Development and Mathematical Analysis of a Modular CNG Valve

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Masters Abstract

Development and Mathematical Analysis of a Modular CNG valve

With the rising cost of oil and uncertainty of supply, there has never been a greater opportunity to offer an alternative fuel into the automotive market than at this present time. Compressed natural gas (CNG) and liquid petroleum gas (LPG) are popular alternatives, producing less green house gasses after the combustion process that add to the raising global warming concern. With high performance fuel injected state of the art engines used in the majority of the late model vehicles, the problem when running on CNG or LPG is poor control of the air/fuel ratio throughout the engine's speed and load range using the conventional zero pressure regulator and mixer combination gas conversion equipment used previously for carburetted engines. This problem is completely eliminated with gas injection system.

The Harrison CNG Electronic Gas injection System control valve is a linear proportional valve. Testing on the valve has found that the response is linear under all operating conditions; however the valve exhibits occasional instances of hysteresis. Due to this unfortunate characteristic further analysis is required, in the form of a mathematical analysis, to determine the exact causes of this problem. Another point of concern is the complexity of the valve, due to the many moving parts, this results in high production costs and increased reliability concerns.

This masters project will include the mathematical analysis of the current Harrison CNG Electronic Gas injection system, further testing and refinement. The objective is to produce a modular system that can be retrofitted to any make of vehicle. Research will be directed in the development of mathematical equations to analyse valve operation for improvement of operation, to increase performance the valve will be redesigned to reduce complexity and ready it for production. The valve will be tested on a variety of vehicles from a 2 litre sedan to a 5.8 litre diesel engine that has been converted to operate on CNG, to prove the versatility of the valve and its ability to tailor the engine torque curve to that required for the vehicles unique operating requirements.

Acknowledgments

I would like to take this opportunity to thanks those that have helped me throughout the project. I would firstly like to thanks my parents, Ian and Lin, and also my brother, Graeme for their continuous support and encouragement. Secondly I would like to thank John Harrison the original inventor of the valve for opportunity to he awarded me when he approached Massey University for further development on the valve. Never in my wildest dreams did I think that I would travel to China to do diesel engine conversions at the beginning of the project. I would like to thank Jou and Lin Lee-Shian Ma for sponsoring the development of the valve and for the trips to China to prove the versatility of the valve and also for their tremendous patience during the development process.

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Lastly but by no means least to my girlfriend, Shu Chin, for all her endless encouragement and never-ending confidence in me, I would like to tribute this poem:

She Takes Me...

by William Thomas Kinsey

She takes me many places
that I have never been:
all I do is look at her
it's then she takes me in.

Into a new and different world
one I've heard mention of;
a world of grace and beauty
a world of endless love.

A world of peace and kindness
(the way the world should be),
with her I see it every day
she is that 'world' to me.

While with her life's a fantasy,
a dream world I would say;
yes, she makes my dreams come true
and she does it every day.

Her world of love is awesome,
it fills the poets pen.
All I do is look at her
and then... she takes me in.

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Chapter 1: Introduction

CNG is seen as the most promising alternative fuel at present, containing between 80% and 95% methane and some other heavier hydrocarbons. Methane has a simple structure and therefore better emissions after combustion; it also has a very high anti-knock value which allows one to increase the thermal efficiency of the engine. Recent surveys have found that the world resources of natural gas are estimated to be 6358.575 trillion cubic feet. At today's current level of consumption this is estimated to last for the next 70 years [3].

Due to methane having a high octane a rating of approximately 120 and also the extended flammability limits of 5-15%, natural gas is well suited to lean burn operation. A lean burn engine operates with an excess air, stoichiometric operation has an air/fuel ratio of 17.2:1 (lambda 1) for methane, if an engine operates with a lean burn condition then the air/fuel ratio is then greater than 17.2:1. The advantages of lean burn are that it decreases fuel consumption and reduces exhaust emissions. Current generator sets operate at lambda 2 (A/F = 34.4:1), but these engines have specially designed combustion chambers having a precombustion chamber that ignites a richer mixture and then the main combustion chamber that contains a very lean mixture. Lean burn operation is not only limited to natural gas, diesel engines operate with excess air due to emissions laws, if diesel engines were to operate with stoichiometric air/fuel mixtures, high particulate emissions (black smoke) are produced which is restricted by current emissions laws. Gasoline engines may also be operated at lean burn using technology known as GDI (Gasoline Direct Injection), this creates a stratified charge within the combustion chamber resulting in a richer mixture present at the spark plug to initiate combustion and a lean mixture throughout the rest of the combustion chamber. Natural gas will result in better emissions than both gasoline and diesel due to its simple structure. Studies have shown that NO_x reductions of 50-80% and particulate matter reductions of 80-90% are possible when heavy-duty vehicles are converted to operate on natural gas instead of diesel.

One of the major disadvantages of natural gas is its low energy density, therefore requiring large heavy storage cylinders. This can be overcome by liquefying natural gas, however natural gas only liquefies at -162°C thereby requiring cryogenic freezing which increases complexity of the system and also the cost. Other technologies have been introduced that use a porous absorbent material that is introduced into the cylinder that increases the energy density of the vessel. This technology uses low pressures of 3.5-5MPa to store 2/3 of the total energy density present in a standard CNG cylinder filled at 20MPa, this allows the saving in cylinder cost, allows the storage devise to have a irregular shape and be made of a lighter material.

Natural gas has had an interesting history in New Zealand that saw New Zealand as a world leader in CNG technology in 1970-1985 after which the infrastructure unfortunately fell apart and today there is very little use or infrastructure present. However Europe and the United States have now seen a resurgence in natural gas technology with international OEM vehicle manufacturers introducing specially designed vehicles to operate with both natural gas and gasoline. With the stringent emissions laws currently in effect and the push by governments to reduce overall green house gases the opportunity for natural gas to become a dominant fuel is almost a reality.

The purpose of this master study was to examine and perfect a CNG (Compressed Natural Gas) fuel metering system. The metering system uses the vehicle's onboard electronic control unit (ECU) used to meter petrol into the OEM (Original Equipment Manufacturer) engine to create a reference signal that is then used to meter the required fuel for a given engine operating condition. This metering system is a mechanical system that has been in development independently for the last few years and was brought to Massey University for further development. The scope of the project included a mathematical analysis of the apparatus to determine the mostly likely cause of action, followed by further testing on a motor vehicle. This thesis will detail the advantages and disadvantages of natural gas and also look at the other alternative fuels that are presently available; it will also discuss some of the testing done to prove the versatility of the metering apparatus and some of the mathematical analysis done. There will be a brief description of the valve operation and a discussion on the results from testing, highlighting achievements and likely shortcomings that future development may eliminate.

The metering apparatus was designed to be a modular system that may be employed on a large variety of vehicles with very little modification to the OEM equipment so that the vehicle may then operate on CNG. The range of engines that can be operated using this apparatus range from a small 1 litre engine to a large 5.7 litre diesel engine that had been converted to operate on diesel. When converting a SI (spark ignition) engine to operate on CNG very little has to be modified to allow for this alternative fuel, the basic components required are:

- CNG cylinder
- Injector emulator
- Ignition advance unit
- CNG metering system
- Mixer unit

The components are easily installed and require no adjustments or calibration by the end users. The advantages of the system are that the vehicle performance is maintained with the alternative fuel operation due to precise metering of natural gas and that the vehicle operator experiences little if no evidence of vehicle operation on natural gas.

Chapter two will introduce natural gas, the history of natural gas in New Zealand, introduce other alternative fuels, testing procedures for emissions standards, exhaust after-treatment, onboard diagnostics, and the advantages and disadvantages of various control systems for engine operation. Chapter three will concentrate on the mechanical development of the valve, firstly describing the operation of the valve and then describe the development to improve the valve. Chapter four will concentrate on the electronic requirement for vehicle conversion to natural gas. Chapter five describes the testing and results of the vehicle conversions. Chapter six will detail future developments of the valve to ready it for production.

Chapter 2: Literature Review

2.1 Overview of CNG

Compressed natural gas (CNG) is seen to be the most promising alternative fuel for automotive applications due to the simplicity of the fuel molecule, after complete combustion natural gas produces less harmful green house gases than other fuels except for hydrogen. CNG is well suited to the Otto cycle due to its high knock resistance and the fact that it easily forms a homogeneous mixture with air. CNG consists mainly of methane (CH_4) with minor amounts of heavier hydrocarbons and some non-hydrocarbons. Table 1 shows the typical composition of natural gas; however the composition varies significantly from region to region. For example, in the United States natural gas contains 85-95% (by volume) methane, in the north sea it may contain up to 20% carbon dioxide and the gas supplied to Finland from Russia contains 98% methane.

Component	Volume (%)	Mass (%)
Methane	92.29	84.37
Ethane	3.6	6.23
Propane	0.8	2.06
Butanes	0.29	0.99
Pentanes	0.13	0.53
Hexanes	0.08	0.39
CO2	1	2.52
Nitrogen	1.8	2.89
Water	0.01	0.01
Total	100	100

Table 2.1: Natural gas composition [1]

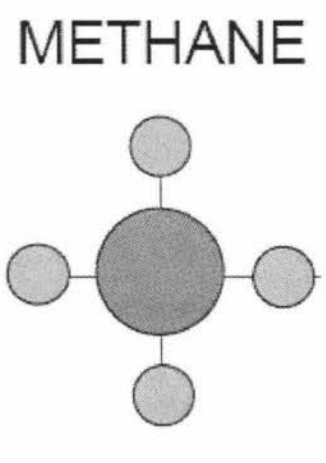


Figure 2.1: Methane molecule [2]

The energy content of natural gas is variable and is dependent on its composition: the greater the percentage of non-combustible gases in a natural gas, the lower the Btu (British thermal unit) value. In addition, the more carbon atoms in a hydrocarbon gas, the higher its Btu value.

The composition of natural gas has a significant impact on the engine performance, equipment durability and exhaust emissions. Water, sulphur compounds, carbon dioxide, oxygen and other impurities in natural gas cause storage tank and fuel system corrosion and corrosion fatigue resulting in the cracking of materials. It is therefore important to minimise these impurities to achieve an acceptable service life of the storage tank and other associated equipment. Another point of consideration is the formation of ice in the regulators, fuel filters and injectors due to the condensation of the water content in the natural gas which would cause rough engine operation. In order to prevent this occurrence natural gas is dried to a dew point under the minimum ambient temperature in which the vehicle/engine may operate.

2.1.1 Origin of Natural Gas Accumulations

The decomposition of organic material in an oxygen-poor environment, with the aid of anaerobic bacteria, results in the formation of methane. Since organic matter is ubiquitous in the younger sediments of the earth, so is methane. If all of the existing methane could be collected, it could provide most of the world's energy for hundreds of years. Unfortunately, most is too diffuse to be commercially recovered. Natural gas, a hydrocarbon mixture consisting primarily of methane and ethane, is derived from both land plant and marine organic matter. Over geologic time, almost all natural gas reaches the earth's surface and is lost to the atmosphere. When its upward migration is interrupted by a geologic trap (an upwardly convex permeable reservoir rock sealed above by impermeable cap rock) commercial quantities of gas can accumulate. This gas is termed non-associated gas. Commercial amounts of gas also can accumulate as a gas cap above an oil pool or, if reservoir pressure is sufficiently high, dissolved in the oil. Such natural gas is termed associated gas. [3]

Natural gas generation and migration occur over an extensive vertical zone that includes shallow biogenic gas, intermediate dissolved gas of the oil window, and deep thermal gas. The production of biogenic methane requires anaerobic microbial activity, and is confined to poorly drained swamps, some lake bottoms, and marine environments below the zone of active sulphate reduction. Gas of predominantly biogenic origin constitutes more than 20 percent of global gas reserves. The mature stage of petroleum generation occurs at depths between about 6,500 and 16,000 feet, depending upon the geothermal gradient. At these temperatures and pressures the full range of hydrocarbons are produced within the oil window and significant amounts of thermal methane gas are often generated along with the oil. Below about 9,500 feet, primarily wet gas that contains liquid hydrocarbons is formed. In the post-mature stage, beneath about 16,000 feet, oil is no longer stable and the main hydrocarbon product is thermal methane gas. This thermal methane gas is a product of the cracking of the existing liquid hydrocarbons. [3]

Gas displays an initial low concentration and high dispersity, making adequate seals very important to conventional gas accumulation. Due to differences in the physical properties of gas and oil, similarly sized oil traps contain more recoverable energy (on a Btu basis) than gas traps, although more than three-quarters of the in-place gas often can be recovered. Less than one percent of the gas fields of the world are of giant size, originally containing at least 3 trillion cubic feet of recoverable gas. These fields, however, along with the associated gas in giant oil fields, account for about 80 percent of the global proved and produced gas reserves. Oil is derived mainly from marine or lacustrian source rocks, but, since gas can be derived from land plants as well, all source rocks have the potential for gas generation. Many large gas accumulations appear to be associated with the coal deposits. [3]

Proved reserves are those that could be economically produced with the current technology. Below is an overview of the current estimates for the proven world natural gas supply, as can be clearly seen Eastern Europe, Former U.S.S.R and the Middle East hold the largest reserve of proven natural gas.

