Liquid Petroleum Gas as Automotive Fuel in South Africa

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Supervisor

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Declaration

I, the undersigned, hereby declare that the work contained in this thesis is my original work and that I have not previously in it's entirety or in part submitted it at any university for a degree.

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Date: 28 November 2001

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Executive Summary

The trends in worldwide fuel consumption and availability were studied, these indicated a bigger growth in gaseous fuel use than that of crude oil over the last decade.

The economics (cost of converting and running vehicles on LPG) were studied and compared with those of petrol and diesel fuels.

The government's approach to LPG taxation and the structure of the fuel price was also considered in an attempt to foresee what the future holds for LPG use in the motor industry.

Gas fuelling systems that are currently available were studied and briefly described. The information obtained from the background study was used to help in the conversion of a two litre petrol engine. The engine was equipped to run on petrol Injection, liquid phase LPG injection and LPG carburettion. In-cylinder pressures, exhaust emissions and fuel consumption were amongst the parameters that were recorded for each fuel. The in-cylinder pressure measurements were used to study the combustion characteristics of petrol and LPG. Computer modeling was also used to investigate the trends that were recorded and this gave valuable insight into the different combustion characteristics of each fuel and the effect of gaseous versus liquid supply.

For the passenger bus market a 12 litre 6 cylinder diesel engine was converted to LPG operation only. This required several changes to the pistons, cylinder head, inlet manifold and the addition of an electronic ignition system. Some changes had to be made to the squish characteristics of the pistons to make it suitable for homogeneous fuel air mixtures. The reasons for this were studied and described. Dynamometer tests revealed inadequacies in the ignition system that still need to be addressed before the engine can be built into a bus.

Recommendations are made as to best utilize LPG in the South African Automotive industry, so as to improve public transport and air quality in some of our cities.

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Opsomming

'n Studie van tendense in wêreldwye energieverbruik en besikbaarheid is gedoen. Dit het aan die lig gebring dat die groei in die gebruik van gasagtige brandstowwe in die laaste dekade die van ru-olie oortref het. Die lewensvatbaarheid van Vloeibare Petroleum Gas (VPG) voertuie, ombouing sowel as lopende koste, is bestudeer en vergelyk met die van Petrol en Diesel voertuie. Die regering se benadering tot belasting op VPG en die struktuur van die brandstofprys is ook ondersoek om te bepaal of die gebruik van VPG in n groter skaal as tans lewenvatbaar is.

Vir tegniese agergrond is gas aangedrewe voertuie wêreldwyd bestudeer om te sien watter brandstof-voorsiening stelsels en enjins gebruik word. Die verskillende stelsels word bondig beskryf. Hierdie inligting is onder meer gebruik in die ombouing van n twee liter petrolenjin na VPG. Die enjin is toegerus om op beide petrol en VPG te loop terwyl die VPG in gasfase met behulp van 'n vergasser of as vloestof deur brandstof inspuiting toegedien kon word.

Ontbrandingskamerdruk, uitlaatgasse en brandstofverbruik is van die parameters wat tydens toetse gemeet is. Die ontbrandingskamerdukmetings is gebruik om die verbrandingskarakteristieke van elke brandstof te bepaal. Nagebootste verbrandingstempos is in n rekeraarprogram gebruik om verskillende verbrandings karakteristieke wat gemeet is te ondersoek en tendense te bevestig.

Vir die swaarvoertuigmark is 'n 12 Liter diesel enjin ombebou na VPG gebruik. Die dieselpomp en inspuiters is vervang met elektroniese vonkontsteking en vonkproppe. Die verbrandingskamer moes verander word om spontane verbranding tydens samepersing te voorkom. Die redes hiervoor is ondersoek en beskryf. Dinamo toetse het tekortkominge uitgewys in die elektroniese vonkontstekingsstelsel wat nog nie ten volle aangespreek is nie.

Aanbevelings is gemaak om die toenemende gebruik van VPG as motorvoertuigbrandstof in Suid Afrika aan te bevorder om sodoende beter gebruik te maak van die beskikbare energie uit ru olie en ander bronne. Aanbevelings is ook gemaak ten opsigte van die gebruik van VPG in openbare vervoer en verbetering van lug gehalte in sommige stede.

Acknowledgements

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Glossary of terms

ADE   :   Atlantis Diesel Engines

AFV   :   Alternative Fuelled Vehicle

BMEP :   Break Mean Effective Pressure

BTDC :   Before Top Dead Center

CAE   :   Center for Automotive Engineering, currently (Stellenbosch Automotive Engineers)

CNG   :   Compressed Natural Gas

ECU   :   Engine Control Unit

EGR   :   Exhaust Gas Recirculation

LPG   :   Liquefied Petroleum Gas

MAP   :   Manifold Absolute Pressure

MEPC :   Mineral and Energy Policy Center

MON   :   Motor Octane Number

MSD   :   Multiple Spark Discharge

NAAMSA: National Association of Automobile Manufacturers of South Africa

OEM   :   Original Equipment Manufacturer

PIC   :   Programmable Integrated Circuit

RACER: Rapid Acquisition of Combustion and Engine Results

RON   :   Research Octane Number

SABS :   South African Bureau of Standards

SAPIA :   South African Petroleum Industry Association

SFC   :   Specific Fuel Consumption

SHED :   Sealed Housing for Evaporative Determination

SPICE :   Simulation Program for Internal Combustion Engines

TDC   :   Top Dead Center

WOT   :   Wide-Open Throttle

1 INTRODUCTION

The use of natural gas and liquefied petroleum gas as vehicle fuel has found wide acceptance worldwide but not in South Africa. It is therefore believed that there is scope for the increased use of gaseous fuels in the local automotive industry. Changes in energy consumption indicate that the use of gaseous fuels is set to increase. The reasons for this are manifold, the main ones being the ready availability of natural gas and the limited availability of crude oil. The increased use of gaseous fuels in turn has led to the rapid development of gas fuel handling systems and gas engines suitable for automotive use. Such is the development of gas fuel engine and vehicle technology that gaseous-fuelled vehicles have gained widespread acceptance and their use has become common in many countries of the world.

To understand why the same trends are not evident in South Africa, fuel production and consumption in South Africa had to be studied. It was found that a great deal of energy in South Africa is derived from coal and crude oil, but very little from gas. The development of gas fields has only recently begun. Previously the only sources of gas have long been the oil refineries, which produce liquefied petroleum gas, and Sasol with its fuel from coal projects, which is a producer of methane-rich gas. Together with a rather lenient fuel specification and no emissions legislation this has done little to encourage the use of gaseous fuels on a greater scale. As a result South Africa lags behind the rest of the world in gas fuelled vehicle technology. Changes however could be eminent; the high price of imported crude oil combined with the development of the Kudu and Pande gas fields as well as the Mossgas project will increase the availability of gaseous fuels. This should favour the use of gaseous fuels in the foreseeable future. It is with this in mind that this study was initiated. Since only LPG was commercially available at the time of this study, it was decided to study its use in motor vehicles as an introduction to the world of gas fuelled vehicles for South Africa.

2 GAS AS AN AUTOMOTIVE FUEL

2.1 Automotive use of crude oil resources

The quest for mobility makes the transport sector one of the greatest energy consumers of our time. So much so that the transport sector consumes about 70 % of the worlds crude derived products. These products, petrol, jet fuel and diesel are popular in transport because of their high energy density, which makes them ideally transportable and simple to store. In 1995 the balance of world energy consumption, illustrated in figure 2.1, consisted of 41 % oil, 23 % gas, 25 % coal, 8 % nuclear and 3 % hydroelectric.
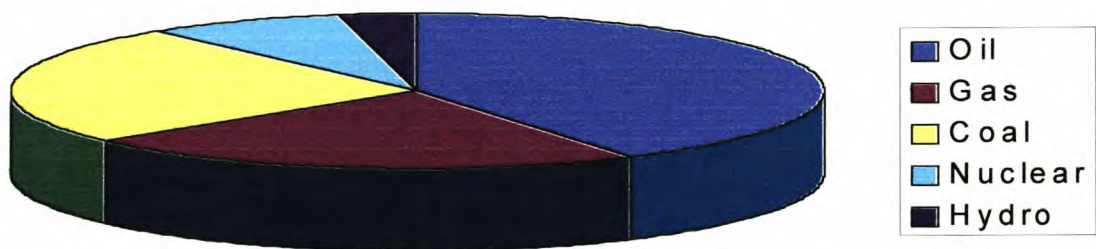
Figure 2.1 World energy consumption in 1995 (Doppengieter and du Toit, 1996)

Of these, the only renewable energy resource is hydroelectric. All the other energy sources that supply the bulk of our energy are of a finite nature requiring the need for alternative sources of energy to be used on a greater scale in future.

2.2 Future energy availability

To give an idea of the expected lifetime of energy reserves, the reserve to production, R/P ratio is used. At 1995 production rates and estimated energy reserves, the expected lifetime for the main energy resources was calculated to be the following:

oil 46 years, gas 64 years, coal 228 years (Doppengieter and du Toit, 1996).

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These may well be extended with discovery of new oil, coal and gas fields since there are large areas that have not been explored sufficiently to ascertain the possible yield of gas and oil. Further improvement in oil recovery methods could also increase the yield of already exploited oil fields.

2.2.1 Changes in energy consumption

Due to the finite nature of crude oil reserves, the transport industry, the major consumer, is challenged to reduce its dependence on this source of energy. Limited availability and environmental concerns have caused a shift towards the use of alternative fuels of which natural gas is one. Figure 2.2 shows the global consumption of natural gas and oil in the period 1989 to 1995 during this time the increase in natural gas was 1.39 % exceeding the 0.75 % of crude oil during the same period.
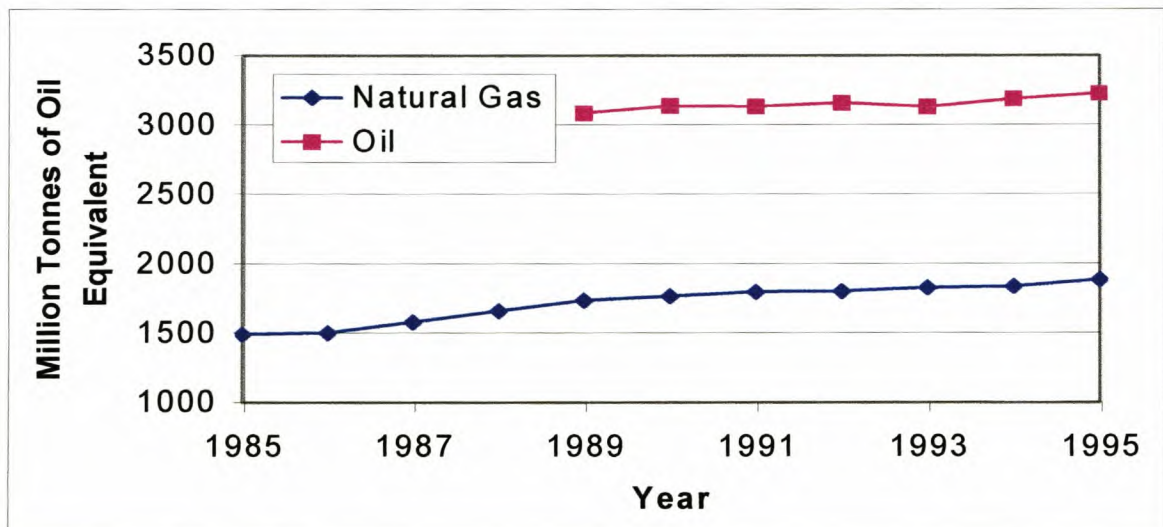
Figure 2.2 World oil and natural gas consumption from 1985 to 1995 (Doppengieter and du Toit, 1996)

On average the natural gas consumption from 1989 to1995 was equivalent to 57 % of the oil consumption during the same time. This indicates how significant the use of gaseous fuels has become during these last years.

2.2.2   Alternative fuels

The following alternative fuels have been identified by the World LPG Association (1995) as being viable for use in the transport industry.

- Compressed Natural Gas (CNG)

- Ethanol

- Liquefied Natural gas (LNG)

- Liquefied Petroleum Gas (LPG)

- Methanol

- Reformulated Gasoline

Hydrogen-fuelled vehicles may soon be added to this list (World LPG Association, 1995).

Figure 2.3 illustrates why diesel and petrol are so popular as fuels in transport. These have the highest energy content per volume of all the fuels, making them ideally transportable.
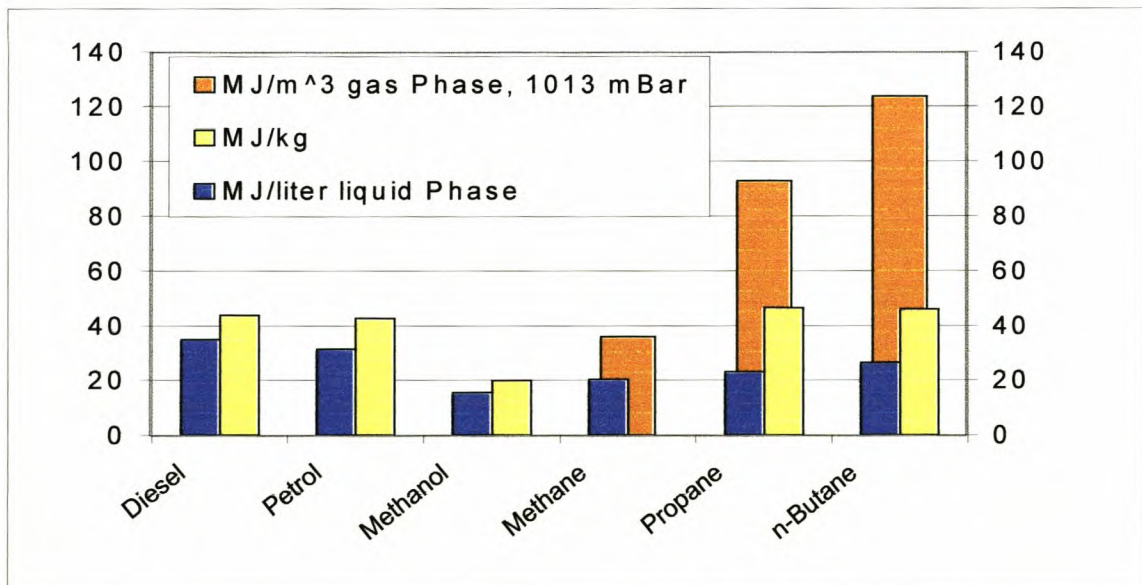
Figure 2.3 Lower heating value of alternative fuels (Bergmann and Busenthur, 1987)

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LPG (propane and butane) has the third highest energy content per volume, making it the most popular of the alternative fuels due to relative ease of transport and acceptable vehicle range (World LPG Association, 1995).

2.2.3 Natural gas

The fastest growing alternative fuel is natural gas which is reflected by a 100 % growth between 1992 and 1997. Natural gas consists mostly of Methane (CH_4). Its growth can be ascribed to the widespread availability of sources and therefore merits special attention amongst the alternative fuels.

Sources of natural gas include gas fields, oil fields, landfill sites and any biological decomposition where it is known as biogas. It is cheap to produce since it appears in its raw state in nature and no costly refining is needed before it can be used as fuel. The main cost associated with natural gas is that of transport due to its low energy density. Natural gas requires a pipe network to transport and supply it in useful quantities.

The density of natural gas at atmospheric pressure is 1000 times lower than that of crude oil. Despite its low density natural gas is finding greater acceptance as an automotive fuel. This is mainly due to its widespread availability as well as stricter exhaust emissions legislation. Table 2.1 shows the use of natural gas vehicles per country in 1991, when there were more than 1.8 million natural gas vehicles in use worldwide.

Table 2.1 Distribution of natural gas vehicles in the world (Shires, 1995)

Country	Number of Vehicles [1991]	Number of CNG Sites [1991]
Former USSR	over 500 000	----------------
New Zealand	600 000	350
Italy	250 000	250
Argentinia	130 000	150
Canada	30 000	150
U.S.A.	30 000	150
Columbia	1000	24
Australia	500	50
Brazil	250	11
Chile	100	1
Total	1850	1136

Natural gas fuel pricing requires long term planning. The real cost of the fuel should be determined by the cost of the infrastructure for transport and the ability to upgrade and renew transport networks. The actual cost of the fuel itself is comparatively low. This is unlike the case with crude oil where the actual cost of the fuel plays a dominant role; and in places this has led to the under pricing of natural gas products, resulting in a long-term detrimental affect on the industry.

2.2.4 Liquefied petroleum gas

Liquefied petroleum gas consists mainly of mixtures of propane (C_3H_8) and butane (C_4H_{10}). The major advantage of LPG is that it lends itself to use in petrol engines without any change to the engine. It is a clean-burning fuel that can be transported in liquid phase under moderate pressures. It has a high energy density making it ideal for automotive applications.

As a result, LPG is the alternative automotive fuel with the biggest market penetration. This is reflected by the almost 4 million LPG fuelled vehicles in the world in 1994 with over 22000 filling sites compared to the 1100 filling sites for natural gas vehicles in 1991.

Table 2.2 Distribution of LPG vehicles in the world, (World LPG Association, 1995)

Country	1994 Consumption [Thousands of tonnes]	Number of Vehicles [Thousands]	Number of LPG Sites
Japan	1814	305	1921
South Korea	1434	278	502
Italy	1202	1050	1900
Mexico	1185	300	1500
U.S.A.	1012	350	3300
Australia	890	330	2450
Netherlands	810	470	2000
Canada	649	140	5000
Various	451	267	3100
C.I.S.	292	450	1000
Thailand	140	19	92
Total	9879	3959	22765

In the Netherlands 7 % of all passenger vehicles run on LPG. This is the result of the ready availability of LPG in the Netherlands. Besides possessing significant gas fields the Netherlands have huge crude oil refineries that supply the rest of Europe with crude-derived products. These refineries produce significant amounts of LPG in the process. In Australia 6 %

of all light duty vehicles are LPG-fuelled. In America 2/3 of the estimated 420 000 AFV's operate on LPG consuming the equivalent mass of 0.18 % of the petrol and diesel consumption combined (World LPG Association, 1995).

2.3 Energy consumption in South Africa

Figure 2.4 shows the energy consumption in South Africa for 1995. It shows that our use of coal by far supersedes any other form of energy with 77.6 %, oil 19.3 %, nuclear 2.9 % and hydro 0.2 %, while gas made up less than 1 % (SAPIA, 1996).
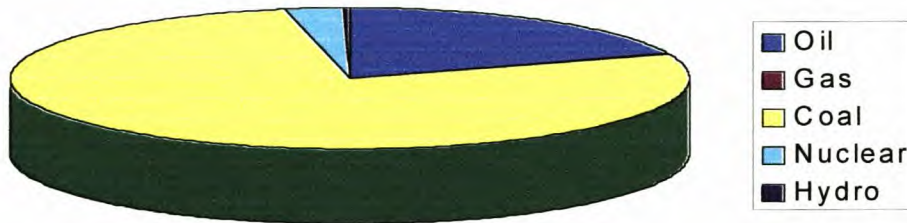
Figure 2.4 Energy consumption in South Africa during 1995 (SAPIA, 1996)

Coal is by far South Africa's largest conventional energy source. South Africa is the sixth largest coal producing country in the world and the eighth largest coal consumer in the world. The availability of coal has lead to the production of liquid fuels from coal in the Sasol plants that at have at times produced 40 % of the country's automotive fuel needs. This process also produces alcohol and methane-rich gases which are being used increasingly for automotive applications as it is generally too costly to export.

Gas fields in the Southern African region include the Pande gas field of Mozambique, the Kudu gas field off the coast with Namibia and the Bredasdorp basin off the East Cape Coast. These are all relatively small fields with yields estimated between 6-60 trillion cubic feet. There are some smaller gas fields in the Karoo and the Kalahari with yields estimated between 17-170 million cubic meters (0.6-6 trillion cubic feet) (US Department of Energy, 1997). The only current gas-producing fields are: the gas field in the Aughallas area serving the Mossgas project,

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and the Pande gas field in Mozambique. The Kudu gas field of Namibia is planned to go into production soon with the possibility of pipelines leading down to Saldanha. This could open up opportunities for heating and power generation with natural gas in the area.

Oil fields correspond with the location of the gas fields since crude oil is actually a mixture of oil and gas. The identified fields are predominantly gas so that oil reserves are considered negligible (Doppengieter and duToit, 1996 and US Department of Energy, 1997).

2.3.1   LPG sources in South Africa

The main source of LPG in South Africa is the refining of crude oil that produces between 1-3 % LPG by mass as by-product in the initial distillation process. The consumption of crude oil therefore can be used to give an indication of the amount of LPG that can be derived from its refinery. South Africa's Oil consumption by SAPIA members reached 425 000 barrels per day in 1995. If 1-3 % of this is converted to LPG it means that, in weight equivalent, 4 250 to

12 750 barrels of LPG are produced daily. During the same time Sasol and Mossgas produced an equivalent of an additional 195 000 barrels per day.
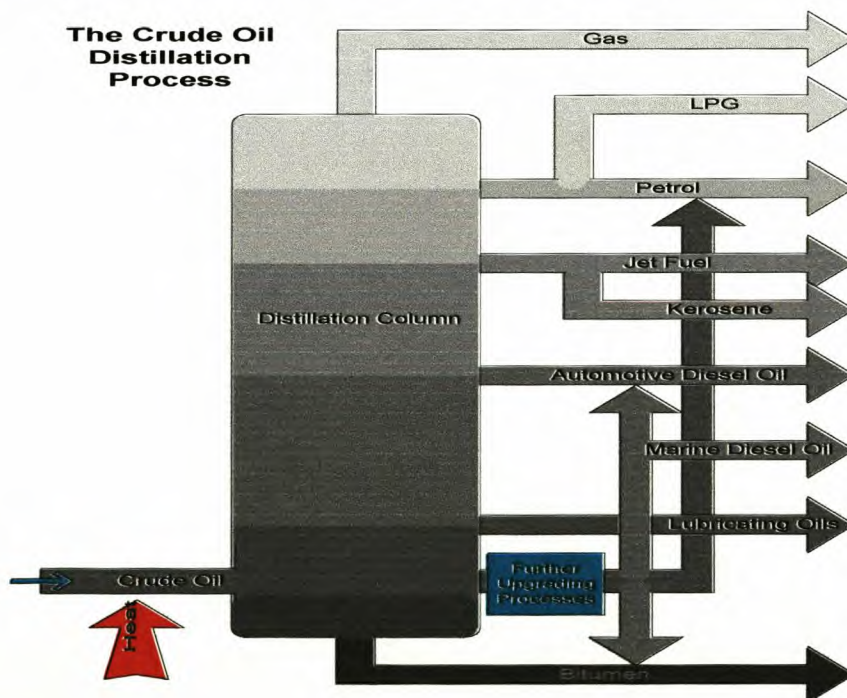
Figure 2.5 The refinery of crude oil (SAPIA, 1995)

Figure 2.6 represents the petroleum products consumption from 1985 to 1995 by SAPIA. It shows great growth in petrol consumption and jet fuel while diesel, paraffin, fuel oil and LPG consumption remained mostly constant over this period.
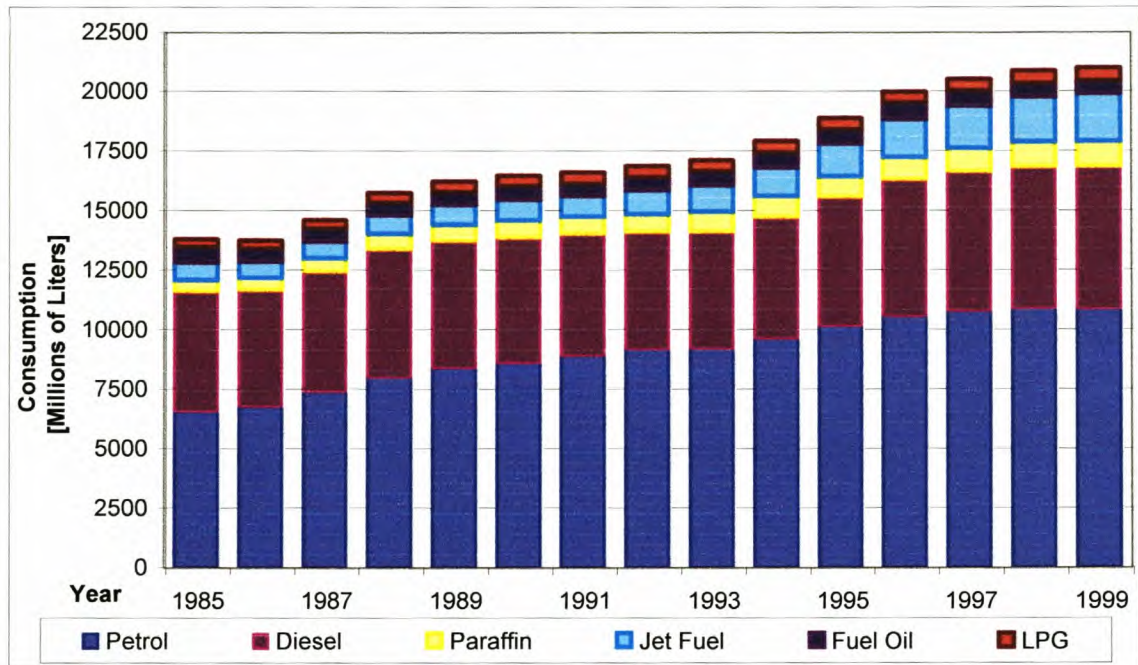
Figure 2.6 Consumption of petroleum products in South Africa from 1985 to 1999 (SAPIA, 2000)

The Mossgas project produces about 10 000 barrels of petroleum products per day from natural gas. This inevitably leads to the production of LPG as a by-product in the order of 5 % by volume as represented in the 1994 figures. This amounted to 36 567 cubic meters of LPG and 724 247 cubic meters of oil. The future of the project seems to be uncertain and will depend on new investments to explore other nearby gas fields (Doppengieter and duToit, 1996).

Availability of LPG

With the new political climate in South Africa from 1992 international travel increased dramatically causing the demand for jet fuel to skyrocket as shown in figure 2.7. Jet fuel consumption doubled from 1992 to 1996 and grew by 160 % from 1985 to 1999, while petrol and LPG consumption grew by 60 % during the same period.
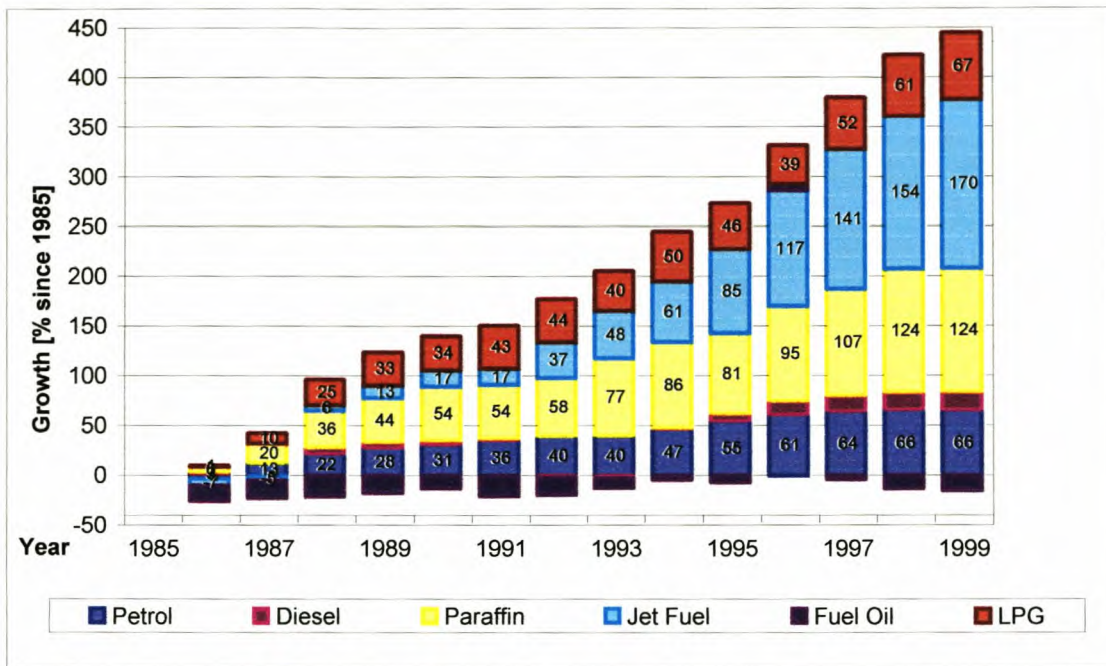
Figure 2.7 Growth in consumption of petroleum products from 1985 to 1999 (SAPIA, 2000)

Figure 2.7 represents the growth in petroleum product consumption from 1985 to 1999. It indicates that LPG consumption showed more or less the same growth, percentage wise, as petrol. The years 1993 to1996 showed a decline in LPG consumption whereafter growth picked up again. This can be ascribed partly to the electrification of houses in previously underdeveloped areas that has decreased the need for cooking with LPG. On average the LPG consumption made up 2.6 % of the total petroleum product consumption during this period.
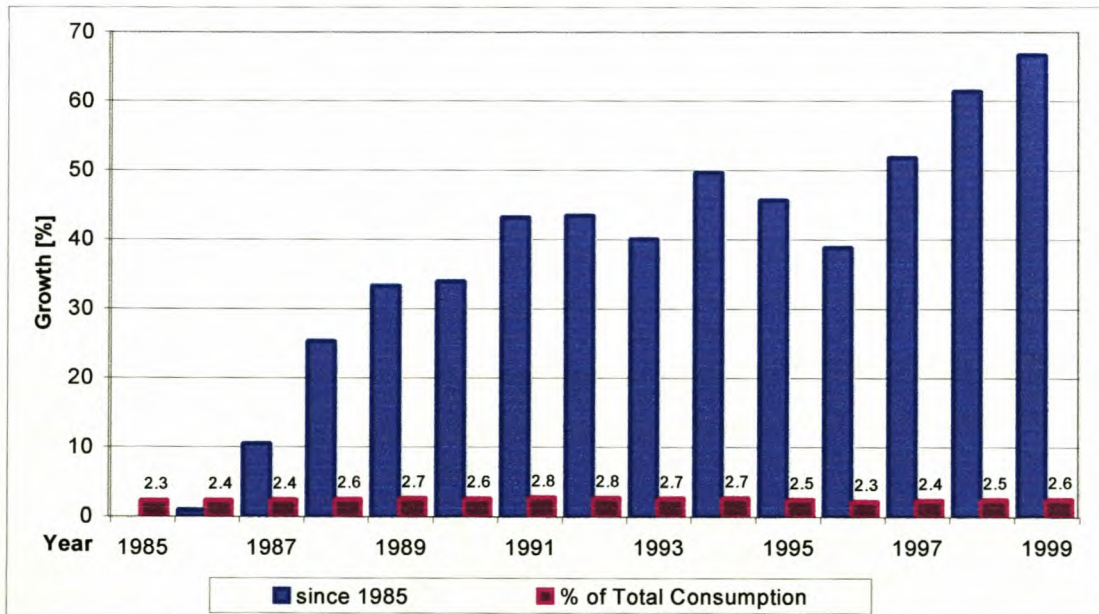
Figure 2.8 Growth in LPG consumption since 1985 (SAPIA, 2000)

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Since LPG is a by-product of crude oil refining this leads to an increasing surplus in LPG. In 1995 when LPG made up 2.5 % of the total market there already existed a surplus of LPG, especially during the summer months. Only during the two coldest months of the year when LPG is used for industrial heating did the demand meet the supply (Searle, 1995).

Automotive use of gas in South Africa

The gas industry in South Africa mainly consists of the Mossgas plant in Mosselbay and the SASOL gas from coal project in Mpumalanga. The only significant market for gas is served by the Gaskor network in Mpumalanga. The methane rich gases produced as by product of the fuel from coal project and natural gas from the Pande gas field in Mozambique are used to power part of the SASOL fleet of vehicles with the intention to convert the complete fleet to run on gas (Shires, 1995).

Forklift operators throughout the country use LPG on a small scale. In most cases no special retail network exists except for large operators that have their own filling stations. Some LPG taxis are operating in the Durban area while LPG taxis in Johannesburg area have been withdrawn from service.

Governing bodies and standards

The Mineral and Energy Policy Center advise government on energy matters. The MEPC is a non-government organization that conducts studies for the government in the mining, electricity and petroleum industries with regards to policy on fuel taxation and pricing. The South African Bureau of Standards lays down standards for LPG use. The LPG association of SA, amongst others, advises them. The LPG association of South Africa interacts with industry to ensure that safety and standards are maintained. This is done by means of workshops for conversion of vehicles to run on LPG as well as the supplying of relevant information to LPG users. Conversion of vehicles to run on LPG has to be carried out by approved personnel. They have to undergo the LPG association training course and need to have converted a certain number of vehicles to run on LPG under the supervision of certified personnel from the LPG Association of South Africa.

2.4 LPG and the environment

2.4.1 Exhaust emissions with LPG as fuel

The chemical composition of LPG makes it a fuel that produces 13 % less CO_2 than petrol or diesel for the same amount of energy produced (Gerini and Monnier, 1996). It has a higher H/C ratio than heavier fuels such as petrol or diesel so that more water is formed relative to carbon dioxide. This makes LPG a low producer of greenhouse gas, as can be seen from emissions tests on modern passenger vehicles in figure 2.9, which show the CO_2 emissions with LPG fuelling on average 15 % less than that of petrol.
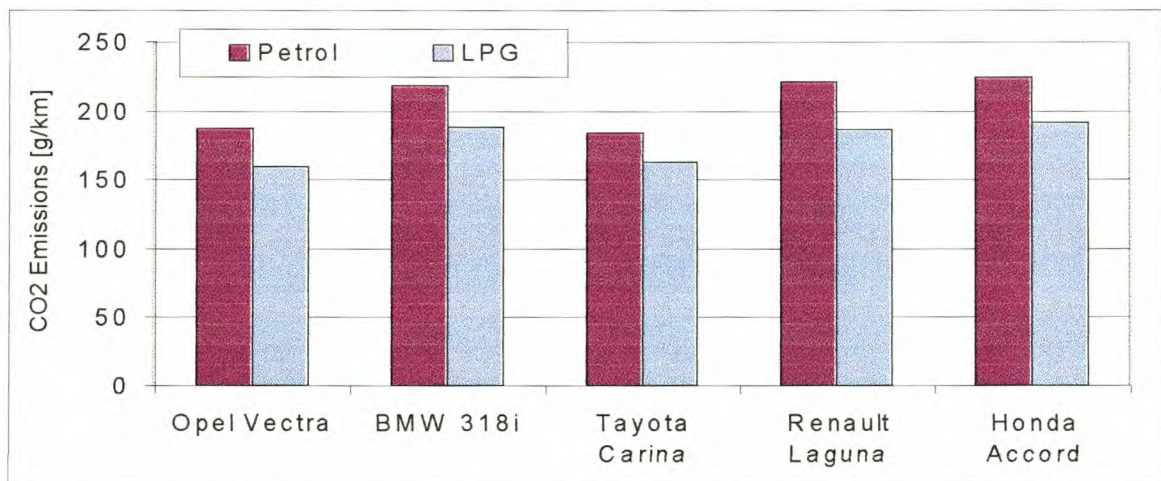
Figure 2.9 CO2 Emissions of petrol versus LPG of five cars meeting 1996 European emissions specifications (World LPG Association, 1995)

The gaseous nature of LPG ensures excellent mixing with air, thus aiding complete combustion. This reduces the presence of partially combusted fuel or hydrocarbons in the exhaust emissions. This is reflected by figure 2.10 which shows how LPG-fuelled vehicles on average produce 12 % less hydrocarbons than petrol vehicles.
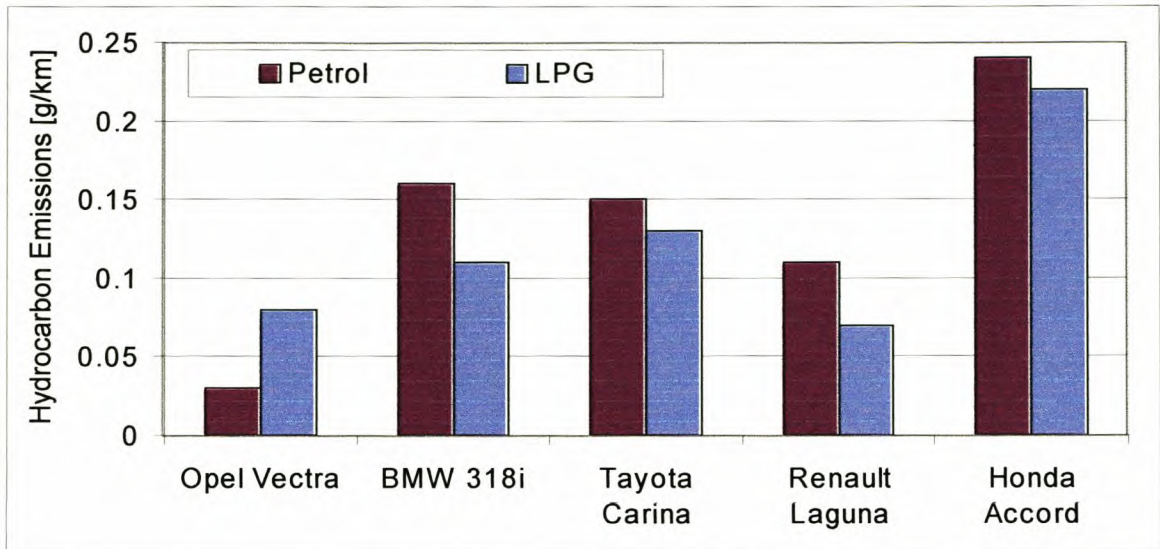
Figure 2.10 Hydrocarbon emissions of LPG versus petrol of five cars meeting European 1996 specifications (World LPG Association, 1995)

Improvements in technology have led to the reduction of NOx emissions from petrol vehicles so that by 1988 these vehicles produced less NOx than their LPG-fuelled counterparts. This in turn neccessitated improvements in LPG-fuelling technology to reduce NOx levels to below 0.5g/km as seen in figure 2.11.
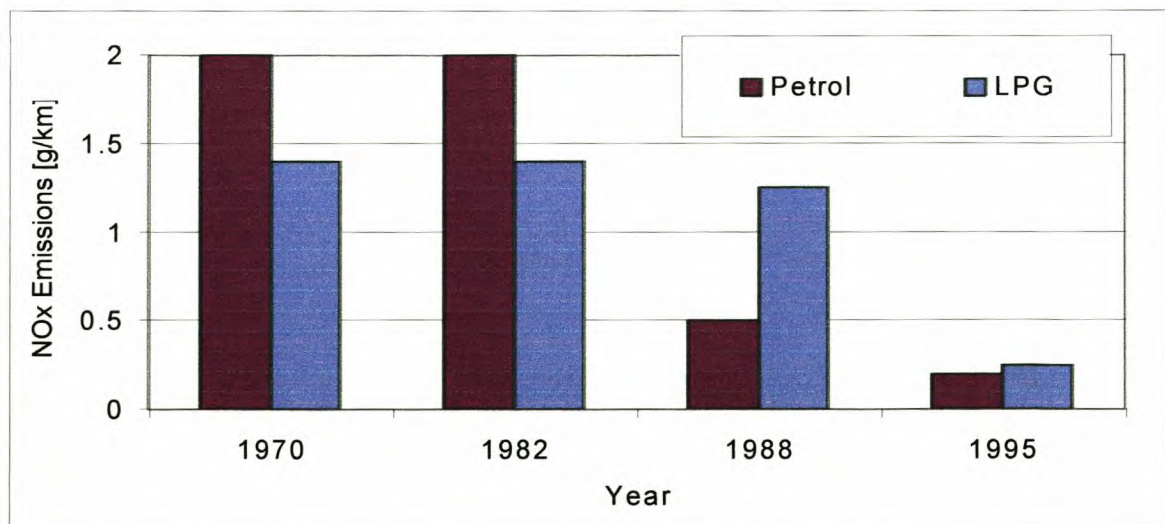
Figure 2.11 Improvement in NOx emissions of LPG and petrol vehicles from 1970 to 1995 (World LPG Association, 1995)

2.4.2　Other environmental advantages

Increased oil life

The low sulphur content of LPG reduces formation of sulphuric acid in engine oil that leads to oil degradation and ultimately damage to engine components. The gaseous nature of LPG significantly reduces wetting of the cylinder walls which robs it of the lubricative film and which can lead to oil contamination by fuel. All these factors contribute to extended oil drain intervals possible with LPG use (World LPG Association, 1995). The use of LPG fuel can extend oil life by 50 % compared with petrol. Increased LPG use will therefore reduce the amount of oil that has to be discarded which, in itself, is a major environmental benefit.

