6	2.4	.9
1600	491.0	82.3	226	645	483.8	81.1	230	635	34.7	1.3	.9
1800	472.0	88.8	232	620	466.7	88.0	234	613	33.9	1.1	.9
2000	444.0	93.0	243	583	441.4	92.5	244	580	32.3	3.0	.9
2200	422.0	97.2	255	554	418.0	96.3	258	549	30.7	5.1	3.7
2400	411.0	103.3	259	540	405.9	102.0	263	533	30.3	4.7	3.5
2500	393.0	102.9	265	516	307.6	101.5	269	509	29.5	4.3	3.6

APPENDIX F   PAPERS PRESENTED

1 FALK, R S. The effect of a BP formulated light diesel on the durability of an ADE 236 diesel engine operating with retarded injection timing. Alternative Fuels Seminar, Pretoria, May 1985.

2 FALK, R S. The effect of an ignition-improved light diesel on the preformance and durability of an engine operating with standard injection timing. Alternative Fuels seminar, Pretoria, May 1986.

3 FALK, R S. Engine operation on ignition-improved light diesel. Annual Transportation Convention, Pretoria, August 1986.

4 FALK, R S. The effect of a diesel blend containing heavy naphtha on the performance and durability of an ADE 236 diesel engine operating with standard injection timing. Alternative Fuels Seminar, Pretoria, July 1987.

5 FALK, R S. The effect of a light diesel blend formulated by BP on the performance and durability of an ADE 314 diesel engine. Alternative Fuels Seminar, Pretoria, July 1987.

6 FALK, R S, and MYBURGH, I S. Engine operation on extended diesel fuels. Annual Transportation Convention, Pretoria, August 1987.

7 FALK, R S. Operation of an ADE 236 diesel engine on light diesel fuels. SAE Fuels and Lubricants Meeting, Portland, October 1988. (SAE Technical Paper 881646)

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