	BP Statistical Review *Year-End 2004* *(trillion ft^3)*	**CEDIGAZ** **January 1, 2005** *(trillion ft^3)*	*Oil & Gas Journal* *January 1, 2005* *(trillion ft^3)*	*World Oil* *Year-End 2003* *(trillion ft^3)*
North America	260.491	259.639	260.494	268.853
Central & South America	250.595	243.956	250.52	240.937
Western Europe	178.305	217.929	182.487	170.054
Eastern Europe & Former U.S.S.R.	2081.433	2043.75	1964.16	2693.227
Middle East	2570.793	2589.649	2522.125	2539.65
Africa	496.43	498.86	476.509	443.2
Asia & Oceania	501.517	504.793	383.913	449.91
World Total	6339.563	6358.575	6040.408	6850.83

Table 2.2: World proven natural gas supply [4]

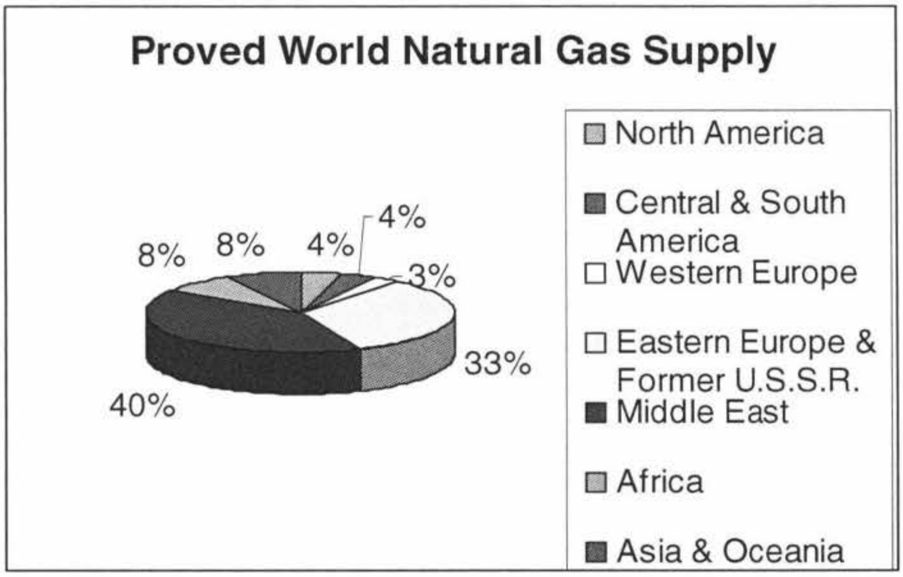


Figure 2.2: Proven world natural gas supply [4]

The Russian Federation holds the world's largest reserve of proved natural gas at 26.7 % of the world supply, followed by Iran at 15.3% and Qatar at 14.4%. At the end of 2003 the world reserve was estimated to be 179.53 trillion cubic meters. The world's reserve-to-production ratio is between 60 and 70 years, this is the estimate of the remaining time that resources would last if production stayed at its present level.

2.1.2 Supply and Demand

At the end of 2004 the Unites States and the Russia Federation accounted for 42.1% of the world's natural gas production, as can be seen in the graph below. Other major producing countries such as Canada, Iran, Algeria, Indonesia, Saudi Arabia, Norway, Netherlands and Uzbekistan contribute to produce 67.8% of the world's natural gas production. The world's production of natural gas amounted to 2691.6 billion cubic meters in 2004; this was a 2.8% increase as compared to that produced in 2003.(Source: BP Statistical Review of World Energy June 2005)

Natural Gas Supply (billion cubic meters), 1970-2004

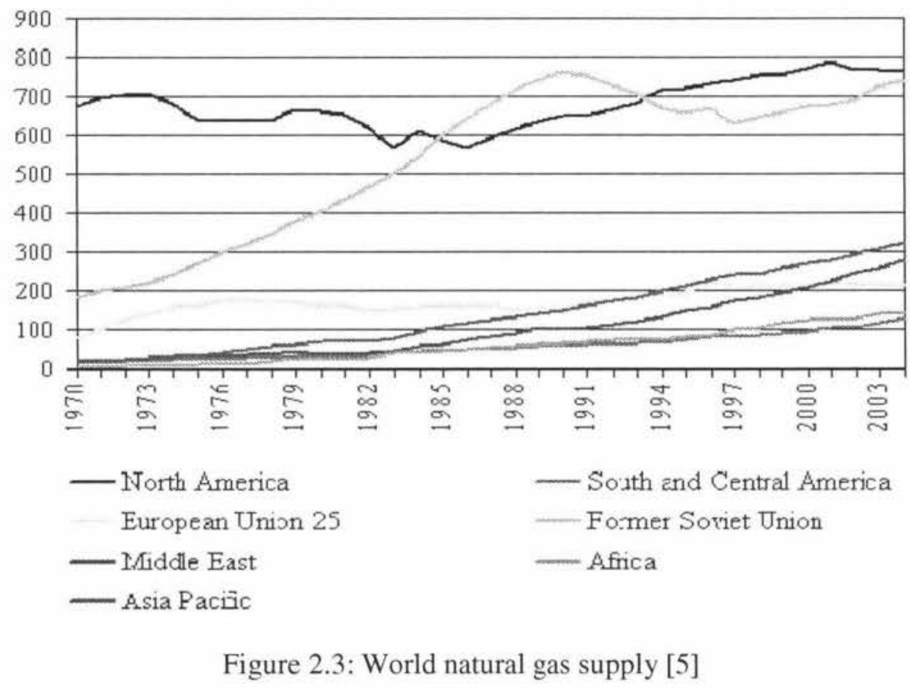


Figure 2.3: World natural gas supply [5]

World natural gas production is expected to grow in the future as a result of new exploration and expansion projects, in anticipation of growing future demand.

Natural gas accounts for approximately one quarter of the world's energy consumption. The graph below indicates that there has been a steady increase in the consumption of natural gas as an energy resource from 1965.

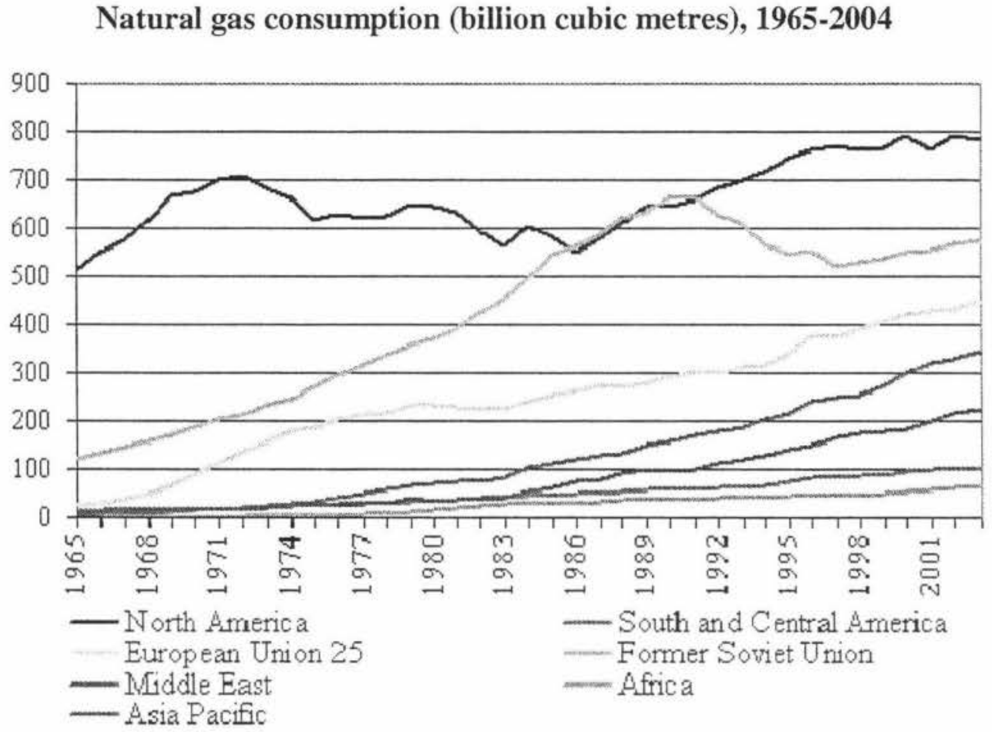


Figure 2.4: World natural gas consumption [5]

The main consuming countries were United States with 24% and the Russian Federation with 15% of the total world consumption. North America, Europe and Eurasia accounted for 70.4% of the world's natural gas consumption. The total world consumption of natural gas was 2689.3 billion cubic meters which was an increase of 3.3% over 2003. South and Central America as well as the Middle East were recorded to have the highest growth rates of 11.4% and 7.2% respectively [5].

2.1.3 Combustion properties of natural gas

The main factors and properties that determine the performance of gas in an engine are as follows:

- Heat of combustion per unit volume
- Heat of combustion per unit weight
- Heat of combustion of the mixture per unit volume
- Wobbe-index
- Self-ignition temperature
- Knock resistance
- Combustion limits

The heat of combustion of mixture per unit volume is an indication as to the power density that can be achieved during combustion and the heat of combustion per unit volume or per unit weight indicates the size and weight of the fuel storage onboard the vehicle.

When switching the engine operation form gasoline to natural gas one has to take into consideration the difference in stoichiometric requirements of natural gas to that of petrol and also the fact that because natural gas is introduced into the engine in a gaseous form, a certain percentage of air will be displaced during this process. To determine the correct air/fuel ratio for stoichiometric operation on natural gas, petrol and diesel the following chemical reactions are used to simulate complete combustion of the various fuels:

Natural gas: $$2CH_4 + 4\left[O_2 + \frac{79}{21}N_2\right] = 2CO_2 + 4H_2O + 4\frac{79}{21}N_2 \qquad [2.1]$$

Petrol: $$2C_8H_{18} + 25\left[O_2 + \frac{79}{21}N_2\right] = 16CO_2 + 18H_2O + 25\frac{79}{21}N_2 \qquad [2.2]$$

$$\text{Diesel: } 2C_{12}H_{26} + 37\left[O_2 + \frac{79}{21}N_2\right] = 24CO_2 + 26H_2O + 37\frac{79}{21}N_2 \qquad [2.3]$$

Using the above balanced chemical equation we are then able to determine the required air/fuel ratio. For natural gas the required air fuel ratio is calculated as follows:

$$A/F_{stoich} = \frac{\left[(6\times 32) + (6\times 28\times \frac{79}{21}\right]}{2(12+(4\times 1))} = 17.2 \qquad [2.4]$$

To determine the percentage of air displaced at stoiciometric operation using natural gas the following equation is used:

$$\text{Percentage displacement air} = \frac{1}{A/F_{stoich}}\frac{MW_{air}}{MW_{fuel}} \qquad [2.5]$$

Where: A/F_{stoich} = stoichiometric air fuel ratio

MW_{air} = molecular weight of air, 28.97

MW_{fuel} = molecular weight of methane, 16.04

Therefore at stoichiometric operation, which is an air/fuel ratio of 17.2:1 for methane, the percentage air displaced is 10.5 %. It is therefore important to remember that for the air/fuel ratio to remain at 17.2:1 the amount of natural gas introduce has to be reduce because of the displaced air.

Using a simple formula it is possible to demonstrate the difference in fuel mass required due to the displaced air.

The air consumed by a 2 Litre engine at 2000RPM with a volumetric efficiency of 80% is:

$$AirConsumed = (RPM)(\frac{Displacement}{2})(V_e)(V_{\mathrm{Red}}) \qquad [2.6]$$

In this case the value V_{red} is equal to 0.905 to compensate for the 10.5% of displaced air. Therefore the amount of air consumed is 1448 L/min.

To convert this to mass

$$Air_{mass} = AirConsumed \times \rho \qquad [2.7]$$

where $\rho = 1.293$ g/m^3 for air

Therefore the air mass consumed is 1872.26 g/min

The mass of fuel is the calculated as follows:

$$F_{mass} = \frac{AirConsumed}{A/F_{stoich}} \qquad [2.8]$$

The result is a fuel mass of 108.85g/min

If however the reduction factor were not accounted for then the fuel mass introduce would be 120.28g/min and would result in an effective air/fuel ratio of 15.56 which is a richer mixture and would result in higher exhaust emissions.

The Wobbe index is used to measure the interchange ability of various gases. The Wobbe index relates to the amount of thermal energy that flows through a fixed orifice size, two gases with the same Wobbe index can be interchanged without any altering of the metering apparatus. If a metering system does not have any feedback, and the engine is tuned for operation on a specific gas, if the vehicle is then refuelled with gas from a different source that has a different Wobbe index the vehicle will subsequently either run lean or rich depending on the difference in the Wobbe index. If however the metering system used a oxygen sensor as a feedback mechanism then the system would be able to compensate for the change in fuel composition.

A diesel engine uses the compression ignition principle where the end temperature and pressure of compression initiates auto ignition of the fuel. For CNG to be used in this manner would require a compression ratio of 38:1 due to the high auto ignition temperatures, this is not feasible and therefore a natural gas engine requires either a spark ignition system or a pilot injection system that uses diesel fuel. The high knock resistance of natural gas is an advantage to the Otto engine, because continuous knocking combustion is destructive for an engine. A high compression ratio is possible approaching 14:1 with natural gas due to the research octane valve of 120, the high compression ratio increases the thermodynamic efficiency of the engine. The thermal efficiency of an engine is directly proportional to the compression ratio, the formula that relates thermal efficiency to compression ratio for an Otto cycle engine is as follows:

$$\eta_t = 1 - \frac{1}{CR^{1-\gamma}} \qquad [2.9]$$

from the equation above one can see that the higher the compression ratio that better the thermal efficiency of the engine.

The knock resistance of a gaseous fuel is related to the methane number. Methane has the highest anti-knock resistance and is therefore given the value of 100, whereas hydrogen has the lowest anti-knock resistance and is therefore given the value of 0. Therefore if a gaseous fuel has a methane number of 60, it therefore has an anti-knock resistance equal to 60% of methane and 40% of hydrogen.