Reduced Noise Pollution

In congested areas and highly populated environments the reduction of noise is an essential part of the improvement in quality of life. Smooth combustion characteristics and therefore reduced vibration and noise is synonymous with gas-fuelled vehicles. The noise-level measurement of a LPG-powered bus engine resulted in 3 dB less noise being produced compared to an identical diesel engine, 78 dB with LPG versus 81 dB with diesel (Bergmann and Busenthur, 1987).

2.4.3　LPG and evaporative emissions

LPG components such as butane are commonly used in petrol to improve cold starting and knock resistance. This enables fuel companies to provide fuel with good anti-knock qualities and high vapor pressures. This is acceptable in cold climates where these qualities are needed to facilitate cold starting but leads to problems in the South African climate. When temperatures in the fuel tank start to exceed the boiling point of the lighter components in the fuel, these components will evaporate. If no evaporative control devices are in place the amount of fuel that gets lost through evaporation can be considerable. To reduce this evaporative loss, most cars are equipped with a pressure release valve which allow a certain pressure to build up in the tank before it is vented to the atmosphere. This is only partially effective. To effectively retain the fuel vapor a carbon canister is required, this device is able to absorb the hydrocarbons in the fuel vapour that would otherwise escape through the tank breather. With the correct equipment and fuel these losses can be contained almost entirely. Most cars in South Africa

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however, are not equipped with such devices so that losses to the atmosphere are considerable. The CAE in Stellenbosch have acquired a SHED and have embarked on a testing program to quantify these losses. Tests carried out with local fuels and vehicles indicate excessive evaporative losses of 7 to 10 times that allowed by European and American standards. This can be attributed to the absence of evaporative control devices in most local vehicles and the presence of volatile components in the fuel. The result is that at least one percent of the petrol sold in South Africa does not reach the engine, but evaporates into the atmosphere. This translates to a loss of 100 million liters of fuel, at the 1995 petrol consumption figure, which translates to a loss of more than R 200 million to the consumer and the economy of the country as a whole (v.d.Westhuizen, 1996).

To adequately address the problem, the manufacturers of new vehicles need to start implementing evaporative emissions controls in their fuel supply systems. This however, is only part of the solution. Fuel specifications need to be altered to limit the components with low boiling points, such as butane, so that fuels will be less prone to evaporation. This will inevitably lead to a greater surplus of lighter fractions, especially butane, so that the market for these fuels, typically LPG, will have to grow to be able to absorb the increased supply.

2.5 Economics of automotive LPG

According to the world LPG association South Africa is identified as one of the countries where automotive LPG use will be the result of industrial or economic considerations. This is mainly due to the lack of emissions regulations that could otherwise be an incentive to use LPG. This situation may change soon due to increased pressure to implement emissions legislation of some kind. It indicates that LPG is recognized as economically viable for South Africa by this body. The availability of LPG, market acceptance, technical support, government involvement and infrastructure for its distribution will determine what market penetration can be achieved (World LPG Association, 1995).

2.5.1 Vehicle conversion cost

Conversion costs of a light commercial vehicle at 1997 prices amounted to R5000 (ADC Engineering, 1997). Minimum expected conversion costs for a heavy duty LPG conversion such as a bus is estimated at R15 000-20 000. For calculations see Appendice A2.

2.5.2 Fuel consumption of LPG fueled vehicles

LPG has the highest energy content per mass of all the fuels used in the automotive sector. Energy content per volume is 65 % and 71 % that of diesel and petrol respectively. This is reduced in practice by the fact that LPG tanks can only be filled to 80 % of their volume to allow for expansion due to temperature variation.

Table 2.3   Fuel consumption for a city bus, LPG versus other fuels (Bergmann and Busenthur, 1987)

	Fuel Consu	Tank Volume	Tank Weight	Consumption		Competitive Cost	
	[l/100km]	[liter/100km]	[kg/100km]	[rel/Diesel,	[rel/petrol]	[rel/Diesel %]	[rel/Petrol %]
Diesel	45	45	50	1.00	0.71	100	140
Petrol	63	63	65	1.40	1.00	71	100
Methanol	100	100	105	2.22	1.59	45	63
CNG	175	175	220	3.89	2.78	26	36
Propane	90	112	80	2.00	1.43	50	70
n-Butane	75	93	80	1.67	1.19	60	84
LPG	82.50	102.50	80.00	1.83	1.31	55	77

Table 2.3 shows average values of fuel consumption for a city bus type 0305 as reported by Bergmann and Busenthur (1987). The LPG values on the bottom line of table 2.3 is for a propane/butane mixture of 50/50 to give properties representing that of a average mixture of commercial LPG. Fuel consumption, tank weight and tank volumes are compared graphically in figure 2.12 using the values given in table 2.3.

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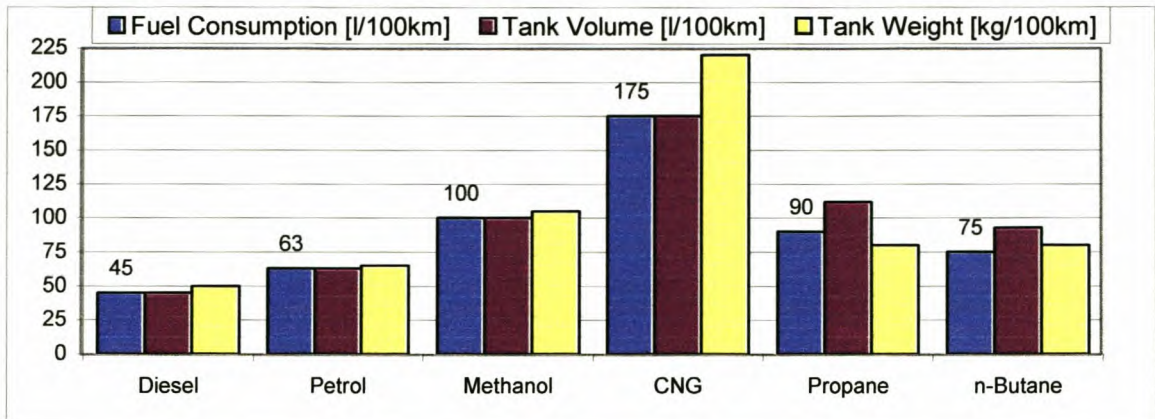
Figure 2.12 Fuel consumption and tank requirements for different fuels (Bergmann and Busenthur, 1987)

For a 50/50 mixture of propane and butane the fuel consumption will be 1.82 times that of diesel requiring a tank that has 2.28 times the volume of the diesel tank at 1.6 times the weight of the aforementioned tank. Compared to petrol, LPG will consume 30 % more fuel by volume requiring a tank that has 63 % more volume weighing 23 % more than the petrol tank. (Bergmann and Busenthur, 1987)

Required cost of LPG to compete with petrol and recover installation costs

Figure 2.13. shows at what cost gaseous fuels have to sell per volume to be able to provide the same running cost as petrol and diesel.
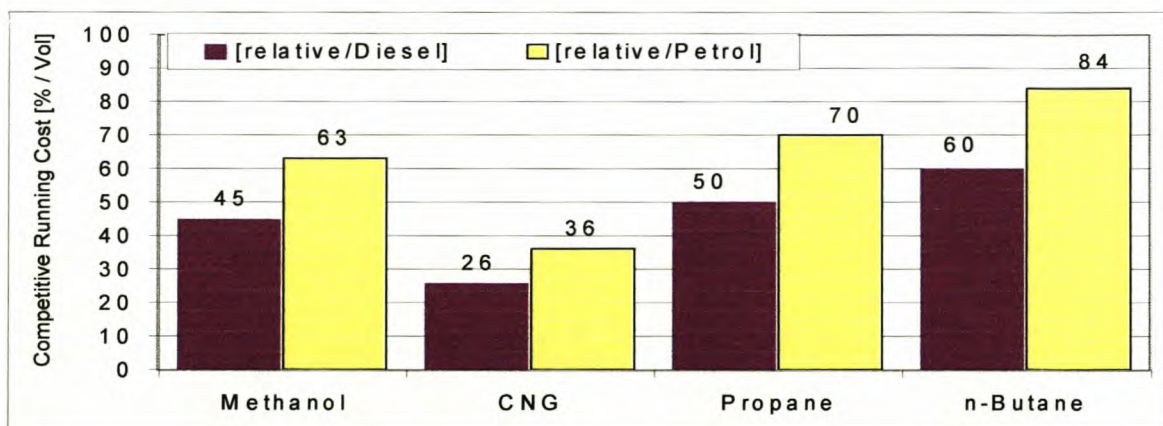
Figure 2.13 Competitive running cost of alternative fuels relative to petrol and diesel (Bergmann and Busenthur, 1987)

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From this information the competitive running cost per volume for a 50/50 mixture of propane/butane will be 55 % of the diesel price and 77 % of the petrol price. At the November 2000 fuel prices this means LPG has to sell at R 2.80 per litre to compete with petrol. At the time of writing LPG sold for R 2.31 bulk which makes it feasible to use LPG as fuel in light passenger vehicles.
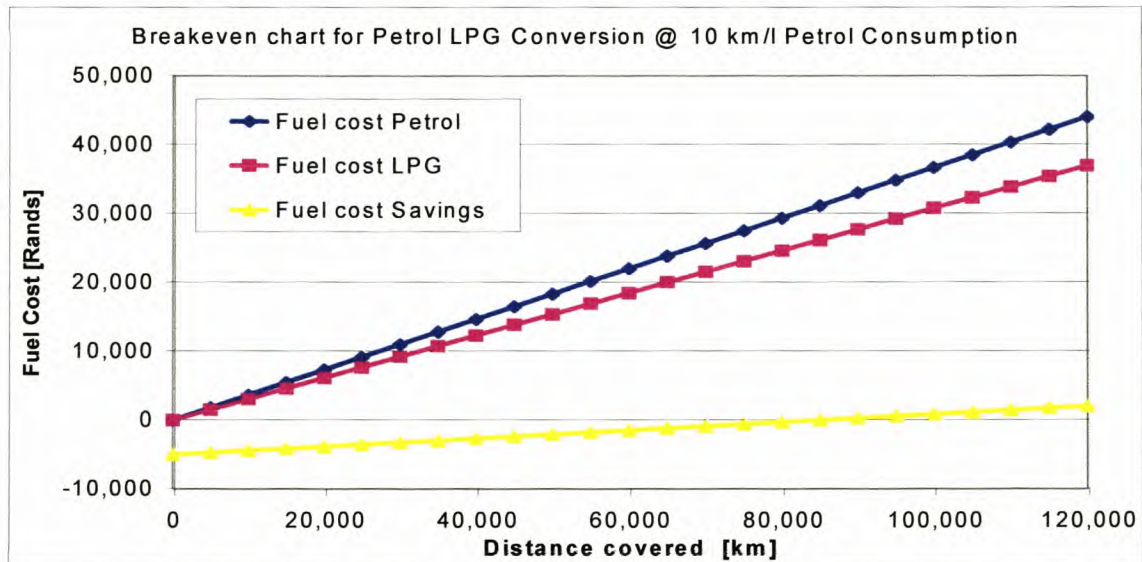
Figure 2.14 Recovery of conversion cost with LPG-fuelled petrol vehicle

Figure 2.14 illustrates the breakeven distance for a petrol vehicle converted to LPG to recover the installation cost of R 5000. The installation cost can be recovered in 85000 km if the best current bulk price of LPG is used, which translates to LPG cost of 60 % the cost of petrol per litre. This does not compare favorably to other countries in the world such as the Netherlands where installation costs can be recovered within 22000 km, partly due to higher fuel costs and a greater price differential between petrol and LPG.

LPG versus diesel

When LPG has to compete with diesel it has to sell at 55 % of the diesel Price or R2.01 per litre, compared to the November 2000 diesel price of R3.67 per litre. This is not the case and it is therefore economically unviable to convert a diesel-fuelled vehicle to run on LPG. This is mainly due to the high efficiency of diesel engines that run at higher compression ratios than petrol or LPG engines. Figure 2.15 shows the cost of running a heavy-duty bus on LPG instead

of diesel. At a diesel consumption of 2 km/litre and with current fuel prices (Nov 2000) it will cost 26.5 cents/km more to run on LPG than diesel, excluding the cost of conversion, that is estimated to be at least R 20000. The details used to calculate the costs can be seen in table 3.3. It is therefore not viable to convert heavy-duty vehicles to LPG operation only.

Figure 2.15 Running cost of LPG /diesel conversion

One other option that remains is to maintain the original diesel engine and add LPG fuelling. In this way 20-25 % of the energy content of the diesel fuel can be replaced by LPG. For such an engine the running cost is such that with the current fuel prices it may be possible to recover the cost of conversion amounting to R 15000 within 150000 km, as shown in figure 2.16.

Figure 2.16 Diesel LPG bi-fuelled engine running cost

For the detailed calculations used for conversion and running cost see Appendix A2.

As far as specific fuel consumption is concerned LPG compares favorably with petrol (lower or similar SFC) due to its higher energy content per mass. Compared to diesel however, higher SFC values may be expected with LPG operation due to the limited compression ratio.

2.5.3 The price of LPG in South Africa

Current Prices

Current fuel prices (December 2000) gives LPG a price advantage of 40 cents per litre cheaper than petrol. At this price difference the recovery distance on a light commercial vehicle is approximately 85000 km which does not make it very attractive to convert to LPG in the short-term.

LPG could however have a greater cost advantage over diesel and petrol fuels, since it is both cheaper to produce and is not taxed like diesel and petrol, of which 45 % of the price is owed to tax. It should therefore be possible to negotiate lower prices as the availability and sales volume of LPG increases once a suitable dispensing network can be put in place.

Changes in Supply and Demand

In 1995 LPG was already in oversupply during the summer months; in effect 10 out of 12 months of the year when it is not used for industrial heating. Subsequently the use of all the

of diesel. At a diesel consumption of 2 km/litre and with current fuel prices (Nov 2000) it will cost 26.5 cents/km more to run on LPG than diesel, excluding the cost of conversion, that is estimated to be at least R 20000. The details used to calculate the costs can be seen in table 3.3. It is therefore not viable to convert heavy-duty vehicles to LPG operation only.

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Breakeven chart for LPG assisted Diesel Engine

Figure 2.16 Diesel LPG bi-fuelled engine running cost

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In 1995 LPG was already in oversupply during the summer months; in effect 10 out of 12 months of the year when it is not used for industrial heating. Subsequently the use of all the

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crude-derived products have increased, excluding LPG, that has seen a decrease relative to other fuels. This is ascribed to the electrification of houses that previously had to rely on gas for cooking. As living conditions improve this trend can be expected to continue. Future stricter fuel specifications may force fuel manufacturers to limit the butane content in petrol thus increasing the further the availability of LPG components. Further exploration of gas fields in Namibia, the Pande gas field and Bredasdorp basin may contribute to the increased availability of LPG as a by-product.

2.5.4 Cost of infrastructure for LPG distribution

Road Transport

LPG may be transported by road tanker in a similar way to petrol and diesel. However, the need of a pressurized container will lead to an increase in the cost of transportation.

Filling stations

It may prove difficult to incorporate LPG facilities within existing petrol and diesel filling stations because of the distance that needs to be maintained by law between the filling station and the storage facility for LPG.

Cost of a LPG filling installation

Due to the relatively low pressures needed to store LPG an LPG filling station costs about one third of a natural gas filling station e.g. US$ 65 000 for a LPG filling station compared to US$ 200 000 for a CNG filling station (World LPG Association, 1996).

2.5.5 Government policy

A clear and predictable government support policy needs to exist for an alternative fuel to enter and compete successfully within the automotive fuel market. Since government involvement is so important it is wise to see what governments are doing across the world to encourage the use of alternative fuels. Various incentives exist to encourage the use of alternative fuels. These can be either purely economic or a result of environmental pressures such as emissions legislation.

The most common way to encourage the use of alternative fuels worldwide is by applying different levels of taxation for different fuels. For LPG these vary from 0 % (Australia) to 31 % (Netherlands) of the tax applied to petrol.

Some other measures used by governments are:

- The production of mono-fuel LPG vehicles by OEM's e.g. 90 % of taxis in Japan are LPG powered. Stringent NO_x limitations necessitate the use of alternative fuelled vehicles (Japan)

- No go zones for non LPG vehicles during days of heavy pollution (Greece and Italy)

- Limit LPG use to public transport vehicles (Greece and Spain)

- Maintaining a consistent LPG tax policy with a five-year advance warning to policy changes providing a LPG selling price at 50-60 % less than gasoline and diesel. (Australia).

- LPG allowed for buses and taxis only and selling price for automotive LPG at one third that of gasoline (South Korea).

- Providing special road taxes and parking privileges for alternative fuelled vehicles.

In South Africa emissions legislation has not yet been implemented and therefore no such driving force currently exists. Vehicle emissions are a growing concern in our cities as proven by the brown haze study which was done for the Cape Town area, which indentified vehicle emissions as the main cause of the brown haze (Wicking de Villiers and Dutkiewicz, 1997). A proposal has been made by NAAMSA members to implement emissions regulation EU1 for non-entry level passenger vehicles by the year 2003. This implies a significant reduction in the emission of CO, which is more easily obtained with the use of gaseous fuels.

2.6 Technical information on LPG

2.6.1   Composition of LPG

Liquefied petroleum gas or LPG forms part of the light hydrocarbon fuels that are a product of oil and gas mining. It consists mainly of mixtures of propane (C_3H_8) and n-butane (C_4H_{10}). In most countiries where LPG is used as automotive fuel a special specification exists for automotive LPG, usually with high propane content to prevent freezing in cold climates and to facilitate cold starting. This may vary from a propane content of 95 % in very cold climates to 40 % propane content in warmer climates such as South Africa.

Table 2.4 Typical composition of South African LPG (Searle, 1997)

Variable	Mean
Density@ 20 Deg C. kg/l	0.55
Methane moll %	0.06
Ethane moll %	3.59
Ethylene moll %	0.05
Propane moll %	31.21
Propylene moll %	12.93
Iso-Butane moll %	25.00
Normal butane moll %	15.12
Butenes moll %	11.31
Pentanes moll %	0.75
Non Volatile residue ul/l	0.00
Vapour Pressure at 37 deg C kPa	896.59
Total Sulphur mg/kg	10.45

The composition of Local LPG may vary between 40 % propane with 60 % n-butane to 60 % propane with 40 % n-butane. This variation is prescribed by the South African Bureau of Standards (SABS). The boiling point for propane and butane is -42°C and -10°C respectively. Cold climatic conditions will require a high propane content to ensure sufficient pressure in the tank while our warm climate can tolerate high levels of butane in the LPG, while still maintaining adequate pressure levels.

2.7 Physical properties

2.7.1 Vapor pressure

Vapor pressure can be calculated using the Antoine vapor pressure correlation for each component. The vapor pressures of mixtures have been calculated as the sum of the partial pressures according to Dalton's Law (Reed, 1986).
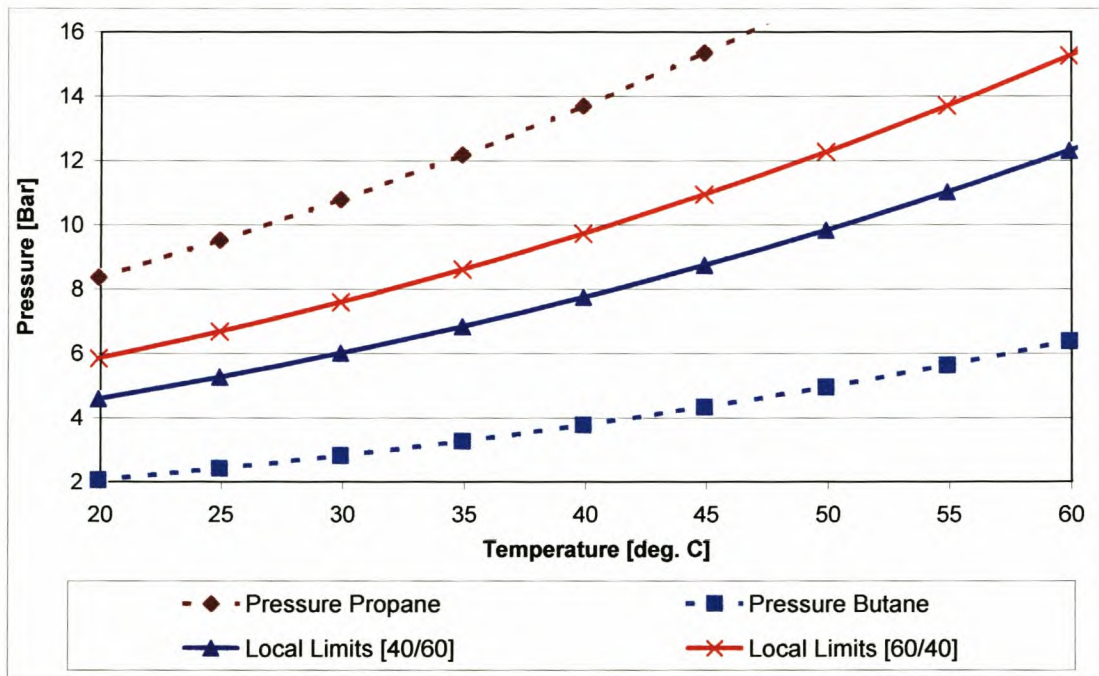
Figure 2.17 Vapor pressures of LPG

Figure 2.17 represents the vapor pressures that will be encountered with LPG as a function of butane-propane mixture and temperature. It is clear that the vapour pressure is very temperature-dependent; the pressure increases exponentially with an increase in temperature. The mixture variations allowed within the specifications allow for a one bar pressure difference on average from one mixture to the next. Vapor pressures can vary between 3bar, and 10bar under temperatures that may occur with normal use. The fuel system therefore needs to be able to handle these variations in pressure.

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Boiling point

The boiling point for propane and butane is -42°C and -10°C respectively.

Latent heat of evaporation

The latent heat of evaporation indicates how much heat needs to be added to LPG in liquid phase to transfer it to vapor phase at a specific temperature. This energy has to be supplied either by the engine coolant when an evaporator is used or by inlet air and contact with engine parts in the case of liquid phase injection. At room temperature it amounts to about 340kJ/kg for propane (Lom, 1982).

To supply a engine delivering 100kW requires approximately 330kW of fuel energy due to the fact that only one third of the energy supplied by the fuel is recovered as useful work. This will requires a minimum flow of 25.5kg/hr of propane requiring 2.4kW of heating to evaporate the propane. The heating requirements to evaporate the fuel are therefore about 2.4 % of the engine output and can easily be extracted from the engine coolant. For butane these values will be higher so that the evaporation of commercial LPG will require in excess of 2.4 % of the engine output. In the case of LPG injection in the liquid phase this energy will be extracted from the inlet air and warm engine components. If the air alone should be used to evaporate the fuel it would lead to a decrease in air temp of around 24°C. This can be useful in improving engine performance, since a decrease in inlet air temperature increases charge density and the amount of fuel that can be burned in an engine, thereby increasing engine power output.

2.7.2   Combustion of LPG

The stoichiometric combustion of propane and butane in air on a volume basis is given below.

Combustion of propane

$$1m^3C_3H_8 + 23.8m^3air \rightarrow 3.m^3CO_2 + 18.8m^3N_2 + 4m^3H_2O + 93.9MJ \qquad \text{Equation 2.1}$$

Combustion of butane

$$1m^3C_4H_{10} + 31m^3air \rightarrow 4.m^3CO_2 + 24.4m^3N_2 + 5m^3H_2O + 121.7MJ \qquad \text{Equation 2.2}$$
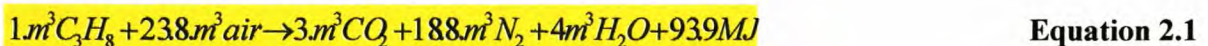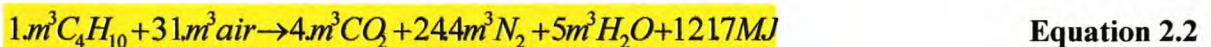
North American Combustion Handbook (Reed, 1991)

The flammability limits in % volume in air at 15°C , 1 atm are illustrated in figure 2.18.
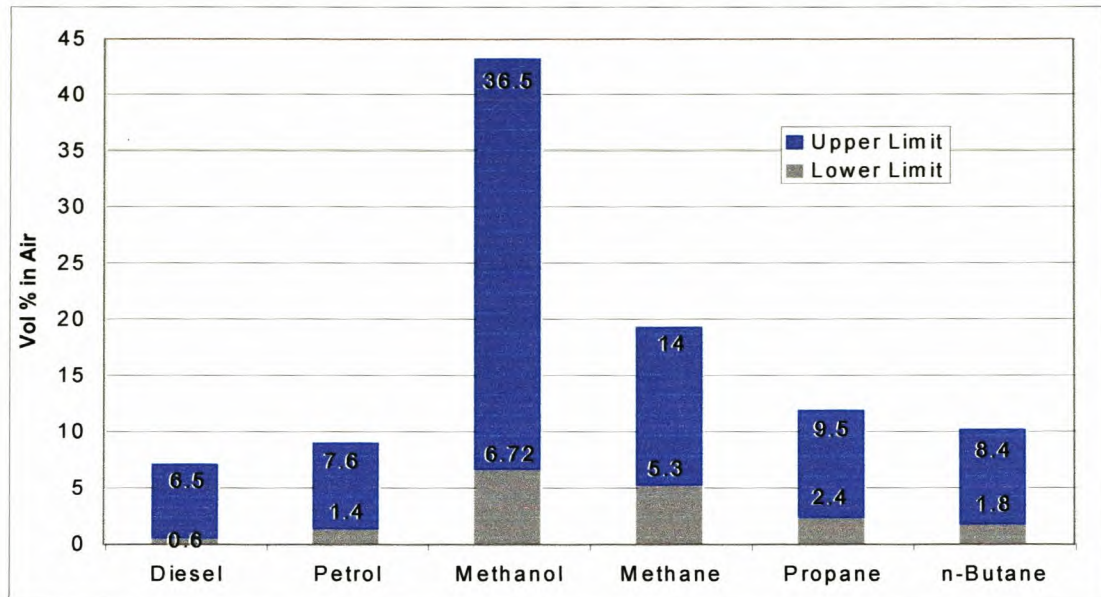
Figure 2.18 Limits of flammability for different fuels (Bergmann and Busenthur, 1987)

The limits of flammability for n-butane and propane indicate a wide tolerance within which combustion can be achieved with the stoichiometric values given in table 2.5.

Table 2.5 Stoichiometric air/fuel ratios for LPG components (Bergmann and Busenthur, 1987)

Stoichiometric Air/Fuel Ratios		
Fuel	mass %	vol %
Propane	15.64	4
n-Butane	15.43	3.2

In order to anticipate the difference in combustion that can be experienced with different alternative fuels it is helpful to have information on flame speed and flame temperatures of different fuels. In a study done by Bergmann he determined the results that are illustrated in figure 2.19.

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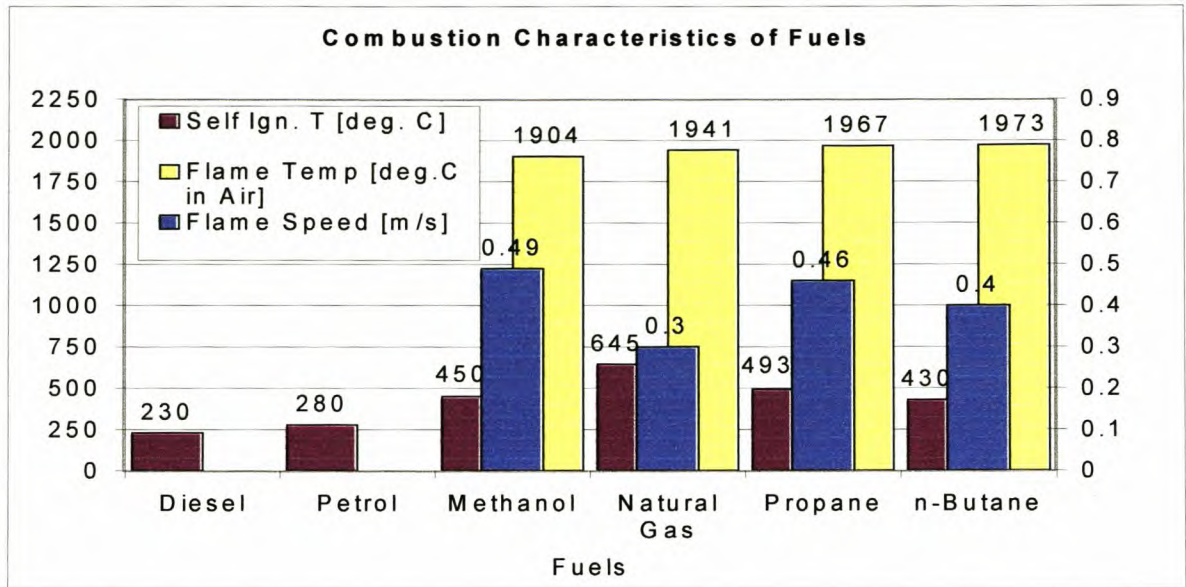
Figure 2.19 Autoignition temperatures and flame speeds of alternative fuels (Bergmann and Busenthur, 1987)
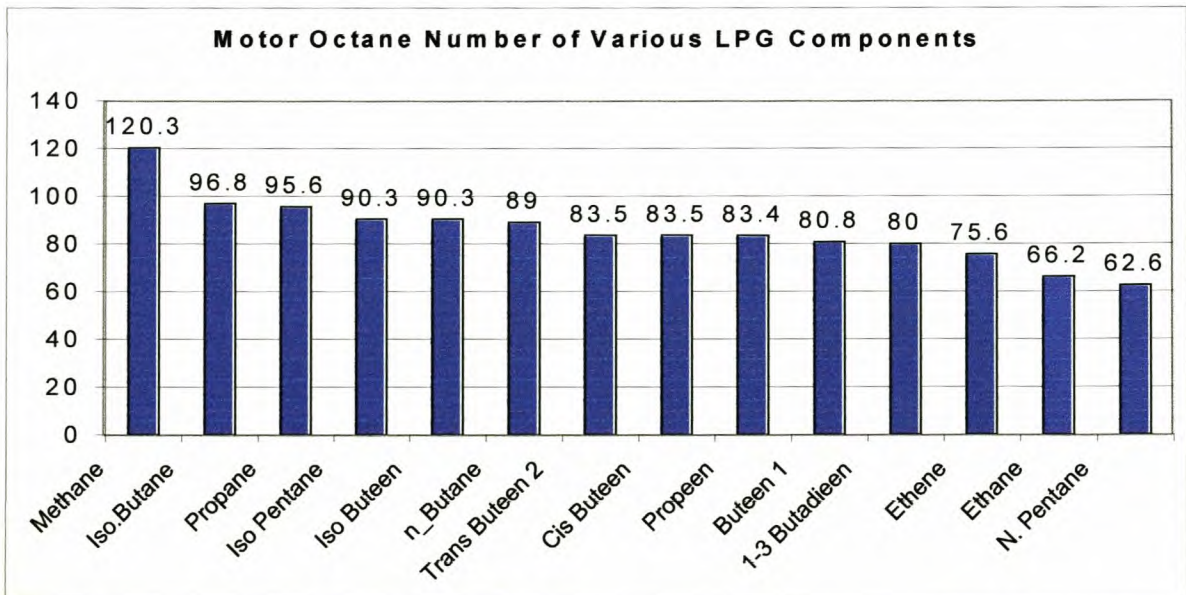
Figure 2.20 Motor octane numbers of various LPG components (Jordaan, 1998)

Figure 2.20 shows the octane numbers for various LPG components. All the components found in LPG have good RON (Research Octane Number) levels, while the saturated hydrocarbons also have good MON (Motor Octane Number) levels. The unsaturated hydrocarbons, those with double bonds, however, have poor MONs with a sensitivity of 15-20 between RON and

MON. The MON levels of between 80 and 85 indicate that while low load conditions will see satisfactory performance, severe knock can occur under high speed and load conditions (Owen and Cowen, 1990). If a LPG evaporator is used there will be no evaporative cooling and the MON will dominate. This seriously affects the maximum allowable compression ratio that can be used, thus lowering engine efficiency. Some manufacturers of LPG conversion kits quote a maximum compression ratio of 9.5:1 due to the sensitivity of these components (Jordaan, 1998). This is however not a problem in engines running on both LPG and petrol since the knock resistance of petrol is lower than that of LPG, so that it is the petrol properties that limit the allowable compression ratio and not the LPG. For information on fuel properties see Appendix A1.

2.7.3 The effect of LPG use on engine efficiency

The maximum attainable engine efficiency can be calculated in the following manner:

$\eta = \dfrac{Work.Out}{Work.In} = 1 - \dfrac{Qout}{Qin}$ The work out and work in for the Otto cycle (Ferguson, 1986) is

related to the compression ratio in the following manner: $\eta = 1 - \dfrac{1}{r^{\gamma-1}}$ **Equation 2.3**

Here η is the efficiency, r the compression ratio and γ the ratio of specific heats. The results of this calculation and that of the efficiency of the diesel cycle can be seen in figure 2.21. The ratio of 2.8 was chosen for the diesel cycle calculations because it is typical of the displacement changes in an engine with a 9.9:1 compression ratio at 50 degrees from TDC, that conservatively represents a 60 to 70 degree combustion duration, if combustion is initiated at 10 to 20 degrees before TDC. This is longer than most typical combustion duration's and therefore should give a conservative estimate of efficiency.
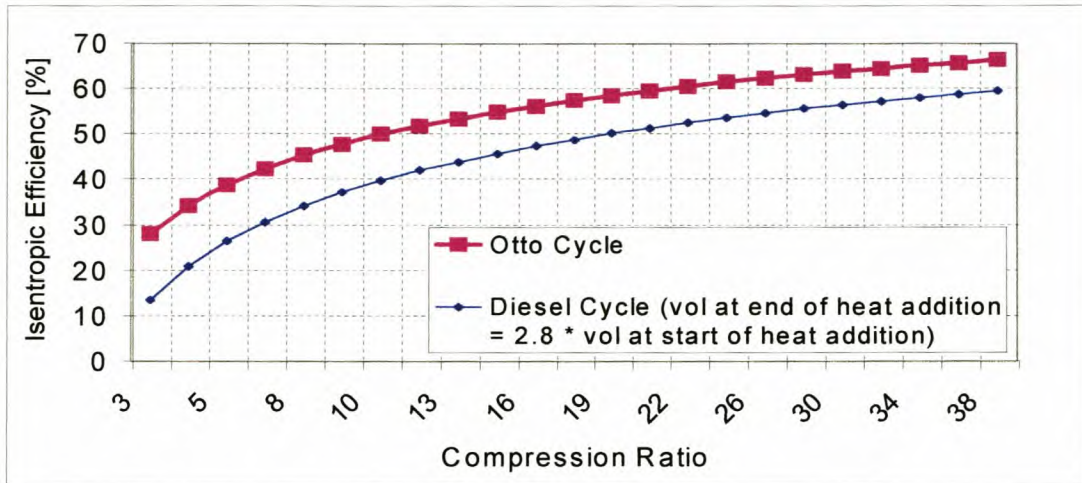
Figure 2.21 Effect of Compression Ratio on Maximum Attainable Engine Efficiency

The real process in an engine is better represented by a combination of the Otto and diesel cycle so that the real achievable efficiency will fall between the two lines in figure 2.21 if heat loss is ignored. Another factor, other than compression ratio, that plays a role in engine efficiency, is combustion duration. Typically, gaseous fuels have a longer combustion duration at high engine speeds than petrol and diesel. The relative efficiency given by Bergmann and Busenthur (1987) in figure 2.22 therefore does not correspond directly to the allowable compression ratio, as one would expect from the calculations shown in figure 2.21.
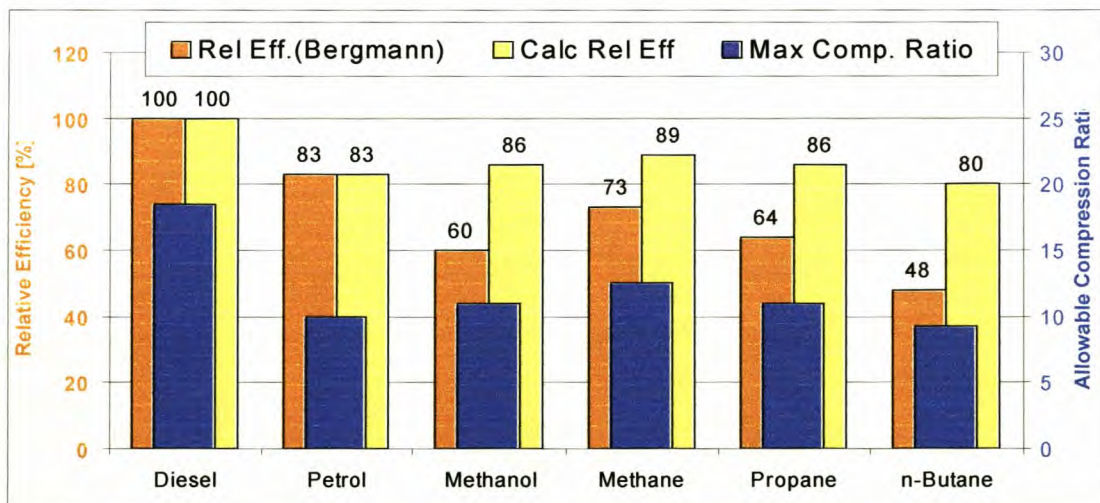
Figure 2.22 Usable compression ratios and efficiency of alternative fuels (Bergmann and Busenthur, 1987)

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It is interesting to note that for liquid fuels the calculations exactly match Bergmann and Busenthur's data. For gaseous fuels however, the calculated values are higher than his findings. This can partially be ascribed to a difference in combustion mechanisms between gaseous and liquid fuels. This indicates that gaseous fuels typically have a longer combustion duration than liquid fuels. This phenomenon is studied in some more detail in chapter 6.

2.7.4 Effect of LPG on engine durability

Two parameters that are directly linked to engine durability are the Break Mean Effective Pressure or BMEP and exhaust temperature. Bergmann compared alternative fuels to diesel, his findings are shown in figure 2.23.
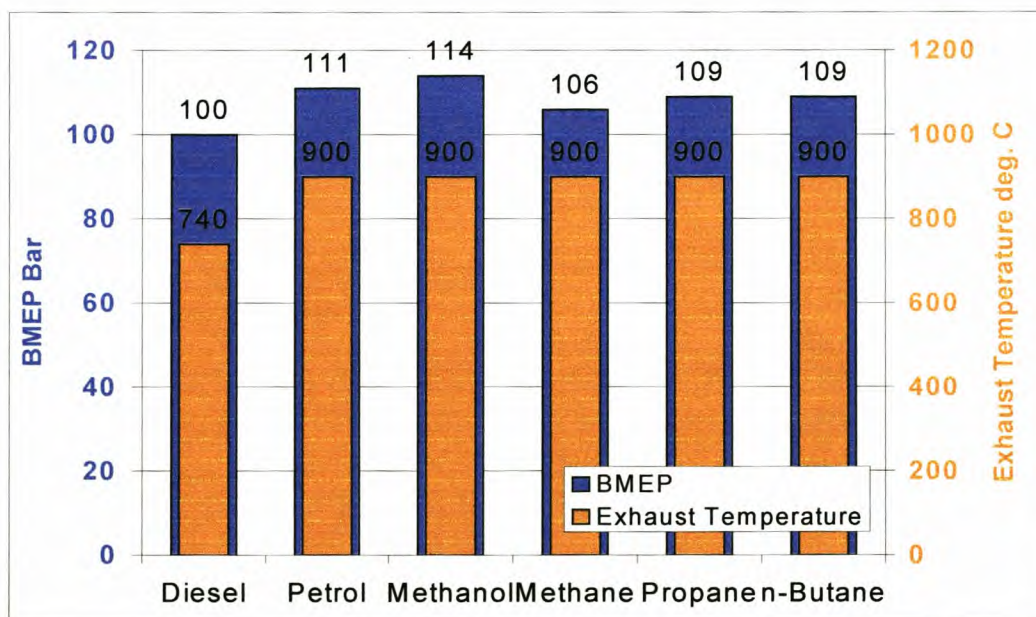
Figure 2.23 The Effect of maximum allowable compression ratio on BMEP and exhaust temperatures

The BMEP is a indication of the mechanical load on the engine as a result of the pressures experienced during compression, combustion and expansion and is therefore a useful parameter in comparing wear and fatigue rates on engine components. Exhaust temperatures are a good indication of the thermal loads and fluctuations the engine is subjected to, since these create thermal stresses and influence component strength and durability.