The combustion limits of natural gas are 5-15% by volume, which differs to that of petrol that has combustion limits of 0.6-8%. This enables one to run natural gas engines at very lean limits in this way cutting down on emissions and also fuel consumption. At lambda 1.4 and 1.5 NO_x emissions are considerably less than that at stoichiometric or slightly leaner mixtures.

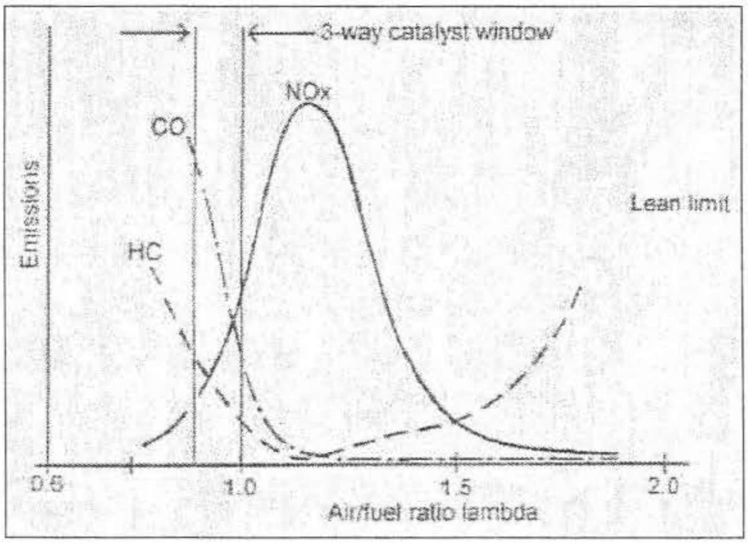


Figure 2.5: Exhaust emissions with respect to lambda [6]

However methane does burn slower than gasoline and when operating under lean conditions the mixture will burn even slower. Therefore the ignition system has to advance the spark according to the air/fuel ratio in order to maintain complete combustion. Another requirement from the ignition system when running at leaner mixture is a greater ignition energy, at stoichiometric operation the spark energy required is at the minimum, if the mixture is then altered leaner or richer the ignition energy required to ignite the mixture increase.

2.1.4 Fuel composition

As has been mentioned before in previous section, performance and emissions of SI gas engines is dependant on good combustion, high knock resistance, high energy content of fuel mixture and optimum combustion rate. In relation to engine performance the following natural gas properties are important:

- Density
- Heating value
- Stoichiometric Air/Fuel ratio
- Knock resistance

All of the above mentioned gas properties are linked to the gas composition; it is therefore evident that any variations in the gas composition will affect the performance and exhaust emissions of the engine. The composition of natural gas is not identical throughout the world as the gas is drawn from different production fields, even gas from a specific production field is prone to variations in composition over time, the report done by Hien Ly illustrates various surveys done in this field [6]. To maintain pipeline safety and longevity quality standards have been introduced to govern the quality of reticulated gas, an example of this type of standard is the New Zealand standard (NZS 5442) for general usage gas. Below are the limits from the New Zealand standard.

Absolute Limit		
Wobbe index	minimum	46.0 MJ/m^3
	maximum	52.0 MJ/m^3
Relative density	maximum	0.8
Oxygen – for gas to be transported through		
MP and LP systems only	maximum	1.0 mol %
Oxygen – in all other cases	maximum	0.1 mol %
Hydrogen	maximum	0.1 mol %
Hydrogen sulphide	maximum	5 mg/m^3
Total sulphur	maximum	50 mg/m^3
Total halogens (as C)	maximum	25 mg/m^3
Water	maximum	100 mg/m^3
Hydrocarbon dewpoint	maximum	2 °C at 5 MPa
Temperature	minimum	2 °C
	maximum	40°C

Table 2.3: NZS 5442 natural gas composition limits [7]

The limits set by he New Zealand standard are based on criteria for safe utilisation of natural gas in a wide variety of gas applications while still remaining its safe transportation through pipelines.

To accommodate the requirements of naturals gas vehicles in terms of engine applications a number of international standards have been established, examples of these are the SAE J1616 and ISO 15403. These standards establish limits relative to the use of CNG as a vehicle fuel. The following areas are addressed in these standards:

- Water Content
- Carbon Dioxide
- Sulphur Compounds
- Methanol
- Oxygen
- Particulate Matter
- Oil Content

- Pressure Hydrocarbon Dewpoint Temperature
- Natural Gas Odorant
- Wobbe Index
- Knock Rating

Not only are there international standards set to ensure the quality of CNG but engine manufacturers also list the recommended natural gas composition for satisfactory operation and durability of their engines dependant on location of use in the world [6].

2.1.5 Effects of fuel composition on engines and vehicles

Water content: The presence of a high water content in CNG fuel may result in the formation of water, ice particles, frost or hydrates when the vehicle is operating at low temperatures and high pressure. It is therefore important to ensure that the pressure water dewpoint temperature is compatible with the geological location in which the vehicle is operated, to ensure that condensation of water will not occur in the on-board CNG storage cylinder.

Heavy hydrocarbons: During peak demand periods some utilities may add propane/air mixtures to natural gas. Due to propane's low vapour pressure, if a significant amount of propane is present in a high pressure and low temperature it will transform into its liquid form. If the cylinder pressure is then to reduce thereby resulting in the revapourisation of the propane liquid, fuel variability will occur, thereby making control of the air/fuel ratio difficult. Not only will the air/fuel ratio be difficult to control, the presence of heavier hydrocarbons reduces the anti knock rating of natural gas and can therefore result in engine damage.

Carried-over oil: Often lubrication oils from pipeline compressors are carried over and are present in trace quantities in natural gas. If a high level of oil is present in CNG there is a possibility that the oil may condensate and pool in the regulators or storage cylinders thereby resulting in operational problems of the engine.

Sulphur compounds: Sulphur compounds in the form of mercaptans, hydrogen sulphide and odorants are found in natural gas. Mercaptans and hydrogen sulphide naturally occur in natural gas and are reduced through treatment processes to reduce the potential of sulphur stress cracking in high pressure transmission pipelines. Due to natural gas being colourless and odourless, odorants are added as a means of easy leak detection prior to distribution of natural gas. Lean burn operation of an engine requires an oxidation catalyst to reduce HC and CO emissions, the presence of sulphur-based odorants such as tetrahydrothiophene (THT) and tertiary butylmercaptan (TBM) in concentrations as low as 10-15 mg/m3 has detrimental effects on the conversion efficiency of the catalyst. The paper presented by Hien Ly illustrates the results of other authors work in this field [6].

Knock resistance: Methane has the highest knock resistance of all hydrocarbon motor fuels, this is because of the low carbon content of the compound. It is therefore evident that if other heavier hydrocarbons are present in the composition of natural gas it will directly affect the knock rating of the natural gas. Kubesh [8] showed that it is possible to determine the octane value of the fuel if the composition is known. This estimation is based on the reactive hydrogen/carbon ratio of a value greater than 2.5 and requires that the inert gas percentage be lower than 5 percent in the natural gas mixture. The equations are as follows:

- $MON = -406.14 + 508.04*(H/C) - 173.55*(H/C)2 + 20.17*(H/C)$ [2.10]
- $MN = 1.624*MON - 119.1$ [2.11]

As mentioned before vehicle manufactures specify minimum knock resistance properties of a fuel to ensure durability and performance of the engine.

If a vehicle has been optimised for maximum performance and efficiency on a certain composition of natural gas and has no form of feedback in terms of the exhaust emissions, then the vehicle may experience poor performance and elevated exhaust emissions when variations are present. Nigel N. Clark et al concluded that "open loop operation at steady state can be accomplished successfully with gases of widely varying concentrations but the same Wobbe number." [9] Liss and Thrasher [10] found that variations in the Wobbe number will affect the equivalence ratio for an open loop engine.

Exhaust Emissions: The emissions components that are monitored by current regulations are CO2, CO, HC and NO_x. The production of NO_x during combustion is proportional to temperature and the availability of oxygen. Therefore if the gas composition changes resulting in higher combustion temperatures or differences in the air/fuel ratio then the production of NO_x will increase. CO is a result of a lack of oxygen and HC are a result of lower temperatures and flame speed causing incomplete combustion, it is therefore evident that if the air/fuel ratio changes this will have an affect on the emissions of CO and HC. Due to the U.S emissions regulations only limiting the non-methane hydrocarbon (NMHC) emissions, differences in the concentrations of heavier hydrocarbons within the fuel must be kept to a minimum, as the proportion of NMHC in the fuel directly affects the levels of NMHC exhaust emissions. The primary effect of increased emissions is due to the variations in the Wobbe index, resulting in variations in of the air/fuel ratio. In order to limit increases in pollutant emissions due to variations in natural gas properties, the Californian Air Resource Board has established limits for natural gas sold commercially as vehicle fuel [11].

2.2 Natural Gas history in New Zealand

In 1969 a gas field was found of the coast of New Zealand and in 1979 a decision was made to develop the field for electricity generation. However in the late 1970's it was found that the demand for electricity was less than was estimated, therefore the use of gas as an alternative was considered. To aid the uptake of an alternative fuel the oil crisis of 1978/79 resulted in restrictions on petrol sales and an increase in petrol prices. In 1979 the New Zealand Government set a target of 150,000 vehicles to be operating on CNG by 1985. However in 1984 the target and incentives were abandoned by the new Labour Government due to a change in economic policy. By 1985 over 100,000 CNG kits had been sold which equated to 67% of the total vehicles that had access to natural gas and natural gas constituted 6% of the transport fuel used in SI engines.

The question is 'How was it possible for a country to covert such a large percentage in such a short period of time?' The answer to this question relates to the local government involvement and the drive to find an alternative. The role of the Government in the CNG program was crucial, the following is a summary of the government involvement:

- Funding in 1978 of R&D to evaluate technical, economic and environmental aspects of CNG
- Formulation of implementation plan in 1979
- Acceptance of plan, setting of targets, setting of incentives in 1979
- Establishment if CNG Co-ordination Committee in 1979
- Establishment if infrastructure of regulations and standards insperirate, etc. 1979-1985
- Training programs foe installers.1989 onwards
- Various promotional and marketing activities 1979-1985
- Marketing studies 1980,1981,1984
- Modification of incentives 1980,1982 and 1984
- Conversion of Government vehicle fleet to CNG 1979-1985
- Funding of engine and related research 1980 onwards
- Monitoring of all aspects if CNG program
- Extensions of natural gas reticulation (through NGC, a Government cooperation)

[12]

Of particular importance during the CNG program were market studies that were done to determine the plans and policies to ensure the success of CNG. As shown above three market studies were done, 1980,1981 and 1984. The 1980 market survey found that the government incentives needed to be increased to maintain the growth in the CNG conversion market. As a result of the survey the Government reaction was to accelerate depreciation for vehicle conversions and altered the rules for the 25% refuelling station grant. This resulted in an increase in kit sales, an average increase to 955 kits per month in the period of January to June in 1981 from an average of 532 per month in July to December in 1980. In 1982 an industry funded CNG voucher was introduced and then later in 1983 the accelerated depreciation was replaced by a 100% Government loan scheme for conversion kits. These steps created an environment that persuaded the public to convert their vehicles to CNG.

From the 1984 market survey it was found that the vehicle conversions were substandard and that there was a need for quality assurance and warranty for CNG vehicle owners. It was also found that people looking to convert their vehicles to CNG gained most of their information from installers and existing converted vehicle owners. Therefore unsatisfied users were able to persuade potential conversion not to be done. Due to the changing engine management systems, cars that used electronic fuel injection suffered from poor performance and high fuel consumption when converted to CNG operation, resulting in government loans not being repaid. The arrival of cheaper petrol and diesel removed the economic advantage that CNG offered.

By 1985 the sale of CNG conversions kits dropped and eventually stopped altogether. In the period of 1979 and 1985 the Government had invested more than $NZ 20 million in the various incentive schemes and loans used for vehicle conversions and refuelling stations on top of the administrative costs involved in implementing the program. With the election of a new Government in 1984 all incentives were withdrawn and the New Zealand CNG industry declined. Deregulation of service stations resulted in oil companies buying the privately owned stations and subsequently removing the CNG equipment. Today there are no CNG refill stations in Auckland, the closest refill stations are one in Pukekohe and another in Huntly.

There are however positive points to be taken from the CNG history in New Zealand, particularly the experience related to the introduction of an alternative fuel into the transport sector. For the successful introduction, acceptance and to maintain a growth of an alternative fuel the following has to be implemented:

- CNG has to be no more than 50% of the retail price of petrol
- Direct Government support is required if CNG is 50% of the retail price of petrol and only indirect support when the cost is only approximately 30%
- A network of refuelling stations that can provide comparable refuelling service to gasoline stations
- High quality OEMs and/or conversions
- Infrastructure – standards, regulation, training
- Private sector and government champions

2.3 Other alternative fuels

This section will review other alternative fuels that are currently being used around the world. It will detail their advantages and disadvantages in respect to both vehicle performance and manufacture.

2.3.1 Alcohols (methanol and ethanol)

Methanol and ethanol are the two alcohol fuels considered as good alternative fuels to diesel and petrol. They are good candidates for alternative fuels due to their favourable physical and combustion properties being very close to that of petrol and diesel therefore being able to use the same fuel delivery systems and engine configuration. Both ethanol and methanol have higher octane ratings than petrol therefore allowing for higher compression ratios and better thermal efficiency; however both methanol and ethanol has lower energy densities than petrol. Methanol requires twice as much fuelling and ethanol one and a half times to equal that of petrol.