From the study by Bergmann it becomes clear that exhaust temperatures are on average 150°C higher with alternative fuels than with diesel. LPG displays similar exhaust temperatures as petrol and BMEP is 2 % lower. This indicates that LPG fuelling should not have a negative effect on the life of a petrol engine.

The only detrimental factor to consider when using LPG in a petrol engine, is the possibility of valve seat wear in older engines designed to operate on leaded fuel, since LPG is lead free, and therefore provides no lubrication for the valve seats. This was studied in some detail by Wiles (1965). Since modern engines are designed to run on unleaded fuel, running on LPG will not cause deterioration of the valve seats.

Compared to a diesel engine there seems to be some discrepancy as to the effect of LPG on engine life. The higher BMEP shown in figure 2.20 would suggest higher mechanical loading and consequently, one would expect a reduced engine life. Measurements on converted diesel engines running on LPG show that wear rates are up to 25 % lower on the sleeves (Bergmann and Busenthur, 1987). The reason for this may be found in the combustion process and in the difference in compression ratios. Combustion with LPG is more gradual than is the case with diesel and compression ratios with LPG fuelling is much lower than with diesel fuelling, resulting in lower peak pressures. This is confirmed by measurements of noise levels of LPG engines that indicate lower noise than their diesel counterparts in the order of 3 dB, 78 dB with LPG versus 81 dB with diesel (Bergmann and Busenthur, 1987).

Oil requirements of LPG-fuelled engines

The oil requirements of a LPG-fuelled engine is similar to that of a petrol-fuelled engine, so that when LPG is used in a petrol engine the same oil that is used for petrol operation can be used.

The oil requirements on a LPG engine is considerably different from that of a diesel engine due to differences in fuel composition and increased combustion temperatures with LPG. Oil with low sulfated ash content has to be used since the ash leads to the formation of deposits that can be harmful to the engine (Automotive Fuels Handbook, 1991).

2.8 Conclusion

From the changes in South African energy consumption it becomes clear that an oversupply in LPG exists and that it can be expected to increase. This combined with rather lenient fuel specifications and old vehicle technology are factors that contribute to evaporative losses of fuel per vehicle, which are orders of magnitude higher than those allowed in Europe and America. This is costing the country in excess of R200 million per year while at the same time it contributes to poor air quality. As vehicle emissions are identified as the major source of pollution in some major cities, implementation of some form of emissions control is becoming a necessity. All the above factors indicate that it is possible for LPG to play a bigger role in automotive applications in South Africa. The quality of petrol will benefit from such a move, as will air quality, due to both the decrease in evaporative emissions as well as the low emissions emitted by vehicles using LPG.

The supply of LPG remains limited so that it is not expected to capture a large enough share of the market to make taxation thereof a big source of income for government. It is expected to find application in selected areas where vehicles follow a set route, such as public transport.

3 GAS FUEL SYSTEMS

3.1 Dedicated gas engines

A dedicated gas engine is designed to operate solely on either natural gas or LPG. Vehicles equipped with these engines are classified as mono-fuel vehicles, since they use only one fuel. The engine can then be optimized to run at the maximum allowable compression ratio for that fuel. Dedicated gas engines are typically found in the heavy-duty sector of the market because diesel engines are not suitable for gaseous fuel use without major modifications.

3.2 Mixed diesel–gas (dual-fuel) engines

With a mixed diesel-gas engine the design of the original diesel engine remains unchanged, but a gas fuel system is added enabling some of the diesel fuel to be supplemented with gas. The correct amount of gas is mixed into the inlet air and burnt in the cylinder, after diesel injection has initiated combustion. Due to the high compression ratio of the diesel engine the fuel/air ratio of the gas must be kept below the lean limit for spontaneous ignition, since combustible mixtures of gas would spontaneously ignite causing severe knock. This puts a limit on the amount of diesel that can be displaced using this system. The lower limits of combustion in % volume in air for methane and propane are 5.3 and 2.4 % respectively.

3.3 Petrol and gas (bi-fuel) engines

Light duty spark ignition engines lend themselves to gaseous fuel use without major modification. It is therefore common for these vehicles to retain their petrol fuel system with the addition of a gas fuel system that can be either LPG or CNG. The driver can then choose which fuel to use and is not completely dependent on a gas supply network. Figure 3.1 displays a simple installation in which the driver manually selects the fuel that is required by switching between LPG and petrol. Solenoid valves are used to connect the fuel supply of the fuel that is selected. In the case of carburetor vehicles a delay time is sometimes incorporated when

switching from petrol to gas to allow fuel from the float bowl to be used before the gas supply solenoid is activated.
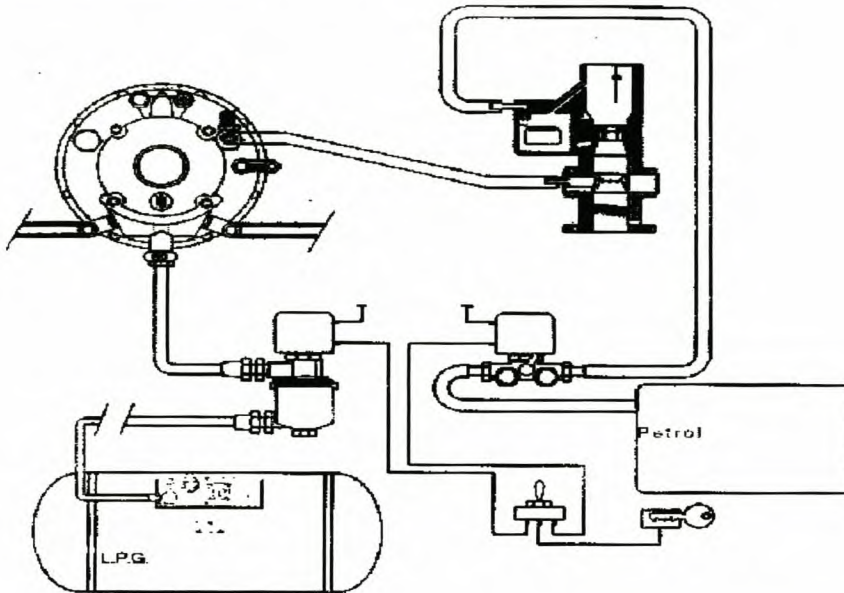
Figure 3.1 Fuel selection with a bi-fuel (petrol/LPG) vehicle (World LPG Association, 1995)

With this arrangement even older engines designed to run on leaded fuel can use LPG fuel, as long as one tank in four is run with petrol to supply the lubrication needed to protect the valve seats (Wiles, 1965). Note that engines designed to run on unleaded fuel will not suffer from valve seat wear when run on LPG.

3.4 Three generations of equipment

Over the years LPG equipment has developed to keep up with the introduction of stringent emissions requirements. This has led to the development of more accurate LPG fuel systems as reflected by the CO of LPG-fuelled vehicles as shown in figure 3.2. High CO emissions indicate a high incidence of incomplete combustion since CO would react with oxygen to form CO_2. This indicates either poor fuel distribution or overfuelling. CO readings are therefore a good indication of the ability of the fuel system to supply the correct amount of fuel and to supply it in such a way that good mixing occurs.

Figure 3.2 shows how the three generations of LPG fuel supply systems have improved the CO emissions from 1970 (generation 1) to 1982 (generation 2) and 1995 (generation 3) of equipment.
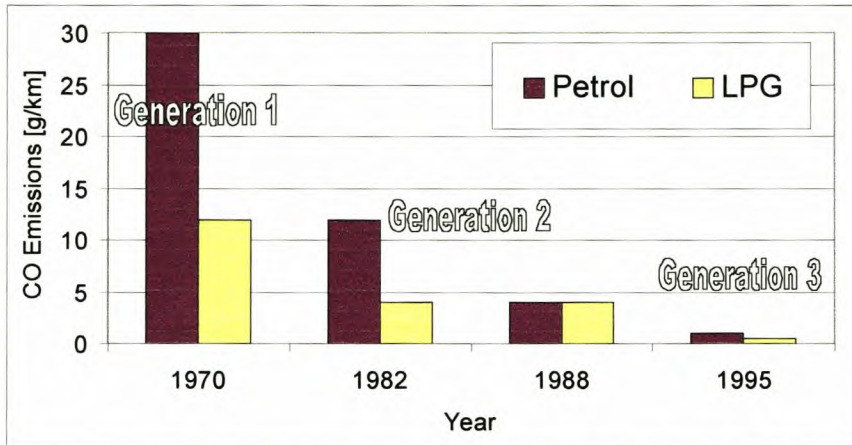
Figure 3.2 CO emissions from LPG fuelled vehicles (World LPG Association, 1995)

3.4.1   Mechanical LPG mixer systems (Generation 1)

This is the most basic system that is suitable in countries where vehicles are not required to meet emission standards. It is an open-loop system without the ability to make adjustments to maintain correct mixtures at all operating conditions. Figure 3.3 shows the components used in a typical generation 1 gas supply system from the tank to the gas mixer.
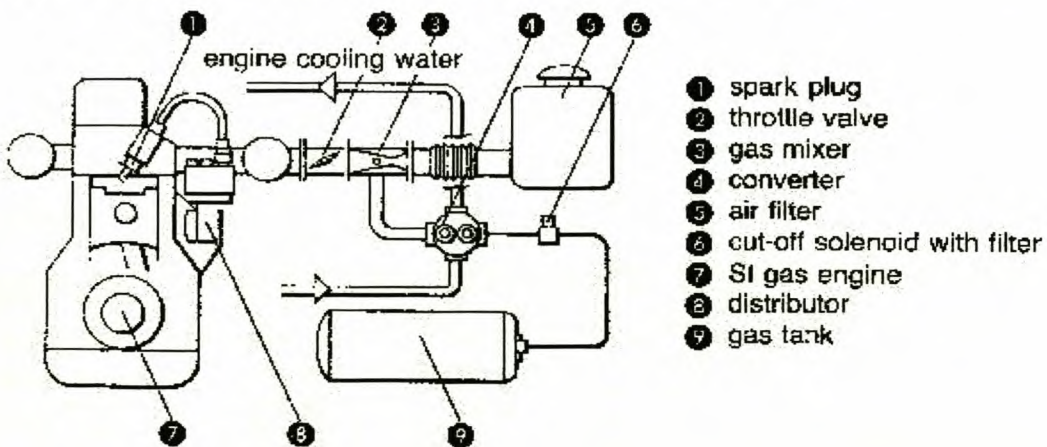
Figure 3.3 First Generation Dedicated LPG Engine (Bergmann and Busenthur, 1987)

3.4.2 Electronically adjustable gas fuel systems (Generation 2)

These systems make use of the basic configuration of the Generation 1 equipment, but the fuel metering is made more precise with the use of electronic control and feedback from the engine to make closed-loop control possible. This has led to a reduction of CO emissions from 12 g/km to 4 g/km in 1988, as can be seen in figure 3.2.

3.4.3 LPG injection systems with self-learning capabilities (Generation 3)

"These LPG systems are microprocessor controlled, self-learning and are without manual adjustment." (World LPG Association, 1995) The third generation of LPG equipment is designed to meet stringent requirements for emissions and fuel consumption. These systems are used in combination with OEM engine management information and require close co-operation with the OEM. The introduction of a self-learning system has major advantages. The system optimizes itself during driving so that the original installer of the equipment only has to make sure that the system is able to function within the requirements of the specific engine, and no time has to be spent mapping the fuel system for each new engine. This significantly attributes to low installation cost since labor is reduced while good performance is ensured since engine aging and other changes are automatically accounted for by the system.

All of the above are discussed in detail in the technical reference paper by the World LPG Association.

3.5 Basic gas fuel system components

Figure 3.4 shows the layout of the equipment used with gaseous fuels such as LPG. It is identical to that used with natural gas, with the exception that natural gas requires two regulators to reduce the pressure from 200bar, instead of the one two-stage regulator used with LPG.
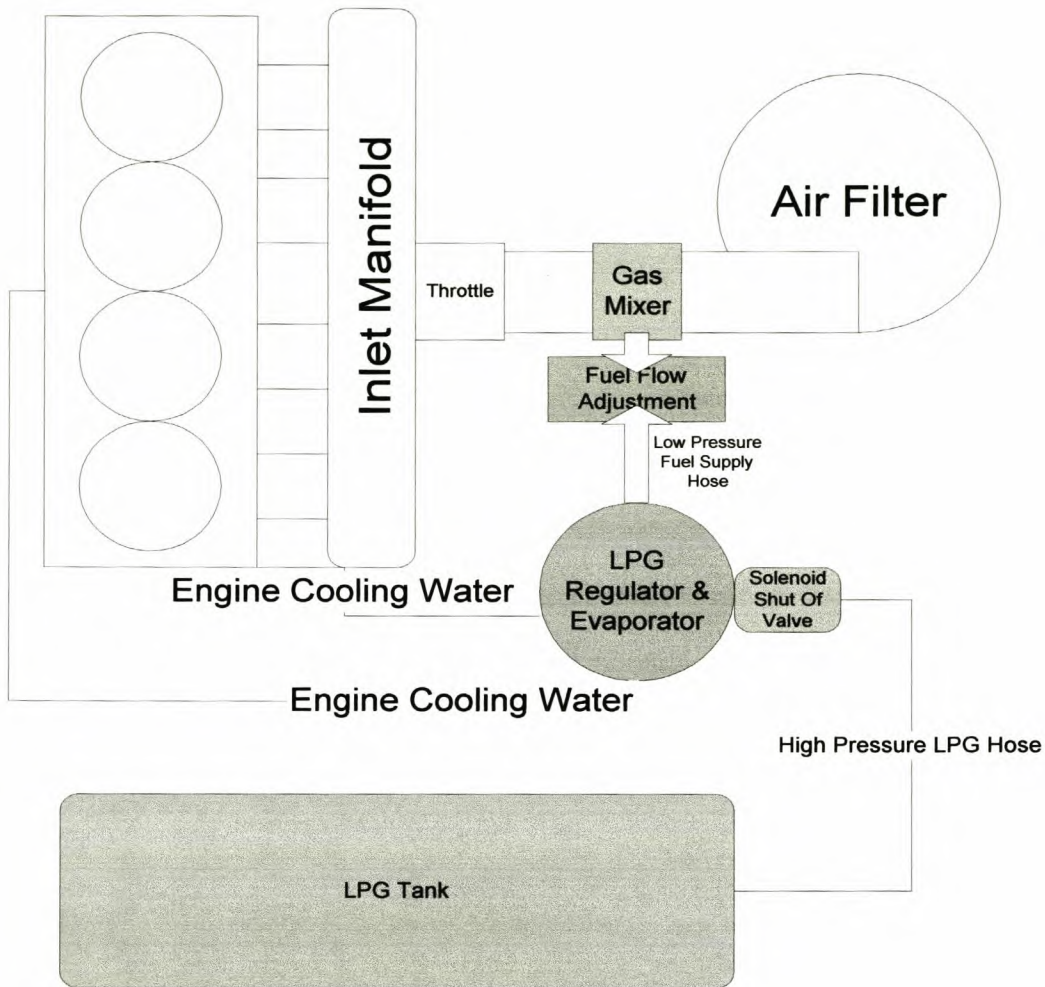
Figure 3.4 Layout of basic LPG equipment

The components used for the LPG conversion are shaded. Fuel is stored at 3-7bar in liquid phase in the LPG tank. From there it flows through a high-pressure hose into the vaporizer/regulator. A solenoid valve acts as a fuel shut-off on the high-pressure side. The vaporizer uses engine coolant to supply the heat needed to vaporize the liquid phase LPG. Gas phase LPG is then inducted into the engine by the gas mixer from the low-pressure side of the regulator.

3.5.1 The LPG fuel tank

a. Filler valve

b. Vapor return valve

c. 10 % outage valve

e. Relief valve

f. Vent to outside of vehicle

g. LPG valve

h. High-pressure hose

k. Fuel gauge
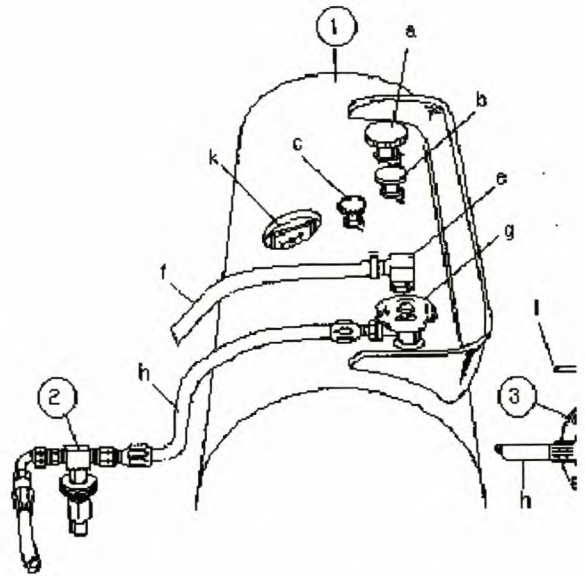
Figure 3.5 an Automotive LPG Tank (World LPG Association, 1995)

The tank has to contain the gas at pressures of up to seven bar, thus approximately 3mm steel is commonly used in the fabrication of a small- to medium-sized tank. The tanks have to comply with regulations TRG 380 regarding pressure vessels. The maximum working pressure is about seven bar but tanks are rated at 18bar and tested to 30bar. Maximum filling is to about 80 % capacity to leave room for expansion. A safety valve is set to open should the pressure exceed 17.2bar. Tank capacities range between 40 to 128 litres for passenger vehicles. (Garret, 1991 and Bosch Automotive Handbook, 1996)

3.5.2 Fuel hoses and fittings

High-pressure LPG fuel lines are either of the steel braided rubber type or seamless 6mm or ¼", 6.2mm copper tubing or stainless steel, especially with natural gas. The European standard is 6mm copper pipe while local suppliers use ¼" copper pipe because most of the local equipment is imported from the United States.

Heavy-duty quick-coupling fittings are used on removable LPG cylinders for forklift truck applications to make the removal of tanks possible, since they are not filled on site but at central

depots. The supply line from the tank and the connection in the tank itself is self-sealing so that once it is removed from its connection on the tank it will not release residual gas that may be either in the tank or supply line.

3.5.3 Gas vaporizer regulator

The vaporizer regulator reduces the pressure of LPG in the tank to a pressure just below atmospheric pressure. This is done in two stages with the first stage being surrounded by engine coolant to supply the energy needed to vaporize the liquid-phase LPG. Some models are also equipped with electric heating to aid cold starting. Common practice in colder countries is to start the engine on petrol and switch over to gas once the coolant has reached operating temperature. In local conditions this is seldom necessary. For good heat transfer the vaporizer is typically manufactured of aluminum and fitted with fins in the area of heat transfer to the LPG after the first pressure reduction. To prevent LPG from leaking from the evaporator the outlet side of modern LPG regulators is also fitted with solenoid shut-off valves, thus shutting off the low-pressure fuel supply to the engine when it is turned-off. Figure 3.6 shows the LPG regulator used on the 12-litre diesel engine conversion.

Figure 3.6 A LPG evaporator / regulator

Liquid-phase LPG enters the regulator from the left (copper pipe). The front part of the regulator is heated by engine coolant (black pipes) to evaporate the liquid-phase gas. Gas-phase LPG is supplied to the engine at low pressure through the steel braided pipes. This supply can be shut-off by the solenoid, illustrated in the top-right corner of the picture.

3.5.4   The gas-air mixer

Two types of gas-mixers exist; fixed geometry and variable geometry mixers. The fixed geometry mixer is the simpler of the two with limited ability to handle variations in airflow. The variable geometry mixer on the other hand, is more complex but able to accommodate a bigger variation of airflow than the fixed geometry mixer.

Variable geometry gas-mixers

With the variable geometry gas-mixer the gas is supplied by the regulator at pressures above atmospheric pressure and the mixer exactly meters fuel by varying the restriction to the gas-flow. Thus, the gas is injected into the airstream at relatively low pressures. This is typical of the IMPCO models where the regulator and mixer form a unit to provide good air/fuel ratios throughout all engine-operating conditions.

Fixed geometry gas-mixers

Fixed geometry gas-mixers rely on induction to supply the correct amount of gas to the engine.

This is probably the most simple type of device for gas-air mixing since it has no moving parts, as the throat size stays the same regardless of the flow that passes through it.

Operation of a fixed geometry mixer

Gas is supplied in the gas phase by a low-pressure regulator at a pressure just below atmospheric pressure. The outlet side of the pressure regulator is connected to the neck of a venturi in the clean air pipe of the engine. The movement of air in the clean air pipe through the venturi creates a vacuum relative to the airflow through it, thereby sucking in varying amounts of gas as the airflow varies.

The theory for operation of a fixed geometry venturi mixer can be derived as follows:

for a fixed control volume the change in density over time plus the sum of the flow out of the boundaries of the control volume must be zero.

$$d\rho / dt + \rho q_{i,i} = 0 \qquad \text{Conservation of mass} \qquad \textbf{Equation 3.1}$$

Assumptions:

Assume steady frictionless incompressible flow without heat transfer over the boundaries, the conservation of mass equation then simplifies to:

$q_{i,i} = 0$ In simple terms this means what goes in must go out or: $V_1 A_1 = V_2 A_2$ **Equation 3.2**

Figure 3.7 shows the control volume for a fixed geometry venturi.

Air enters at D and passes through the throat section at d. From conservation of mass the speed of air at d relative to D can be calculated from the area ratios: $A_1 / A_2 = D^2 / d^2$.   **Equation 3.3**

This gives the following speed ratio: $V_2 / V_1 = D^2 / d^2$ **Equation 3.4**
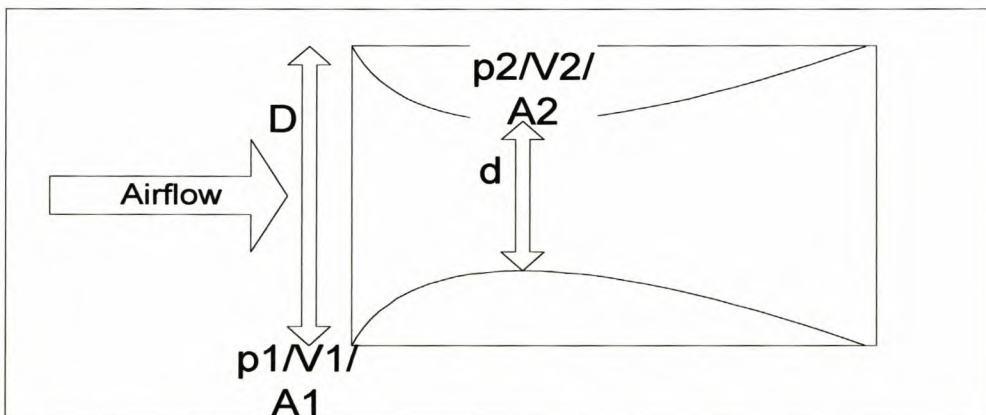
Figure 3.7 Fixed geometry venturi

The vacuum created in the venturi by movement of air through it causes flow of gas into the air stream. The theory behind it lies in the Navier Stokes equation that correlates pressure and velocity for a Newtonian. In the case where the fluid is frictionless and incompressible without work or heat transfer this can be simplified to the following:

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$$\frac{p_1}{\rho_1} + \frac{1}{2}V_1^2 = \frac{p_2}{\rho_2} + \frac{1}{2}V_2^2 \qquad \text{or} \qquad p_1 - p_2 = \frac{1}{2}(\rho_2 V_2^2 - \rho_1 V_1^2)$$

This can be further simplified by assuming constant density so that:

$$p_1 - p_2 = \frac{1}{2}\rho(V_2^2 - V_1^2) \qquad \qquad \textbf{Equation 3.5}$$

One way of achieving high vacuum without restricting the inlet excessively is the use of staggered venturies one after another such as can be seen in figure 3.8.

The advantage of this system is that the exit of the smaller venturi is situated at the throat of the other venturi. This creates a low pressure at its exit that helps to accelerate air through the small venturi even with moderate airflow while the obstruction to the air passage is limited.
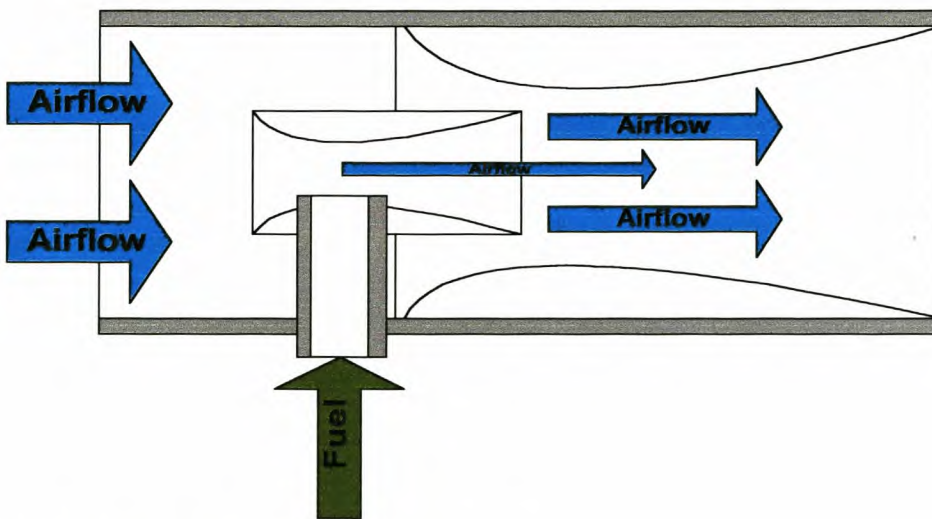
Figure 3.8 Two Stage Venturi System

The venturi would be made to fit the inlet air pipe so that the only parameter left to change would be the diameter ratio of the narrow section of the venturi compared to the feed-pipe diameter. The required vacuum can be achieved as a function of diameter ratio.

3.5.5   Fuel mixture adjustments with fixed geometry venturi

Two mechanical fuel-flow mixture adjustment methods are used. The first is simply a variable restriction in the gas-supply line as shown in figure 3.9. This is sometimes called the power screw since it is normally adjusted at open throttle until maximum power is attained.
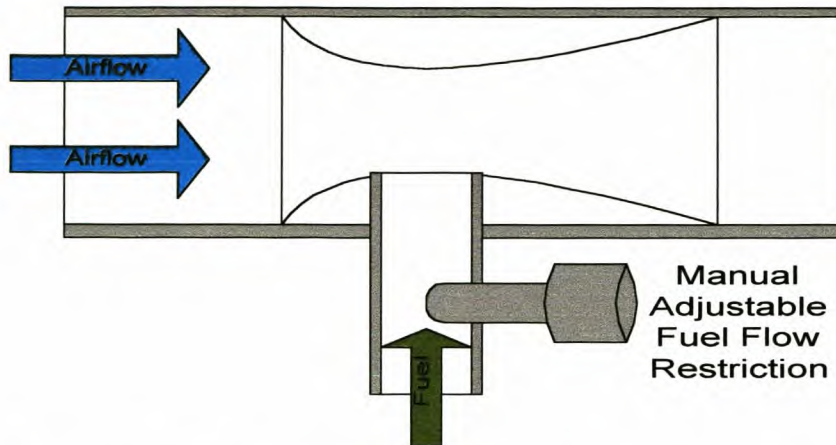
Figure 3.9 Fixed Geometry Venturi with Manual Mixture Adjustment

The limitation with the manual mixture adjustment is that it is only set for one load condition. This may not be the optimum for part load performance.

<u>Back pressure mixture regulation for a fixed geometry venturi</u>

To enable lean-out at part load, a system is used that varies the back-pressure on the diaphragm of the pressure regulator. The bigger the pressure difference, the bigger the gas-flow. A schematic of such a system is shown in figure 5.6.

The back-pressure regulation is an excellent way to vary gas flow and lends itself better to electronic control than the manual adjustment. The weights applied to each pressure can be varied by varying the jet sizes that are used as indicated in figure 5.6.

3.5.6 Effect of gas-supply line diameter

During tests with a converted engine, it was found that the venturi was not able to supply enough vacuum to supply sufficient gas to the engine. In order to improve the fuelling a study was done to see what parameter, if changed, would have the greatest effect on fuelling. It was found that, with this system, the diameter of the low-pressure gas-supply hose has the greatest effect on the amount of gas that will be mixed into the inlet air and is therefore the first part to be changed. Most other parts, such as supply-line length, have a linear effect of fuel-flow restriction, but the gas-supply pipe diameter affects it to the fifth power d^5.

The derivation follows:
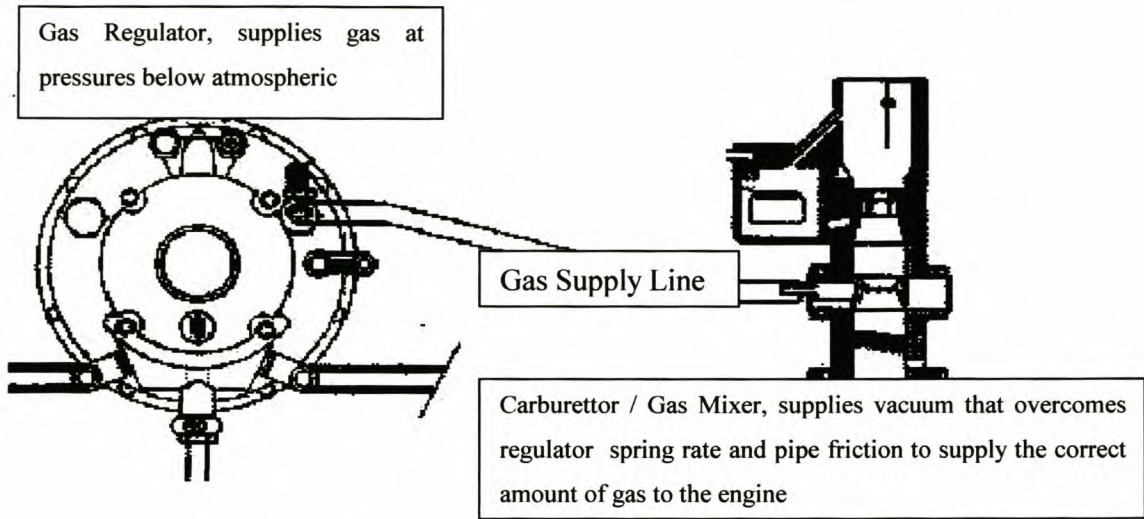
Figure 3.10 The gas supply hose from regulator to gas mixer (World LPG Association)

The speed of gas-flow in the supply-line is directly related to the diameter squared

$$Velocity(V) = \frac{Volume/sec\left(\overset{\bullet}{v}\right)}{Area\left(\pi\left(\frac{d}{2}\right)^2\right)} \Rightarrow V = \frac{\overset{\bullet}{v}}{\pi(\frac{d}{2})^2} \Rightarrow V = 4\times\frac{\overset{\bullet}{v}}{\pi(d)^2} \qquad \textbf{Equation 3.6}$$

The friction head of the gas in the supply-line in turn is directly related to the speed in the

supply-line squared $h_f = f\dfrac{LV^2}{d2g}$ **Equation 3.7**

(valid for laminar and turbulent flow), f is the Darcy friction factor (p203 F.M. White, 1988).

Substituting Equation 3.3 into Equation 3.4 gives $h_f = f\dfrac{L\times16(\dfrac{\overset{\bullet}{v}^2}{\pi^2 d^4})}{d2g} \Rightarrow$ simplified this

gives: $h_f = f\dfrac{L8\overset{\bullet}{v}^2}{\pi^2 d^5 g}$. **Equation 3.8**

It indicates that the single parameter that has the greatest effect in the head loss and therefore the amount of gas that will flow to the mixer is the diameter of the supply-line and all it's fittings. It is of interest to note that often the fittings, entries and exits present the greatest amount of resistance to flow, greater than that of the pipe friction. This shows that a lot can be done to improve this by smoothing pipefittings, entries and exits.

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3.6 LPG injection systems

In modern multi-point port fuel injected engines the inlet manifold is designed only to transport air and optimized for volumetric efficiency. This in turn has led to the development of port gas-injected systems which prevent formation of a combustible mixture in the inlet manifold, since backfiring into the inlet manifold may cause damage to the engine management and air induction system. In response to this, many LPG injection systems have been developed which works on similar principles. Most inject LPG in the gaseous-phase, but liquid-phase LPG injection is possible. The following configurations exist:

- Single Point Injection
- Multi Point Injection
- High Pressure Direct Injection

3.6.1 LPG injection (gaseous versus liquid)

The mixing of LPG in the gas-phase with inlet-air typically replaces between 3.2 and 4 % of the inlet air with gas. This potentially reduces the available power by 3.2 to 4 %, since less air is available for combustion. One way to overcome this is by injecting LPG in the liquid-phase.

3.6.2 Evaporative cooling with liquid-phase LPG injection

With liquid-phase injection the evaporation of the LPG will withdraw heat from the inlet air and the engine components that it comes in contact with to evaporate. This in turn will cool the inlet air so that the density increases and more air is drawn into the engine. To determine the maximum performance achievable with LPG it was decided to study the possible advantages obtainable as a result of evaporative cooling with liquid-phase LPG injection.

Figure 3.11 shows the temperature-dependant nature of the latent heat of evaporation of propane, that is given as 340kJ/kg at room temperature. This information together with the heat capacity of air was used to determine the evaporative cooling effect of a stoichiometric mixture of propane with air. For more detail see Appendix A3.

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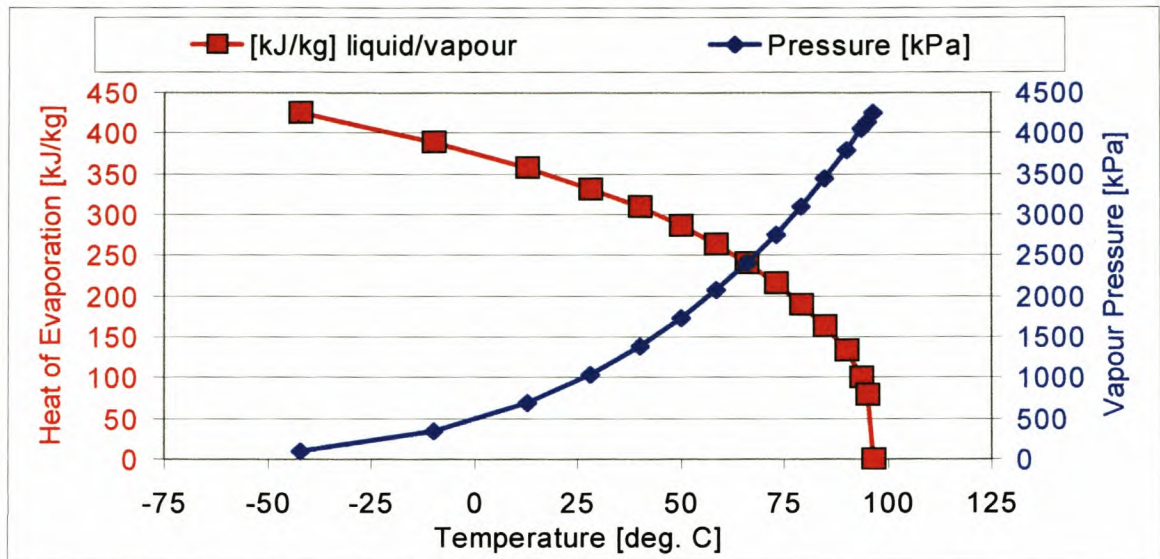
Figure 3.11 Latent heat of evaporation of propane

For the purpose of this evaluation a fuel temperature of 28°C was used with a air temperature of 27°C to calculate the cooling effect on the inlet air with a air/fuel ratio of 15.64 % by mass. Figure 3.12 shows the effect of inlet air cooling on charge densities.
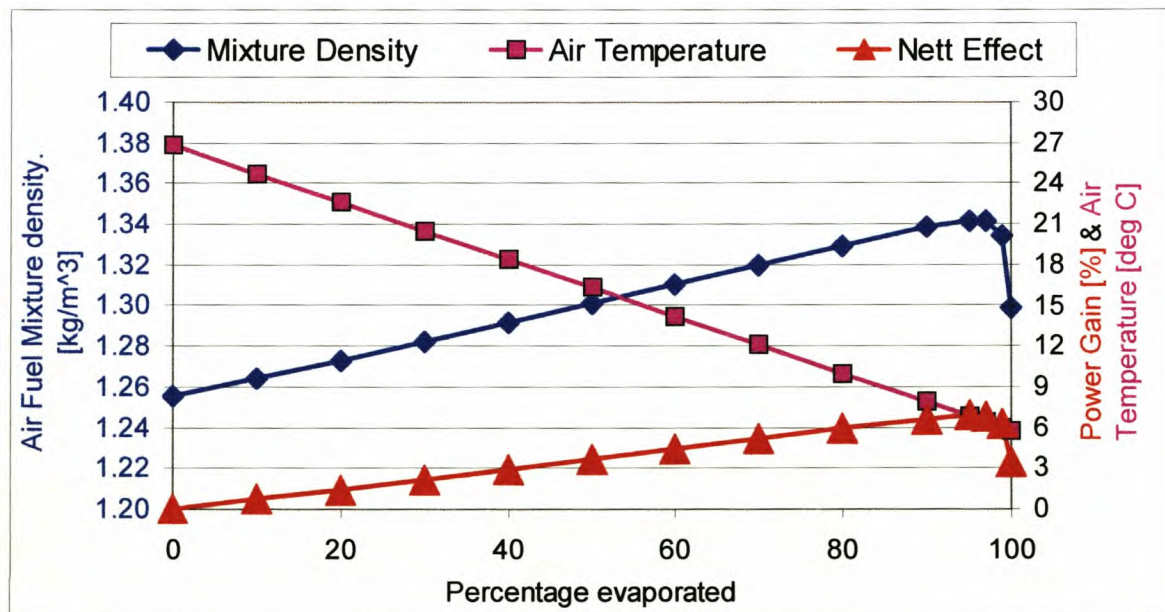
Figure 3.12 The effect of evaporative cooling with liquid phase LPG injection

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The mixture density essentially is an indication of the energy obtainable from a specific volume of fuel and air. As the LPG evaporates the overall fuel density decreases. The cooling effect increases the air density from 1.18 to 1.27kg/m^3, an increase of 6 %. Considering that complete evaporation of the fuel will replace 4 % of the air, the evaporative cooling has the potential to increase power by 2 % when compared to LPG supplied in the gaseous-phase. It is interesting to note that greater gains than this may be possible if only 95 % of the fuel is evaporated before the inlet-valve closes. The combined effect of increasing air density and decreasing fuel density shows that the overall charge density is at a maximum when 95 % of the fuel has evaporated. It then has a potential performance gain of 6.8 %. Once more than 95 % of the fuel has been evaporated the volume of the fuel starts to play a role in air displacement and the energy density of the fuel-air mixture decreases. Complete fuel evaporation under the conditions mentioned will cool the inlet air by 21°C as shown in figure 3.12. The calculations for evaporative cooling can be seen in Appendix A3.