Methanol is a colourless liquid with a faint odour that is commonly produced by a process of steam reformation of natural gas. This is when natural gas is reacted with steam in the presence of a catalyst under high heat and pressure to for carbon monoxide and hydrogen. The most common use of methanol is to make methyl tertiary butyl ether (MTBE) that is used as an oxygenate for addition to gasoline for winter time oxygenate programs and to make reformulated gasoline [13] Another disadvantage of methanol is that formaldehyde is produced as a combustion by-product in larger quantities than that produced by petrol. Other exhaust emissions from methanol combustion are lower NO_x due to the lower combustion temperatures and because it contains oxygen it results in a leaner combustion resulting in less CO emissions.

The major disadvantage of methanol is that in its pure form it is the most aggressive to materials, both metals and elastomers. Methanol will corrode magnesium and aluminium, corrosion has been found to be rapid when aluminium is exposed to methanol vapours. Corrosion of aluminium results in gelatinous precipitates that clog filters and injectors and ultimately result in increased engine wear. Stainless steel is the metal least affected by methanol with carbon steel also being one of the least affected metals. Due to the high polar nature of methanol it results in methanol being a good solvent which in turn cause material compatibility problems with elastomers causing them to swell, shrink, crack, soften or harden. Methanol has also been found to react with certain lubricating oil additives and causes them to leach out of the oil [13]. Due to methanol's material compatibility problems vehicles that are designed to operate on petrol of diesel have to be modified to ensure that no materials are present that may be corroded by methanol. Modern vehicles are being made with aluminium blocks, heads and automotive plastic inlet manifolds to reduce weight and also for thermodynamic properties, however if used with methanol in duel fuel operation will have dramatic effects on the engine. Therefore anticorrosive coatings have to be applied to the engine components to ensure engine life.

Ethanol is primarily produced by using fermentation technologies. The feedstock used is corn, other grains, potatoes, beets, cheese whey, sugar cane or grapes, almost any source of starch or sugar may be used as a feedstock for ethanol production. There are three primary ways that ethanol may be used as a transportation fuel:

1. As a blend with gasoline
2. As a component of reformulated gasoline both directly and/or transformed into a compound such as ethyl tertiary butyl ether (ETBE)
3. Used directly as a fuel with 15% or more of petrol known as E85

Not only does ethanol require 1.5 times the storage capacity of petrol, but acetaldehyde is a by-product of combustion when using ethanol.

Ethanol is recognised as being less aggressive than methanol towards metals and elastomers. The metals recommended for use with ethanol are carbon steel, stainless steel and bronze [13]. Ethanol will also react with elstomers however less aggressively than methanol. Most of the common elastomers will work satisfactorily with ethanol such as Viton, neoprene, natural rubber and flourosilicones.

2.3.2 M85 and E85

M85 and E85 came about because of the low vapour pressure and high latent heat of methanol and ethanol cause cold-start difficulties in SI engines, to overcome this problem and to increase the flame visibility it was decided that 15% volume gasoline be added to methanol and ethanol. This resulted in M85 and E85. The addition of petrol increases the percentage of carbon, decreases the percentage of oxygen and doesn't influence the hydrogen percentage. The addition of petrol does however drastically change the boiling characteristics resulting in a wider boiling range. The heating values of M85 and E85 are increased, resulting in only 1.75 litres of M85 and 1.4 litres of E85 to equal one litre of petrol. One disadvantage of introducing petrol is that the flash point is lowered to a value close to that of petrol and also the autoignition temperature is lowered.

2.3.3 DME

DME (dimethyl ether) has recently emerged as a fuel option. DME is produced via synthesis gas, this DME synthesis (oxygenated synthesis) has a slightly higher efficiency that methanol synthesis. DME is non toxic and is presently used as propellant in aerosol canisters. DME is similar to LPG in the sense that it liquefies at moderate pressures (6 bar), however is had excellent ignition properties (high centane number) and is therefore suited to the diesel process. DME has a low viscosity and lubricity and also reacts with certain elastomers. Due to the high vapour pressure of DME fuel transfer pumps must be located in the fuel tanks. This would require the design and construction of a high pressure injection system separately from that of diesel. New refuelling systems and new vehicles would also have to be developed.

The advantage of DME is that it gives the same or even better efficiency than conventional diesel. However the drawback is the energy loss in the fuel conversion process.

2.3.4 LPG

LPG (Liquefied petroleum gas) consists of several light hydrocarbons which have the main distinguishing characteristic of becoming a liquid when put under a pressure less than 300psi. Propane and butane are the most common LP gases, for transport LP gas propane is the main constituent. LPG is produced in two methods: 1) with the production of natural gas and 2) as a result of crude oil refining. Propane has similar advantages as natural gas in that it has no evaporative or running losses, unburned hydrocarbons are from propane are easier to oxidise in oxidation catalyst than methane and therefore results in lower hydrocarbon emissions.

Propane is fairly benign which allows tanks to be made of inexpensive steels and of aluminium. Propane is a good solvent for hydrocarbons and the plasticizers used in elastomers. It is important the right fuel hoses be used for propane. Hoses made from nitrite or neoprene should be used and are compatible with LP gas [13].

2.3.5 Vegetable oils

By reacting vegetable oils with methanol or ethanol, esters are formed which have been given the generic label "Biodiesel". Combustion of Biodiesel results in favourable emissions with reduced smoke, particulates and gaseous emissions. Other than emissions biodiesel has other favourable characteristics in that it contains no toxins and quickly biodegrades when spilt on the ground or in water. The only drawback of using biodiesel is due to the economic impact. Neat biodiesel has approximately 13% less energy than typical diesel, due to the oxygen content of about 10% in the fuel. Although due to biodiesels higher specific gravity of approximately 0.88 compared to that of 0.82 of conventional diesel, results in a 7% loss of energy per unit volume. Biodeisel has a higher viscosity and higher pour points compared to typical diesel which could affect operation in cold temperatures. However there are additives available to decrease the pour point.

Vegetable oils and their methyl and ethyl esters are physically very similar to diesel fuel. There has been no indication of any metals currently used for distribution, storage, dispensing or onboard storage of diesel fuels not being compatible with vegetable oil fuels. However, there are reports of some signs of incompatibilities with fuel transfer hoses, and nitrile and butadiene elastomers with methyl esters. Elastomers with high fourine content have not exhibited any problems to date [13].

2.3.6 Hydrogen

Hydrogen is the ideal alternative fuel, which can be used in internal combustion engines or in fuel cells. When hydrogen is combusted in an internal combustion engine the only emissions are water and oxides of nitrogen depending on the heat of the combustion process. When hydrogen is oxidised in a fuel cell the only product is water. Hydrogen is primarily produced by steam reformation of natural gas, although it may be produced from any source that contains hydrogen. The major drawback of hydrogen is the storage medium, compared to all other fuels hydrogen has the lower energy storage density. Hydrogen can be stored in compressed cylinders similar to CNG or liquefied that same as LNG, it can also be stored in metal hydrides or in carbon absorbents. Hydrogen as a liquid has about 27% of the energy per litre of petrol, and about 23% of energy per litre of diesel. As a compressed gas at 3000psi, hydrogen has only about 5% the energy of gasoline per litre. Thus to equal the energy storage volume of gasoline, hydrogen will need at least 4 times the fuels storage volume if stored as a liquid, and 20 times the storage volume if stored as a compressed gas [13].

Due to hydrogen molecules being the lightest and smallest of all fuel molecules some materials compatibilities with both metals and non-metals exist. Long term exposure of hydrogen to carbon steel can cause hydrogen "embrittlement" which makes the steel more susceptible to stress fractures. Areas most prone to hydrogen "embrittlement" are portions of steel that have been strain-hardened such as bends in piping that has been welded. High-strength steel alloys have been found to be more susceptible to hydrogen embrittlement rather than low-strength alloys. Hydrogen embrittlement is increased when hydrogen is stored under pressure, when purity of hydrogen is high, and when temperatures are near ambient. Hydrogen is compatible with mostly all elastomers posing no material compatibility concerns [13].

2.4 Exhaust emissions and testing procedures

Since the first emissions-control legislation came into effect in the 1960's there have been continuously tighter restrictions place on the permissible exhaust emission limits. During this time all industrial countries have introduced an emissions-control legislation that specifies the permissible exhaust emissions limits for gasoline and diesel vehicles. Today there are four primary exhaust emissions-control legislations, these are:

- CARB legislation (California Air Resource Board)
- EPA legislation (Environmental Protection Agency)
- EU legislation (European Union)
- Japanese legislation

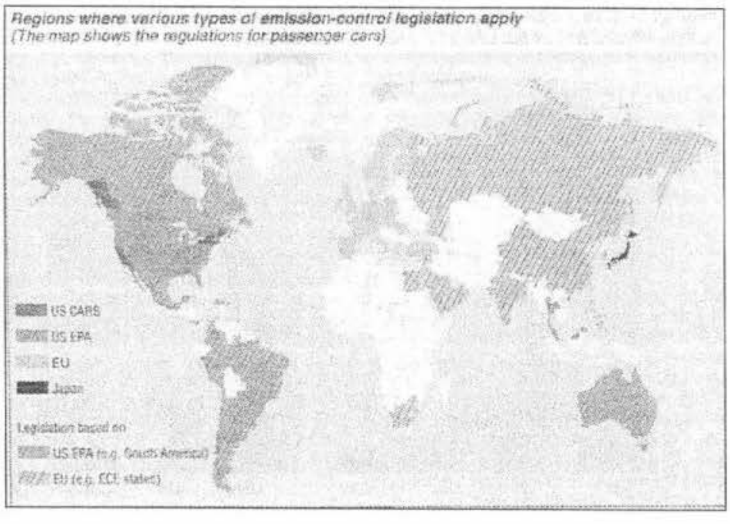


Figure 2.6: World regions implementing emission regulations [14]

Exhaust emissions are limited in two ways: firstly limits are put on the exhaust pollutants or particulates produces by a vehicle and also the evaporative hydrocarbon emissions, secondly the composition of fuels is regulated such that elements that are related to exhaust emissions, exhaust toxicity or performance of exhaust gas after treatment are minimised. These regulated components are:

- Benzene
- Total aromatics
- Polyaromatics
- Olefins
- Lead
- Sulphur

The exhaust emissions that are regulated are:

- Carbon monoxide (CO)
- Total hydrocarbons or non-methane hydrocarbons (THC or NMHC)
- Nitrogen oxides (NO_x)
- Particulates (PM)

Exhaust emission test procedures consists of the following elements:

- Requirements for ambient conditions
- A defined speed/load pattern for the vehicle or engine
- Reference fuel
- Measuring apparatus
- Gaseous component concentration
- Particulate mass emission of smoke density

- Exhaust gas flow
- Calibrations gases
- Calibration procedures

[2]

2.4.1 Test methods

Vehicles are categorized into different classes for testing:

- Passenger cars
- Light commercial vehicles (Under 3.5-3.8 tonne depending on legislation)
- Heavy commercial vehicles (Over 3.5-3.8 tonne depending on legislation)
- Off-highway (construction vehicles, agricultural and forestry machinery)

Exhaust emissions testing of passenger cars and light commercial vehicles is done on a chassis dynamometer. The measuring equipment used is a CVS (constant volume sampling) system; this measurement system may be used for emissions measurement for the US, EU and Japanese legislations. Each of the legislations requires the passenger cars and light commercial vehicles to perform dynamic test cycles. These test cycles are categorized into two different types dependant on the method in which they are performed:

- Test cycles derived from recordings made in actual on-road trips (FTP test cycle)
- Test cycles (synthetically generated) made up of road sections with constant acceleration and speed (MNEDC)

The following section will detail the various emissions-control legislations currently being used.

2.4.2 Passenger cars and light duty vehicles

2.4.2.1 CARB

On the 1 January 2004 LEV II came into force, this legislation includes all vehicles up to the permissible weight of 3.85 tonnes. Exhaust emissions are measured in the FTP 75 driving schedule (Federal Test Proceedure). The following are classifications dependant on the vehicle emissions and may be used by manufacturers:

- LEV (Low-Emissions Vehicle)
- ULEV (Ultra-Low-Emissions Vehicle)
- SULEV (Super Ultra-Low-Emissions Vehicle)
- ZEV (Zero-Emissions Vehicle)
- PZEV (Partial ZEV)

In order to gain approval for vehicle types, manufacturers must certify that the vehicles exhaust emissions will not exceed that of the emissions limits of:

- 50,000 miles or 5 years ("intermediate useful life"), or
- 120,000 miles or 10 year ("full useful life")

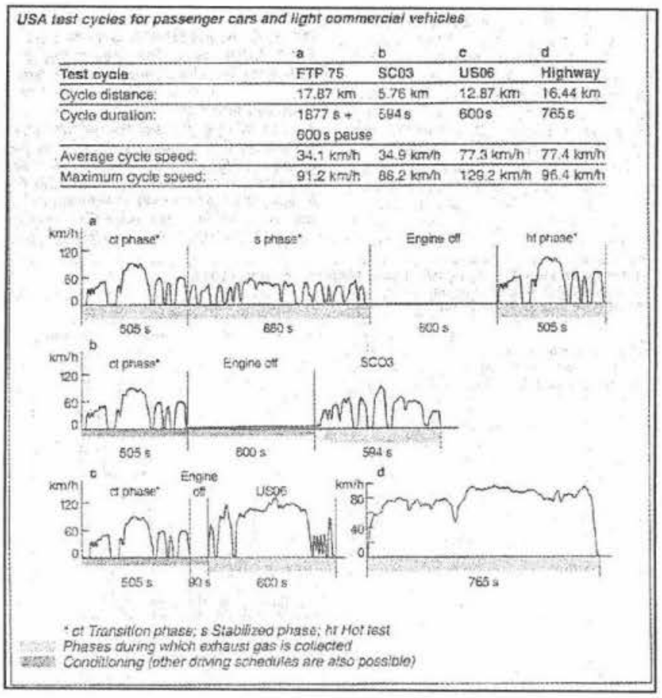
The manufacturer must supply two vehicle fleets from its production run to perform durability tests. One fleet must have driven 4000 miles before testing and the other is used for endurance testing to determine the deterioration factors of individual components in a continuous duty test.

California LEV II Emission Standards, Passenger Cars and LDVs < 8500 lbs, g/mi										
	50,000 miles/5 years					120,000 miles/11 years				
	NMOG	CO	NOx	PM	HCHO	NMOG	CO	NOx	PM	HCHO
LEV	0.075	3.4	0.05	-	0.015	0.090	4.2	0.07	0.01	0.018
ULEV	0.040	1.7	0.05	-	0.008	0.055	2.1	0.07	0.01	0.011
SULEV	-	-	-	-	-	0.010	1.0	0.02	0.01	0.004

Table 2.4: LEV II emissions standards for passenger cars and light commercial vehicles [15]

The FTP 75 test cycle consists of three test sections, these sections represent actual speeds measured in the U.S on the streets of Los Angeles during morning commuter traffic. The test procedure is as follows:

- Conditioning (vehicle allowed to stand for 12 hours at room temperature, 20-30°C)
- Collection of pollutants:
    - CT Phase - collection of diluted exhaust gas in bag 1 of CVS during cold transition phase
    - S Phase – after 505 seconds changeover to sample bag two for stabilised phase, engine then switched off for 600 seconds
    - HT Phase – Engine restarted, speeds in this phase are the same as that in the CT phase
- Evaluation



	a	b	c	d
Test cycle	FTP 75	SC03	US06	Highway
Cycle distance:	17.87 km	5.76 km	12.87 km	16.44 km
Cycle duration:	1877 s + 600 s pause	594 s	600s	765 s
Average cycle speed:	34.1 km/h	34.9 km/h	77.3 km/h	77.4 km/h
Maximum cycle speed:	91.2 km/h	86.2 km/h	129.2 km/h	96.4 km/h

Figure 2.7: USA test cycles for passenger and light commercial vehicles [14]

The individual results of the three phases are weighed by factors of 0.43 (ct phase), 1 (s phase) and 0.57 (ht phase) to get an overall result. The weighed sums of all pollutant masses from all three bags are correlated with the distance travelled during the test and then expressed as pollutant emission per mile.

2.4.2.2 EPA

EPA legislation principally applied to the states of the U.S, excluding California. The EPA legislation for passenger cars and light duty trucks (LDT) is not as strict as that of the CARB. In 2004 the Tier 2 standard was introduced that specifies emission limits for:

- Carbon monoxide (CO)
- Nitrogen oxides (NO_x)
- Non-methane organic gases (NMOG)
- Formaldehyde (HCHO)
- Particulates (PM)

The exhaust emissions are measured during the FTP 75 driving schedule, sampled and correlated with the route driven to then be expressed as grams per mile.

The emissions limits are divided into 10 (passenger cars) or 11 (HLDT, heavy LDT) emissions standards (Bin). Bin 9 and 10 are interim bins that will cease to apply after 2007.

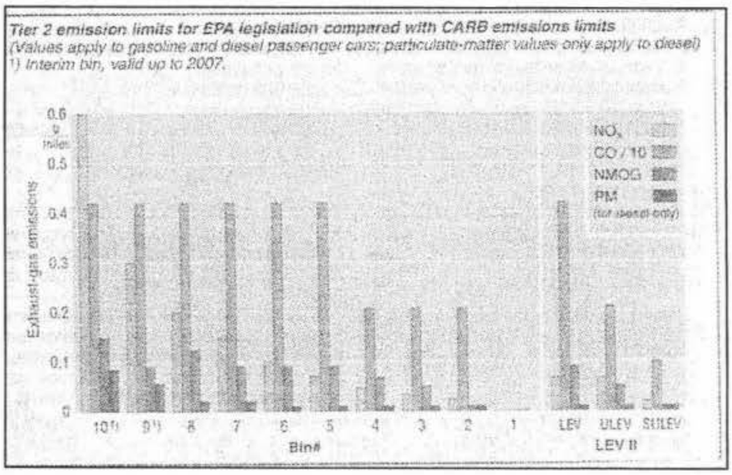


Figure 2.8: EPA emission limit compared to that of CARB [14]

2.4.2.3 EU

Emission limits for passenger cars and light commercial vehicles are contained in the emission-control standards:

- EU1 (from 1 July 1992)
- EU2 (from 1January 1996)
- EU3 (from 1 January 2000)
- EU 4 (from 1 January 2005)

The EU standards specify emission limits for the following pollutants:

- Carbon monoxide (CO)

- Hydrocarbons (HC)
- Nitrogen Oxides (NO_x)
- Particulates, only for diesel vehicles

The permissible emission limits are based on a distance travelled and are expressed in g/km. The exhaust gas measurement is made on a chassis dynamometer using the MNEDC (Modified New European Driving Cycle).

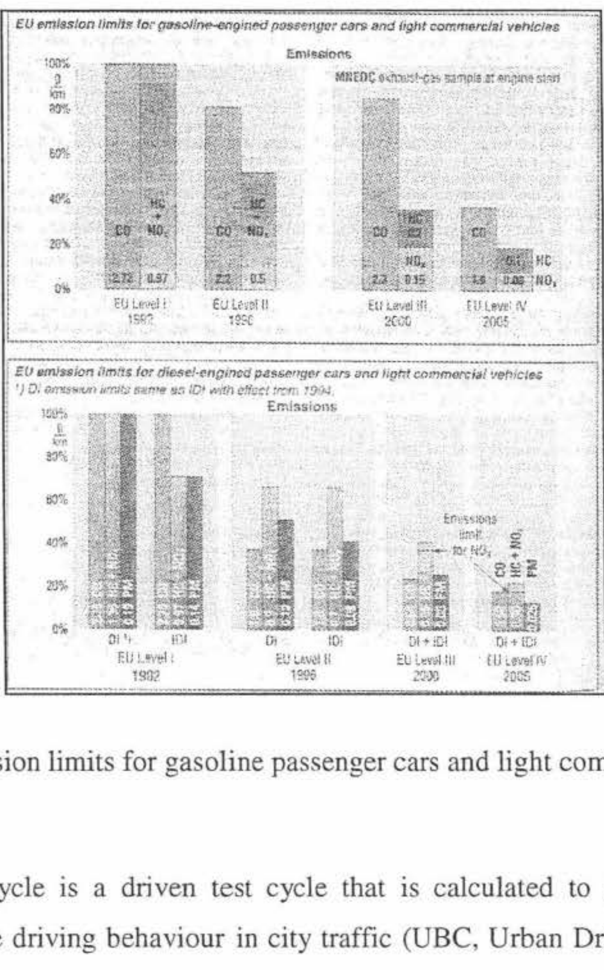


Figure 2.9: EU emission limits for gasoline passenger cars and light commercial vehicles [14]

The EU/ECE test cycle is a driven test cycle that is calculated to provide a reasonable approximation of the driving behaviour in city traffic (UBC, Urban Driving Cycle). In 1993 the cycle was supplemented by a highway section, and since EU 3 the lead-in time of 40 seconds has been omitted, thereby including the cold start condition. The MNEDC test is carried out as follows:

- Conditioning – the vehicle must be at a specific temperature for at least 6 hours with the engine switched off, 20-30°C

- City driving cycle – Comprises of four identical sections each lasting 195 seconds and driven without a break, rout distance 4.052 km
- Highway driving cycle – Directly after city driving cycle, lasts for 400 seconds and covers a distance of 6.955 km
- Evaluation

Emission Standards for Diesel and Gas Engines, ETC Test, g/kWh							
Tier	**Date & Category**	**Test Cycle**	**CO**	**NMHC**	**CH_4[a]**	**NOx**	**PM[b]**
Euro III	*1999.10, EEVs only*	*ETC*	*3.0*	*0.40*	*0.65*	*2.0*	*0.02*
	2000.10	ETC	5.45	0.78	1.6	5.0	0.16 0.21[c]
Euro IV	2005.10		4.0	0.55	1.1	3.5	0.03
Euro V	2008.10		4.0	0.55	1.1	2.0	0.03

a - for natural gas engines only

b - not applicable for gas fueled engines at the year 2000 and 2005 stages

c - for engines of less than 0.75 dm^3 swept volume per cylinder and a rated power speed of more than 3000 min^{-1}

Table 2.5: EU emission limits [15]

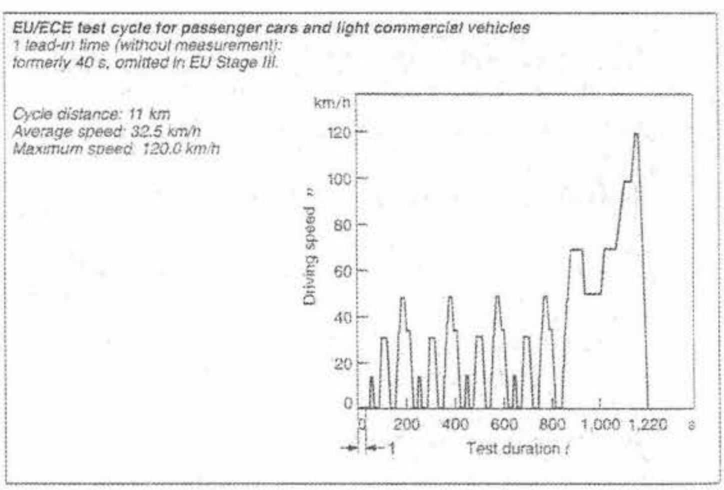


Figure 2.10: EU/ECE test cycle for passenger cars and light commercial vehicles [14]

2.4.2.4 Japanese Legislation

Japanese legislation specifies emission limits for the following pollutants:

- Carbon monoxide (CO)
- Hydrocarbons (HC)
- Nitrogen Oxides (NO_x)
- Particulates, only for diesel vehicles
- Smoke, only for diesel vehicles

Besides passenger cars, Japan has vehicle categories for LDV (Light-Duty Vehicle) up to 1.7 tonne and MDV (Medium-Duty Vehicle) up to 2.5 tonne permissible weight. Exhaust emissions are measured in the 10.15-mode test and the 11-mode test for gasoline engines. Two test cycles are combined to provide a complete test for gasoline vehicles. After a cold start the 11-mode cycle is run four times, with evaluation of all four cycles. The 10.15-mode test is run once as a hot test. This test cycle simulates the characteristic drivability in Tokyo and has been extended to include a high-speed component [14]. Preconditioning for the hot test also includes the stipulated idle emissions test and is conducted as follows: After warming up the vehicle at 60km/h for approximately 15 min the HC, CO and CO2 concentrations in the exhaust pipe are measured. After a second warm-up period of 5 min at 60 km/h, the hot test is then started [14].

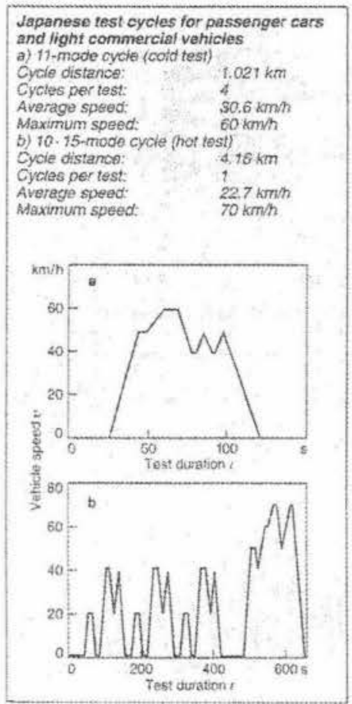


Figure 2.11: Japanese test cycles for passenger cars [14]

2.4.3 Heavy duty vehicles

2.4.3.1 U.S legislation

Emissions certification of heavy-duty applications is done by running the engines on engine dynamometers. The EPA legislation defines heavy commercial vehicles as vehicles that have a total weight over 8500 lbs (3,850 kg). With Tier 2 being introduced in 2004, vehicles between the weights of 8500 lbs and 10,000 lbs used for passenger transport are categorised as light-duty trucks and are therefore tested on a chassis dynamometer. Californian and EPA legislations are almost identical, except for an additional program for city buses required by Californian legislation.

The US legislation specifies limits on the following pollutants for diesel engines:

- Hydrocarbons (HC)
- Some NMHC

- Carbon Monoxide (CO)
- Nitrogen Oxides (NO_x)
- Particulates
- Exhaust-gas capacity

These limits are based on engine performance and are expressed as g/KW.h, and are measured using the HDTC (Heavy-Duty Transient Cycle). Since Tier 2 came into effect in 2004 HC and NO_x have been combined to form a composite limit, CO and particulate emissions have remained the same as that from 1998. Further drastic reductions of limits will take effect starting with the 2007 model year. The new particulate emission limits are lower than pervious limits by a factor of 10. NO_x and NMHC emission limits will be phased-in between model years 2007 and 2010 [14].

The U.S HD Transient Test is based on highway operation under real world conditions, and has significantly more idle times than the ESC test. This is a complicated test that covers numerous load points, as can be seen in the figure below.