Due to the possibilities of evaporative cooling, it was decided to convert a light-duty engine, on which liquid-phase LPG injection could be compared with gas-phase LPG induction. The result of the test is described in Chapter 6 of this report.

3.7 LPG ignition systems

3.7.1 Optimum spark timing

The following can be applied to spark advance timing according to Heisler (1995). Because of the high octane rating of LPG, performance improvements can be obtained by advancing the timing by about 10 % at low engine speeds. However, since flame propagation with LPG is slower than with petrol at high engine speeds, ignition must be retarded by about 15 % at the upper-speed range to prevent knock. Figure 3.13 illustrates the typical spark advance curves for LPG versus petrol fuelling.

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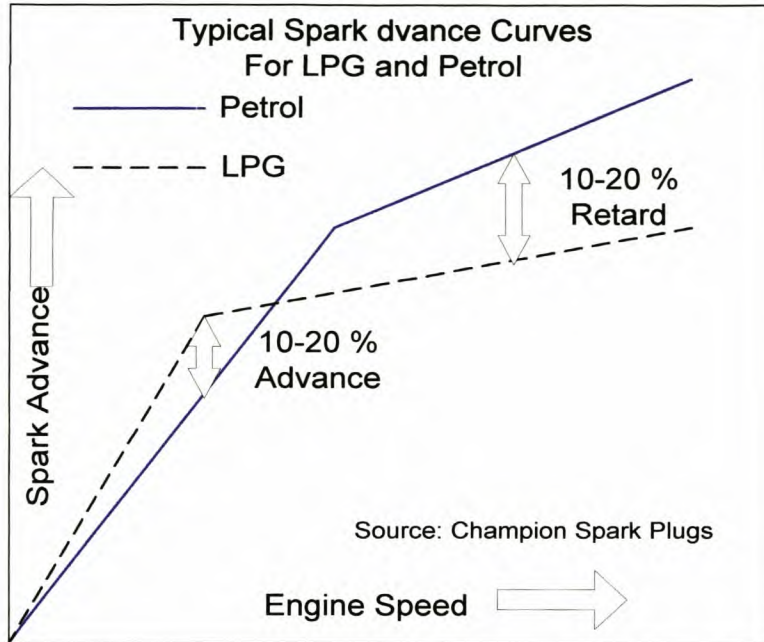
Figure 3.13 Spark advance of LPG versus petrol

This was confirmed in tests by the excellent low speed performance obtainable with LPG while the spark advance had to be increased at high engine speeds to reach optimum performance.

3.7.2   Spark voltage requirements

The spark voltage requirements given by Garret (1991) were as follows: the Spark voltage requirements for LPG is higher than that of petrol. In the order of 2 kV more than the 5 to 6 kV of petrol at idle e.g. 7 to 9 kV at idle and 5.5 to 6.5 kV at full load compared to 3.5 to 4.5 kV required for petrol at full load. This indicates an increase of 30 to 40 % that is needed because of the higher resistance to electrical current in gaseous or dry fuels as opposed to liquid fuels.

During engine tests with liquid-phase LPG injection it was found that it was more difficult to induce stable combustion at the mid-speed ranges where normally the best torque would occur. This was ascribed to the fact that the better breathing at those speeds combined with the evaporative cooling of LPG leads to greater charge density (and resistance to spark) than is the case with normal petrol injection. According to Heisler (1995) this is indeed the case, as the engine's spark voltage requirements tend to follow the torque curve. The point of maximum torque, and therefore charge density, is the point at which the spark voltage requirements are also at a maximum.

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3.7.3 Coil type

The available ignition voltage is mainly determined by the type of coil used, the amount of time available to charge or discharge the coil and the routing of the coil discharge energy from the coil to the spark plug. The main limitation with a single coil system is the time available for charging. This can somehow be overcome by using coils with short charge times such as racing type coils that have a rapid charge and discharge rate. Much energy (up to 60 %) is lost through the distributor and high tension leads. For this reason most modern engines have done away with distributors and use more than one coil to supply energy to the spark plugs. The most common system is the double-ended coil that has a high-tension lead at each end, allowing one coil to supply two cylinders. This allows twice the charge time that would be available if one coil were to serve four cylinders. Shorter high-tension leads without distributor losses and longer charge times per coil all contribute to the availability of high-energy spark at the spark plug. Some engines go one step further by using one coil per spark plug and no high-tension leads at all, thereby further reducing coil size and giving good spark energy. These systems are popular with dedicated gas engines because of the high-energy spark they deliver. In the light-duty engine conversion described in chapter 5, a double coil pack was used on a four cylinder motor. A single coil distributor type used on a 6 cylinder LPG engine described in chapter 5 proved insufficient for the ignition requirements. This illustrates that systems that are sufficient for petrol engines may not be able to deliver the spark energy that is required to ensure good combustion initiation with gas engines.

3.7.4 Electronic coil drivers

The next most important component to coil type is the type of coil driver used. Most electronic coil drivers with solid state switching are capable of handling 12 A, three times the current that traditional points can handle. To protect the coil they are equipped with a current limiter so that a low resistance coil may cause the coil driver to shut-off and it will not operate satisfactorily. This also means that a ballast resistor is not needed when using an electronic coil driver since it will compensate for voltage fluctuations and supply the coil with the correct voltage.

Two types of distributors are used to supply the coil driver with signal.

The inductive coil driver

The inductive type of pickup supplies a sinusoidal curve to the coil driver that has to be processed by the coil driver to determine the correct discharge time for the coil. The sinusoidal signal has to be conditioned to determine a clear cut-off point were switching can take place. This reduces the time that is available for the charging of the coil.

The Hall sensor coil driver

The Hall sensor type of coil pickup gives a square wave type of signal from the distributor that requires no conditioning to trigger the solid state switching to the coil. It also provides a more accurate switching than is possible with the sinusoidal wave from the inductive type.

For these reasons the hall type of pickup is used in most applications today (Heisler, 1995).

Capacitor discharge type

With these systems a capacitor is used to discharge to the coil at high voltages so that the primary voltages in the coil are about 400 V. This enables very rapid charging of the coil so that even at high engine speeds the dwell time is sufficient to give a high-voltage spark.

The disadvantage of this system is that the spark duration is relatively short compared to the other systems (Heisler H, 1995).

Multiple Spark Discharge (MSD) Systems

The MSD system is a system whereby capacitor discharging is used to provide a high-voltage spark, but the capacitor is charged and discharged several times so that multiple sparks of high voltage are given over a period of 20 degrees crank angle instead of one. These types of systems are used in drag racers or where misfiring may be a problem. It is interesting to note that ignition induction periods may be as long as 20 crank angle degrees long, thus this system may have it's merits.

3.8 LPG spark plug requirements

3.8.1   Heat range

Spark plug heat range is an indication of the *temperature of the central electrode* of the spark plug. It is desirable to have the central electrode hotter than the side electrode since the central

electrode act as the cathode. When the cathode is hotter than the anode it is more willing to release electrons, thus reducing spark voltage requirements.

Another reason for using hot spark plugs is to prevent the formation of fuel and oil deposits on the spark plug electrodes, that may lead to fouling and formation of insulation around the electrodes. For this reason engines that run cold use hot plugs while engines that run hot use colder plugs to prevent the spark plug from overheating. Should the engine be running at low speeds and low load a hot spark plug is required to achieve the required electrode temperature. With a racing engine operating at high speeds and temperatures, a cold spark plug is required to prevent the electrodes from overheating.

The temperature of the central electrode is determined by the length of the heat conduction path from the electrode to the body of the spark plug. In modern engines the side electrode may at times get so hot that it exceeds the temperature of the central electrode. To counter this, the electrodes of cold spark plugs have copper inserts in the side electrode to increase heat conduction to the body of the spark plug.

With LPG-fuelling of petrol engines, it is common practice to use spark plugs one range hotter than the standard for that specific engine, to reduce the spark voltage requirements.

3.8.2   Spark plug gap

To adapt an engine that has a ignition system for petrol-fuelling it is common to reduce the spark gap by about 0.2mm to overcome the higher resistance posed by the use of a gaseous fuel (Williams and Lom, 1982).

A common policy when fitting spark plugs is that it is better to start with the minimum gap needed for ignition to allow for electrode wear since by the time the spark plugs need to be replaced the voltage requirement may have doubled due to the increase in electrode gap. A large spark plug gap results in higher intensity initial spark and also increases the wear rate of the spark plugs (Heisler, 1995).

3.8.3   New developments

Plasma jet igniters

A plasma jet igniter is like a spark plug with a recessed central electrode and an annular anode around it. Spark energies of several Joules, up to 20J, normal spark ignition energy is 0.18J, is

used to ionize a small pocket of gas inside the igniter which then forms a plasma which introduces a greater source of combustion than those obtainable with a common spark plug. These are especially used in lean burn applications with turbocharged natural gas engines or with high EGR rates where greater energy is required to prevent misfire (Gardiner and Mallory, 1995).

3.9 Safety in handling of LPG

Handling of LPG in liquid phase requires special care since any length of pipe can contain large amounts of energy that will evaporate quickly in the eventuality of a leak. Once it has evaporated it occupies up to 400 times the volume of the liquid that has evaporated, forming a combustible mixture of air and fuel that is invisible. Considering that 2.4 to 9 % of propane by volume is required to form a combustible mixture in air, it can fill a combustible volume 5000 to 16000 times that of the liquid-phase LPG that has escaped. To help detect gas build-up it is dosed with odorants that can be detected at 20 times below that of combustible levels.

To prevent leaks LPG fittings that are removable have to be of the self-sealing type. Once a connector is removed from its connection the ends of the pipe are sealed to contain whatever gas may be left in the pipe. Even with these precautions it is important to release the fittings quickly, since there is a time, while the fitting is half connected, when gas does escape. The use of insulating gloves when connecting or disconnecting LPG pipes under pressure is essential. A liquid-phase leak on open skin, or even a well-insulated glove, may cause frostburn or freezing of valves or connectors in the open position if not properly attached. Since LPG is heavier than air, it will sink to the ground when released in the atmosphere. Floor-level ventilation is therefore required when LPG is used in enclosed spaces. (Williams and Lom, 1982)

4 LIGHT DUTY ENGINE CONVERSION

A 2-litre engine was used as a test platform. The engine was equipped to run on different technologies available for LPG conversion and on petrol injection. The purpose of the conversion was to get insight into the characteristics of LPG as an engine fuel and compare the performance attainable with LPG-fuelling to petrol-fuelling. To obtain information about combustion characteristics a pressure-sensor was fitted in one of the combustion chambers. The pressure data was used to determine the different combustion characteristics of the different fuel systems (liquid versus gaseous) and different fuels (petrol versus LPG).

4.1 Test engine

A two-litre Nissan NA 20 spark ignition petrol engine was used as a test platform. The engine was fitted with a redesigned inlet manifold, ports and a combustion chamber that was reshaped to reduce the octane requirement of the engine. The carburettor was replaced by a fuel-injection system that could be controlled by an electronic engine-management system interfaced with a PC. The basic engine specifications can be seen in table 4.1

Table 4.1 NA 20 engine data

Bore	Stroke	Compression Ratio	Displacement		
[mm]	[mm]	US24 Head	[ml]		
86	86	9.21	1998		

Valve Data (8 valves)							
IVO	IVC	Duration	Lift	EVO	EVC	Duration	Lift
BTDC	ABDC	deg. CA	mm	BBDC	ATDC	deg. CA	mm
2	50	232	9.5	40.5	19.5	240	9.5

Fuel injection	Non Sequential Port injection		
Engine Management	DUPEC		
Ignition System	DUPEC Twin Coil with NGK BP6ES Spark Plugs		

Four test configurations were used, these are given below with their abbreviations.

- Petrol injection (P-INJ)

Petrol injection without LPG mixer fitted to engine.

- LPG injection (LPG-INJ)

Liquid phase LPG injection without the LPG mixer fitted.

- LPG induction (LPG-IND)

LPG induction in the gas phase through the LPG mixer.

- Petrol injection with LPG mixer fitted (P-INJ+GC)

This was done to compare with LPG induction to see the effect of the mixer on engine performance due to the restriction it poses to the flow of inlet air.

4.2 Design of a LPG-induction fuel system

4.2.1 Equipment used

The gas conversion was done with a Beam 120 regulator vaporizer capable of supplying a 90kW engine. Engine coolant from the inner loop of the engine was tapped-off to supply the heat necessary to vaporize the liquid-phase LPG. The gas was fed from the regulator to the gas-mixer via two pipes, one of which was equipped with an electronically controlled flow-restriction valve to make fine mixture adjustment possible.

Figure 4.1 shows the layout of the gas-induction equipment from the LPG tank to the gas-mixer upstream of the throttle.
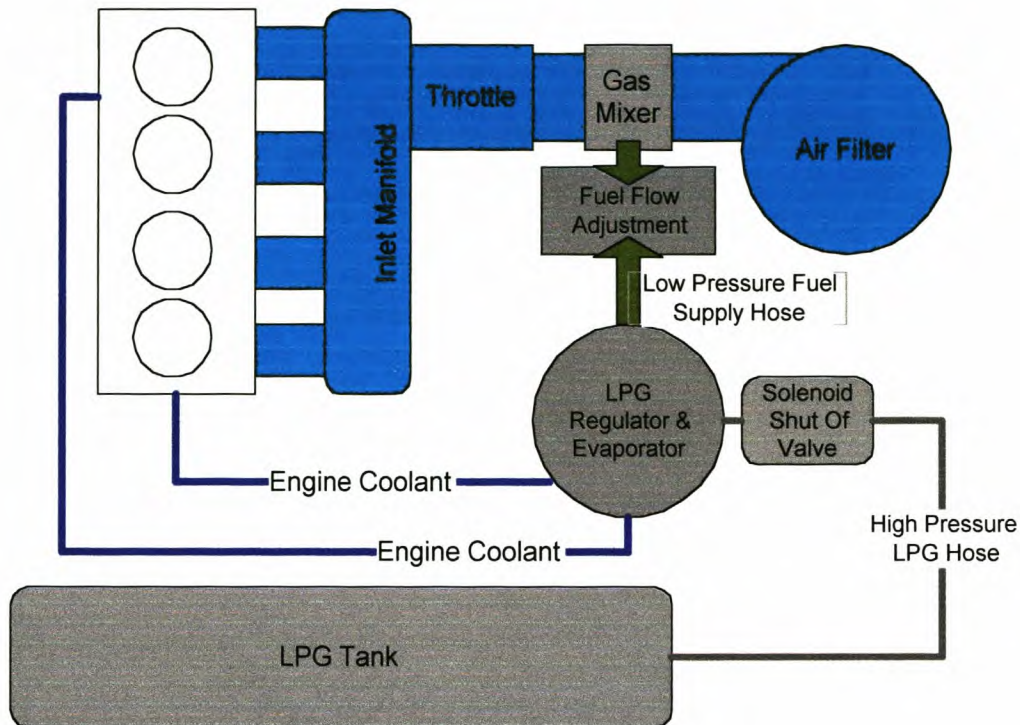
Figure 4.1 Equipment used for gas phase LPG induction

The solenoid shut-off valve was connected to the fuel pump circuit to prevent fuel-flow when the engine does not run, except for a brief period before cranking. As a further safety measure the regulator requires vacuum from the engine before the second stage can supply fuel to the engine. This second system was, however, found to be insufficient to ensure complete shutdown of gas flow.

4.2.2   Design of the gas-mixer

The gas-mixer was designed according to the principles described in Chapter 3. The vacuum required for maximum fuel-flow was used as a guideline for the throat to inlet ratio that was required. Pressure loss over the gas-supply pipes was calculated using data for smooth pipes and loss coefficients (k factor) for restrictions as given by (White, 1988) as shown in table 4.2. The properties of a 50/50 propane/butane mixture was used in the calculations.

Table 4.2 Properties used in mixer design

Fluid Properties				Engine parameters		Supply pipe		
Density [kg/m^3]	Mix %	Viscosity		litres	Inlet Dia	Diameter mm		12.5
Propane	1.9	50	[Stokes]	2	60	Length m		1.5
Butane	2.55	50		Throat Dia		Joints #		2
Mix	2.225		0.04		40	K factor		1.8
Air	1.2							

The mixer was then designed to fit into the existing air-supply pipe that has an inside diameter of 60mm. A throat diameter of 40mm was chosen to supply the required vacuum. The importance of the restriction posed by the gas-supply line becomes evident when figure 4.2 is studied. It shows how the use of two supply pipes lowers the pressure loss to 25 % of the head loss with 1 pipe. For this reason two pipes were required to make supply of the correct amount of gas to the engine possible.
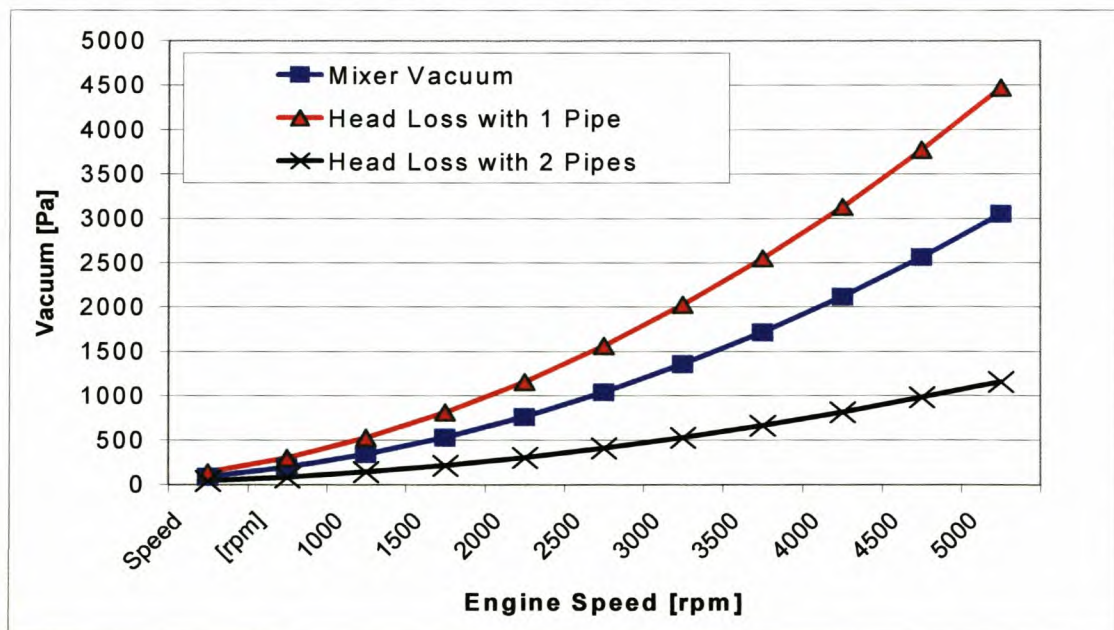
Figure 4.2 Mixer Characteristics matched to vacuum requirements for LPG induction

The mixer design was then finalized using a 60mm entry diameter with a 40mm throat diameter and a divergent exit to ensure pressure recovery. Gas was fed through two

12.5mm (inside diameter) pipes entering at 180° relative to each other. These pipes fed into a common annular ring to ensure good mixing with inlet air.

4.2.3 Fuel-flow control mechanisms

One approach ensuring that minimum adjustments are needed to supply the correct fuel-air mixture, is to design a fuel-supply system that will supply a near correct mixture without intervention. In some cases this is achieved using two supply pipes, one of which carries the main stream of gas and the other being used only for enrichment or leaning of the mixture (World LPG Association, 1995). Much the same strategy was used here, although for practical reasons the pipes are of the same diameter. Only one needed to be adjustable to achieve fine-tuning.

Throttling of one pipe was achieved by using a valve that could be adjusted electronically during mapping and the valve position recorded. With mapping completed this data could be stored in a PIC to automatically fine-tune the mixture under different load conditions (much like a open-loop fuel injection system with electronic mapping).

Automatic compensation for air filter clogging or obstruction of inlet air was achieved by referencing the pressure after the air filter to the back pressure of the regulator.

4.2.4 Calibration with LPG gas-induction

The LPG supply was matched to the engine requirements and mixer characteristics by varying engine load and speed and adjusting the valve position until the desired CO levels were recorded. The parameters used for the calibration were MAP (Manifold Absolute Pressure) and engine speed in rpm. Figure 4.3 shows the valve position, measured in volts (V), with 5 V representing the open position while 0 V indicates the closed position. It shows how low engine speed and low load required an open valve position due to the low vacuum created by the venturi under these conditions. It also indicates the complexity of the system needed to supply the engine with the correct amount of fuel under all load conditions. Note that this was done at steady load and speeds.
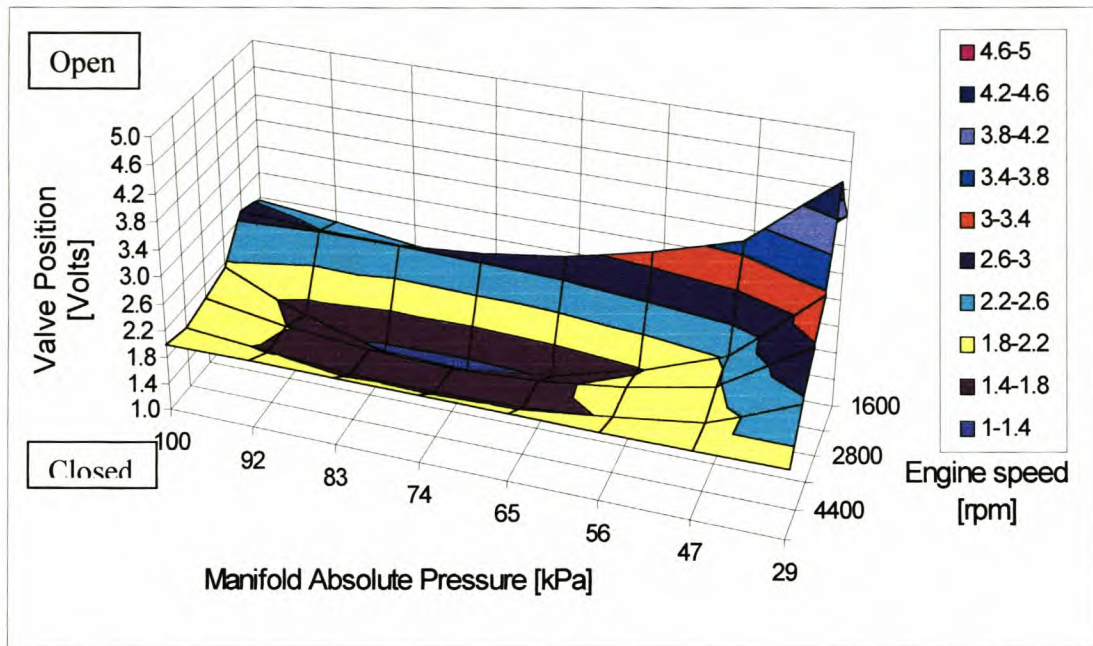
Figure 4.3 Valve position versus engine speed and MAP

The different valve positions shown in figure 4.3 show that at low engine speeds and low manifold pressure there is little airflow through the engine. The valve opening was increased under these conditions to supply sufficient amounts of gas due to the low vacuum created by the mixer.

4.2.5 Dynamic response of the system

During transient testing it was found that the engine would run rich for a brief period after reduction in load and lean after increase in load. This was ascribed to the inertia of the gas-supply system that partly depends on the length and diameter of the gas-supply pipes. Under full load at 5000 rpm the gas would flow at 15 m/s in the gas-supply pipes. This represents considerable kinetic energy so that after reduction in load the inertia of the system would lead to overfuelling. The same reasoning accounts for increase in load where energy is required to increase the kinetic energy in the supply-pipe before the desired flow will be reached.

To improve the dynamic response of the system it would be advisable to use supply-pipes with larger diameter and shorter lengths so that the kinetic energy and inertia in the supply-pipe would be reduced.

4.3 Design of a liquid-phase LPG injection system

To overcome the problems of power loss with gas-phase LPG-mixer systems and their limited controllability, as well as the dependence on sometimes expensive hardware it was decided to evaluate the use of liquid-phase LPG injection with the possible use of more standard equipment. It was decided to use off-the-shelf equipment wherever possible and integrate this into a liquid-phase LPG injection kit that could be used with the ECU of the OEM.

4.3.1 LPG injection system components

The components used in the conversion to LPG injection can be seen in figure 4.4.

Figure 4.4 does not show the engine-management system, since that was identical to the one used with petrol-injection with the option to choose which fuel was being used.
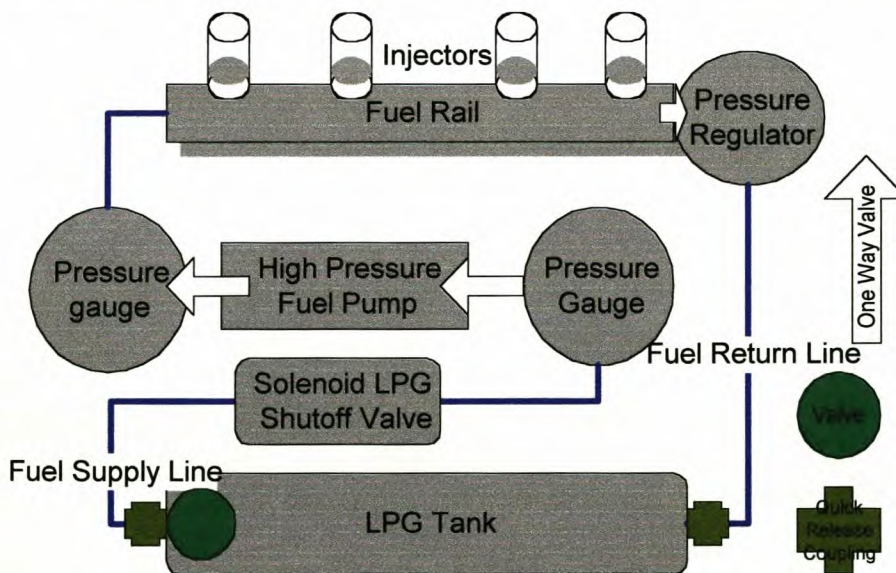
Figure 4.4 Layout of LPG injection equipment

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The fuel pump, fuel rail and injectors had to be compatible to both LPG and petrol since both fuels would be supplied through the same fuel rail and injectors alternately. To achieve this, the vapor-pressure of propane (at the temperatures that may be experienced in the engine compartment) had to be taken into account and the whole system had to be able to withstand pressures of up to 15bar. A high-pressure petrol fuel pump was encapsulated in a steel container to contain the high pressures and make connection of high-pressure hose to the pump inlet and outlet side possible.

Electronic fuel injectors (pulse-width modulated), a high-pressure fuel pump and a fuel rail capable of controlling fuel pressure up to 8bar was used. An electronic engine control unit ECU was utilized to vary spark advance and injector pulse duration as a function of engine speed and manifold absolute pressure.

In order to minimize the possibility of LPG vaporizing in the fuel rail a special fuel rail was developed that could be cooled or kept at a fixed temperature for the test bench tests, thus minimizing the amount of variables that had to be accounted for.

4.3.2   Injectors

Electronic fuel-injectors (semi-sequential, pulse width modulated) were used. These had a petrol flow-rate of up to 1.6 cm^3/s at 8bar with a 50 % duty cycle at 200Hz. This is the frequency at which the injectors would operate at 6000 rpm for this engine. This proved to be sufficient for the 2-litre engine which was tested with LPG at engine speeds up to 5000 rpm. The fuel-pressure was kept at 8bar for both petrol- and LPG-injection. Whilst the density of LPG is lower than that of petrol, the absolute viscosity of LPG (at 20°C, vapor pressure) is considerably lower than that of petrol. This difference in viscosity, 1.27E-4 N.s/m^2 for LPG compared to 2.92E-4 N.s/m^2 for petrol at 20°C compensated for the difference in fluid density so that it was found that with identical fuel pressures the difference in injection timing between the two fuels was small (Jenkin, 1962).

4.3.3   Fuel pump and fuel rail pressure regulator

A high-pressure fuel pump was used in conjunction with a high-pressure regulator that could be controlled to keep the fuel rail pressure at 8bar. The high pressure was needed

to keep the LPG from gasifying in the fuel rail. The fuel pump was tightly encapsulated in a container to prevent leakage or damage to the high-pressure connections.

The pressure-regulator could be manually adjusted to maintain a constant pressure difference between the fuel rail and the inlet manifold. This was achieved by referencing the inlet manifold pressure to the diaphragm that controlled the amount of fuel returning to the tank from the fuel rail. At one stage the pressure in the LPG tank was used as reference pressure in an attempt to increase the fuel rail pressures. This proved to be unnecessary as the regulator could be adjusted manually to achieve the desired pressure difference.

Engine control unit

A locally developed DUPEC control unit (ECU) was used to control injector-pulse duration as a function of engine speed and manifold absolute pressure (MAP). The advantage of this ECU was that it could be optimized separately for each fuel. Switching from one fuel to the next could be done without compromising performance.

4.4 Test equipment

Gas Analyzers

Siemens gas analyzers were used to measure NO_X, Hydrocarbons, CO, CO_2 and O_2. The analyzers were calibrated using calibration gas before and after each test. Results were logged automatically on a time stamped basis together with other engine data to a PC.

In-cylinder pressure transducer

No. 4 cylinder was fitted with an AVL 12QP pressure transducer with a sampling frequency of 110 kHz for sampling of in-cylinder pressures throughout the cycle at engine speeds of up to 5000 rpm.

4.4.1 Fuel-flow measurement

Petrol flow was measured using a gravimetric fuel-flow meter consisting of a 3 litre flask on a mass balance which was connected to the data acquisition system. The fuel mass difference over 0.5 second intervals was used to calculate fuel-flow and averaged over the period during which readings were taken at a specific load point once stability had

been reached. The measurement of fuel flow with gas was somewhat more difficult due to the mass of the gas cylinder that made it difficult to get accurate fuel-flow readings over a short period of time. To overcome the low resolution of the LPG mass-flow measurements a 25 kg container (when full), was used and was filled from a 48 kg cylinder as represented in figure 4.5.
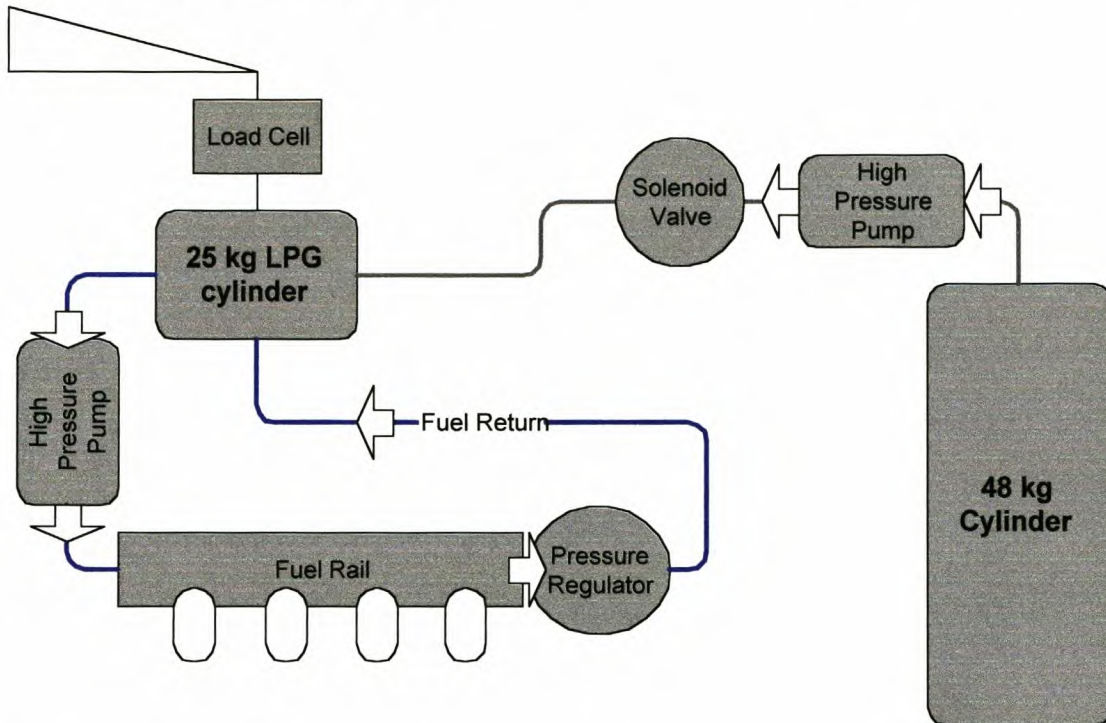
Figure 4.5 LPG fuel-flow measurement

The small LPG cylinder (25kg when filled to capacity) was suspended from a load cell to determine fuel-flow during testing. With this arrangement, relatively accurate fuel-flow measurements could be made over a period varying from 30 seconds to 2 minutes, depending on the fuel consumption at the time. The gas mass was recorded together with the real time on the data logging system. The fuel-flow was calculated from this after the test. The small cylinder could be filled from a bigger cylinder using a high-pressure pump whenever refilling was required.

4.5 LPG injection system tests

Initial test runs were promising except for the occasional misfire that could be audibly detected and which prevented tests from being run at speeds above 3000 rpm. The misfiring was initially attributed to LPG evaporating in the injector fuel rail and the fuel-pressure was increased from between 4-5bar to between 8-10bar to compensate for this.

With 8bar pressures results were markedly worse, since the engine would start misfiring badly with any increase in speed above 1800 rpm. It was unclear whether the misfiring was due to the mixture being too rich or too lean, although CO readings were available, high CO readings could either indicate misfiring from too little fuel or partial combustion due to overfuelling.

The misfiring and bad engine performance was clearly related to engine speed. The only factor that could initially be traced to speed dependence was the injector behavior at different frequencies. For this reason it was decided to test the injectors at ranges of frequencies and pressures to quantify their performance.

4.5.1 Injector tests

Injector test equipment

A test rig was constructed in order to test various injectors, one at a time, under different conditions. The results of the injector tests are shown in Appendix B4.
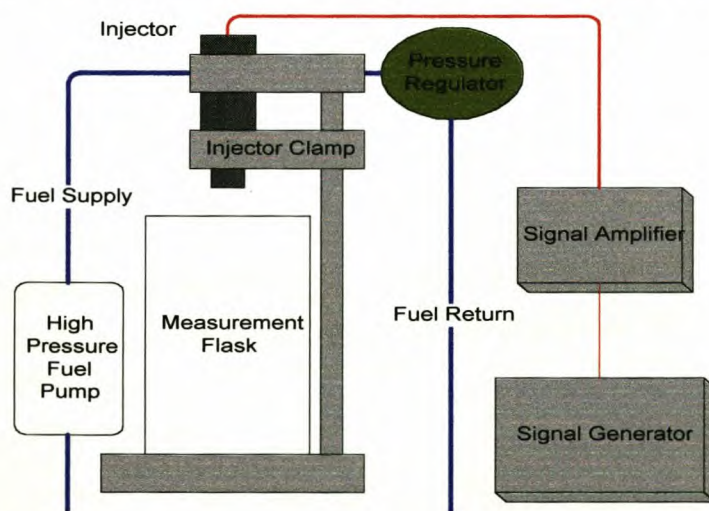
Figure 4.6 Layout of injection test rig

The injectors were pulsed using suitable adapted electronic circuitry. A high-pressure Bosch fuel pump and a high-pressure regulator were used to test to pressures up to 8bar.

Injector test criteria

Test criteria were at first set at the maximum frequency that is to be expected with that type of engine and injection system i.e. 200 Hz at 6000 rpm. A duty cycle of 50 % was chosen.

The fuel-injection system used is of the sequential type. This means that fuel is injected at all inlet ports simultaneously with the result that each injector injects four times per inlet stroke for one cylinder.

Injector Test Results

Two types of injectors were tested, a standard injector with two spray holes that had previously been used in the engine and a single hole test type with a higher flow-rate. The results can be seen in figure 4.7.
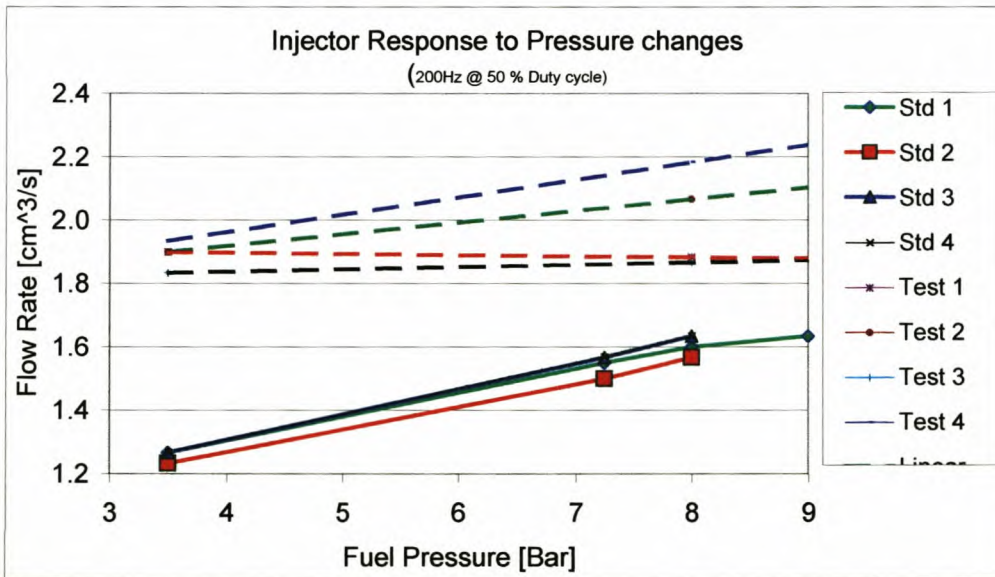
Figure 4.7 Injector flow rate response to increasing fuel pressures

The tests on various pressures indicated that the single hole test injectors (dotted lines) show greater variation in flow-rate at high pressures than the standard type which showed very little variation up to pressures of 8bar.

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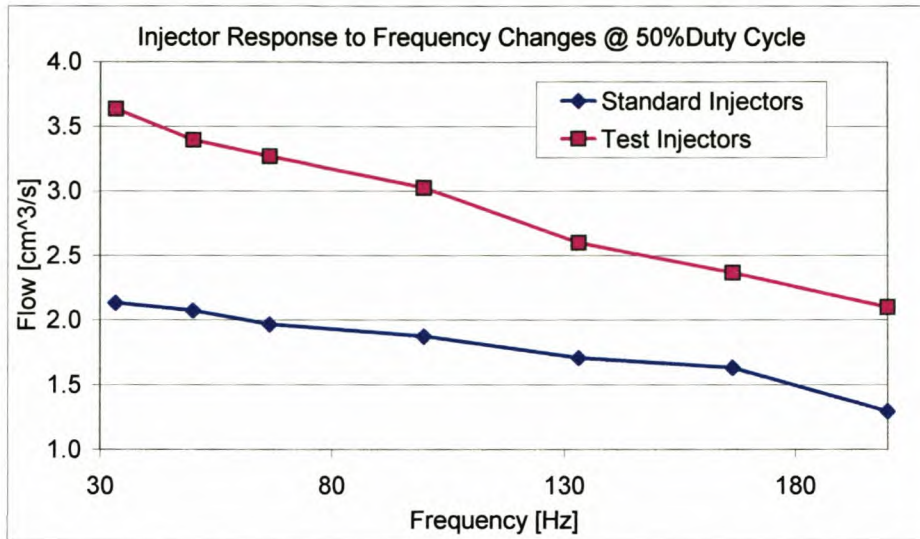
Figure 4.8 Injector flow rate response to frequency changes

The tests were done in the range of 30Hz to 200 Hz which is representative of engine speeds of 800 to 6000 rpm. The frequency was varied while the duty cycle was maintained constant so that the injector delay could be determined. This was calculated to be 0.084 ms for the standard injectors and 0.153 ms for the test injectors.

Conclusion of Injector Tests

The injector tests were correlated to engine requirements using the engine displacement volume and the injector characteristics and calculated delay time with the engine-management system's injector pulse-timing map for each engine speed. Good correlation was found for petrol, proving the validity of the test procedure used. The engine requirements were determined for LPG in the same manner as for petrol in order to know in advance what type of injector duration should be used.

Some backwards calculations were done for interest sake by using the injector duration of the ECU and determining the engine volumetric efficiency with the fuel entering the engine known once the injector characteristics were known.

Table 4.3 clearly indicates that the engine had not yet been mapped properly for different engine speeds as the injection duration was the same from 2000-7000rpm. It does however show that the values used are realistic at approximately 84 % volumetric efficiency.

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Table 4.3 Comparing injector data to engine performance

US 20 Values determined from Fuel injection map

Engine speed		Frequency	Inject dur.	Correct dur	Open T/s	Fuel Flow	Fuel Flow		Vol eff.
[rpm]	[m sek]	[Hz]	[ms]	[ms]	[ms]	[cm^3/s]	[l/hr]	[kg/hr]	[%]
1000	30	33.3	2.50	2.42	80.5	0.2203	3.2	2.38	48.95
2000	15	66.7	4.25	4.17	277.7	0.7597	10.9	8.20	84.40
3000	10	100.0	4.25	4.17	416.6	1.1395	16.4	12.31	84.40
4000	7.5	133.3	4.25	4.17	555.5	1.5194	21.9	16.41	84.40
5000	6	166.7	4.25	4.17	694.3	1.8992	27.3	20.51	84.40
6000	5	200.0	4.25	4.17	833.2	2.2791	32.8	24.61	84.40
7000	4.3	233.3	4.25	4.17	972.1	2.6589	38.3	28.72	84.40

The tests concluded that the injectors, although rated at three bar, could operate at eight bar with good repeatability. It was decided to operate at eight bar since this would help to prevent evaporation of the fuel in the fuel rail. The standard injectors were used since they displayed the least variation with changes in pressure.

4.5.2 Fuel rail modifications

It was found that, during testing, the engine operating temperatures could cause the fuel rail to be exposed to temperatures above 60°C for prolonged periods of time. At 60 °C the vapor pressure of commercial LPG exceeds 12bar requiring very high-pressures to prevent evaporation in the fuel rail.

In order to minimize this possibility, a special fuel rail was designed that could be cooled or kept at a fixed temperature by means of external coolant for the purpose of laboratory tests. The supply side of the fuel rail was set inside another pipe so that coolant could be circulated around the fuel-supply to shield it from radiation of engine heat.
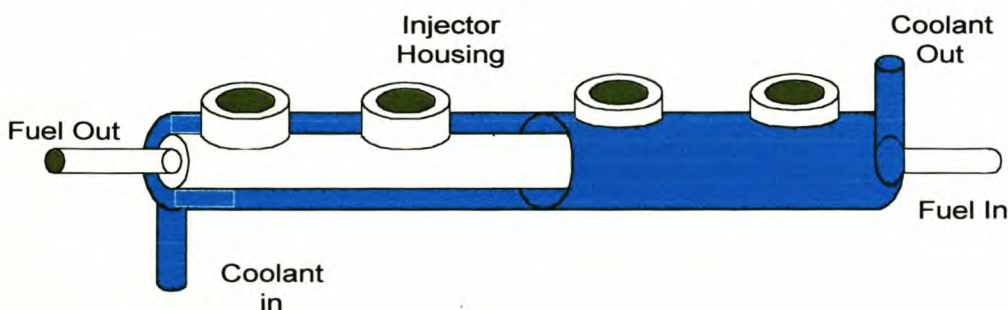
Figure 4.9 Modified fuel rail for LPG injection

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For testing purposes water from the outer loop of the heat exchanger of the engine was used for cooling of the fuel rail. This proved to be effective since no more problems with evaporation of fuel were encountered.

4.6 Engine test procedure

The purpose of the test was to evaluate emissions, fuel consumption and power output under various loads and to study combustion mechanisms with the different fuels and fuel systems. These requirements resulted in a 21 point test as seen in table 4.4 which includes a power curve from 2000-5000 rpm and part load tests at 2800 and 3600 rpm. Emissions and fuel-consumption readings were taken at all test points while in-cylinder pressure measurements were taken at selected points. These are shaded in purple in table 4.4. The manifold absolute pressures of 56 and 74 kPa were chosen to test part-load conditions.

Table 4.4 Layout of Test Points

Manifold Pressure [kPa]	Legend:	Test without pressure measurements				Tests with pressure measurements			
100	21	20	17	16	5 pts timing swing fuel loop	4	3	2	1
74				18	14				
56				19	15				
Engine speed [rpm]	2000	2400	2800	3200	3600	4000	4400	4800	5200

4.6.1   Combustion analysis

Two approaches can be used for combustion analysis; one where the engine is optimized for each fuel and another where some important parameters such as spark timing are deliberately kept the same for different fuels to gain insight into induction period or ignition delay. The first approach was used in this study.

4.6.2　Test reproducibility

Atmospheric conditions

Since the tests were conducted over a period of time, corrections had to be made to the data to compensate for changes in atmospheric conditions. In this study the standard SABS correction factors were used for corrected power and torque.

Engine condition

In order to see if variations being recorded were engine-related, the engine was tested in standard fuel-injection configuration with the LPG mixer before and after all the tests, to monitor any changes in the engine condition. This indicated that the engine had lost compression before the last two tests with LPG induction and petrol injection with the LPG-mixer. These therefore could not be compared directly to the first tests with petrol injection without mixer and LPG liquid-phase injection. The loss of compression was more pronounced at low engine speeds than high and was attributed to leaking valves. This deterioration in engine condition cannot be attributed directly to the use of LPG. The engine was also used for other work during the period in which these tests were conducted, amongst others, knock tests, which are more likely to have caused damage than LPG fuelling.

4.6.3　Mapping procedure

Test cell temperature

It was found that the temperature in the test cell would increase gradually during tests. To optimise timing and fuelling over the range of temperatures experienced in the test cell, would be a mammoth task. An air box was therefore used to supply air at a controlled temperature to the engine. It consisted of a large sealed box with an air cooler inside that maintained air temperature to an accuracy of plus minus 10 degrees from setpoint.

Ignition timing and fuelling adjustments

It became clear that engine response to sub-optimal timing or fuelling was less critical under part-load than full-load. It is known that the lower charge densities at part-loads require lower spark voltages to initiate combustion than that of high charge densities at

full-load. If the spark ignition system is therefore on the limit of the engine spark requirement it helps to first map the engine at part-load where the resistance to sparking is less critical.

Knowing this, it was possible to map the engine in areas that were previously inaccessible due to persistent misfiring. Once the desired speed had been reached the throttle could be opened steadily while constantly making adjustments to keep CO % between acceptable limits. It was found that with LPG a CO reading of below 0.3 % led to misfiring and loss of power. For mapping data see Appendix B1.
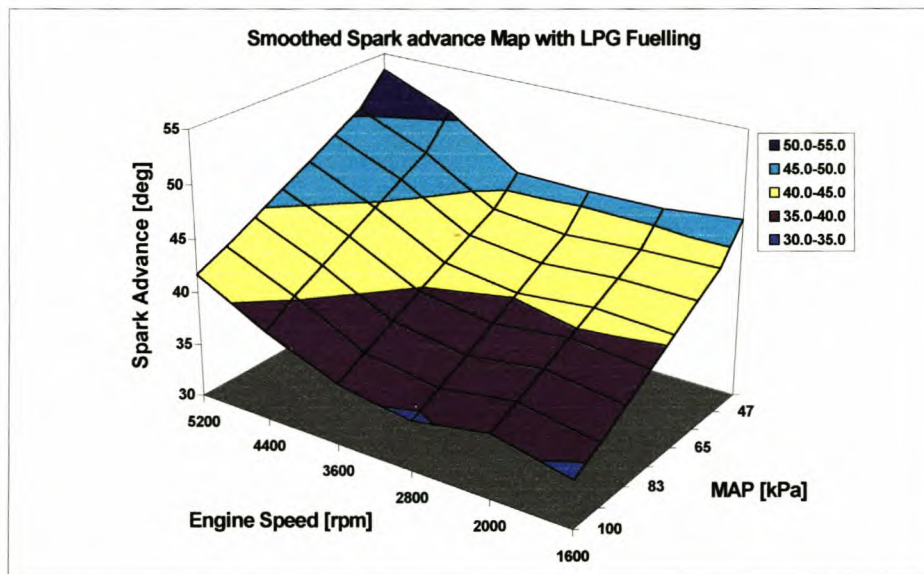
Figure 4.10 Spark advance map for LPG-fuelling

Figure 4.10 shows the spark advance map that was obtained for gas-phase LPG-fuelling. From this the relationship between engine speed, manifold pressure and spark advance became clear. It showed a linear relationship between manifold pressure and spark advance with lower manifold pressures requiring more spark advance. The relationship between engine speed and spark advance is less clear. The general trend was that higher engine speeds require more spark advance while the optimum spark advance is least at engine speeds where the most torque is developed.

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4.7 Conclusion

The conversion to liquid-phase LPG injection, as well as gas-phase LPG induction and petrol injection on one engine, provided a valuable platform for the study of the different fuel systems. This was achieved in controlled conditions and with off the shelf equipment normally used in petrol-fuelled vehicles. The results are discussed in Chapter 6.

5 HEAVY DUTY LPG ENGINE CONVERSION

It is believed that there is a need for clean engines for public transport vehicles, such as busses, that operate in city centers and densely populated areas. LPG engines are both quieter and cleaner than their diesel counterparts and are therefore ideal for this application. A bus company in the Cape Town area was approached with an offer to provide them with a LPG engine that could be fitted into one of their busses. This would serve as a trial project and be used to increase public awareness to the use of LPG as automotive fuel. Once the proposal was accepted a heavy-duty diesel engine was converted to LPG use for use in a bus.

5.1 Conversion of the ADE 447 engine to LPG

Mercedes Benz has developed a natural gas conversion kit for the ADE 447 engine. It is a six cylinder in line engine rated at 150kW at 2200rpm and a maximum torque of 880Nm at 1000rpm. Parts for the kit were available from ADE South Africa. These were adapted to make the engine suitable for LPG use.

Parts that were not available were machined from standard diesel engine components. The converted engine was mounted on a test bed for testing and optimization of the LPG fuel and ignition systems.

5.1.1 Standard engine specifications

The engine used was a 6 cylinder naturally aspirated in-line ADE 447 with 12-litre displacement.

Bore: 128mm

Stroke: 155mm

Compression ratio: 17:1

Rated power: 150kW at 2200 rpm

Max torque: limited to 800Nm at 1000rpm

5.1.2 Changes to standard engine:

The following changes were made to the standard diesel engine to convert it to LPG operation.

- The diesel pump was replaced with a distributor and spark plugs were fitted in place of diesel injectors.

- The diesel inlet manifold was replaced with one that was optimized for air-flow rather than swirl.

- The manifold is fitted with a throttle that was activated by a link to the mixture control. This was mounted in place of the injector pump so that the same diesel actuator could be used.

- A gas-mixer with load dependent mixture control was fitted in the clean air pipe just before the throttle.

- Turbo cylinder heads with hardened valve seats were fitted to handle the higher temperatures that would be experienced with LPG operation.

- The injector holes were machined and threaded to accommodate spark plugs.

- The compressor air intake had to be re-routed from the manifold to the clean air pipe before the throttle.

- The piston bowls and top were machined to lower compression ratio to 9.5:1 and reduce the squish velocity.

The following were the performance requirements of the LPG engine:

Rated power: Limited to 145kW at 2200rpm, Torque de-rated to 694Nm at 1400rpm to match the engine to the gearbox and transmission of the vehicle it was designated for.

Replacement of diesel injection pump with distributor.

Figure 5.1 Shows the distributor and mixture control device that replaced the injector pump. The distributor drive fitted into the gear that drives the injection pump. Ignition timing could be adjusted while the engine was running by loosening and rotating the distributor housing. For greater adjustment the gear that drives the distributor could be

loosened and rotated. The vacuum modulation valve (with all the pipes connected to it) regulated the vacuum to the LPG regulator thereby adjusted the mixture. Load dependant control was done via a cam-driven device that mechanically controlled the vacuum to the regulator as well as ignition timing, depending on throttle position. A proximity switch senses closed throttle for overrun fuel cut off while an electric motor acts against the spring that operates the throttle to prevent overspeed.
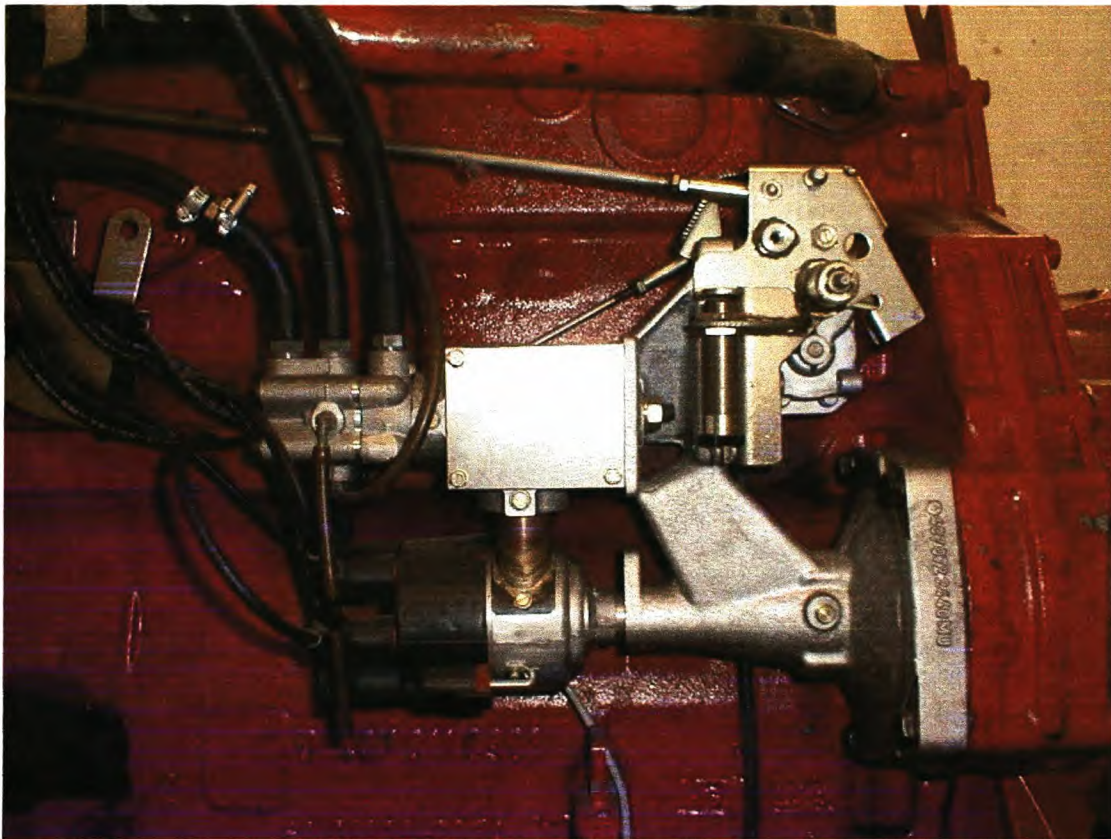
Figure 5.1 Distributor housing with throttle linkage and mixture control

The distributor is of the inductive type that is used with an electronic coil driver.

Spark ignition system

A Motorolla ignition unit with accompanying coil was used. This came from a 6-cylinder petrol engine. The system required a 12V supply and the inductive signal from the distributor. A high-voltage cable runs from the coil to the distributor to be distributed to the correct cylinder by a mechanical rotor. The high-tension leads supplied by the

conversion kit are made to fit tightly into the spark plug housing to seal it against moisture. The spark plug used was the Motorcraft AGPR 22C, which is a medium heat range spark plug and the Champion C6YCC which is a cold spark plug. These were chosen because of their different heat ranges. The Champion spark plugs had an excellent heat transfer capacity to keep the electrodes cool and prevent hot spots that could lead to spontaneous combustion.

Timing adjustment

Once the timing has been set to approximately 15-20° BTDC, adjustments can be made by either loosening the gear (inside the engine) on the distributor and adjusting the rotor of the distributor or by loosening the stator housing of the distributor and doing minor adjustments there. The latter can be done while the engine is running.

5.1.3   Machining of pistons to lower compression ratio

Natural gas pistons were obtained from ADE. These were intended for a natural gas conversion and had to be machined to lower the compression ratio from 12.5:1 to 10.35:1. The compression ratio was lowered by increasing the combustion bowl volume, later modifications led to a further reduction in compression ratio as shown in Appendix A5.

At first machining was done on a NC machine, requiring long machine times and producing a slightly rough surface finish. For later changes a lathe was used with satisfactory results, although it required a small amount of hand finishing which lead to minute variations that were regarded to be within acceptable limits.

Cylinder head modifications.

Figure 5.2 shows all the changes that were put into effect by Mercedes Benz Brazil to convert the engine from diesel to gas. Much of this can be achieved by employing turbo-charged engine cylinder heads. This was the route followed in this project. This represented a compromise, since these heads produce high swirl which is not desirable in a gas engine. The reason for this being that in gas or spark ignition engines the source of combustion is at the spark plug, and flame propagation needs to occur outwards radially

from that point. Swirl is not conducive to radial flame propagation, it is built into direct-injection diesel engines to improve mixing of the injector spray which sprays radially outward from the injector.

① MoCr-alloyed cast iron
② heat and wear resistant valves
③ heat and wear resistant exhaust valve insert
④ omission of ring carrier
⑤ spark plug connector with seal
⑥ induction port for optimized volumetric efficiency
⑦ spark plug
⑧ increased quenching gap
⑨ increased combustion chamber
⑩ spark plug trough
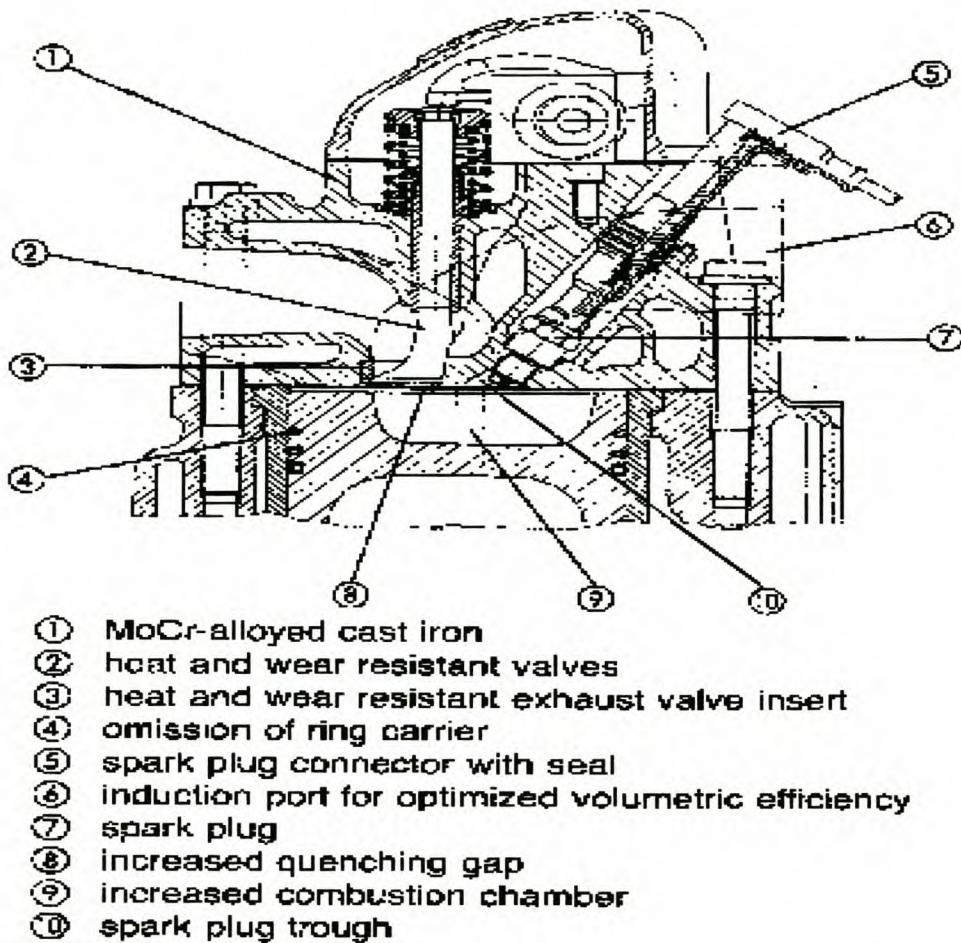
Figure 5.2 Changes to the cylinder head of the standard diesel engine (SAE 952289)

The engine had to be de-rated to 694Nm and 145kW in order to match the strength of the gearbox and drive train of the bus it was designated for. Therefore the compromise of using a high swirl head was accepted and a minimum of changes was made to the cylinder heads. The only machining done was to the injector holes to take spark plugs.

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5.2 Gas-supply system

5.2.1   Installation of LPG regulator vaporizer

A LPG regulator and evaporator capable of supplying a 140kW engine was mounted on the inlet manifold side of the engine on the test bed. The coolant connections with 15mm inside diameter were fitted to the engine coolant inlet before the water pump and the first tapping on the engine block behind the air compressor. This connected the regulator directly to the internal loop of the engine coolant.

5.2.2   Gas-mixer design

The gas mixer was designed to be machined in two parts to be press-fitted together with the two fuel-supply pipes entering at 180 degrees relative to each other. Figure 5.3 shows the assembly. The exit cone is shaded and the most important dimensions are indicated.

The detailed calculations for the gas-mixer design can be seen in Appendix A4.
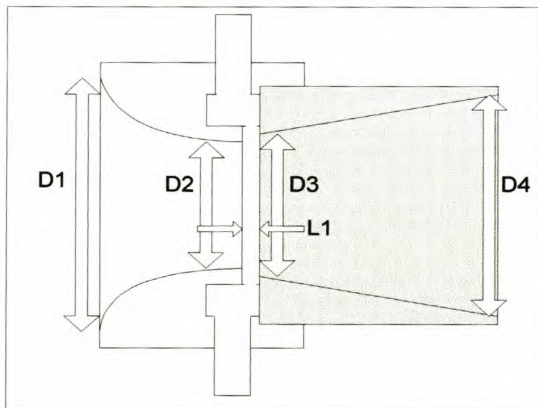
Figure 5.3 Layout of gas mixer design with main dimensions
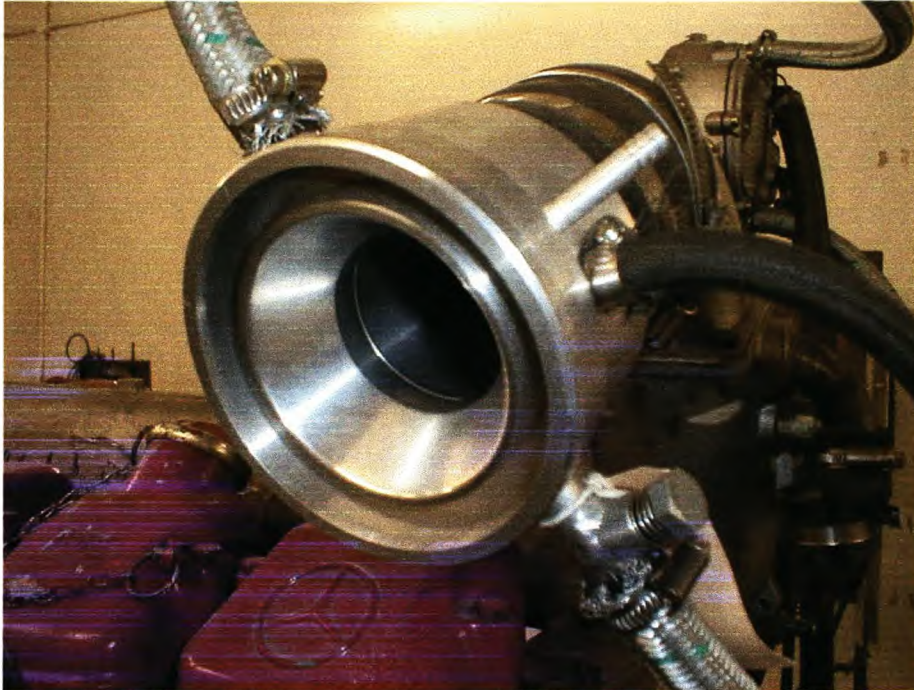
Figure 5.4 Gas mixer showing inlet cone, gas supply pipes and pressure tappings

The outer dimensions of the mixer were chosen to fit the throttle body on the exit side, and the air-supply pipe that is used in the bus on the inlet side. The dimensions of the two fuel hoses that enter the mixer were chosen to match the fittings that fit on the gas regulator. The inner dimensions were matched to suit the air and gas requirements of the engine using all the other dimensions mentioned above. This led to the design as shown in table 5.1 and figure 5.4.

Table 5.1 Mixer and fuel supply pipe specifications for the 447 engine

Mixer Calculations		Mixer Characteristic in mm		Supply pipe	
D1	90	Entry Diameter	Area Ratio		
D2	54	Throat diameter	2.778	Diameter	15
D3	56	Gas entry diameter		Length	1
D4	80	Exit diameter		Entry K factor	0.9
L1	4	Gap Width		K factor	1.9

Since the engine had to be de-rated, it made the design of the mixer less critical in terms of pressure loss over the mixer so that an entry to throat area ratio was chosen that would ensure sufficient vacuum for the engine. For detail see figure 5.5. The exit cone of the mixer was designed with a 7 degree exit angle to provide pressure recovery.

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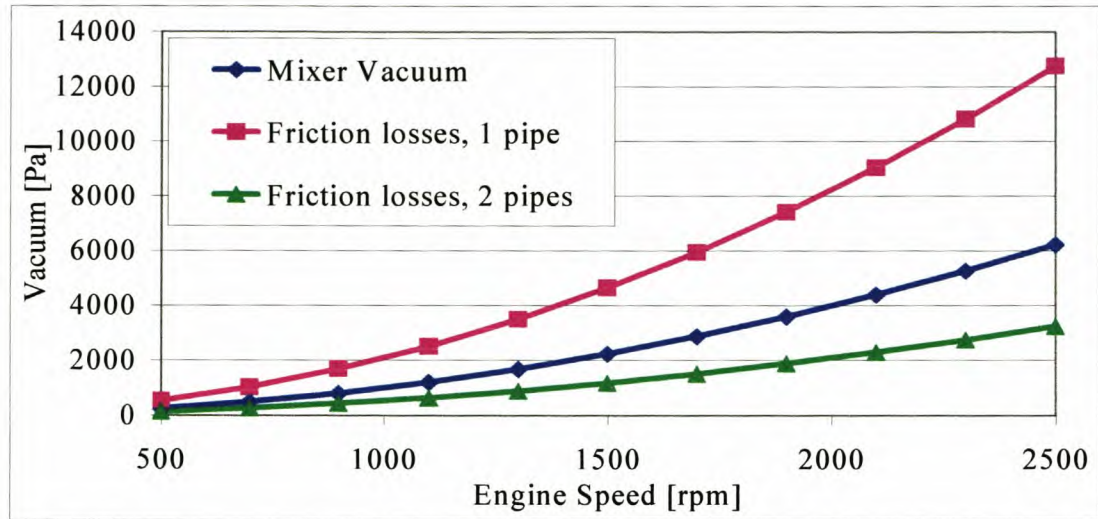
Figure 5.5 Mixer characteristics versus engine fuel requirement for 447 engine

Figure 5.5 shows the safety margin built into the mixer design by ensuring that the mixer is able to supply a vacuum exceeding the requirements of the fuel supply system of the engine. This was done to allow for better mixture control in order to improve performance under part-load and low engine speeds where airspeeds are low. One of the supply pipes was fitted with a manually variable restriction for rough mixture adjustment, but the back pressure mixture regulation proved to be so superior to this method that it was later abandoned.

5.2.3   Mixture regulation system

The gas conversion kit supplied by Mercedes Benz Brazil contained a cam-driven mixture regulation system that assembles to the engine with the distributor as shown in figure 5.1. This system is connected to the inlet manifold pressure via a variable restriction that is throttle position dependent. It is simultaneously referenced to the inlet air-pressure to compensate for blockages in the inlet or air-filter, as well as a reference pressure at the throat of the venturi to make fine-tuning possible. To override the system an adjustable bleed-valve to atmosphere was used to facilitate mapping during preliminary tests.
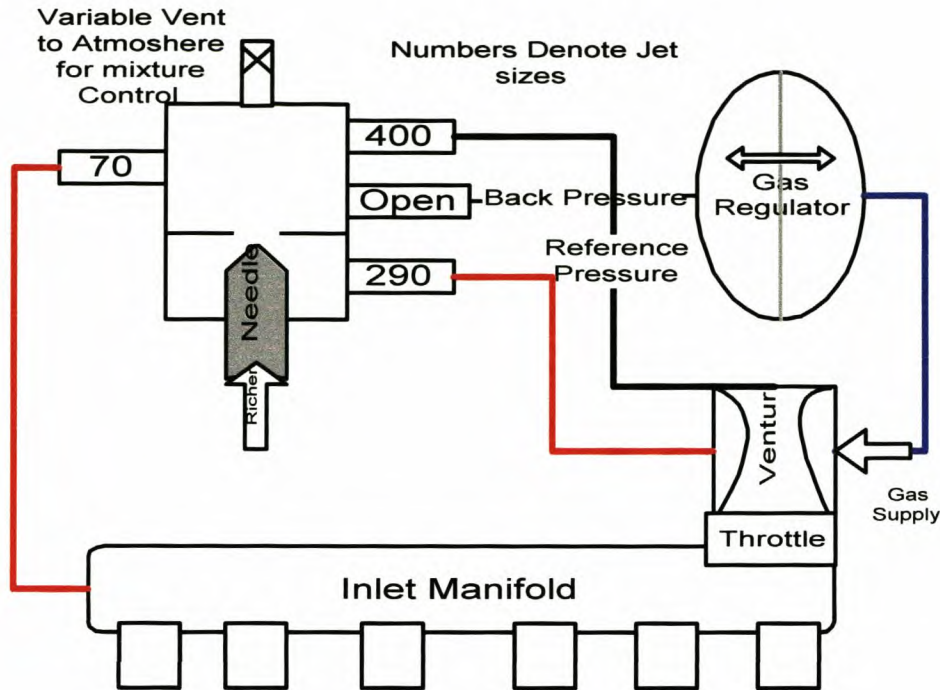
Figure 5.6 Mixture regulation device on 447 engine

The system is used to change the back pressure on the regulator diaphragm and thereby reduce or increase fuelling. The gas-flow through the regulator is directly related to the pressure difference between the venturi and the back pressure making this an excellent control system. During idle, when very little air enters the engine, a constant flow of fuel is provided by a needle valve, since the mixer does not supply sufficient vacuum under these conditions to ensure adequate fuelling.

5.3 Combustion chamber study

In a diesel engine the squish gap is made as small as possible to create high bowl entry velocities in order to improve turbulence and combustion. Tests with a diesel engine converted to run on LPG showed that the engine would shudder violently once certain manifold pressures were reached. It was suspected that this might be an indication of spontaneous combustion in the squish zone. To determine if the squish characteristics of the engine were causing this, the following study was made.

Research by Bergmann and Busenthur (1987) indicated that when running on natural gas a quencing gap of 2.5mm "has proven to be superior" to that of the 0.8-1.2mm of a similar diesel engine.

The calculations and detail of geometry are shown in more detail in Appendix A5.

Squish velocity
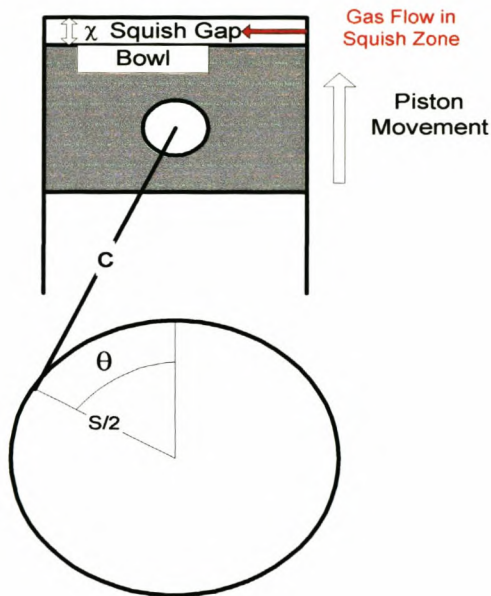
Figure 5.7 Parameters describing piston movement

Piston movement can be described in terms of the following:

$c = conrod$ _ $length$
$S = piston$ _ $Stroke$
$\theta = angle$ _ $BTDC$ _$[radians]$
$x = piston$ _ $crown$ _ dis tan ce _ $from$ _ $cylinder$ _ $head$

$$x = \frac{S}{2}(1 - \cos\theta) + c(1 - c \times \cos[a\sin(\frac{S \times \sin\theta}{2c})])$$

Equation 5.1

$$\frac{dx}{d\theta} = \frac{1}{2}S \cdot \sin\theta + \frac{S^2 \sin\theta \cdot \cos\theta}{4(c[1 - \frac{1}{4}S^2 \frac{(\sin\theta)^2}{c^2}]^{\frac{1}{2}})}$$

Equation 5.2

The change in volume above the piston can be described in terms of the equation above.

$$\frac{dx}{dt} = \frac{dx}{d\theta} \times \frac{d\theta}{dt}$$

Equation 5.3

while: $\quad \dfrac{d\theta}{dt} = \varpi \quad\quad$ ϖ is the rotational velocity in radians per second $\quad$ or

$$\varpi = rpm \cdot \frac{2\pi}{60}$$

Equation 5.4

This gives the change in squish volume per time as:

$$\frac{dV}{dt} = \frac{dx}{d\theta} \times \varpi \times A_{Squish}$$

Equation 5.5

where $\quad A_{Squish} = \dfrac{\pi}{4}(D^2 - d^2)$

Equation 5.6

is the area above the squish zone of the piston.

The squish velocity as a function of diameter with a piston (with central bowl) is:

$$V_d = \frac{dV}{A_d dt} \quad \text{where} \quad A_d = \pi(x + \gamma)d$$

Equation 5.7

γ = the clearance above the piston when on TDC

Fully expanded this is:

$$V_d = \{\frac{1}{2}S \cdot \sin\theta + \frac{S^2 \sin\theta \cdot \cos\theta}{4(c[1 + \frac{1}{4}S^2 \frac{(\sin\theta)^2}{c^2}]^{\frac{1}{2}})}\} \cdot rpm \frac{2\pi}{60} \cdot \frac{(D^2 - d^2)}{(x + \gamma)d}$$
Equation 5.8

Geometric simplification

The equation above is for a symmetrical arrangement but the piston bowl is normally not in the center of the piston. The geometry was adjusted according figure 5.8 below to simulate the worst case scenario by centering the bowl so that the maximum squish distance is the same as that of the offset bowl in the real engine.
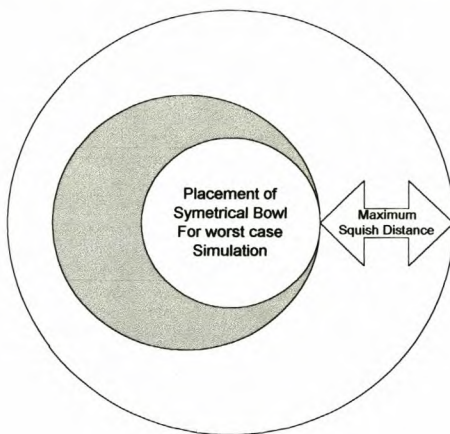
Figure 5.8 Simplified geometry of squish area

With this arrangement the only area in which the equation applies is in the region of the arrow shown above. Since it is only the maximum squish that we are interested in, this can be used with reasonable certainty.

The results shown in figure 5.9 for entry speed into the bowl were obtained using equation 5.8.
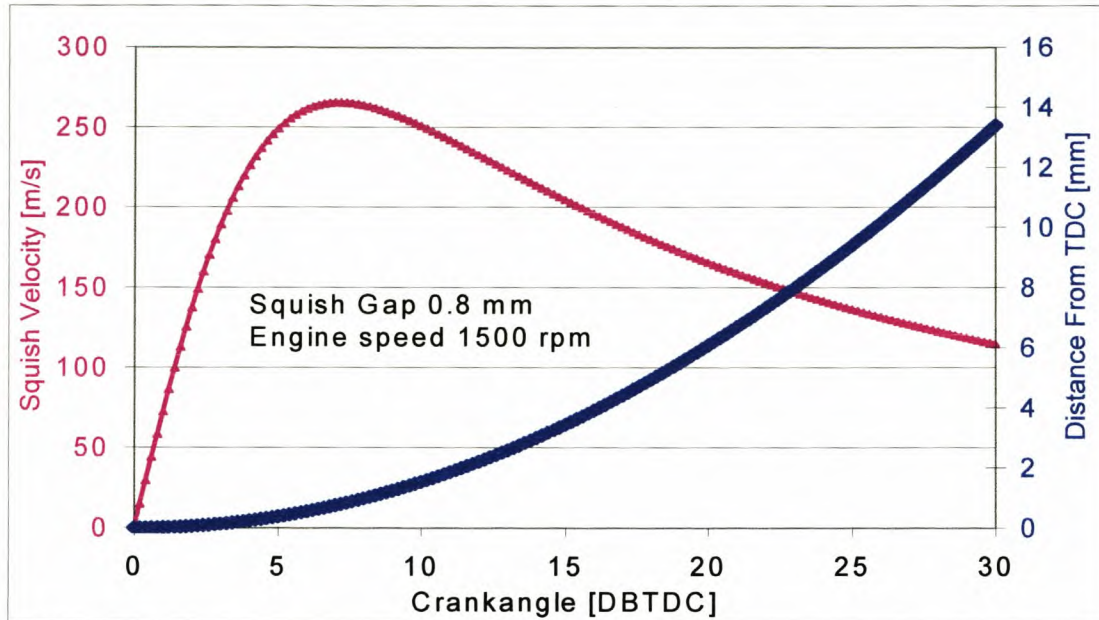
Figure 5.9 Squish velocity as function of crank angle

Figure 5.9 shows that the maximum squish velocity occurs at about 7 degrees BTDC. From the velocity the pressure in the region near the cylinder wall can be calculated with the use of the energy equation.