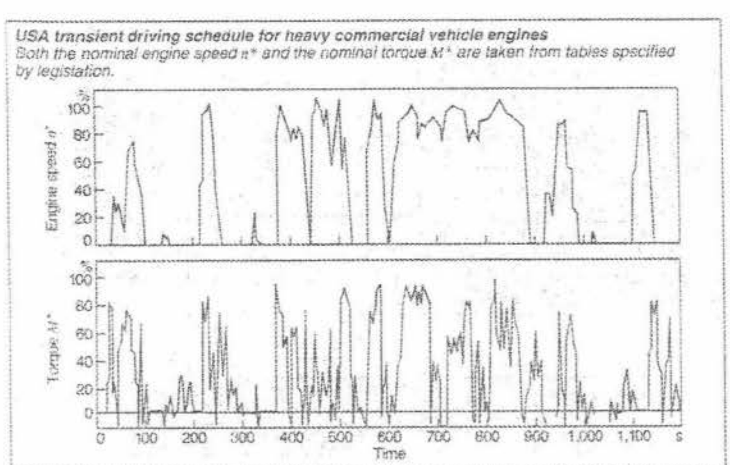


Figure 2.12: U.S HD transient test [14]

Starting from model year 2007 U.S emissions limits must also be met by the European 13-stage test (ESC).

2.4.3.2 EU legislation

European legislation states that vehicles with the total weight over 3,500 kg and designed to transport more than 9 people are considered as heavy commercial vehicles. Since the introduction of Euro 4, three tests have been put in place to measure exhaust emissions, these are:

- European Steady Cycle (ESC)
- European Load Response Test (ELR)
- European Transient Cycle (ETC)

The ESC test is used to measure the exhaust emissions, this test requires measurement at three engine speeds plus idle. The operating points are determined from the engine full-load curve. The test sequence stipulates a series of thirteen different steady state operating modes. Factors are used to weight the measured gaseous emissions and particulates as well as the power output at each operating point [14]. The ESC allows certification staff to choose three additional load points to measure NO_x. These additional load points may only differ by a small amount from those at the adjacent operating points.

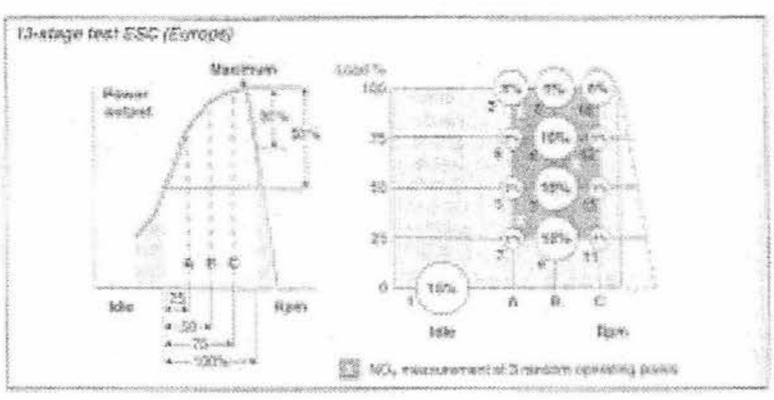


Figure 2.13: ESC test cycle [14]

The ETC is the transient test, similar to that of the U.S HD Transient Test, is used for measuring the gaseous exhaust emissions and particulates. The test cycle is derived from real on-road trips and is divided into three sections:

- City section
- Highway section
- Expressway section

The test lasts for 30 min.

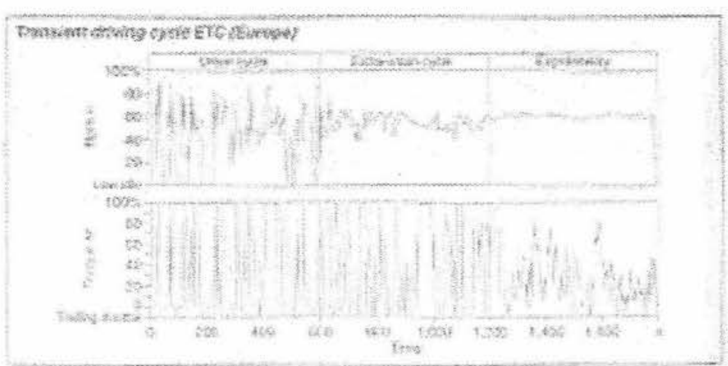


Figure 2.14: ETS cycle test [14]

The ELR test is used for the measurement of exhaust gas opacity.

2.4.3.4 Japanese Legislation

All vehicles that have a permissible weight over 2,500 kg and are capable of transporting more than ten people are considered as heavy commercial vehicles. The Japanese legislations stipulate limits for HC, NO_x, CO and particulates. The emissions are measured using a stationary Japanese 13-stage test (hot test) and the exhaust gas opacity is subject to the Japanese smoke test. The Japanese 13-stage test is similar to that of the ESC with the exception that the operating points, their order and their relative weighting differing from that of the ESC. The 13-stage Japanese test focuses on lower engine speeds and loads due to the high traffic density in Japan.

2.4.4 Unregulated Emissions

Exhaust gases from motor vehicles contain a high number of different chemical compounds and emissions legislations only regulate percentage of the total emissions.

A FID (Flame ionisation detector) is used to measure the amount of hydrocarbons emitted from the exhaust. These hydrocarbons include less harmful compounds such as methane, ethane and propane as well as very harmful compounds such as benzene and polyaromatic hydrocarbons. It is therefore evident that to minimize adverse heath effects and reactivity of exhaust emissions speciation of hydrocarbons is needed.

The measurement of nitrogen oxides is done using a CLD (chemiluminescent detector) instrument, this instrument only measures nitrogen oxide and dioxide, it does not however measure other nitrogen containing compounds such a nitrous oxide and ammonia.The measurement of particulate matter does not contain any further information as to the chemical composition of the particulate matter nor the particulate size distribution.

Many unregulated exhaust components are considered harmful to human health. Some of the most important air toxics are:

- Benzene
- 1,3-butadiene
- formaldehyde
- acetaldehyde
- polycyclic organic matter associated with particulates

2.5 OBD (On-Board Diagnostics)

In order for vehicles to maintain the exhaust emission limits stipulated in the various legislations all components related to fuelling and exhaust after treatment have to be monitored to detect any malfunctions. The OBD ties together the different subsystems of the engine [2]:

- Fuel system
- Lambda control system
- Ignition system (misfire)
- Catalyst
- Particulate trap
- EGR system (functionality)
- Engine management

In 1988 the first stage of CARB legislation came into force in California with OBD I. OBD I required monitoring of the following:

- Exhaust gas related electrical components, e.g. short circuits and line interrupts, and then store them in the ECU
- A fault indicator lamp (MIL- Malfunction Indication Light), mounted in the dash to detect faults to the driver
- An On-board means, Blink code, to provide readout of which components that had malfunctioned

In 1994 OBD II was introduced in California, in addition to the requirements in OBD I, OBD II now monitors the system functionality, e.g. checks that sensor output are plausible. OBD II requires that all exhaust-gas systems and components should be monitored for a malfunction in one of these systems or components causes a significant increase in noxious exhaust gas emissions (OBD emission limits) [14]. The threshold is currently set so that the MIL will be illuminated if the emissions exceed 1.5 times the applicable emission standard. All components and systems must be inspected at least once during an exhaust test cycle, eg FTP 75. From model year 2005 onwards, a specific monitoring frequency (In-use Monitor Performance Ratio) is required for many monitoring functions in normal vehicle operation [14].

OBD adapted to European conditions is known as European On-Board Diagnosis (EOBD) which is based on the EPA OBD and has been in force since 2000.

Typically there can be up to 10 monitors operating in OBD II. These are:

- Catalyst monitor
- Fuel system monitor
- Misfire monitor
- Evaporative system monitor
- Oxygen sensor monitor
- EGR monitor
- Secondary Air system monitor
- Comprehensive component monitor
- Thermostat monitor
- PCV monitor

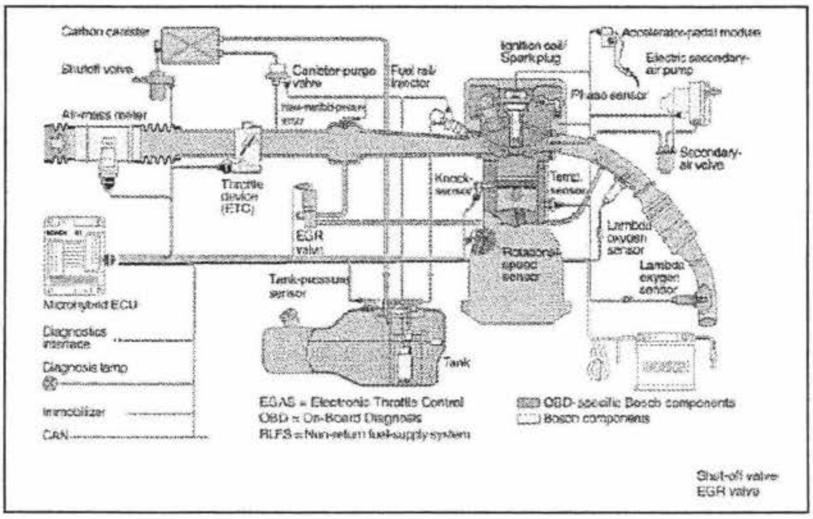


Figure 2.15: Bosch Motronic ME7 engine management system with OBD [2]

One of the major issues with OBD systems is that they have been designed to operate with gasoline as a fuel. Therefore if an ODB II equipped vehicle is operated on an alternative fuel such as natural gas, it is then likely that the monitor will sense that a fault has occurred and illuminated the MIL and set a false fault code even though the vehicle may be operating properly. Up to 2004 it was possible for manufacturers of alternative fuel system to request for certain specific monitoring requirements to be disabled that would not be reliable when the vehicle is operated on natural gas. However it is now the responsibility of alternative fuel manufacturers to ensure that the alternative fuel systems fitted to model year 2005 and above vehicles be fully functional OBD II systems. As stated by the EPA, "By holding aftermarket alternative fuels converters to the same regulatory requirements that OEM's meet, EPA is maintaining compliance equity among manufacturers", they also state that "Having a fully functional OBD system provides the EPA with some assurance that alternative fuel converted vehicles have durable emission components and are complying in use" [16].

2.6 Exhaust gas after treatment

To meet the new exhaust emission legislations the exhaust gases require an after treatment process to reduce the legislated compounds. Both lean burn and stoichiometric engines require this to reduce emissions. The following reaction equations describe the basic function of an automotive catalytic converter:

Oxidation of CO and HC in CO_2 and H_2O

$$C_yH_n + \left(1+\frac{n}{4}\right)O_2 \rightarrow yCO_2 + \frac{n}{2}H_2O \qquad [2.12]$$

$$CO + \frac{1}{2}O_2 \rightarrow CO_2 \qquad [2.13]$$

$$CO + H_2O \rightarrow CO_2 + H_2 \qquad [2.14]$$

Reduction of NO/NO_2 into N_2

$$NO(orNO_2) + CO \rightarrow \frac{1}{2}N_2 + CO_2 \qquad [2.15]$$

$$NO(orNO_2) + H_2 \rightarrow \frac{1}{2}N_2 + H_2O \qquad [2.16]$$

$$\left(2+\frac{n}{2}\right)NO(orNO_2) + C_yH_n \rightarrow \left(1+\frac{n}{4}\right)N_2 + yCO_2 + \frac{n}{2}H_2O \qquad [2.17]$$

[17]

Precious metals such as platinum (Pt), palladium (Pd) and rhodium (Rh) are used to catalyse these reactions. These precious metals are dispersed on a substrate oxide with a large surface to attain the highest possible efficiency, these substrate materials are typically inorganic materials such as Al_2O_3, SiO_2 or TiO_2. The catalytic substrate is produced in an aqueous solution with a solid content of 30-50% [14]. This solution is then coated onto either a metal or ceramic honeycomb-shaped monolith. This honeycomb-shaped monolith allows for the largest possible surface area for the catalytic reaction in a small space.

Stoichiometric engines use a three-way catalyst (TWC) which is able to reduce NOx and oxidise CO and HC. A TWC utilises CO and HC to reduce NOx, as illustrated in the reaction equations above, it then uses the discharged oxygen to oxidise both CO and HC. To maintain these oxidation reactions and reduction equations it is important to maintain the optimum air/fuel ratio, this is at stoichiometric ($\lambda = 1$). On a stoichiometric gas engine a TWC is capable of reducing these components buy 90-95%.

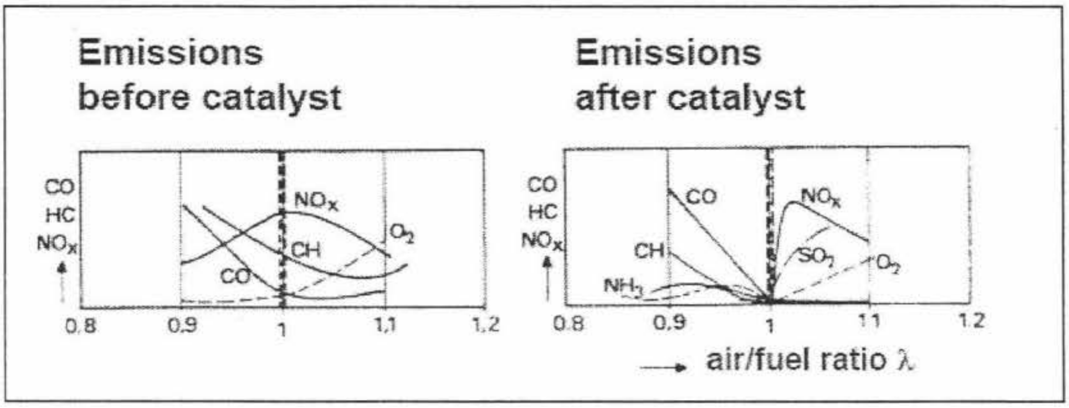


Figure 2.16: The effect of a TWC with respect to air/fuel ratio [2]

Lean burn engines operate with excess oxygen in the exhaust, this makes it substantially more difficult to convert pollutants in the exhaust. HC and CO are preferably converted by the catalytic converter as a result of faster reaction speeds. Consequently the previously converted reaction partners are missing to reduce NO_x. It is therefore required that new technologies are developed to allow for efficient exhaust gas treatment, especially of NO_x, in lean burn applications. The following are the different approaches used to reduce NOx in lean exhaust:

- Direct NO decomposition
- Plasma technologies
- Selective Catalytic Reduction (SCR)
- NOx storage catalytic converters

Catalytic converters that are able to directly break down NO into N_2 and O_2 are the ideal product for exhaust gas after treatment of SI lean burn engines. Although NO decomposition is thermodynamically preferred and the basic chemistry has been revealed in the R&D laboratories, transforming it to a real engine or vehicle operation has been unsuccessful to date [17].