The effect of compression on temperature

Isentropic compression was used to simulate the effect of compression on cylinder pressure and temperature according to the following relationships:

$$\frac{T_2}{T_1} = r^{(\gamma-1)} \quad \text{Temperature relation} \qquad \frac{P_2}{P_1} = r^{\gamma} \quad \text{Pressure relation} \qquad \textbf{Equation 5.9}$$

The compression ratio could therefore be used for predicting in-cylinder temperatures without heat transfer as a first order approximation to see if the squish effects are large enough to cause auto-ignition. This was done for all crankangles near TDC where auto-ignition is most likely to occur.
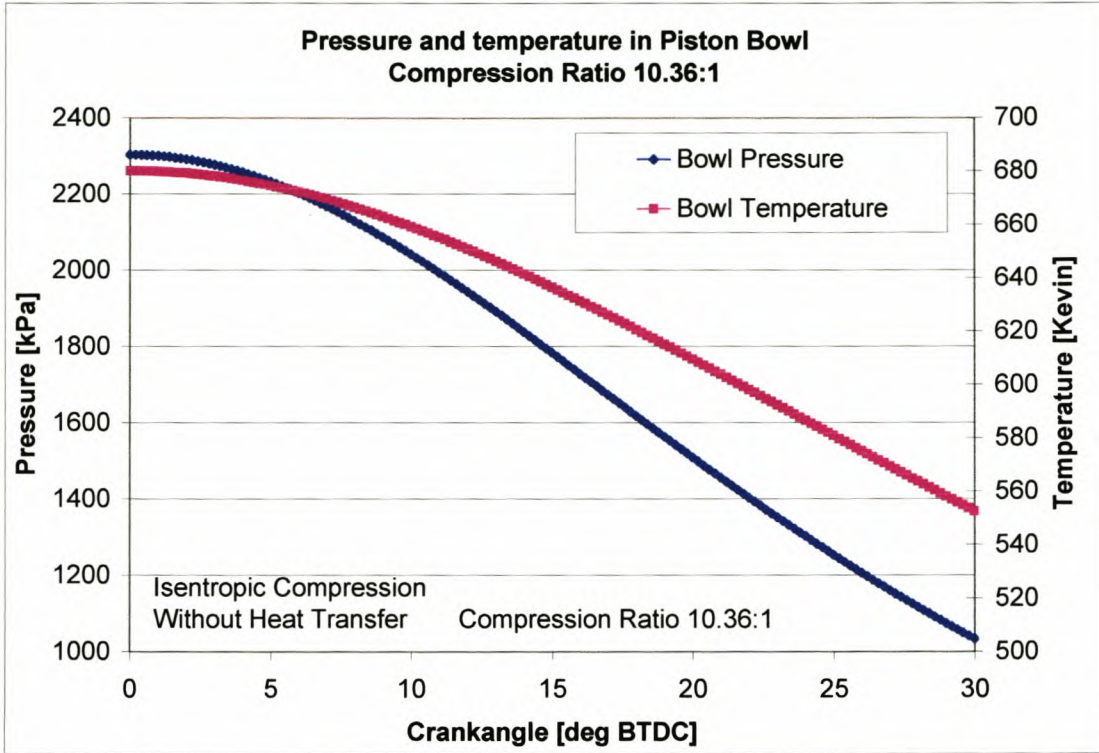
Figure 5.10 Effect of compression on gas temperatures

Squish effects have the greatest influence at about 7 degrees BTDC, whilst the compression effect is at its highest at TDC.

Pressures in the squish zone

To calculate the pressures that may occur in the squish zone we use the energy equation:

$$\frac{p_1}{\rho_1} + \frac{V_1^2}{2} = (\frac{p_2}{\rho_2} + \frac{V_2^2}{2}) + w_f + Q \qquad \textbf{Equation 5.10}$$

Q = Heat transfer $\qquad w_f = Friction\_Work$

Assumptions

As a first order approximation, this can be simplified by *neglecting friction and heat transfer*, thus:

$$p_1 + \rho_1 \frac{V_1^2}{2} = \rho_1(\frac{p_2}{\rho_2} + \frac{V_2^2}{2}) \quad \text{With } V_1 = 0 \text{ where } V_1 \text{ is the velocity at the cylinder wall.}$$

If incompressible flow is assumed: $\rho_1 = \rho_2$ this simplifies to the following equation which can be recognised as Bernoulli's equation for *incompressible flow*.

$$p_1 = p_2 + \rho \frac{V_2^2}{2}$$

Equation 5.11

To determine if the flow can be regarded as incompressible, the flow mach number was calculated for each crank angle using equation 5.11.

The maximum mach number of 0.514 was calculated at 7.3 degrees BTDC with a squish velocity of 275 m/s and a bowl temperature of 666.7 Kelvin. The engine conditions were 1500 rpm with a quenching gap of 0.8mm. At these mach numbers (0.3-0.8), flow is in the subsonic region and compressibility does play a role. The assumption of incompressibility is therefore not sufficiently accurate and can at best be seen as a first order approximation.
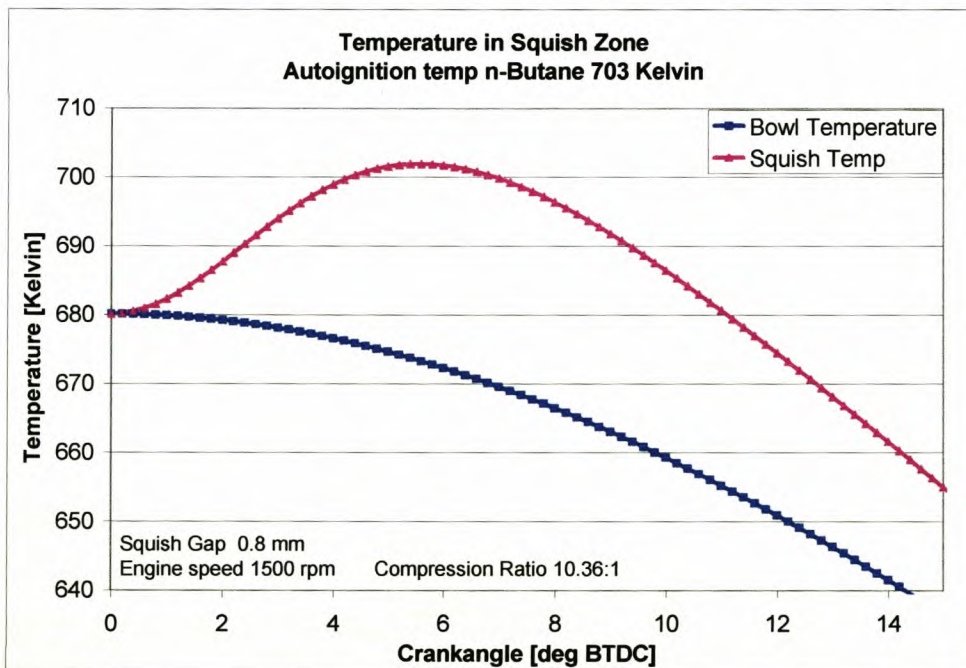
Figure 5.11 Gas temperatures in the squish zone

As can be seen from figure 5.11, the temperatures in the squish zone are approximately 20 degrees higher than those in the piston bowl, due to the effect of the squish velocity gradient from the wall of the cylinder to the piston bowl.

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<u>Increasing the quenching gap</u>

The only way to reduce the squish velocities is by increasing the quenching gap. This requires removal of material from the piston head. For practical purposes a gap of 2.5mm is chosen.

With the quenching gap set to 2.5mm at 1500 rpm the maximum Mach number is 0.297 so that the assumption of incompressibility, used for the calculations, is acceptable.
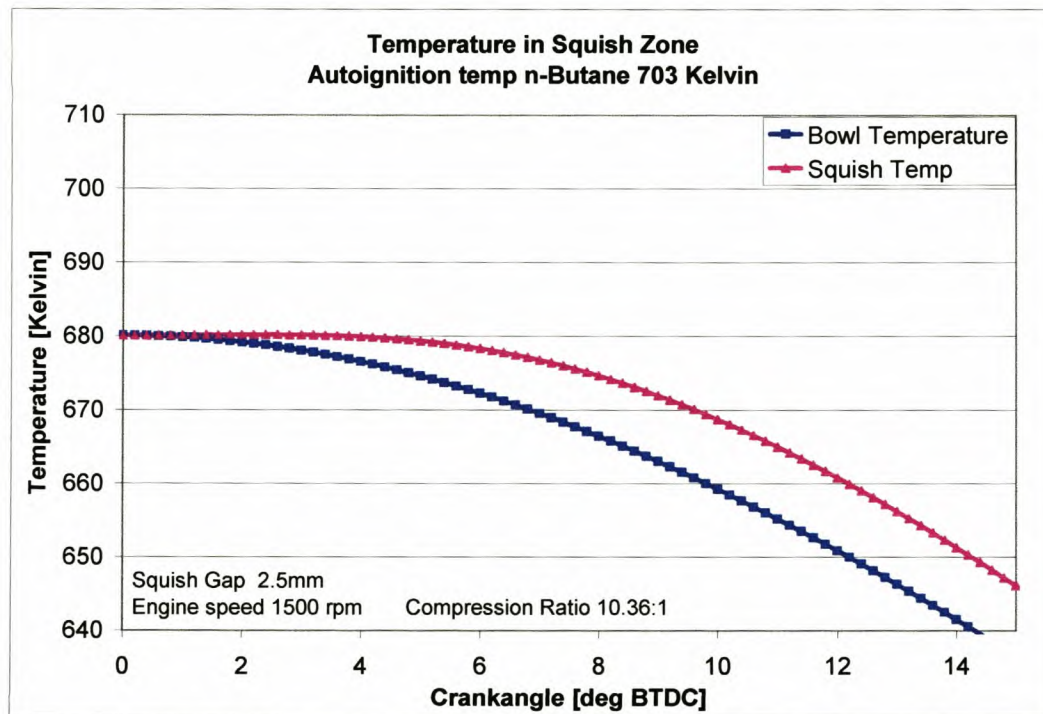
Figure 5.12 Temperature with reduced squish

Figure 5.12 shows how increasing the quenching gap to 2.5mm has the effect of lowering the pressures in the squish zone so that these do not exceed those that occur outside the squish area as a result of compression. Note that although this has been demonstrated at one speed only, the trend applies generally for all speeds. This illustrates that the increased quenching gap has had the effect of keeping the temperatures in the squish zone below those where auto-ignition may occur.

5.3.1 The effect of compressibility, heat transfer and friction

The neglect of compressibility has already been partially discussed, but to see the effect

of including it, the basic equation has to be studied: $p_1 + \rho_1 \dfrac{V_1^2}{2} = \rho_1(\dfrac{p_2}{\rho_2} + \dfrac{V_2^2}{2})$

with V_1 (the velocity against the cylinder wall) being zero this becomes:

$$p_1 = \rho_1(\frac{p_2}{\rho_2} + \frac{V_2^2}{2}) \hspace{3cm} \textbf{Equation 5.12}$$

Due to compression $\rho_1 > \rho_2$ so that the result will be a higher pressure in the static zone. This indicates that neglecting compressibility will lead to a conservative estimate of the temperatures in the end-gas region.

When friction is included, the energy equation indicates that the pressures in the end-gas regions would be higher due to the work done by the gas to overcome friction and still exit to the bowl at the same velocity, which is the requirement of conservation of mass.

$$p_1 = p_2 + \rho(\frac{V_2^2}{2} + w_f) \hspace{3cm} \textbf{Equation 5.13}$$

$$Friction\_Factor : \tau_w = \frac{1}{8}f\rho V^2 \hspace{3cm} \textbf{Equation 5.14}$$

A friction factor of 0.015 can be used for fully turbulent flow with a low relative roughness (White, 1998, p313).

The heat transfer in the squish zone would be high due to the large area to volume ratio and high gas velocities. Since the velocities in the end-gas region are zero, the heat transfer there will be the lowest. The piston surface temperature is considerably lower than the mixture temperature so that the heat transfer to the piston top will have the effect of quenching the flame. This is why it is sometimes called the quench gap. Neglecting heat transfer, therefore, will have the effect of over-predicting the effect of squish on the temperatures in the squish zone.

5.3.2   Results of squish study

It was not considered to be in the scope of this project to determine the exact temperatures in the squish zone but rather to evaluate whether squish could lead to auto-ignition. From this preliminary study it becomes clear that the squish of a diesel engine could cause auto-ignition due to high local pressures in the squish zone. Maximum pressures in the squish zone occur at about 6 degrees BTDC. Heat transfer to the piston would lessen this effect while inclusion of friction and compressibility would increase it. Increasing the quenching gap will alleviate the problem although squish is still desirable for increasing combustion tempo. The optimum quenching gap is a function of fuel type, maximum engine speed, compression ratio and piston geometry so that it would be different for each engine. For the ADE 447 engine Bergmann has found a 2.5mm quenching gap to be optimal.

5.4 Final engine configuration

As a result of the squish study, the squish gap was modified by removing material from the cylinder heads. This was done in such a way as to remove a minimum of material resulting in a gradual removal of material from the side of the piston crown, increasing to a maximum at the piston bowl as shown in figure 5.13. The final compression ratio with the reduced squish piston was 9.5:1.
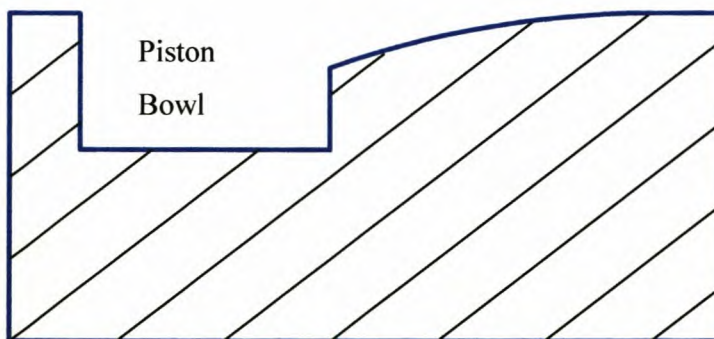
Figure 5.13 Cross-section of modified piston crown

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6 LIGHT DUTY ENGINE TEST RESULTS

6.1 Test experience

LPG injection

The initial poor performance with LPG injection was attributed to misfiring due to insufficient spark energy. The standard spark plug gap for the engine was 0.75-0.80mm. This was first lessened to 0.65mm upon which the performance was markedly worse! When the gap was widened to 0.90mm combustion became much more stable. This was attributed to the fact that larger spark plug gap requires higher voltages to initiate spark. This would lead to an increased spark intensity that would improve the initiation of combustion.

LPG induction

With LPG induction it was not necessary to change the standard gap of the petrol engine to get good combustion. It is common practice to decrease the spark plug gap for LPG use by about 0.2mm to overcome the higher resistance to sparking with the gaseous fuel.

The spark plugs used in the standard engine were NGK BF6ES which is a medium heat range spark plug. Spark plugs fitted for testing were NGK BF5ES which are one range hotter than the standard. The spark plug on number two cylinder melted during testing with LPG. It had survived running under similar conditions on 95 octane petrol. The problem was traced back to a broken thread insert on no. 2 cylinder that had created an air gap between the spark plug and the cylinder head, leading to a reduction in heat rejection from the spark plug that caused it to overheat. The effect of this was not immediately evident on engine performance. It is believed that the hot spark plug caused ignition of the mixture but at non-optimum timing. The loss in power was more pronounced at low engine speeds than at high engine speeds. Power was reduced by about 20 %. The molten metal caused no observable damage to the cylinder liners.

To rectify the problem a new insert was fitted and spark plugs used that are two ranges colder than the original, namely NGK BF8ES.

6.1.1 Torque

Figure 6.1 shows the torque recorded with all the different combinations of fuels and fuel-supply methods.
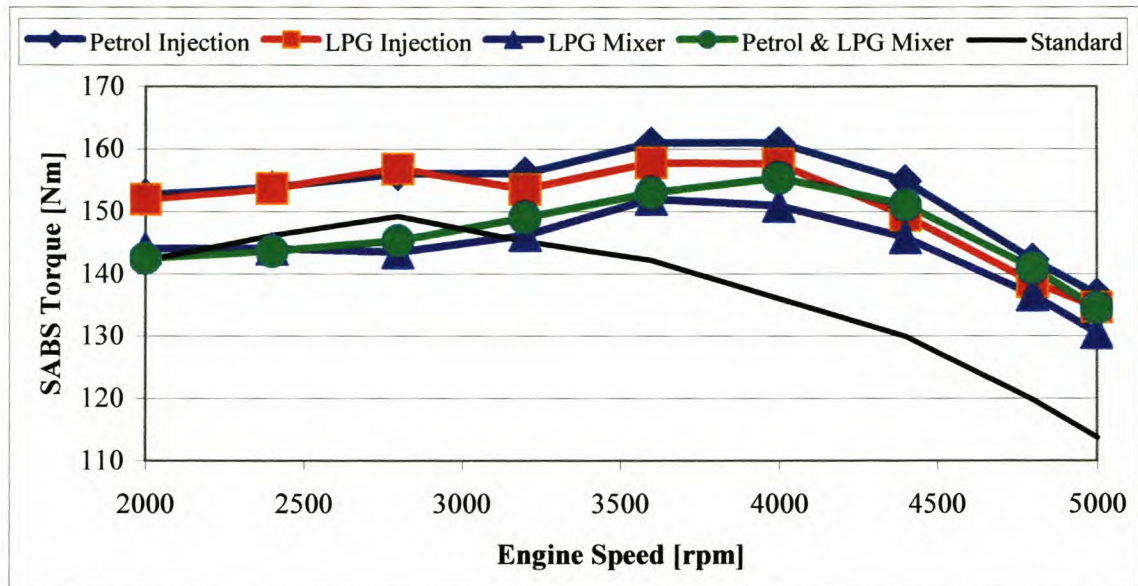
Figure 6.1 Petrol and LPG torque comparison

A maximum torque of 161Nm was recorded with petrol injection at 3600-4800rpm. The LPG injection achieved 158Nm, a 2 % decrease, whilst LPG induction obtained a maximum of 152Nm. When compared to the standard carburetor NA 20 engine which used a different manifold, torque had improved by 6 to 20 % across all engine speeds. Engine performance calculations were corrected for atmospheric conditions using SABS torque corrections. The good mixing achieved with LPG induction in the gas-phase lead to good low speed torque characteristics. This can be seen from figure 6.1 where LPG carburettion produced more torque than petrol injection at 2000 rpm. This advantage is lost at higher engine speeds due to increased turbulence that leads to better mixing of fuel air, thereby accelerating the combustion process. An interesting fact is that up to 3600 rpm the difference in torque between LPG and petrol is so small that driveability would not be hampered by LPG use. Only at higher engine speeds will a difference between the fuels become noticeable.

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NOTE: The tests on petrol injection with LPG mixer and LPG induction indicate lower torque than the other tests. This is due to tests being done in-between causing valve seat wear and loss of compression. The effect of valve leakage seems to be more pronounced at low engine speeds than high engine speeds so that performance at high engine speeds is less affected. At low engine speeds it is not accurate to compare the two sets of tests directly, due to this effect

6.1.2   Power output

LPG injection (liquid-phase) and petrol injection produced almost identical power output at engine speeds up to 3000 rpm. At engine speeds above 3000 rpm petrol injection outperformed LPG injection by up to 2 % at 5000 rpm.
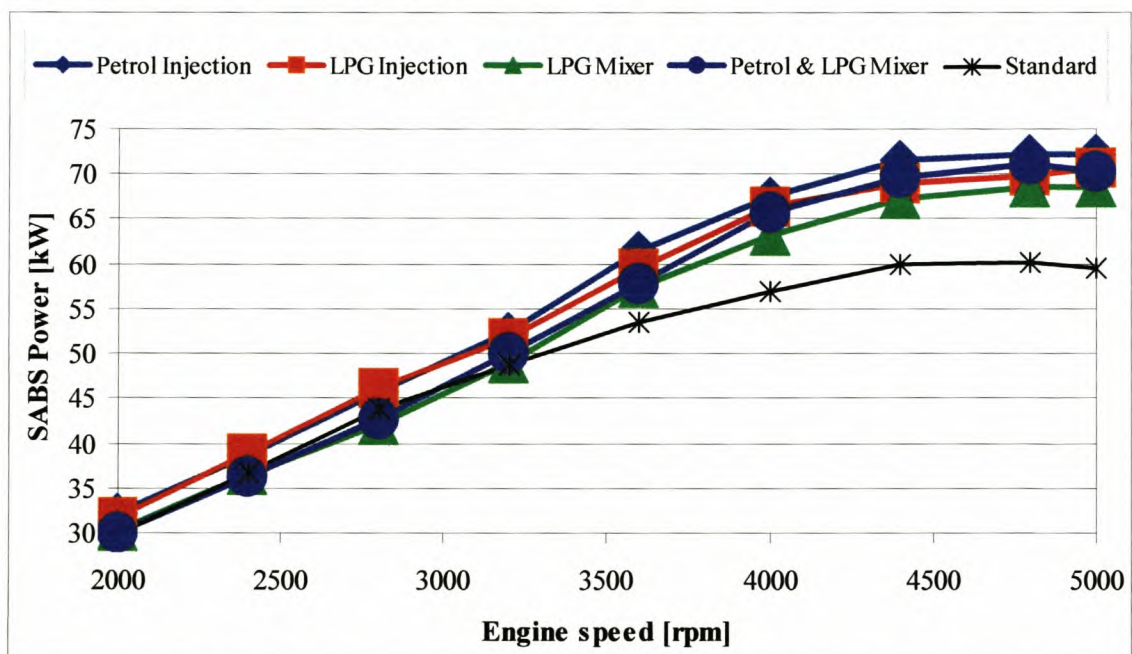
Figure 6.2 SABS corrected power for US20 with different fuel systems

A maximum power output of 72kW was attained at 4800rpm on petrol injection. The fitting of the LPG mixer reduced this to 71kW at 4800rpm while LPG injection delivered 70kW at 4800rpm and LPG induction 68.5kW at 4800rpm.

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This represents a reduction of 2.8 to 4.1 % for LPG injection and LPG induction when compared to petrol injection.

6.1.3   Specific fuel consumption (SFC)

Specific fuel consumption varied between 0.24 and 0.35 kg/kWhr at Wide Open Throttle (WOT). Average values over the test can be seen in table 6.1.

Table 6.1 Average specific fuel consumption

Averaged Values	STD	P-INJ	LPG-INJ	LPG-IND	P-INJ+GC
SFC [g/kWhr]	346.3	260.7	263	262.5	317.1
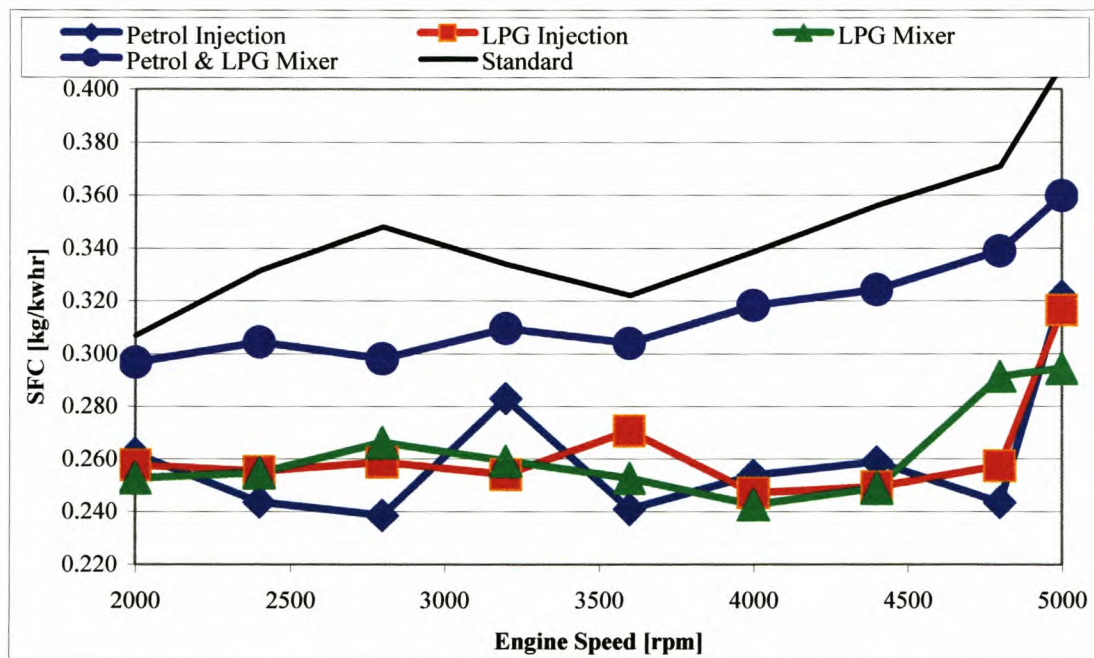
Figure 6.3 SFC for US20 with different fuel systems

The changes to the standard engine lowered the average specific fuel consumption by 25 % while the difference in SFC between LPG and petrol fuelling amounts to a mere 0.9 %. It is further expected that the SFC with LPG can be lowered below that of petrol with better mapping, since the engine was running with excess fuel at most test points with LPG. The test results can be seen in more detail in Appendix B2.

6.2 Emissions with different fuel systems

NOTE: all the emissions data given here are at WOT unless otherwise specified. The results can be seen in more detail in Appendix B2.

6.2.1 Hydrocarbon emissions
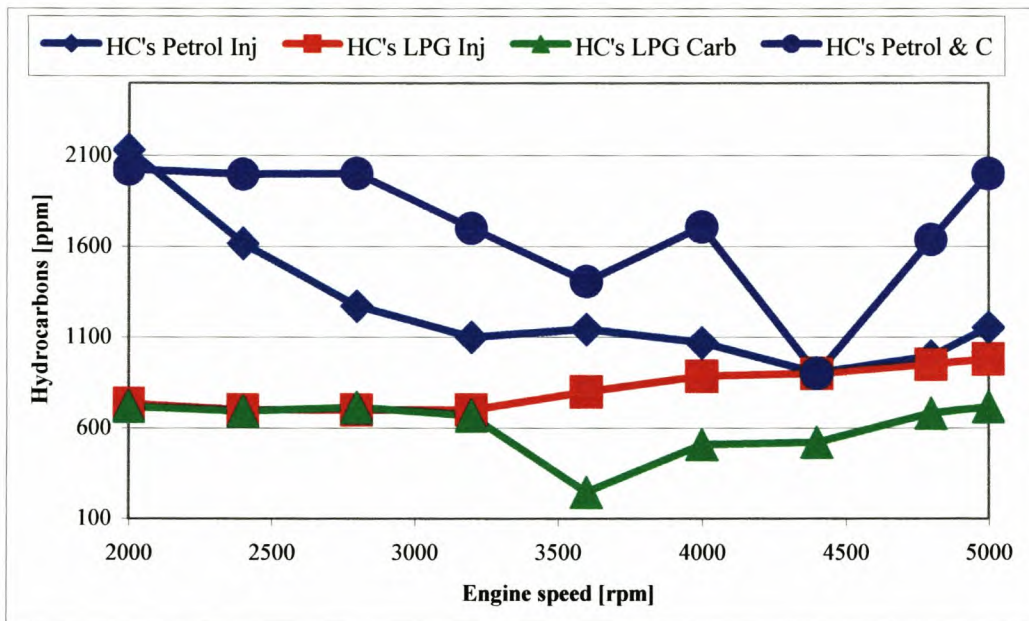
Figure 6.4 Hydrocarbon emissions for US20 with different fuel systems

The lowest hydrocarbon emissions were recorded using LPG induction in the gas-phase. Values ranged from 300 to 700 ppm. LPG injection produced the second lowest hydrocarbon emissions with values between 700 to 1000 ppm while petrol injection produced values between 900 to 1900 ppm. Averaged values can be seen in table 6.2.

Table 6.2 Average hydrocarbon emissions

AVERAGED VALUES	P-INJ	LPG-INJ	LPG-IND
HYDROCARBONS [ppm]	1264	816	606

Of interest here is that, above certain engine speeds, LPG injection displays hydrocarbon emissions similar to that of petrol injection. This is an indication that at high engine

speeds evaporation of the liquid LPG has not been complete by the start of combustion and liquid-phase fuel gets trapped in the crevice volume. This fuel does not combust completely due to quenching in the crevice volume. With petrol injection the hydrocarbon emissions decrease with increase in engine speed due to higher turbulence and more effective fuel-air mixing at high engine speeds. With LPG induction even high CO readings failed to produce high hydrocarbon emissions. At low engine speeds LPG injection produced low hydrocarbon emissions, possibly because there is sufficient time for liquid-phase fuel to evaporate, and thus it does not get trapped in the crevice volume.

6.2.2   NO$_x$ emissions

Emissions of NOx and CO are represented in figure 6.5, it illustrates the correlation between CO and NOx, with low CO giving high NOx readings and visa versa.
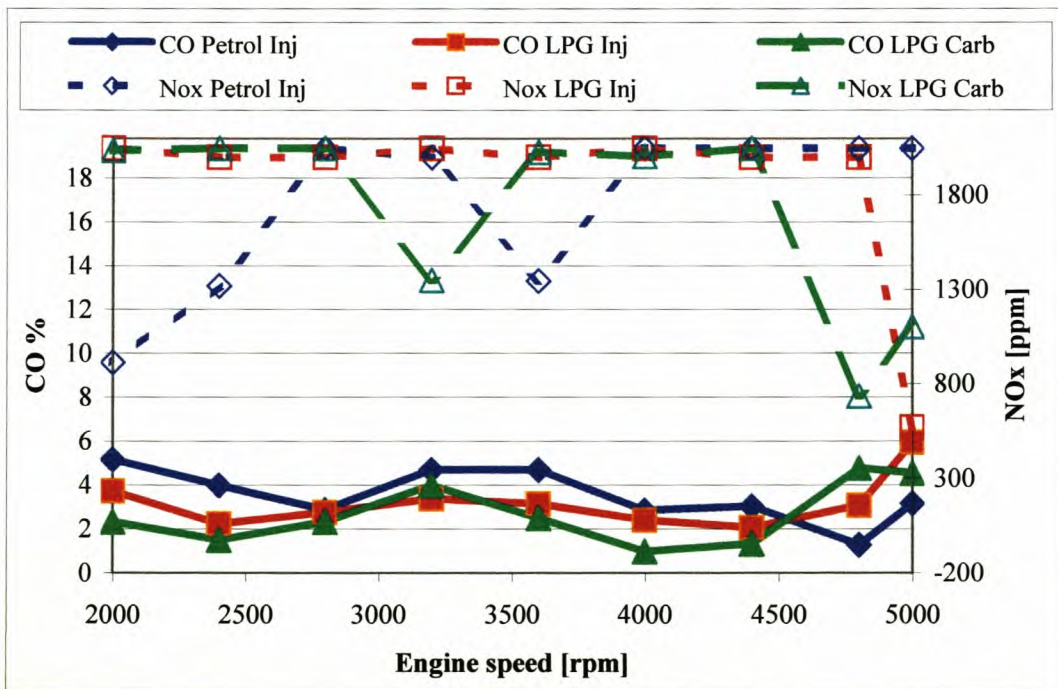
Figure 6.5 NOx emissions of US20 with different fuel systems

The upper limit of the analyzer was 2000 ppm. Figure 6.5 indicates that most of the values were off the scale of the analyzer and no comparison is possible. On average, LPG injection produced the most NOx, followed by petrol injection and LPG induction.

This may well indicate that the maximum pressures were experienced in the engine with LPG injection, due to greater charge densities at start of compression.

Table 6.3 Average NO$_x$ emissions

AVERAGED VALUES	P-INJ	LPG-INJ	LPG-IND
NOX [PPM]	1757	1856	1711

Of interest is that for different fuels the NO$_x$ levels would decrease at different CO readings. With LPG induction in the gas-phase levels would drop at a CO of about 4 %. With petrol-fuelling NO$_x$ emissions levels would only reduce at CO readings of 4.5 % and higher.

6.2.3   O$_2$, CO and CO$_2$ emissions

O$_2$ levels varied between 0 and 0.3 % with little discrimination possible between the different fuels and systems.
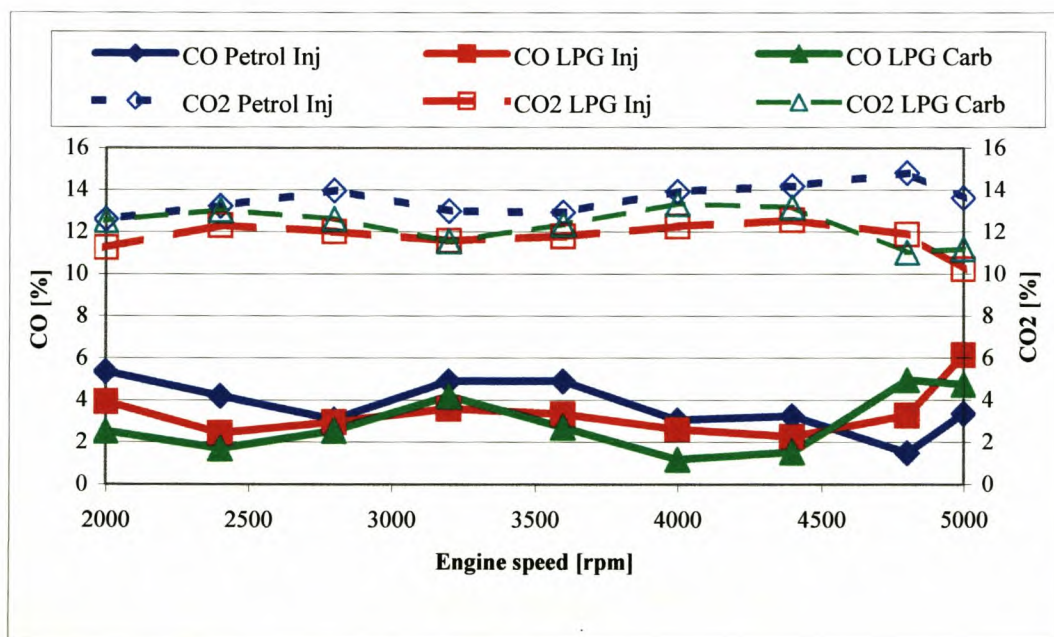
Figure 6.6 CO and CO2 emissions of different fuel systems

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Table 6.4 Average O₂, CO and CO₂ emissions

AVERAGED VALUES	P-INJ	LPG-INJ	LPG-IND
O2 [%]	0.14	0.19	0.16
CO [%]	3.72	3.39	2.89
CO2 [%]	13.59	11.77	12.31

CO_2 readings varied between 11 and 15 % with petrol-fuelling yielding the highest of those values.

6.3 Combustion analysis

Rapid Acquisition of Combustion and Engine Results in short RACER is software developed at the University of Stellenbosch for combustion analysis using in-cylinder pressure measurements (M.R. McLaren, 1995). Both one-zone and two-zone combustion models can be analyzed with this code. For this work the one-zone model was used. For inputs it receives crankangle position and TDC signals together with the amplified pressure pulse recorded in cylinder no. 4. The reference plates for crankangle position and TDC position monitoring was placed on the flywheel side closest to cylinder 4 to minimize the effect of incorrect readings due to crankshaft distortion.

The method used in the analysis was to average pressure data over 200-270 cycles to eliminate cyclic variation, depending on the stability of combustion. The combustion chamber is seen as a control volume in which energy input versus energy output can be calculated from cylinder pressures and the instantaneous heat transfer rate using the equation of Woschni (1967). This was used to calculate how much energy is released by combustion at each crankangle, as well as total heat transfer and in-cylinder temperatures.

6.3.1 In-cylinder pressures

In-cylinder pressures were measured at every crankangle and averaged over 200-270 cycles at all the prescribed test points. For reference, a 120-tooth wheel was mounted close to the cylinder in which the pressures were measured to minimize the effect of crankshaft torsion. To reference TDC another plate and pickup was used.

Pressures were referenced manually by using the inlet manifold pressure as a reference for in-cylinder pressure at the close of the inlet-valve.

A typical example of pressures measured for the different fuels are shown below.
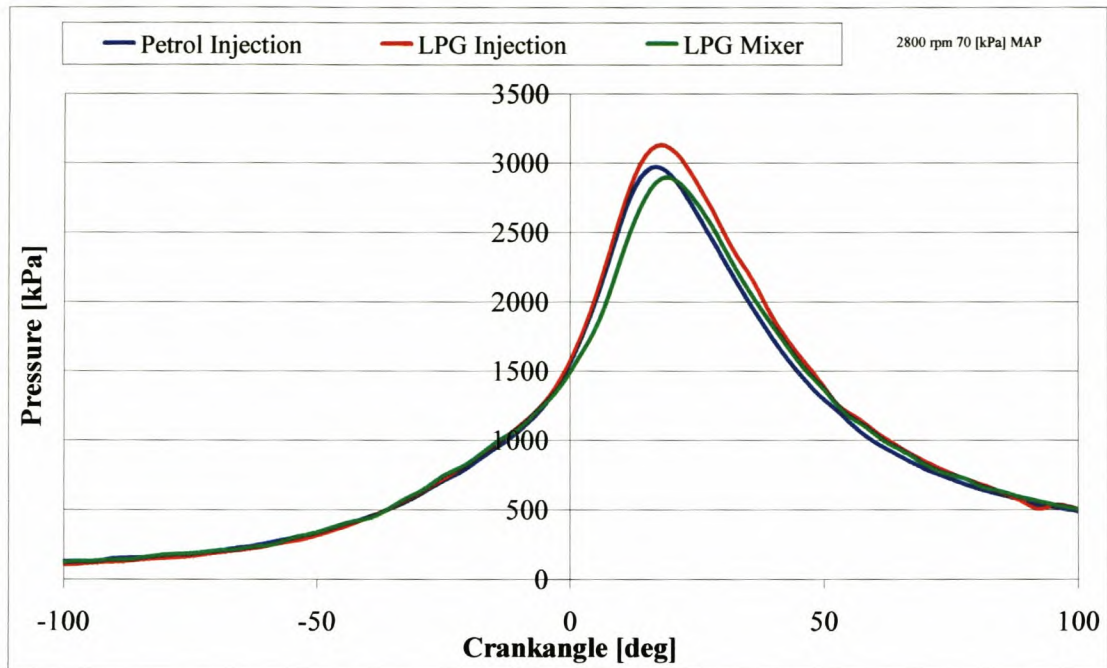
Figure 6.7 Measured in cylinder pressure versus crankangle for different fuel systems

From these measurements combustion analysis was done using the RACER software developed at the Center for Automotive Engineering (CAE).

6.3.2   Heat release rates

From the measured in-cylinder pressures the heat release rate, as a result of combustion, can be calculated if the heat transfer from the combustion chamber to the coolant is known. For the purposes of this calculation, the instantaneous heat transfer coefficient as determined by Woschni (1967) was used in software developed by M'cLaren (1995). Examples of the resultant calculated heat release rate is shown in figure 6.8.
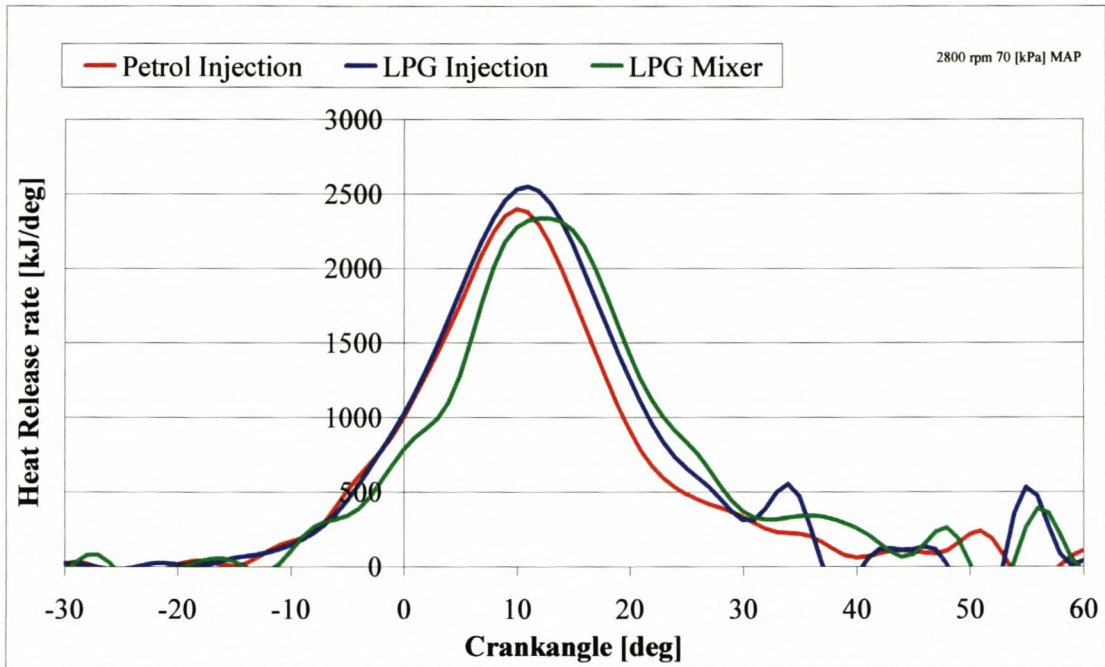
Figure 6.8 Calculated heat release rates of different fuel systems

From figure 6.8 it appears that the heat release rate of LPG injected in the liquid-phase is much higher than that of any other fuel at this engine speed. This can be ascribed to the nature of the fuel-air mixture that indicates that the gaseous fuels show improved combustion at low engine speeds due to good mixing with air. The in-cylinder pressure results can be seen in more detail in Appendix B3.

6.3.3   In-cylinder temperature and total heat released

Total heat released is calculated from the heat release rate integrated over the total cycle during which heat is released from start of combustion to end of combustion. Figure 6.9 shows the calculated gas temperatures and the accumulative heat release for the cylinder pressures shown in figure 6.7.
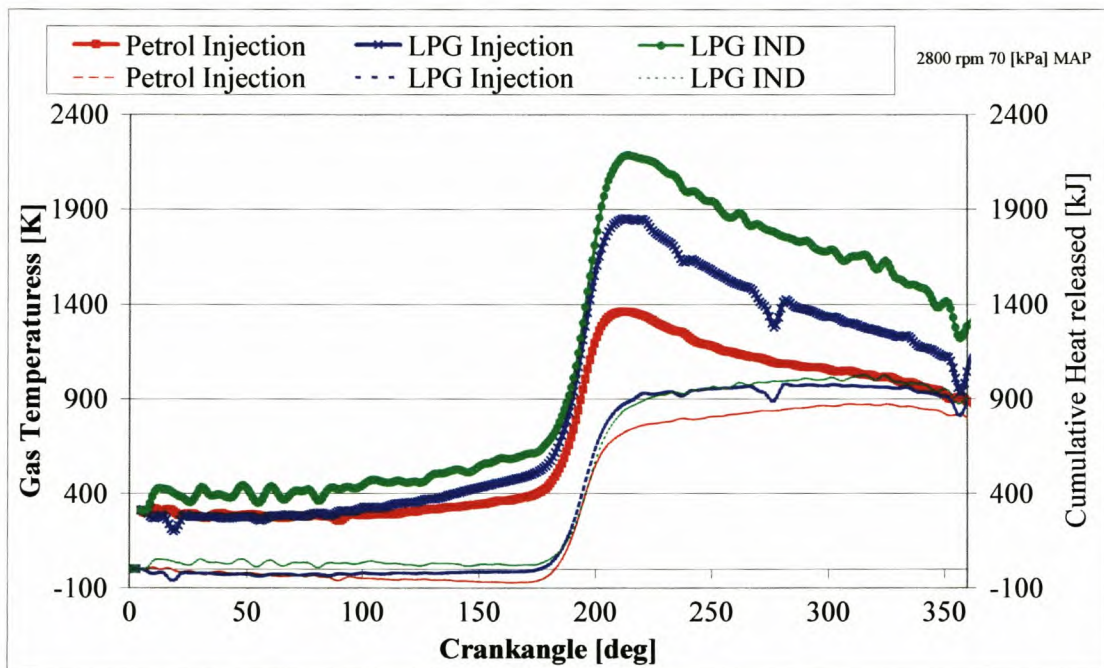
Figure 6.9 Calculated gas temperature versus total heat released for different fuel systems

The calculated gas temperatures, which are calculated as a function of pressure, indicate clear differences between fuels that require evaporation and those that do not. Gas induction in the gas-phase gave much higher cylinder temperatures than liquid-phase LPG and petrol injection. This indicates that the temperature of the fuel-air mixture is lower with fuels that are evaporated by inlet air, and the charge air mass in the cylinder is higher for these fuels.

6.3.4 Maximum in cylinder pressures

The maximum pressure of 62bar was recorded with LPG injection at 3600 rpm. This is at the point of maximum engine volumetric efficiency and when the cooling effect of LPG evaporation gives highest charge densities. This value is just higher than the value of 60.6Bar recorded with petrol injection at 5000 rpm. LPG induction produced a maximum of 57.5bar at 3600rpm. The recorded maximum pressures are shown in figure 6.10.
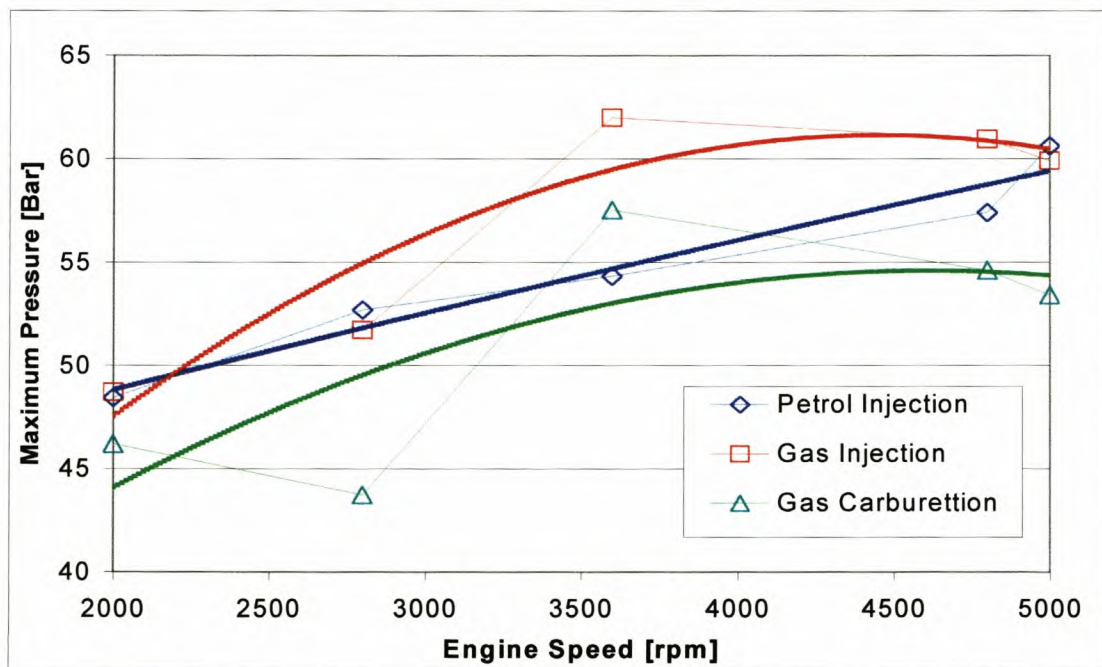
Figure 6.10 Maximum pressures with different fuel systems

Due to scatter in data, trend lines were fitted to make the general trends clearer. Trends are shown in dotted lines. It indicates that pressures are generally at a maximum at speeds where torque is at a maximum. Evaporative cooling with LPG injection increased charge densities and therefore maximum pressures. The trends consistently indicate that liquid-phase LPG injection cause the highest resultant pressures followed by petrol injection, with LPG induction displaying significantly lower in-cylinder pressures than both of these.

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6.3.5   Maximum in-cylinder temperatures

The highest maximum temperatures were calculated with LPG induction, where the calculated values were about 200 degrees higher than those with either petrol injection or liquid-phase LPG injection. This was also observed through the tests in which one spark plug with a broken insert was molten with LPG induction after surviving a full test on petrol injection. The maximum calculated in-cylinder temperatures are shown in figure 6.11.
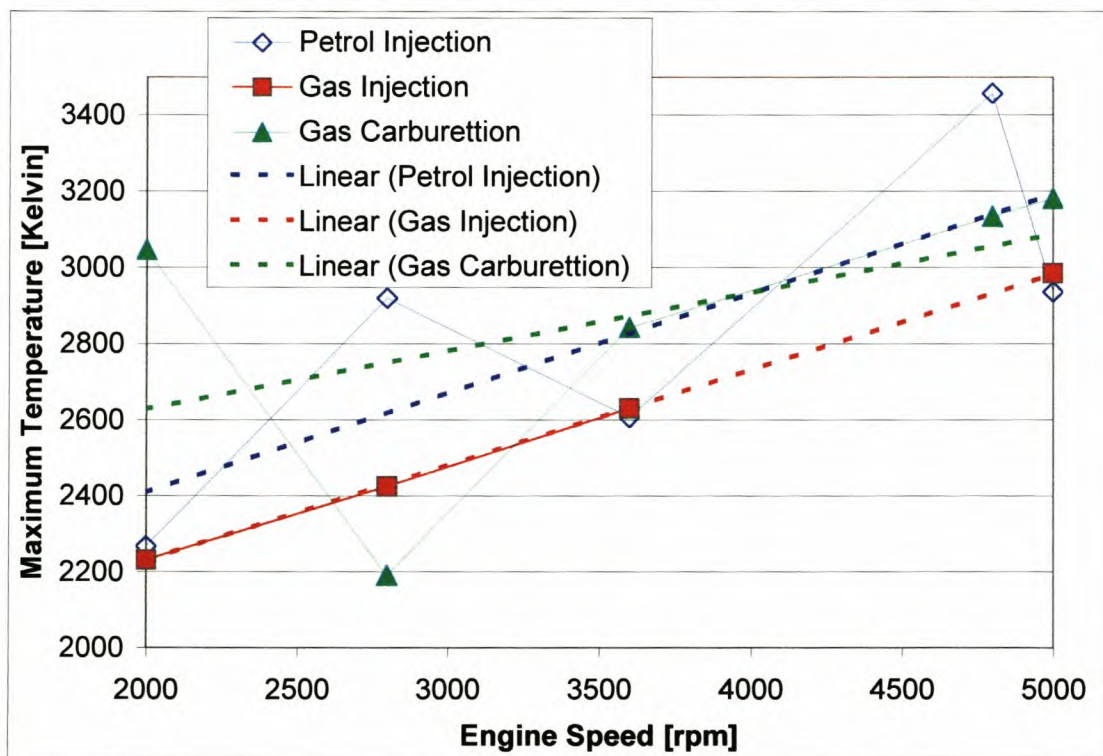
Figure 6.11 Calculated maximum combustion temperatures with different fuel systems

The calculated maximum gas temperatures show great variation with LPG induction and petrol injection, while LPG injection indicate a perfect linear correlation between engine speed and maximum temperatures. It is therefore suspected that the great variations displayed are due to experimental variations and trend lines were fitted through the data to show the general effects. Figures 7. 9 and 7. 10 illustrate the effect of evaporative

cooling with LPG injection that increased the maximum pressure and lowered the maximum gas temperature by as much as 200°K compared to petrol injection. It is suspected that the temperatures for LPG induction shown here are not entirely correct. LPG induction will produce higher temperatures than petrol injection at all engine speeds, as is indicated by tests with both fuels, where spark plugs that survived a test on petrol injection melted during a test with LPG induction. This inconsistency is related to differences in fuel-flow which indicate that the fuel-flow with LPG induction was lower than it should have been at some test points as can be deduced from figure 6.12.
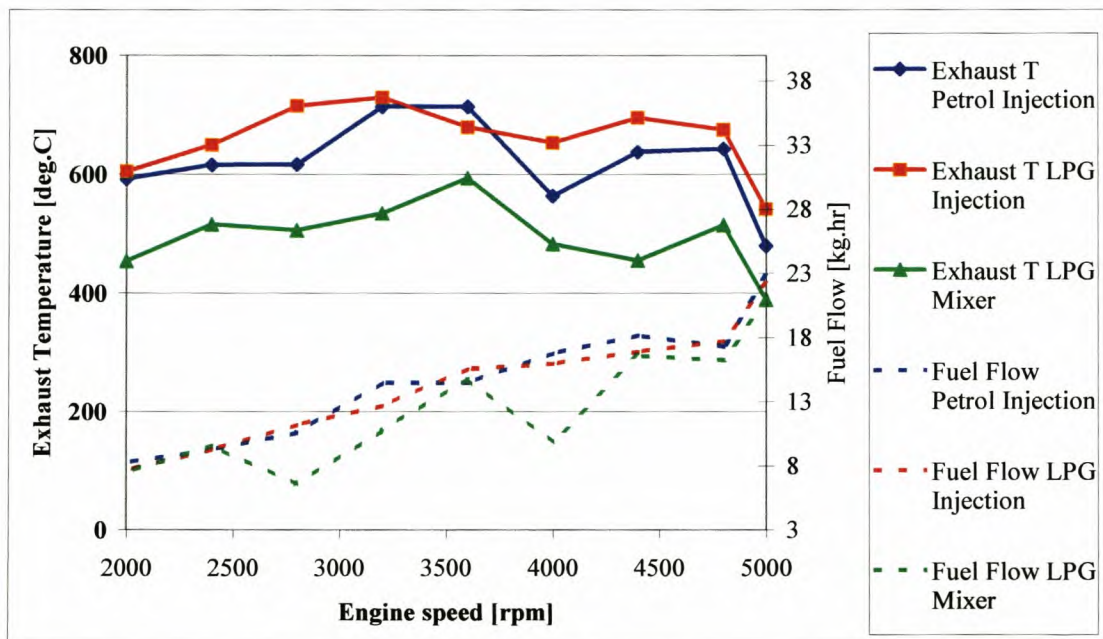
Figure 6.12 Fuel flow and exhaust temperatures for different engine speeds

The exhaust temperatures can be clearly related to fuel-flow. The drop in exhaust temperatures at 5000 rpm are not representative, since the test was started at high speed and kept there for only a short duration, which did not give the exhaust time to warm up completely.

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6.3.6 Calculated combustion duration with different fuel systems

Combustion duration is measured as the angle of crankshaft movement that is required from start of combustion to the end of combustion. Since the initial slope of the heat release curve is small it is difficult to determine the exact start of combustion. Therefore, the interval 2-98 % or 5–95 % of total heat released is used for practical purposes. In this study the 2-98 % interval are given.
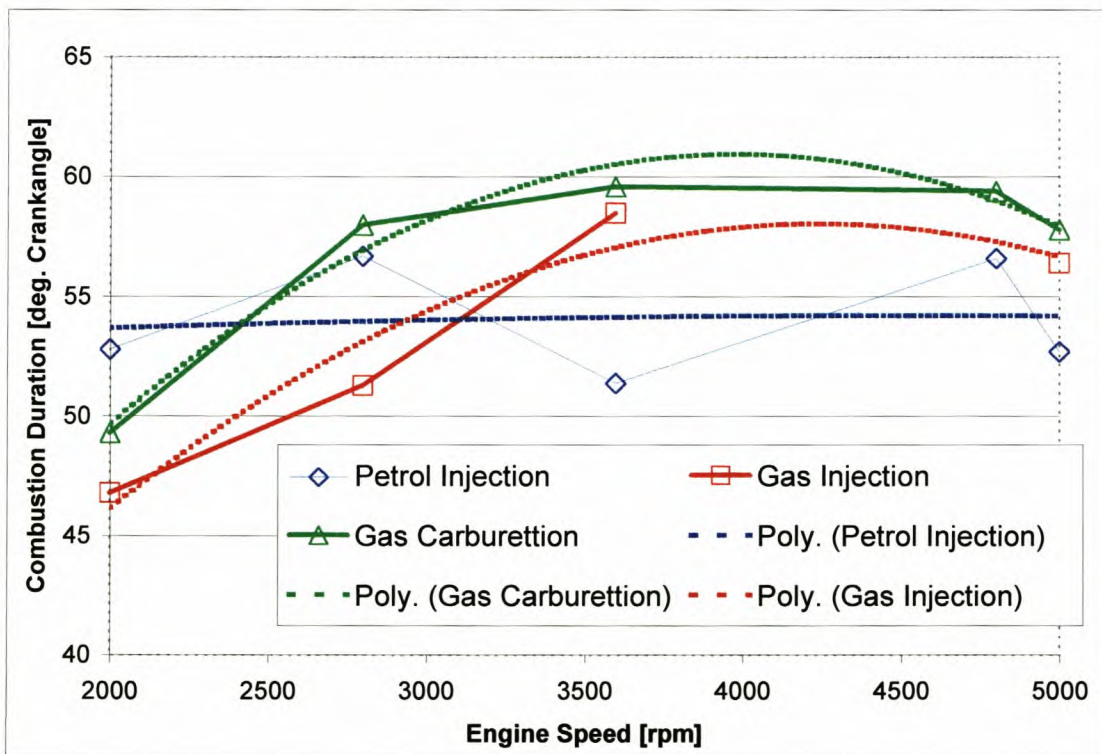
Figure 6.13 Combustion duration (2-98 %) with different fuel systems

The combustion duration values displayed in figure 6.13 illustrate the difference between the different methods of fuelling and the different fuels used. The trend-lines show how combustion duration for petrol-fuelling remains constant with varying engine speed while combustion duration with gas-fuelling starts off with short combustion duration at low engine speeds, and increases at higher engine speeds.

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6.4 Computer simulation of combustion effects

Computer modeling was used to predict the effect of changes to the combustion process. The program used was written for a simple one cylinder one-zone model with isentropic compression and expansion. The program code and some of the results generated can be seen in Appendix C. The program uses the instantaneous heat transfer coefficient as described by Woschni (1967) to determine in-cylinder pressures due to compression and heat released during combustion. The model does not consider the gas exchange processes, this had to be specified by the user as volumetric efficiency.
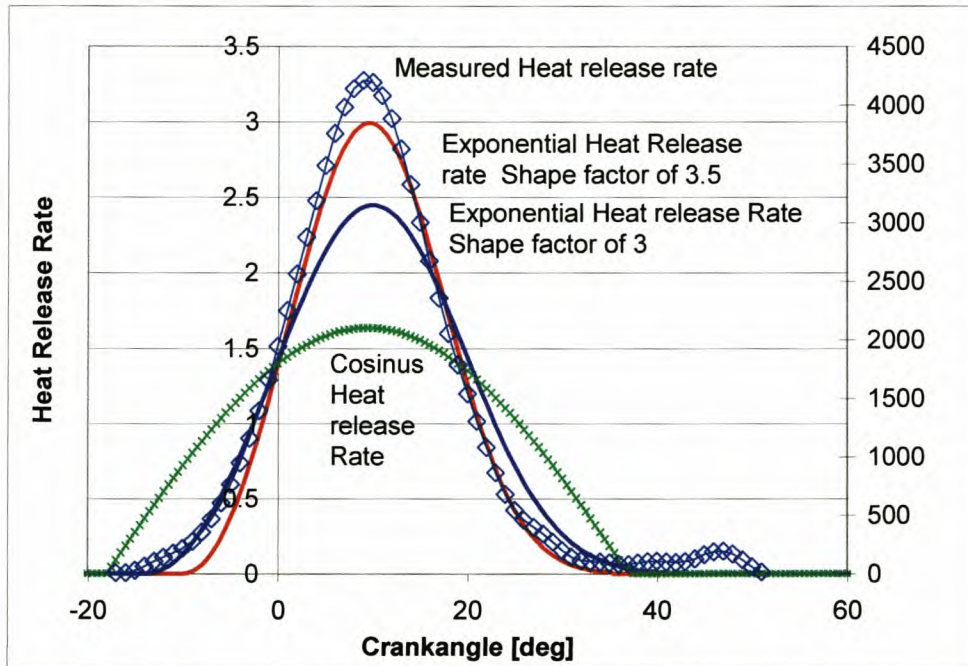
Figure 6.14 Different heat release curves used for combustion simulation

Figure 6.14 illustates how the measured heat release rate is simulated for a 55° burn angle by using different mathematical functions that can be tuned to fit the heat release measured on the engine. The values on the axes differ because the one represents actual values measured on the engine, while the other is in non-dimensional units that are used to determine the shape of the heat release curve. The different heat release rates that are shown in figure 6.14 were derived from the formulas described by Ferguson (1985). The heat release rate χ can be determined as a cosine function, $\chi = \frac{1}{2}\{1 - \cos[\frac{\pi(\theta - \theta_s)}{\theta_b}]\}$ or as an

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exponential function, $\chi = 1 - \exp[-(\frac{\theta - \theta_s}{\theta_b})^n]$ where n is the shape factor, θ_s is the start of combustion and θ_b is the combustion duration. The heat release rate has to be multiplied by the fuel-heating value to get the actual heat release rate for a specific fuel. All the curves in figure 6.14 were drawn for the same combustion duration. The exponential curve with a shape factor of 3.5 fits the measured data the best. This is the heat release function that was used in the computer simulation to match this engine.
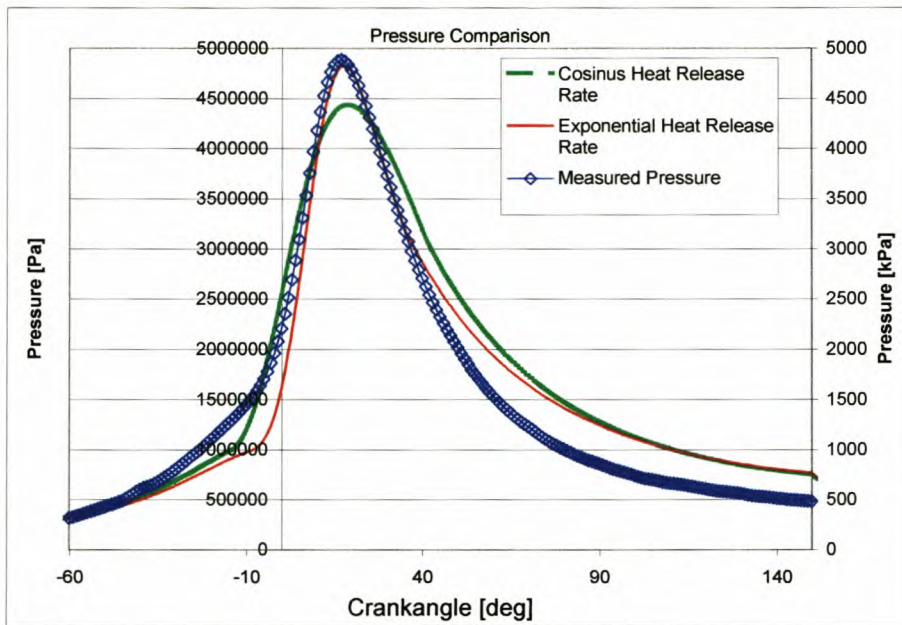
Figure 6.15 Simulated pressures using different heat release rates

Figure 6.15 represents the pressures generated by the computer simulation using the sinusoidal and exponential heat release rates. It indicates that the exponential heat release rate predicts the maximum pressures closely, whilst it deviates from the measured pressure in the compression and expansion during and after combustion. This indicates that the coefficients used in the Woschni equation were not yet properly matched to this engine. The performance results obtained with the model however, correlated well enough with those in the engine. So much so, that it was considered adequate to use the model to study the trends that can be expected with varying engine parameters, such as combustion duration and spark advance.

6.4.1　The effect of combustion duration on engine performance

The motivation for this study is to be able to explain the effects that fuel composition and combustion properties can have on the maximum attainable engine efficiency. Since different fuels display different combustion durations, this effect was simulated using the simple single-zone combustion program mentioned in section 7.4. The results are plotted in figure 6.16, and it is useful to be able to see how this affects engine performance.

For this study the combustion duration was varied from 30° crankangle to 65° crankangle. The spark advance was optimized for each combustion duration and the best results used to describe the effect of combustion duration on engine efficiency.

The chart is titled with x-axis "Combustion Duration [deg.]" and y-axis "Energy Balance [%]". The fitted trend line equation is $y = 0.0031x^2 - 0.4796x + 46.962$. Legend: Shaft Work, Coolant loss, Exhaust loss.
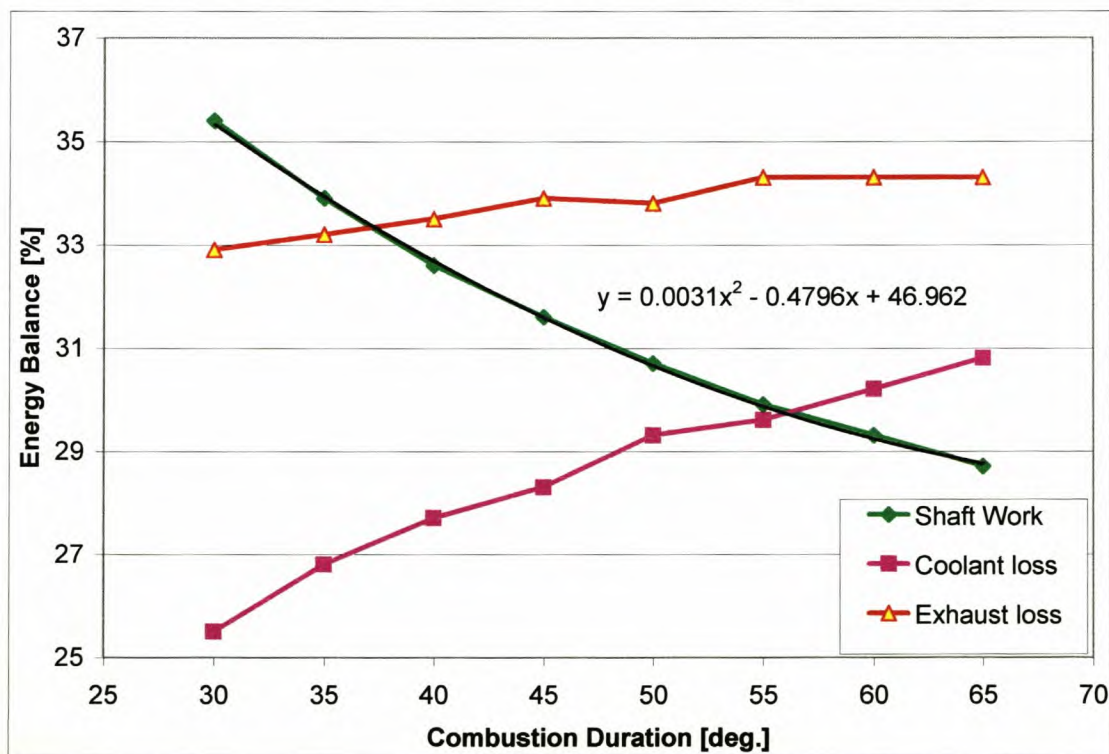
Figure 6.16 Effect of combustion duration on engine efficiency

Figure 6.16 clearly illustrates the advantages of using fuels that have a short combustion duration. Rapid combustion minimizes the loss of energy to the coolant system and the exhaust, thus producing better engine efficiency. A trend line was fitted to the data to quantify the effect numerically and the equation is displayed on the chart. Combustion

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durations measured during testing varied between 48-60 crankangle degrees when the 2-98 % mass burned criterion was used.

6.4.2   The effect of spark advance on engine performance

To see if the effect of combustion duration is more pronounced than the effect of running the engine at non-optimum timing, a combustion duration of 55° was chosen and the timing varied similar to a timing swing in a real engine.

The results are displayed graphically below:
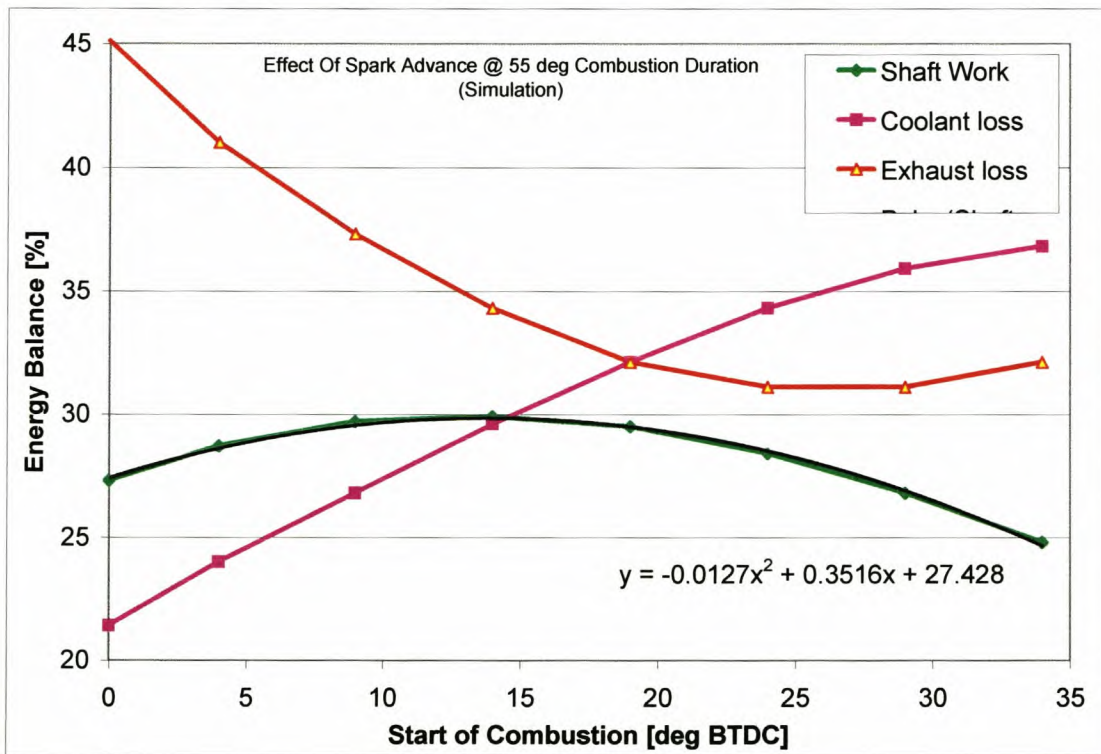
Figure 6.17 Effect of spark advance on engine efficiency

Note than the x- axis on figure 6.17 displays start of combustion and not spark advance, since there is a period after spark when pre-combustion reactions take place which may vary. When this is compared to the real engine it is found that optimum spark advance occurs at about 23° BTDC. This indicates that the induction period is in the order of 10 degrees of crankangle at this engine speed.

7 HEAVY DUTY ENGINE TEST RESULTS

7.1 Summary of results

The results of the tests done on the 12 litre dedicated gas engine, which was converted from diesel, can be seen in short in figure 7.1. It indicates torque relative to engine speed for different engine configurations that were tested on different dates. The legend shows the date of each test. The first of the tests done on 9 August 1998 was with a compression ratio of 10.5:1 and a maximum torque of 550Nm was attained during that test. This is just more than half of the 800Nm that the engine should be able to deliver. Problems with auto-ignition and high squish lead to a lowering of compression ratio to 9.5:1 and the use of a larger squish gap. The other tests were then done with the reduced compression ratio.
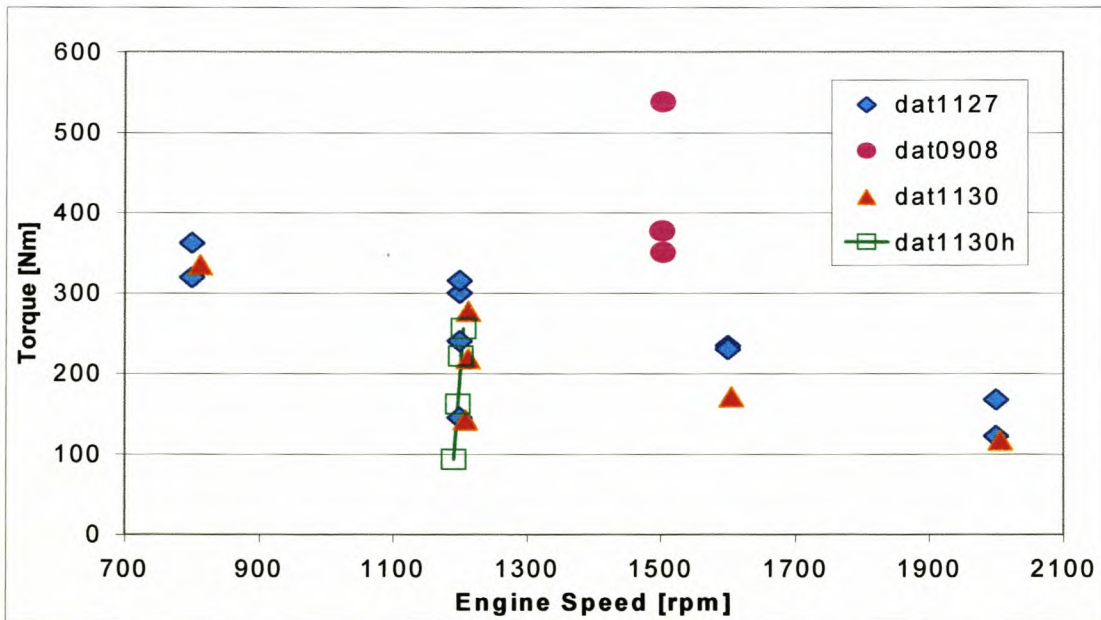
Figure 7.1 Summary of 447 test data

During the first test with a high compression ratio, the engine would shake violently once certain inlet manifold pressures were reached. The nature of the disturbance was such

that it was considered to be spontaneous combustion. Literature on the conversion of diesel engines to gas-fuelling revealed that a bigger squish gap is required with gas-fuelling. It was therefore thought that the high squish with the diesel engine could cause spontaneous combustion of a homogeneous fuel air mixture, thus the bigger squish gap. This was studied in some detail, as described in the conversion of the engine. The investigation in Chapter 5.3 proved that this could indeed be the case.

To prevent spontaneous combustion the squish gap was enlarged and the compression ratio lowered to 9.5:1. Tests at this compression ratio showed a great reduction in torque that was greater than the effect that the compression ratio reduction on its own could have on the engine. This led to the conclusion that auto-ignition with the higher compression ratio aided combustion, thus improving torque.

7.2 Discussion of results

From the test results it is concluded that the high squish contributed to auto-ignition of gas in the squish zone. This had two effects; one was to cause rapid combustion that would tend to make the engine vibrate and pre-ignite to the point where it would almost run in reverse. The other was that it acted as a source of ignition that enabled the engine to run at manifold pressures, where the spark ignition voltage was insufficient to initiate combustion. Hence the great reduction in torque once the squish gap had been enlarged. Another theory is that the high squish is detrimental to the pre-combustion reactions that have to take place in the spark plug gap, due to high gas velocities in that region. To evaluate this, the spark plugs were recessed further into the cylinder head. Since no difference in performance could be detected, this could not be confirmed.

After tests at a wide range of air-fuel ratios, the spark voltages were recorded. It was found that half of the spark voltage was lost in the distributor. It was therefore concluded that the spark plug voltage was insufficient to initiate combustion. Since no distributorless system could be obtained at the time, this could not be confirmed.

8 CONCLUSION AND RECOMMENDATIONS

It is recommended that the use of LPG as automotive fuel in South Africa should be encouraged. In this way the refineries will be able to sell the surplus of butane that currently exists. With a bigger market for butane, the volatility of petrol would decrease, resulting in smaller evaporative fuel losses with estimated savings of R100 million per year. This calculation is based on reducing evaporative emissions by half, i.e. that only 0.5 % of petrol would evaporate instead of the current estimate of 1 %.

It is estimated that LP gas vehicles will make up a small percentage of the light vehicle market, about 3 %. Ideally these vehicles should be used in areas where air quality is poor, thereby contributing to a cleaner environment where it is needed most. LPG-fuelling would be most suited to vehicles that return daily to a depot to be refilled. It would be advantageous if these depots were situated near crude oil refineries, as this would reduce transport costs. Ideal places to implement LP gas vehicle use would be along the coast in the major cities, such as Cape Town and Durban.

The use of LP gas vehicles will have another advantage; namely to introduce the use of gaseous fuels in South Africa's automotive industry. It can be used as a stepping stone to the use of natural gas vehicles which has the potential of capturing a greater percentage of the vehicle market than LPG, due to it's greater availability.

8.1 Development of LPG as automotive fuel in South Africa

8.1.1   Economical incentive for automotive LPG use

The use of LPG in South Africa will have to be economically-driven, since no other driving force exists at present. LPG can present an economical alternative to petrol depending on the amount of taxation placed on it. With the current taxation LPG compares favorably with petrol-fuelling.

8.1.2   The role of government

For LPG to be accepted as automotive fuel on any scale will require a clear and consistent Government policy towards the taxation thereof. Currently no tax is levied on LPG and the Government is not set to change this in the immediate future. Prospective users of LPG for automotive applications require assurance that this will remain the case for at least 5 years if it were to be attractive to convert vehicles for LPG use. It is therefore recommended that the government give at least a 5-year warning in lieu of prospective changes in policy towards LPG taxation. When tax is levied on LPG it should be of such a nature that the selling price for automotive LPG should not be more than 60 % of the selling price of petrol. Usually the Mineral and Energy Policy Center (MEPC) will advise the government with regards to decisions concerning fuel policies such as these.

8.1.3   Target market for LPG use

The focus market for LPG should be vehicles that are used within cities such as taxis, busses and municipal vehicles. This would lead to a reduction in evaporative, tailpipe and noise pollution in densely populated areas, where it is most required. Ideally these vehicles should form part of a captive fleet, as this will make it easier to set up a filling station at a central depot where vehicles can be filled on a regular basis. This will also make it easier to ensure that the vehicles are well maintained and do not become a safety risk.

8.1.4   Application of different LPG fuelling technologies

LPG carburettor systems
In general, LPG carburettor systems are the easiest to install and can be installed to vehicles equipped with carburettor engines, since these inlet manifolds are designed to transport both fuel and air. It may however, present a problem in fuel-injected vehicles where the inlet manifold is designed to transport air only. In these engines backfiring in the inlet manifold due to the presence of LPG air mixture may damage some of the sensors of the engine management system.

<u>LPG injection systems</u>

LPG injection systems can be grouped broadly into either liquid- or gas-phase systems. Liquid-phase injection systems can provide higher power output and faster response than gas-phase systems. On the other hand, liquid-phase systems are more complex than gas-phase injection systems and are therefore less common.

Both the carburettor and injection type systems have reached different stages of development leading to three generations of equipment, each with its specific advantages.

Generation 1 type systems, i.e. systems without any feedback, should be used in vehicles with similar technology; namely, carburettor vehicles and open-loop fuel-injected vehicles.

Generation 2 type systems (closed-loop systems) should be used with vehicles with similar technology, i.e. vehicles with lambda sensors and closed-loop fuel-delivery control.

Generation 3 type systems, (self-learning systems) should be used with vehicles with similar technology, such as the latest generation of vehicles where the engine-management adapts to the driving style of the driver, and the system is self-learning. These systems are particularly useful because of the low manpower required, since it does not need calibration to operate effectively.

When compared to similar technology petrol engines, emissions from a LPG engine, including CO_2, are substantially lower than those achieved with gasoline or diesel fuel. (Bosch Automotive Handbook, 4'th ed. p502).

8.2 Current activities

SASOL of South Africa is paving the way for use of natural gas as vehicle fuel by creating the infrastructure to pipe gas from the Pande gas field off the Mozambique coast to Gauteng, the industrial heart of the country. This will make large quantities of natural gas available in South Africa. The Department of Trade and Industry has declared its intention to encourage the use of natural gas as a vehicle fuel. It is also reported that Nissan may be releasing a LPG/NG version of the new Nissan Almera for the South African market. (Cartoday.com). The CAE is currently working on two projects together with British Petroleum (BP) South Africa to study and promote the use of LPG to

stimulate use of LPG in the Cape Town area. These projects are described in more detail in section 8.2.1 and 8.2.2.

8.2.1   Light vehicle trial

A Toyota Condor 1.8-litre utility vehicle has been equipped for LPG-fuelling.   The vehicle is used for commuting on a daily basis to gain experience in automotive LPG use. The vehicle is clearly marked as LPG powered to increase public awareness of LPG-fuelled vehicles.

8.2.2   Bus trial

Currently the CAE is preparing an engine for use in a 65-seater bus. The bus will be used in the Cape Town area as a trial vehicle to promote LPG use in public transport.