Laboratory prototypes of plasma technologies with heterogeneous catalysts were tested in engine exhausts with varying results. It is presently uncertain whether this technology will become a standard application in SI lean burn engines.

Selective catalytic reduction is the term given to catalysts that are specifically tailored to NO_x conversion in lean conditions. With the addition of suitable reducing agent N_2, CO_2 and H_2O are the end products. SCR catalysts may be classified as either active or passive catalysts. Active SCR catalysts require the introduction of reducing agents; typical reducing agents are ammonia or urea. Passive SCR catalyst do not require the addition of reducing agents, instead passive catalysts uses components that are exclusively in the exhaust for reduction of NO_x. An example of a passive SCR converter is an iridium catalytic converter with a downstream three-way catalytic converter.

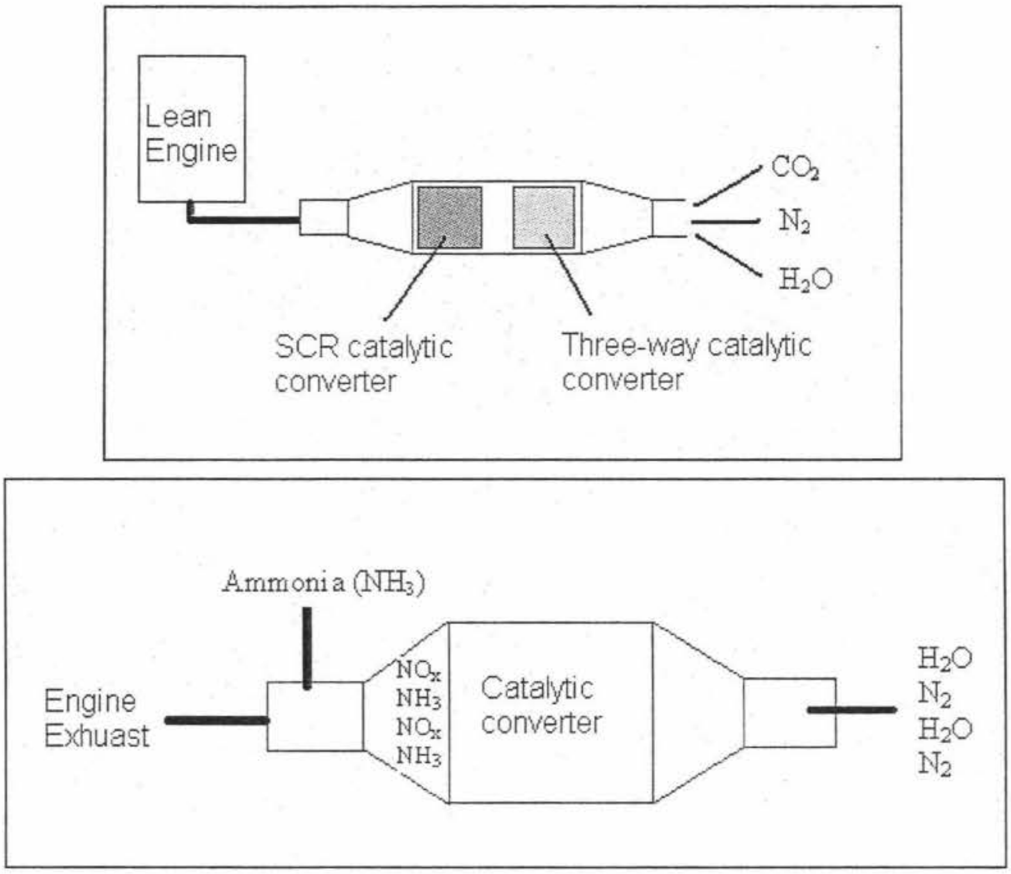


Figure 2.17: Passive and active SCR converter

The most promising treatment for reducing NO_x is the use of a NO_x storage catalytic converter, known as NO_x absorbers or lean NO_x traps (LNT). The operation of converting NO_x to N_2 for this type of catalytic converter can be described using four basic steps:

During lean operation, the NO in the exhaust oxidises at the precious metal in the catalytic converter by reacting with oxygen and forms NO_2:

$$NO + \frac{1}{2}O_2 \rightarrow NO_2 \qquad [2.18]$$

The NO_2 then reacts with the metal oxides deposited in the catalytic converter that are used as storage materials, with the formation of a corresponding storage material nitrate:

$$NO_2 + MeO \rightarrow Me - NO_3 \qquad [2.19]$$

Since this reaction is not catalytic, but rather stoichiometric, the storage material is consumed. As the amount of stored NO_2 increases, the effectiveness of the nitrate formation decreases [17]. Finally a stage of saturation is reached, it is therefore required that the storage material be periodically regenerated to maintain its efficiency. To do this the engine is briefly run at stoichiometric (rich) operation. Under stoiciometric operation the temperature stability of the nitrate is lower than when operating in lean condition, thereby allowing the nitrate to decompose into NO and MeO:

$$Me-NO_3 \rightarrow MeO+NO+\frac{1}{2}O_2 \qquad [2.20]$$

The reducing agents HC and CO present under the rich operation are then used to convert NO into N_2.

$$NO+HC/CO \rightarrow \frac{1}{2}N_2+H_2O/CO_2 \qquad [2.21]$$

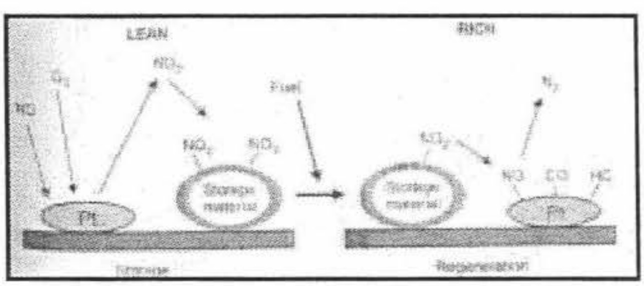


Figure 2.18: Model example of NO_x storage and regeneration [17]

Catalysts are more effective for heavier hydrocarbons than for methane. Due to methane's particularly stable molecule, the catalyst requires a high level of loading and needs to be operated at high temperatures with a low space velocity to be efficient.

For lean burn engines methane reduction of oxidation catalysts are as low as 30-50% [2]. Deactivation of these catalysts is caused by sulphur based odorants such as THT (tetrahydritiofen) present at levels of only 10-15 mg/m^3. It is therefore evident that the composition of natural gas must be strictly monitored to ensure the conformance to emissions regulations.

2.7 Fuel systems

A natural gas fuel system consists of the following components:

- Storage cylinder
- Pressure regulators
- Metering and mixing device

The storage cylinder arrangement is dependant on the vehicle configuration, dictating the size and number of cylinders that can be fitted to the vehicle. The following table illustrates the typical storage capacity at 200 bar for a range of cylinder sizes:

Water capacity, litres	Gas capacity, m^3	Gasoline litres equivalent
50	12.5	15.6
60	15	18.7
70	17.5	21.8
100	25	31.2
120	30	37.4

Table 2.6: Natural gas storage tank equivalence to that of petrol [11]

As can be seen in the table above the equivalent energy storage volume of natural gas is approximately 4 times that of gasoline at 200 bar. Another measure useful for calculating storage quantities is that each litre cylinder capacity holds 180 grams of natural gas [11].

There are two methods to meter natural gas into an engine, i.e. mechanical or electrical. Depending on the fuel system, the gas engines can be divided into three categories:

- 1st generation: Mechanical fuel metering, no feedback
- 2nd generation: Mechanical metering, with closed-loop electronic lambda control
- 3rd generation: Fuel injection, closed-loop control
- 4th generation: fuel injection, OBD capabilities

The first generation system used a venturi or similar mechanical device to meter fuel into the engine. This system was fully mechanically controlled and had no lambda feedback. Therefore during varying running conditions such as engine speed, engine load and tank pressure and varying environmental conditions such as temperature and barometric pressure, the air/fuel ratio would not be consistently at stoichiometric. This system is unable to meet current American and European regulations.

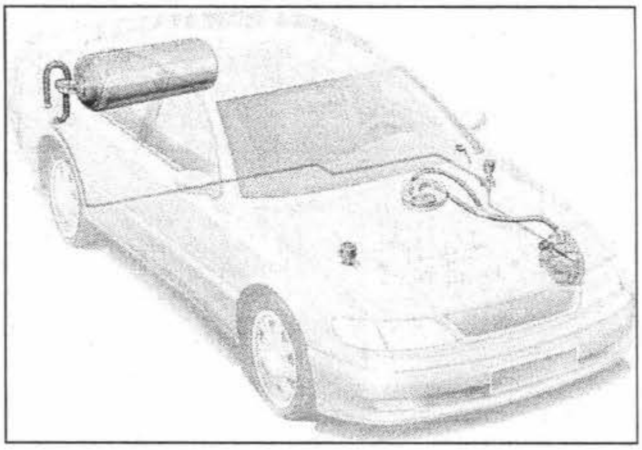


Figure 2.19: First generation CNG fuel system [18]

Second generation fuel systems include lambda feedback to precisely control the air/fuel ratio to ensure the effectiveness of the TWC (three way catalyst). Sophisticated techniques and software have been developed to maintain the stoichiometric air/fuel ratio and result in vehicles able to achieve SULEV emission standards. The use of stepper motor valves or modulation of gas supply pressure in conjunction with wideband oxygen sensors has also allowed lean burn operation of engines. These types of systems have limitations in terms of speed, accuracy and emissions stability.

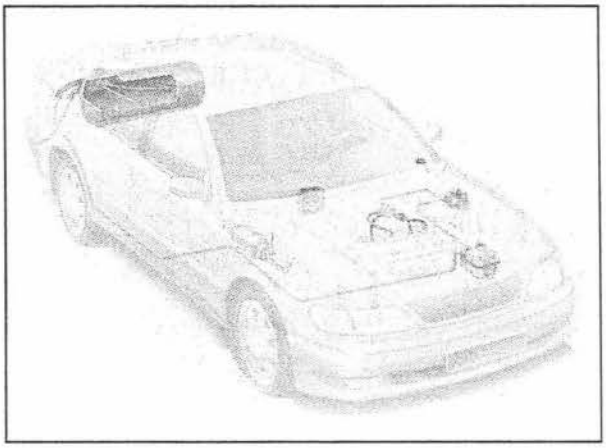


Figure 2.20: Second generation CNG fuel system [18]

Third generation systems achieve good dynamic emissions performance. Due to the American and European test cycles requiring emissions testing of transient performance of engines, precise control of gas is required. This precision gas control is achieved by single or multi-point injection with sequential injection being the ultimate. The valves are controlled by a pulse width modulated signal that is received from a electronic control unit (ECU). The ECU interprets signals from the oxygen sensor, temperature sensor, manifold pressure sensor, etc and determines the required fuel for the condition.

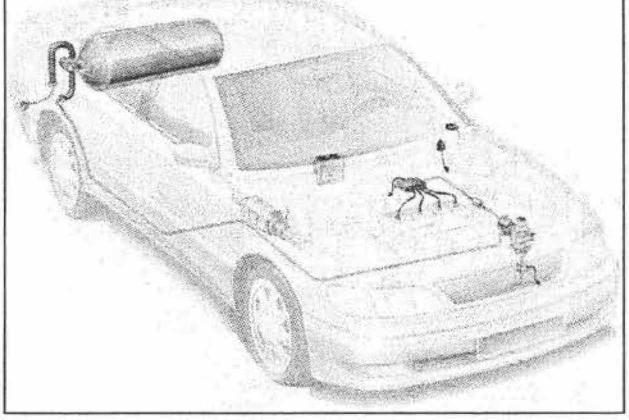


Figure 2.21: Third generation CNG fuel system [18]

2.8 Stoichiometric and lean burn operation of gas engines

2.8.1 Engine operation overview

The major goals in the development of internal combustion engines are high efficiency, in terms of low fuel consumption, low emissions, and high power output.

The auto-ignition temperature of air/fuel mixture dictates the knock resistance, therefore a high auto-ignition temperature will result in a higher knock resistance. The auto-ignition temeperature of a give air/fuel mixture is a function of octane number (ON), mixture pressure and lambda. The equation is as follows:

$$T_{ai} = 300 + \left[\frac{60}{P_{mi} + 0.1} - 6P_{mi} + 5ON\right]\left[1 + 7(\lambda - 1.08)^2\right] \qquad [2.22]$$

This equation is parabolic and is a minimum at λ = 1.0-1.1. Figure 22 below illustrates the effect of ON on the auto-ignition temperature of a air/fuel ratio.