Appendices

Appendice A: Information and calculation spreadsheets

A1: Fuel Properties

A2: Vehicle conversion cost calculations

A3: Evaporative cooling calculations

A4: LPG mixer calculations

A5: ADE 447 squish calculations

Appendix A1

Fuel Properties **Bergmann and Busenthur (1987)**

Air Fuel Ratios, vol. % in Air			
Fuel	Lower Limit	Stoiciometric	Upper Limit
Diesel	0.6		6.5
Petrol	1.4		7.6
Methanol	6.72		36.5
Methane	5.3	11.7	14
Propane	2.4	4	9.5
n-Butane	1.8	3.2	8.4

Air Fuel Ratios by mass in Dry Air	
Fuel	Stoiciometric Value
Propylene	14.77
Propane Commercial	15.64
Butane Commercial	15.43
Butylenes	14.77

Stoiciometric Fuel/Air Ratios		
Fuel	mass %	vol %
Propane	15.64	4
n-Butane	15.43	3.2

Lower HeatingValues of Fuels			
Fuel	MJ/liter liquid Phase	MJ/m3 (gas phase, Std Atm.)	MJ/kg
Diesel	34.9		43.8
Petrol	31.5		42.7
Methanol	15.6		19.9
Methane	20.4	35.9	
Propane	23.2	92.9	46.5
n-Butane	26.5	123.7	46.0

Composition of local LPG	
Variable	Mean
Density@ 20 Deg C. kg/l	0.55
Methane moll %	0.06
Ethane moll %	3.59
Ethylene moll %	0.05
Propane moll %	31.21
Propylene moll %	12.93
Iso-Butane moll %	25.00
Normal butane moll %	15.12
Butenes moll %	11.31
Pentanes moll %	0.75
Non Volatile residue ul/l	0.00
Vapour Pressure at 37 deg C kPa	896.59
Total Sulphur mg/kg	10.45
Caltex Refinery, Cape Town	

Spontaneous Combustion Factors			
Fuel	Self Ign. T [deg. C]	Flame Speed [m/s]	Flame Temp [deg.C in Air]
Diesel	230		
Petrol	280		
Methanol	450	0.49	1904
Natural Gas	645	0.3	1941
Propane	493	0.46	1967
n-Butane	430	0.4	1973

Appendix A2

Cost estimation for LPG conversion for dual fuel operation   **ADC EQUIPMENT**

no	Component	Description	Unit Cost	No.	Total	Engine only	Taxi
1	BEZA 12-A 156 litre	Automotive tar	1973.82	1	1973.82		1973.82
2	Impco EB2	Regulator	1185.95	1	1185.95	1185.95	0.00
3	Solenoid	Fuel Lock	483.38	1	483.38	483.38	0.00
4		Mixer	400	1	400.00	400.00	0.00
5	Flow control	electronic valve	1015.44	0	0.00	0.00	0.00
6	High pressure	Fuel Line	46.03	3	138.09	46.03	92.06
7		Reducing Bush	19.57	1	19.57	19.57	0.00
8		Hose ends	17.96	2	35.92	35.92	0.00
9		General fittings	200	1	200.00	0	200.00
10	Changeover switch		60	1	60.00	60.00	0.00
11	Electrical wiring		20	6	120.00	120.00	0.00
					L. 4,436.73	R 2,170.85	R 2,265.88

ADC quote	15-05-97	R 7,496.71

BRC EQUIPMENT

377.8 lire/rand	04/06/97

no	Description	Price	
		Lire	Rand
1	Reducer 140 kW	78840	208.68
2	LPG Solenoid valve	14268	37.77
3	Petrol solenoid valve	10260	27.16
4	BRAVO electronic control unit	39360	104.18
5	Level sensor for ecu	22190	58.73
6	Petrol pipe dia 7*13	2136	5.65
7	Water pipe 1.5 m	4028	10.66
8	Copper pipe dia 6	10524	27.86
9	Small parts & accessories	6750	17.87
10	Kit box	2124	5.62
11	Mixer for Toyota 4Y 2.2 litre	38690	102.41
12	Gas proof casing	6522	17.26
13	Multivalve with safety valve	41640	110.22
14	Refuelling point	5892	15.60
15	Breather pipe	1338	3.54
16	Vents	444	1.18
17	Armoured pipe	3630	9.61
18	Couple of bands for tank fitting	5250	13.90
19	Couple of tank supports	4380	11.59
20	LPG Tank 62 liter	102000	269.98
		L. 400,266	R 1,059

Assume that import tax can double the cost of imported goods while transport may cost R2500 per kit

This brings the final Cost estimate to: **R 4,618.93**

Dock shipping co
021-217600

Appendix A3, Evaporative cooling calculations

Properties used for calculation of evaporative cooling effect of Propane

Mixing Characteristics of Propane and Air			
A/F Ratio	Ambient T	Fuel T	Delta T
[mass]	[Celcius]	[Celcius]	100% evap
15.64	26.850	-42.1	27.08
[volume %]		-9.9	24.74
4		12.8	22.69
		28.2	21.09
		40.3	19.66
		50.3	18.25
		58.7	16.76
Values in green are used for calculation			

LPG Cooling Properties   1cal=4.184

Properties of Propane			Enthalpy		Enthalpy		Change
Temp	Pressure	Pressure	[kcal/kg]		[kJ/kg]		[kJ/kg]
[Celcius]	[kg/cm^2]	[kPa]	liquid	vapour	liquid	vapour	liquid/vapour
-42.1	1.033	101	64.6	166.4	270.3	696.2	425.9
-9.9	3.51	344	81.8	174.8	342.3	731.4	389.1
12.8	7.03	690	95.7	181	400.4	757.3	356.9
28.2	10.54	1034	105.6	184.9	441.8	773.6	331.8
40.3	14.06	1379	113.7	187.6	475.7	784.9	309.2
50.3	17.57	1724	120.8	189.4	505.4	792.4	287.0
58.7	21.09	2069	127.8	190.8	534.7	798.3	263.6
66.3	24.6	2413	134.1	191.5	561.1	801.2	240.2
73.1	28.12	2759	140	191.6	585.8	801.7	215.9
79.3	31.63	3103	145.3	190.8	607.9	798.3	190.4
84.9	35.15	3448	150.3	189.4	628.9	792.4	163.6
90.2	38.66	3793	155.5	187.3	650.6	783.7	133.1
93.9	41.33	4054	161	185.1	673.6	774.5	100.8
95.3	42.18	4138	164.7	183.8	689.1	769.0	79.9
96.8	43.37	4255	175	175	732.2	732.2	0.0

From: Liquified Petroleum Gases
A.F. Lom 1982 ISBN 0-85312-360-8

Effect of LPG addittion in airstream with evaporative cooling i.e. air density changes

Evaporated	Delta T	Air Dens	Air Vol	Fuel vol.	Mixture vol	Fuel Dens	Air Content	Nett Effect
[%]		[kg/m^3]	[m^3/kg]	[m^3/kg]	[m^3/kg]	[kg/m^3]	[kg/m^3]	Cooling
0	0	1.180	0.8474	0.0020	0.7966	508.000	1.2553	0.000
10	2.11	1.188	0.8415	0.0022	0.7910	457.403	1.2642	0.708
20	4.22	1.197	0.8355	0.0025	0.7854	406.806	1.2732	1.426
30	6.33	1.206	0.8295	0.0028	0.7798	356.210	1.2823	2.154
40	8.44	1.214	0.8235	0.0033	0.7742	305.613	1.2916	2.891
50	10.55	1.223	0.8176	0.0039	0.7687	255.016	1.3009	3.637
60	12.66	1.232	0.8116	0.0049	0.7631	204.419	1.3104	4.392
70	14.77	1.241	0.8056	0.0065	0.7576	153.822	1.3200	5.152
80	16.88	1.251	0.7996	0.0097	0.7522	103.226	1.3295	5.911
90	18.98	1.260	0.7937	0.0190	0.7471	52.629	1.3385	6.627
95	20.04	1.265	0.7907	0.0366	0.7454	27.330	1.3416	6.878
97	20.46	1.267	0.7895	0.0581	0.7455	17.211	1.3413	6.853
99	20.88	1.269	0.7883	0.1410	0.7494	7.092	1.3344	6.303
100	21.09	1.270	0.7877	0.4921	0.7699	2.032	1.2988	3.467

Effect of LPG addittion in airstream without evaporative cooling i.e. air density remains constant

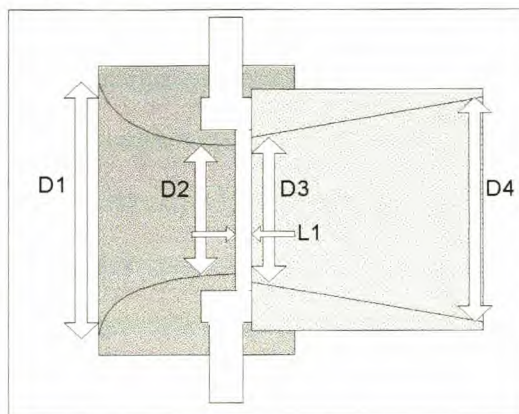| Evaporated	Delta T	Air Dens	Air Vol	Fuel vol.	Mixture vol	Fuel Dens	Air Content	Nett Effect
[%]	no cooling	[kg/m^3]	[m^3/kg]	[m^3/kg]	[m^3/kg]	[kg/m^3]	[kg/m^3]	Air displ.
0	0	1.180	0.8474	0.0020	0.7966	508.000	1.2553	0.000
10	0	1.180	0.8474	0.0022	0.7966	457.403	1.2553	-0.002
20	0	1.180	0.8474	0.0025	0.7967	406.806	1.2552	-0.004
30	0	1.180	0.8474	0.0028	0.7967	356.210	1.2552	-0.006
40	0	1.180	0.8474	0.0033	0.7967	305.613	1.2552	-0.010
50	0	1.180	0.8474	0.0039	0.7968	255.016	1.2551	-0.015
60	0	1.180	0.8474	0.0049	0.7968	204.419	1.2550	-0.022
70	0	1.180	0.8474	0.0065	0.7969	153.822	1.2549	-0.034
80	0	1.180	0.8474	0.0097	0.7971	103.226	1.2546	-0.058
90	0	1.180	0.8474	0.0190	0.7977	52.629	1.2537	-0.128
95	0	1.180	0.8474	0.0366	0.7987	27.330	1.2520	-0.260
97	0	1.180	0.8474	0.0581	0.8000	17.211	1.2500	-0.422
99	0	1.180	0.8474	0.1410	0.8050	7.092	1.2423	-1.038
100	0	1.180	0.8474	0.4921	0.8261	2.032	1.2105	-3.566

Appendix A4, LPG mixer calculations

Green shaded areas can be filled in for changes

Calculations for ADE 447 LPG engine

Mixer Calculations		Mixer Characteristic in mm			Supply pipe	
D1	90	Entry Diameter		Area Ratio	Supply pipe	
D2	54	Throat diameter		2.778	Diameter	15
D3	56	Gas entry diameter			Length	1
D4	80	Exit diameter			Entry K factor	0.9
L1	4	Gap Width			K factor	1.9

Fluid Properties					Engine parameters	
Density [kg/m^3]		Viscosity	Mix %		litres	Type-Stroke
Propane	1.9	[Stokes]	50		12	4
Butane	2.55		50		Mixer Caracteristic	
Mix	2.225	0.04			single	2.778
Air	1.2					

Speed	Air Flow	V1	V2	Vacuum	A/F	Gas Flow
[rpm]	[l/s]	[m/s]	[m/s]	[Pa]	ratio [Vol]	[l/s]
500	50	7.9	21.8	249	27	1.85
700	70	11.0	30.6	488	27	2.59
900	90	14.1	39.3	806	27	3.33
1100	110	17.3	48.0	1205	27	4.07
1300	130	20.4	56.8	1683	27	4.81
1500	150	23.6	65.5	2240	27	5.56
1700	170	26.7	74.2	2877	27	6.30
1900	190	29.9	83.0	3594	27	7.04
2100	210	33.0	91.7	4391	27	7.78
2300	230	36.2	100.4	5267	27	8.52
2500	250	39.3	109.2	6223	27	9.26

Areas in m^2			
A1	0.00636	A3-A2	0.00017
A2	0.00229	Ratio Air/Ga	13.255
A3	0.00246	Idle/Throat	2.195
A4	0.00503		
A5	0.000678584	A2/A5	3.375

1 Supply pipe

Speed		Reynolds	Pipe loss	Minor loss	Total loss	
[rpm]	[m/s]	no.	[Pa]	[Pa]	[Pa]	[mm H2O]
500	10.48	1964876	68.74	464	533	54
700	14.67	2750826	123.87	910	1034	106
900	18.86	3536777	192.29	1504	1696	173
1100	23.05	4322727	273.19	2247	2520	257
1300	27.25	5108677	365.96	3138	3504	358
1500	31.44	5894628	470.10	4178	4648	475
1700	35.63	6680578	585.22	5367	5952	608
1900	39.82	7466528	710.97	6704	7415	757
2100	44.01	8252479	847.06	8189	9036	923
2300	48.20	9038429	993.24	9824	10817	1105
2500	52.40	9824379	1149.28	11606	12756	1302.9

2 Supply pipes

Speed	Gas Flow	Reynolds	Pipe loss	Minor loss	Total loss	
[rpm]	[m/s]	no.	[Pa]	[Pa]	[Pa]	[mm H2O]
500	5.2	982438	20	116.1	136	14
700	7.3	1375413	37	227.5	264	27
900	9.4	1768388	57	376.0	433	44
1100	11.5	2161363	81	561.7	643	66
1300	13.6	2554339	109	784.6	893	91
1500	15.7	2947314	140	1044.6	1184	121
1700	17.8	3340289	174	1341.7	1516	155
1900	19.9	3733264	211	1675.9	1887	193
2100	22.0	4126239	252	2047.3	2299	235
2300	24.1	4519214	295	2455.9	2751	281

Mixer Characteristics vs. System Requirements

- Mixer Vacuum
- Head Loss with 1 Pipe
- Head Loss with 2 pipes

Engine Speed [rpm]
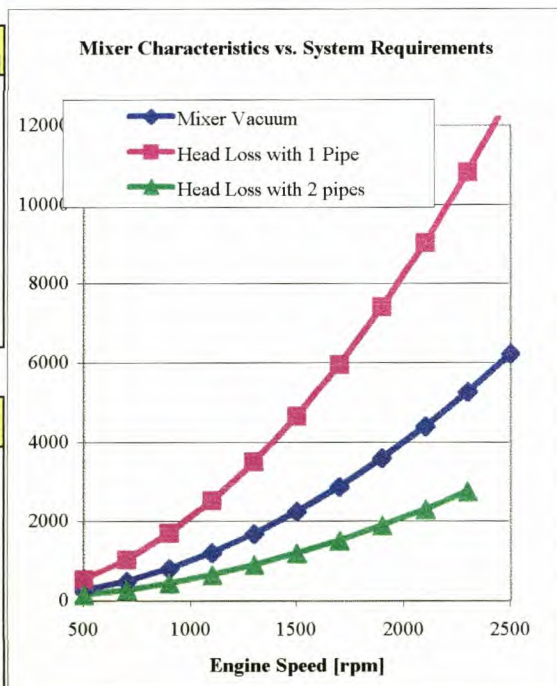
Appendix A5, ADE 447 squish calculations

Gas Piston Detail

Piston Dia [mm]	Bowl dia [mm]	Bowl offset [mm]	Squish gap	Bowl Area [mm^2]	Squish Area [mm^2]	Bowl Depth [mm]
127	68	22	0.8	3632	9036	33.5

Valve Diameters [mm]			Conrod Len [mm]	Stroke [mm]	Ratio	Sum [mm]		Engine Speed [rpm]	[deg/s]		Speed of Sound		455.660
Inlet valve	Exh valve										Inl T [C]	TDC T [K]	Sound Spd
58	50		400	150	1.3333	475.8		1500	9000		25	729	846

Calculation of Squish Velocities

Crankangle [deg]	[radians]	L1	L2	Dist to Top [mm]	Vol bowl [ml]	DV bowl [ml/mm]	Vol Sq [ml]	DV Sq [ml/mm]	Area Sq [mm^2]	V Sq mm/mm	V Squish [m/s]	Mach Sq	Piston Spd [m/s]
20	0.349	70.477	386.370	18.95	190		171		4049				
19	0.332	70.914	387.692	17.19	184	3.632	155	9.036	3673	2.4601	22.141	0.026	15.831
18	0.314	71.329	388.948	15.52	178	3.632	140	9.036	3316	2.7249	24.524	0.029	15.039
17	0.297	71.723	390.137	13.94	172	3.632	126	9.036	2978	3.0342	27.308	0.032	14.243
16	0.279	72.095	391.259	12.45	167	3.632	112	9.036	2659	3.3984	30.586	0.036	13.445
15	0.262	72.444	392.314	11.04	162	3.632	100	9.036	2359	3.8308	34.477	0.041	12.644
14	0.244	72.772	393.302	9.73	157	3.632	88	9.036	2078	4.349	39.141	0.046	11.840
13	0.227	73.078	394.222	8.50	153	3.632	77	9.036	1816	4.9763	44.787	0.053	11.034
12	0.209	73.361	395.075	7.36	148	3.632	67	9.036	1573	5.7442	51.698	0.061	10.226
11	0.192	73.622	395.861	6.32	145	3.632	57	9.036	1350	6.6954	60.259	0.071	9.416
10	0.175	73.861	396.578	5.36	141	3.632	48	9.036	1145	7.8892	71.003	0.084	8.603
9	0.157	74.077	397.227	4.50	138	3.632	41	9.036	960	9.4079	84.671	0.100	7.789
8	0.140	74.270	397.809	3.72	135	3.632	34	9.036	795	11.367	102.302	0.121	6.974
7	0.122	74.441	398.322	3.04	133	3.632	27	9.036	649	13.927	125.345	0.148	6.157
6	0.105	74.589	398.767	2.44	131	3.632	22	9.036	522	17.307	155.766	0.184	5.338
5	0.087	74.715	399.144	1.94	129	3.632	18	9.036	415	21.782	196.042	0.232	4.519
4	0.070	74.817	399.452	1.53	127	3.632	14	9.036	327	27.63	248.667	0.294	3.699
3	0.052	74.897	399.692	1.21	126	3.632	11	9.036	259	34.923	314.308	0.371	2.877
2	0.035	74.954	399.863	0.98	125	3.632	9	9.036	210	43.04	387.359	0.458	2.056
1	0.017	74.989	399.966	0.85	125	3.632	8	9.036	181	50.016	450.141	0.532	1.234
0	0.000	75.000	400.000	0.80	125	3.632	7	9.036	171	52.872	475.850	0.562	0.411

Appendice B: Test data

B1: Light duty engine mapping

B2: Light duty engine test history

B3: In cylinder pressure measurement results

B4: Injector test results

Appendix B1: Light duty engine mapping

Timing Swing & Fuel Loop US 20

Fuel Type:
LPG

Fuel System:
LPG Mixer

Source Data US 20 Book from recordings made during engine mapping

Fuel Type:
LPG

Fuel System:
LPG Mixer

Speed	MAP	Timing	Torque
[rpm]	[kPa]	[Deg]	[Nm]
2400	100	28	154.8
		31	155.5
		33	155.7
		35	155.3
2800	95	29.3	153
		32.5	153.2
		34.2	153.5
		36	153.2
3200	98	29	154.75
		30	154.85
		31	155
		32	154.96
		35	154.22
3600	99	26	152.76
		28.6	154.4
		30.4	155
		32	155.5
		33.5	155.4
		35	154.75
4400	98	30	137
		32	138
		35	138
		37	137.5
4800	95	33	124
		35	124.8
		36	125.75
		37	124.5
		38	125
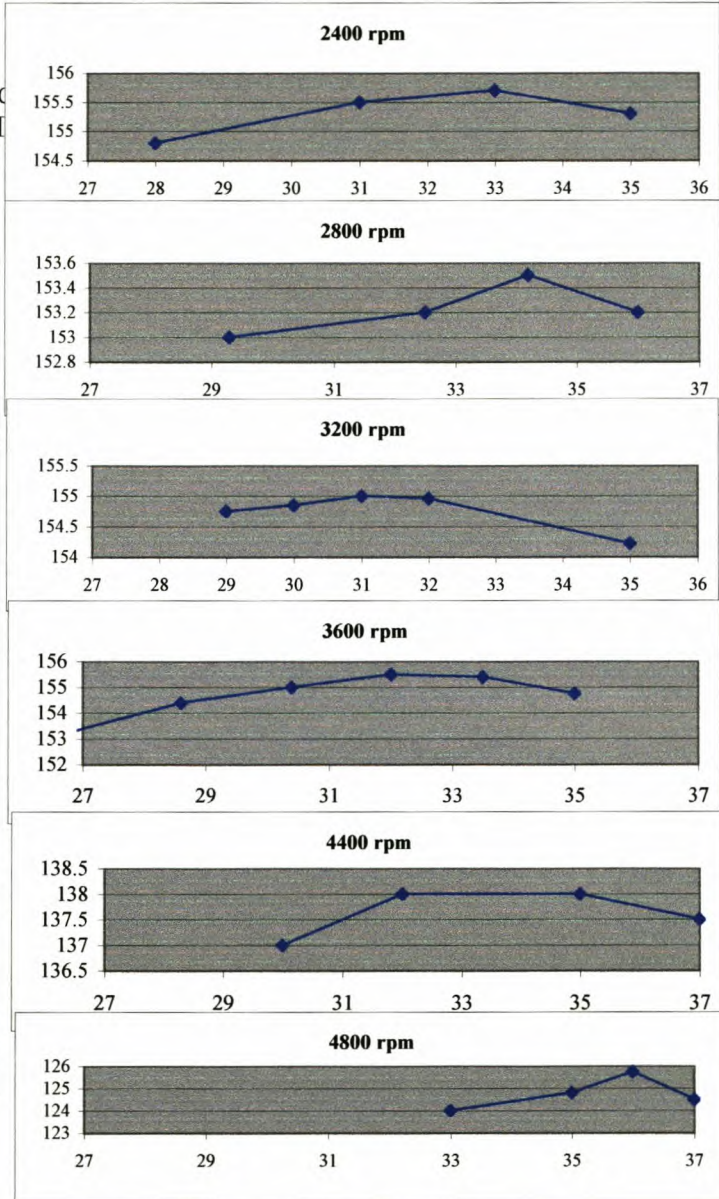
Appendix B2: Light duty engine test history

Engine asssembled on 15-04-97   Engine Code: US 20 / ex Nissan NA 20

Date	Activity	Fuel Type	Special components	Results/ Files etc.
24-11-97	Did LPG carburettion mapping to find correct valve position	LPG	LPG mixer	Saved as.us20map2.xls
15-12-97	Did LPG carburettion mapping to find best spark advance	LPG	LPG mixer	Saved as.us20map2.xls
18-12-97	Petrol injection mapping to find optimum timing	95 Octane	LPG mixer	in US20 Book
	December Break			
9-01-98	Engine refitted valve timing advanced by one tooth	95 Octane	LPG mixer	in US20 Book
9-01-98	Fuel loop with Petrol injection & Racer	95 Octane	LPG mixer	in US20 Book
12-01-98	Test with pertol injection	95 Octane		011298.xls
13-01-98	LPG Mixer mapping 1600-5200 rpm	LPG	LPG mixer	in US20 Book
15-01-98	LPG test with Racer leading to molten spark plug			Dat0115.xls
	Problems with molten spark plug. New insert & reseating of valves			
5-02-98	Test Fuel loop with petrol @ 2860 rpm	95 Octane		
11-02-98	Experimenting with LPG injection	LPG	High pressure injection	
	Unstable combustion leads to manufacture of cooled fuel rail			
08-04-98	Tests with cooled fuel rail Change spark plug gaps + & -	LPG	High pressure injection	GA0804.xls
	Use of Keithley to measure engine parameters			GB0804.xls
	Study literature to find problems with unstable combustion			GC0804.xls
15-04-98	LPG injection problems with mid speed range	LPG	High pressure injection	Gi150498
	Refined mapping procedure sort out unstable mid range performance			
20-04-98	LPG injection without mixer / new spark plugs	LPG	High pressure injection	
21-04-98	LPG injection without mixer / Full test with Racer & emissions	95 Octane	High pressure injection	us20_0421.xls
22-04-98	LPG Carburettion / Full test with Racer & emissions	LPG	LPG mixer	LPGcarb2204.xls
24-04-98	Petrol injection / Full test with Racer & emissions	LPG	High pressure injection	us20_2404.xls
05-05-98	Redo some LPG injection points for Racer Data	LPG	High pressure injection	
06-05-98	Redo some points with LPG carburettion for Racer data	LPG	LPG mixer	
06-05-98	Do Bracketing test with Petrol injection with LPG mixer	95 Octane	LPG mixer	uspc0605.xls
06-05-98	Racer Knock test Detect problems with no3 inlet valve	95 Octane		Dat0705.xls

Appendix B3: In cylinder pressure measurement results

Combustion Summary US20

Entity

Spark advance

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	20	20	23	23	23	28	26	26	28	26	26	28	29	29	32
	75															
	55															

Burn Angle

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	52.8	46.8	49.3	56.7	51.3	58	51.4	58.5	59.6	56.6		59.4	52.7	56.4	57.8
	75	48.3		51.3	52.1	61.2	59.9	55.6		61	58.2					
	55						50	58.8		57.6	61.6		66.2			

Induction Period

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	12.7	12.6	14	12.2	15	18	18.3	14.4	16.7	14.4		18	18.4	17.6	23.1
	75	15.6		14.5	23	15.3	22.2	18.4		16.1	15.4					
	55						23	19.9		14.9	14.4		12.1			

End of Combus

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	65.5	59.4	63.3	68.9	66.3	76	69.7	72.9	76.3	71	0	77.4	71.1	74	80.9
	75	63.9	0	65.8	75.1	76.5	82.1	74	0	77.1	73.6	0	0	0	0	0
	55	0	0	0	0	0	73	78.7	0	72.5	76	0	78.3	0	0	0

Max Temperature

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	2267	2231	3046	2920	2426	2191	2607	2631	2843	3457	-	3134	2936	2985	3181
	75	2504		1968	1361	1853	2185	2685		3211	3168					
	55						2040	1182		2764	3013		4222			

Max Pressure

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	4844	4872	4621	5270	5173	4375	5433	6202	5755	5741	-	5462	6062	5992	5341
	75	2835		2409	2972	3131	2898	3960		4190	4516					
	55						1529	2862		3173	3339		3562			

Total Heat Released

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	1413	1408	1413	1213	1536	1252	1635	1678	1609	1674	-	1630	1637	1657	1628
	75	1009		912	798.3	946.7	950.7	960.6		994	1312					
	55						579.4	698.4		882	950.9		1029			

Max rate of Heat Release

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	3724	4412	4058	3098	4640	3588	4425	5438	4370	4431	-	3861	4883	4783	4178
	75	2299		1874	2399	2549	2335	2547		2348	3203					
	55						1134	2127		2296	2262		2771			

Stability of Combuston [%]

	Speed	2000	2000	2000	2800	2800	2800	3600	3600	3600	4800	4800	4800	5000	5000	5000
	Load	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc	Pi	Gi	Gc
	98	20.5	51.7	22.7	9.8	21.2	38.7	19.7	30.1	12.5	15.1	-	9.8	55.7	34.6	24.1
	75	29.8		37.9	27.1	51.4	30.2	21.9		13.2	18					
	55						50.4	26		20.3	22.5		15.4			

Appendix B4: Injector test results
Injector test sheet

Date: 11-3-98
Test Person: GAVDH
Type of Test: Injector Frequency response   Duration 60 sec

No.	Pressure	Frequency	Duty cycle	Flow Rate	Period	Std	Repetitions	Flow/rep	Corrected flow	
4 Std	[Bar]	[Hz]	[%]	[cm^3]	[ms]	[cm^3/s]		[cm^3/s]	[cm^3/s]	Delay [ms]
	6	33.33	50	128	30	2.133	2000	0.001067	2.30	0.084
		50.00	50	124.5	20	2.075	3000	0.000692	2.33	
		66.67	50	118	15	1.967	4000	0.000492	2.30	
		100.00	50	112.5	10	1.875	6000	0.000313	2.38	
		133.33	50	102.5	7.5	1.708	8000	0.000214	2.38	
		166.67	50	98	6	1.633	10000	0.000163	2.47	
		200.00	50	77.7	5	1.295	12000	0.000108	2.30	

		Frequency	Duty cycle	Flow Rate	Period	Single	Repetitions	Flow/rep	Corrected flow	
4 Single	[Bar]	[Hz]	[%]	[cm^3]	[ms]	[cm^3/s]		[cm^3/s]	[cm^3/s]	Delay [ms]
	6	33.33	50	218	30	3.633	2000	0.001817	3.94	0.153
		50.00	50	203.5	20	3.392	3000	0.001131	3.85	
		66.67	50	196	15	3.267	4000	0.000817	3.88	
		100.00	50	181.5	10	3.025	6000	0.000504	3.94	
		133.33	50	156	7.5	2.600	8000	0.000325	3.82	
		166.67	50	142	6	2.367	10000	0.000237	3.90	
		200.00	50	126	5	2.100	12000	0.000175	3.94	

Injector response to Pressure Changes

	Type							
Pressure	Std 1	Std 2	Std 3	Std 4	Single 1	Single 2	Single 3	Single 4
[Bar]	[cm^3/sec]	[cm^3/sec]	[cm^3/sec]	[cm^3/sec]	[cm^3/sec]	[cm^3/sec]	[cm^3/sec]	[cm^3/sec]
3.5	1.267	1.233	1.267	1.267	1.9	1.9	1.833	1.933
7.25	1.550	1.500	1.567	1.567				
8	1.600	1.567	1.633	1.633	1.883	2.067	1.867	2.183
9	1.633							

Flow/sec	[cm^3/sec]	1.267
Repetitions		200
Flow/cycl	[cm^3]	0.006333
Open time [ms]		2.42
Flow/Open time/cycl		0.002617

Appendice C: Computer simulation program listing

Appendix C1

Program CombustionCycle;

{" BINNEBRAND ENJIN SIMULASIE PROGRAM   Aug 1997   G.A. van der Ham"}
uses wincrt;
    var N,rpm,EVO,VB,B,S,conrod,BA,SA,R,silinders,AF,QF,Patm,Ta,Tg,Tw,Tb,Th,P,Ph:real;
    Tref,Pref,Pmot,Pinl,Pexh,J,Vd,Vtdc,Vref,eps,rho,FuelMass,m,Wn,Qn,DQ,Winst,Wpump,
    crankangle, Pcyl,Pb, theta,V, DV, DX, DP,k1,k2,k3,k4: real;
    C,Cm,w,x,h, Pm, OpMax :real;
    mySpark, Endspark,top : integer;
    tf:text;

    const pi=3.141592654;
        G=1.395; {Ratio of specific heats for fuel-air vapor}

{**}

Procedure initvariables;
Begin
    N := 1.5   {Stepsize in degrees of crankangle movement};
    rpm := 4500;
    EVO := 150;
    VB := 0.98 {Volumetric efficiency};
    B := 0.08 {Bore};
    S := 0.08 {Stroke};
    conrod := 0.1348 {conrod length};
    BA := 70 {Combustion Duration, degrees crank angle movement};
    SA := myspark {Start of combustion};
    R := 19.21   {Compression ratio};
    silinders := 4;

    AF := 14.5   {'AIR FUEL RATIO};
    QF := 41.0 {'Fuel energy content MJ/kg};

    Patm := 101300; {Ambient Pressure mBar}
    Ta := 25 + 273 {'Ambient temperature};
    Tw := 250 + 273 {'Wall temperature};
    Tg := Ta;
    Cm:=2*S*N/60; {Mean piston speed}
    Pinl := VB * Patm {VB * rpm};
    Pexh := Patm + 0.2 * rpm + 0.001 * rpm*rpm;

    J := N * pi / 180;
    Vd := pi * B*B * S / 4; {Cylinder displacement}
    Vtdc := Vd / (R - 1);
    eps := S / (2 * conrod);
    rho := Pinl / (287 * Ta);
    FuelMass := Vd * VB * rho / AF;
    QF := QF * 1000000 * FuelMass;
    m := Vd * rho * VB;
    Wn := 0;Qn := 0;
end;

{**}

```
Function power(x,y:real):real;
begin
power:= exp(y*ln(x));
end;

{*************************************************************************}

Function dVdtheta(Vd, R, theta: real):real;
Begin
 dVdtheta := Vd * (R - 1) / 2 * SIN(theta) / R;
END;

{*************************************************************************}

Function dXdtheta(BA, SA, crankangle, pi:real):real;
Begin;
 dXdtheta := (pi / (BA * pi / 180) * SIN(pi * (crankangle - SA) /BA)) / 2;
END;

{*************************************************************************}

Function Vol(Vtdc, R, theta, eps: real):real;
var xx,a,temp:real;
Begin;
 a:= (eps * SIN(theta));
 xx := (1 - a*a);
 Vol := Vtdc * (1 + (R - 1) / 2 * (1 - COS(theta) + (1 / eps) * (1 - power(xx,0.5))));
END;

{*************************************************************************}

Function DPdtheta(G, Pcyl, V, DV, QF, DQ, DX, theta: real):real;
Begin;
 DPdtheta := -G * Pcyl * DV / V + (G - 1) * (QF * DX + DQ) / V;
END;

{*************************************************************************}

Procedure CUTTA;

begin
  IF (crankangle < SA) THEN
    begin
    DX := 0; C:= 2.28*Cm;
    end;

  IF (crankangle > SA) and (crankangle < (SA + BA)) THEN
    begin

    DX := dXdtheta(BA, SA, crankangle, pi);
    Pmot:= Pinl*power(Vd/V,G)-700000;
    C:= 2.28*Cm + 3.24*V/Vd*Ta/Pinl*(P-Pmot)/50000;  {check this out}
    end;

  IF (crankangle > (SA + BA)) THEN
```

```pascal
  begin
   DX := 0; C:= 2.28*Cm;
  end;

 Pcyl := P;

 V := Vol(Vtdc, R, theta, eps);
 DV := dVdtheta(Vd, R, theta);
 k1 := DPdtheta(G, Pcyl, V, DV, QF, DQ, DX, theta);

 theta := crankangle* pi / 180 + J / 2;
 V := Vol(Vtdc, R, theta, eps);
 DV := dVdtheta(Vd, R, theta);
 Pcyl := P + k1 * J / 2;
 k2 := DPdtheta(G, Pcyl, V, DV, QF, DQ, DX, theta);

 Pcyl := P + k2 * J / 2;
 k3 := DPdtheta(G, Pcyl, V, DV, QF, DQ, DX, theta);

 theta := crankangle* pi / 180 + J;
 V := Vol(Vtdc, R, theta, eps);
 DV := dVdtheta(Vd, R, theta);

 Pcyl := P + k3 * J;
 k4 := DPdtheta(G, Pcyl, V, DV, QF, DQ, DX, theta);
 DP := (k1 + 2 * k2 + 2 * k3 + k4) / 6;
 P := P + J * DP;
   {writeln('P ',P/100000:4:3,'  Pmot ',Pmot/100000:4:3,'  C ',C:2:6);}
END;

{***********************************************************************}

{Begin loop}
procedure loop;
var theta,Watts,Pin,Pfriction,Qnett,Efficiency,SFC,TORQUE,PExhaust:real;
begin
 assign(tf,'pv.txt');
 reWrite(tf);
 writeln (tf,'crankangle ','  DQ ',' DV   ',' DP  ',' Vol  ','P     ',' Tg ',' Winst ',' Wn');
 Wn := 0; Qn := 0;
 crankangle :=-360;

   repeat
     theta := crankangle * pi / 180  ;

     IF crankangle < -180 THEN
      begin P:= Pinl;          C:= 6.18*Cm;
      end;

     IF (crankangle < EVO) and (crankangle > -180) THEN  CUTTA;

     IF (crankangle > EVO) and (crankangle < EVO + 120) THEN
      begin P:= P-(P-Pexh)/(50*N); C:= 6.18*Cm;
      end;

     IF crankangle > EVO + 120 then
```

```pascal
    begin    P:= Pexh;          C:=6.18*Cm;
    end;

    V:= Vol(Vtdc, R, theta, eps) ;
    Tg:= P * V / ((m + FuelMass) * 287);
    {Woshney's heat transfer equation}
    h:=126.2*power(B,-0.2)*power(P/100000,0.8)*power(C,0.8)*power(Tg,-0.52);
    DQ := (4 * V / B + pi * B*B / 2) * h * (Tw - Tg);
    DV := dVdtheta(Vd, R, theta);
    Winst := (P - Patm) * DV * N * pi / 180;
    Qn := Qn + DQ * N * pi / 180;
    Wn := Wn + Winst;

    IF P > Ph THEN  Ph := P;
    IF Tg > Th THEN  Th := Tg;

writeln (tf,crankangle:3:3,' ',DQ:6:1,' ',DV:1:6,' ',DP:9:0,' ',V:1:6,' ',P:9:1,'    ',Tg:4:1,' ',Winst:8:3,'
',Wn:8:3);
        crankangle := crankangle + N

    until crankangle >= 360 ;
    close(tf);
END;

{*****************************************************************************}

procedure display;
var

theta,V,P,IMEP,PMEP,FMEP,BMEP,Watts,Pin,Pfriction,Qnett,Efficiency,SFC,TORQUE,PExhaust,Ppum
p,Balance:real;
    begin

      WriteLn;
        Wpump := ((Patm - Pinl) + (Pexh - Patm)) * Vd;
        IMEP := Wn / Vd;
        PMEP := Wpump / Vd;
        FMEP := (0.97 + 0.15 * (rpm / 1000) + 0.05 * power((rpm / 1000),1.2)) * 100000 * silinders / 4;
        BMEP := IMEP - FMEP - PMEP;
        Watts := BMEP * Vd * silinders * rpm / 120000;
        Pin := QF * silinders * rpm / 120000;
        Pfriction := FMEP * Vd *silinders * rpm / 120000;
        Ppump:= PMEP * Vd * silinders * rpm/120000;
        Qnett := -Qn * silinders * rpm / 120000;

        PExhaust := Pin - Watts - Pfriction  - Ppump - Qnett;
        Tb := PExhaust /( 1005 * m * silinders * rpm)* 120000;


        Efficiency := Watts / Pin;
        SFC := FuelMass * silinders * rpm / 60 * 3600 / Watts;
        TORQUE := BMEP * Vd * silinders / (4 * pi);
        Balance:= Pin - Watts - Pfriction - PExhaust - Ppump - Qnett;

        WriteLn  ;
```

```
        Writeln (' Engine Speed   ',rpm:4:2,' rpm          Engine Displacement ',Vd*1000*silinders:4:3,'
litre');
        writeln;
        WriteLn (' IMEP ',IMEP/100000:8:3, ' bar      Fuel Energy     ', Pin:8:3,' kW   100 %');
        WriteLn (' FMEP ',FMEP/100000:8:3, ' bar      Engine Output   ' ,Watts:8:3,
        ' kW', Watts/Pin*100:8:1, '%');
        WriteLn (' PMEP ',PMEP/100000:8:3, ' bar      Cooling Loss    ',Qnett:8:3,
        ' kW', QNett/Pin*100:8:1, '%');
        WriteLn (' BMEP ',BMEP/100000:8:3, ' bar      Friction Losses ' ,Pfriction:8:3, ' kW',
        PFriction/Pin*100:8:1, '%');
        WriteLn (' Pexh ',Pexh/100000:8:3,' bar      Exhaust Losses  ',PExhaust:8:3, ' kW',
        PExhaust/Pin*100:8:1, '%');
        WriteLn (' Pinl ',Pinl/100000:8:3,' bar      Pumpwork        ',Ppump:8:3,' kW',
        PPump/Pin*100:8:1, '%');
        writeln;
        writeln (' Texh   ',Tb-273:8:1,' Celcius      Energy Balance  ',Balance:3:3,' kW');
        writeln (' Torque ',TORQUE:8:1,' Nm       Combustion Duration   ',BA:3:1,' Degrees ');
        Writeln (' Thermal Efficiency ', Efficiency:1:4,'          SFC ', SFC:8:3,' kg/kWhr');
        writeln;
        WriteLn (' Cylnder Wall Temperature ',Tw - 273:8:1, ' Celcius');
        WriteLn (' Maximum Temperature     ' ,Th - 273:8:1, ' Celcius');
        WriteLn (' Maximum Pressure        ' ,Ph / 100000:8:1, ' Bar');

end;

{*******************************************************************************}

{Hoofprogram}
Begin
WriteLn('*******************************************************************');
WriteLn('*      Prepared by G.A.van der Ham                    *');
writeln('*    Last modified in Dec 2000                        *');
WriteLn('*******************************************************************');

myspark := -28;
 repeat
 Pm := Wn;  {Wn = Netto Work}
 initvariables;
 loop;
     writeln(' Spark Timing BTDC ',myspark,'    Nett. Work ',Wn:3:2);
     myspark := myspark + 1;
until Pm > Wn;

myspark := myspark-2;
initvariables;
loop;
writeln(' Optimum Timing is ',-1 * myspark,' Degrees BTDC');
display;

end.

{*******************************************************************************}
```

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