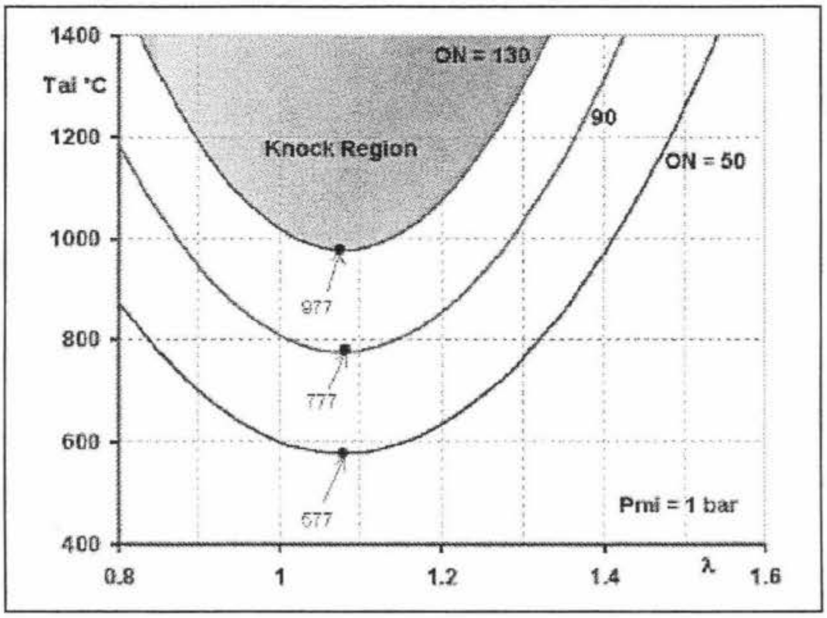


Figure 2.22: Effect of ON on auto-ignition temperature [19]

In the figure above the mixture pressure is set at 1 bar, this however does not represent the final pressure after compression, if a value of 81 bar is substituted into the equation the auto-ignition temperature for an air/fuel mixture with $\lambda = 1.0$ and ON of 130 is reduced to 636°C.

Equation 2.22 and figure 2.22 illustrates that leaner mixtures have greater knock resistance to that of stoichiometric mixtures and are therefore better suited to turbo charged applications. This then points out the principle in lean burn engine operation. Lean burn operation attempts to introduce the maximum amount of air to create the leanest possible mixture that can still be successfully ignited or where misfire does not occur.

Misfire occurs when flame propagation stops, this will occur when the temperature falls below a certain value. This temperature can be calculated using the following equation.

$$T_{mf} = \left[\frac{60}{P_{mi} + 0.1} - 6 \cdot P_{mi} + 350 \right] \cdot 5 \cdot (1 - \lambda)^2 \qquad [2.23]$$

Using the equation for misfire temperature and the following equation for estimating the mixture temperature we are able to illustrate the effects of ignition timing with respect misfire and knock.

$$T_{mi} = T_{AC} \cdot (CR)^{n-1} \cdot f(CA) \qquad [2.24]$$

The mixture temperature is a function of inlet temperature (T_{AC}), mixture pressure and crank angle (CA).Using the three equations 2.22, 2.23 and 2.24 we are able to create the following figure.

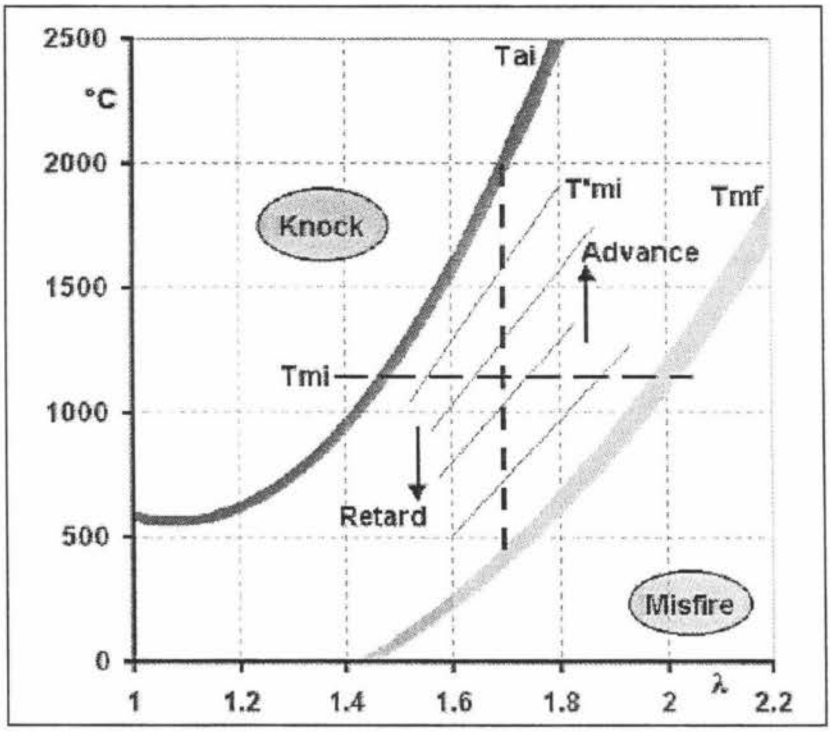


Figure 2.23: Timing effect on combustion [19]

From the figure above we can see that advancing the timing too far will result in the end zone temperature increasing above the auto-ignition temperature and thus result in knock. The desired knock timing margin is within 2 to 3°CA of the knock zone to ensure maximum performance is achieved.

To determine the optimum point of ignition we require the flame propagation velocity, to determine this we use the flame velocity of the gas which can be determined as follows.

$$C_{fuel} = \frac{44}{ON} = \frac{44}{130} = 0.3385 m/s \qquad [2.25]$$

The mean flame propagation velocity (FPV) can be estimated using the heat release rate (HRR). The FPV is affected by the size of the combustion chamber and the ratio of misfire to knock margin.

$$FPV = \frac{\Delta T_{kn}}{\Delta T_{mf}} \cdot C_{fuel} \cdot 0.026 \cdot B^{1.5} \qquad [2.26]$$

Where: B = bore diameter of combustion chamber

$\Delta T_{kn} = T_{ai} - T_{mi}$ = Knock margin

$\Delta T_{mf} = T_{mi} - T_{mf}$ = misfire margin

It has been observed that optimum torque is generally obtained in an SI engine when the combustion pressure peak occurs at around 12 to 15 degrees after TDC. This useful rule of thumb would allow some estimate of the optimum spark timing, if the information of the flame nucleus and the speed of the flame were known [20].

The power output of an internal combustion engine is proportional to the mean effective pressure P_{me}, the speed n and the total piston displacement V_H.

$$P_e = p_{me} \cdot n \cdot V_H \cdot \frac{1}{2} \qquad [2.28]$$

$$P_{me} = \rho \cdot \lambda_l \cdot \eta_e \cdot \frac{H_u}{\lambda \cdot L_{min}} \qquad [2.29]$$

Where: P_e = effective power

P_{me} = mean effective pressure

n = speed

V_H = piston displacement

ρ = density after charging

λ_L = volumetric efficiency

η_e = effective efficiency

H_u = net calorific value

λ = excess air factor

L_{min} = minimum excess air factor

[17]

The density of the air depends on the charge pressure and charge air temperature.

$$\rho = \frac{P}{R \cdot T} \qquad [2.30]$$

p = charge pressure

R = gas constant

T = temperature of charge

It is therefore evident that an efficient intercooler system is required when the engine is using forced induction, i.e. turbocharger or supercharger, as this will directly affect the power output of the engine.

A useful relative engine performance indicator is that of brake mean effective pressure (BMEP) that was derived earlier in this section. This may also be done by dividing the work done per cycle by cylinder volume displacement per cycle.

$$BMEP = \frac{T \cdot 2\pi}{V_s \cdot i} \qquad [2.31]$$

T = Torque

V_s = swept volume

i = working cycle per revolution (0.5 for 4 stroke and 1 for two stroke)

Another good indication of how efficient the engine is operating is that of brake specific fuel consumption (BSFC).

$$BSFC = \frac{M_{fuel}.N}{P_{brake}} \qquad [2.32]$$

M_{fuel} = mass flow rate of fuel in k/m^3

N = RPM

P_{brake} = brake power

$$P_{brake} = 2\pi NT \qquad [2.33]$$

N = RPS, revolutions per second

T = Torque, Nm

Below is a table indicating examples of the effective mean pressure modern engines.

Engine Type	Effective mean pressure (bar)
	Up to
Motorcycle engines	12
Racing engines (Formula 1)	16
Car SI engines (without turbocharger)	13
Car SI engines (with turbocharger)	17
Truck diesel engines (with turbo)	22
Car diesel engines (with turbo)	20
Larger high-speed diesel engines	30
Medium-speed diesel engines	25
Crosshead engines (two stroke diesel)	15

Table 2.7: Effective mean pressure of modern engines [17]

Now that we have a reasonable understanding of the requirements of a SI engine to be efficient in relation to emission, fuel consumption and power output, and we are also able to compare various engine types irrespective of their configuration. The next part of this section will detail the advantages and disadvantages of stoichiometric and lean burn operation of natural gas engines.

Figure 2.24 illustrates the relationship of emissions to air/fuel ratio. All gas engines for automotive applications (cylinder diameter less than 150 mm) have an open-type combustion chamber [2]. Prechamber arrangements are used on larger engines only. Spark ignition engines can be divided into three basic categories dependant on the relative air/fuel ratio:

Stoichiometric	($\lambda = 1$)
Lean-burn	($\lambda \geq 1.5$)
Engines optimised for low fuel consumption	($1.1<\lambda<1.3$)

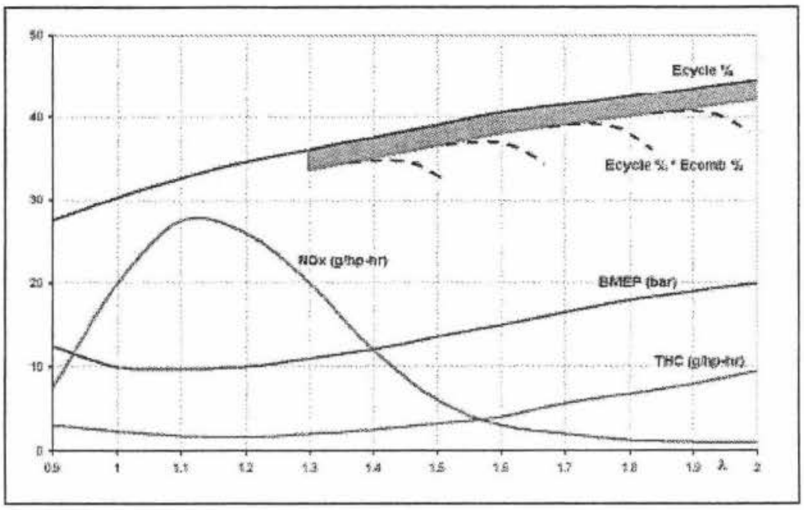


Figure 2.24: Effect of lambda on natural gas engine performance [19]

2.8.2 Stoichiometric engines

Stoichiometric engines are equipped with a closed-loop control system and three way catalysts (TWC) to maintain low emissions. These engines operate with $\lambda = 1$ and slightly richer.

Benefits of stoichiometric concept are:

- the possibility of achieving extremely low exhaust emissions
- stable engine operation
- moderate requirements on the ignition system
- high BMEP for naturally aspirated engine

The drawbacks are:

- a closed loop fuel system and a three-way catalyst are needed for emission control
- The emissions are highly dependent on the reliability of the oxygen sensor and the control system

- high thermal loadings compared to diesel or lean-burn operation
- restricted possibilities for turbocharging
- in some cases a fuel consumption penalty compared to lean-burn operation
- not suitable for fuels with low knock resistance

[2]

2.8.3 Lean-burn engines

Lean burn engines operate with excess air; this allows one to control the exhaust emissions by controlling the combustion temperature, lower combustion temperatures result in lower NO_x production. Lean burn engines run typically at λ 1.5 -1.6, this enables lean-burn natural gas engines to produce roughly 1/4 to 1/3 the amount of NO_x emission compare to that of an equivalent diesel engine. The relationship between λ, exhaust gas NO_x concentrations and thermal efficiency of a gas engine is illustrated in the figure below.

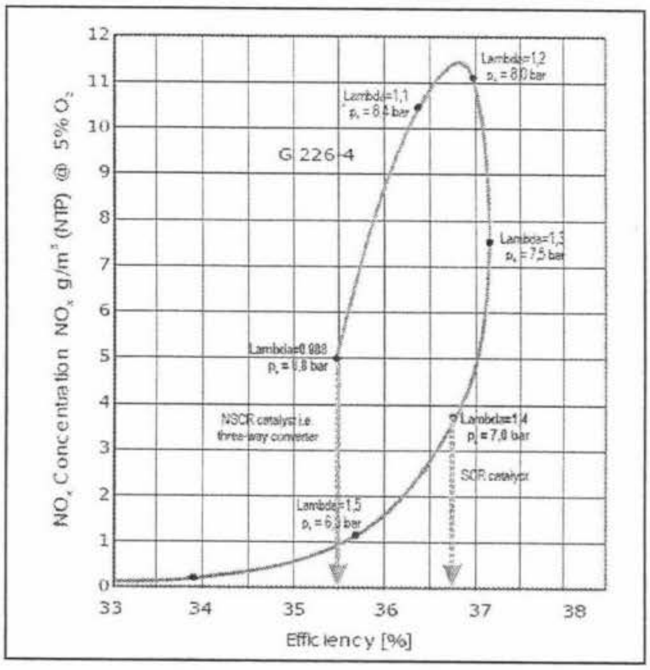


Figure 2.25:Trade off of NO_x emissions and engine efficiency [2]

As can be seen in the figure above the highest NO_x production is near $\lambda = 1.2$. At $\lambda = 1.5$ it can be seen that NO_x emissions have dropped dramatically, however so has the efficiency of the engine, it is now close to that of a stoichiometric engine. NO_x emissions can also be plotted with respect to ignition timing and air/fuel ratio, the figure below illustrates the relation between ignition timing and air/fuel in the production of NO_x.

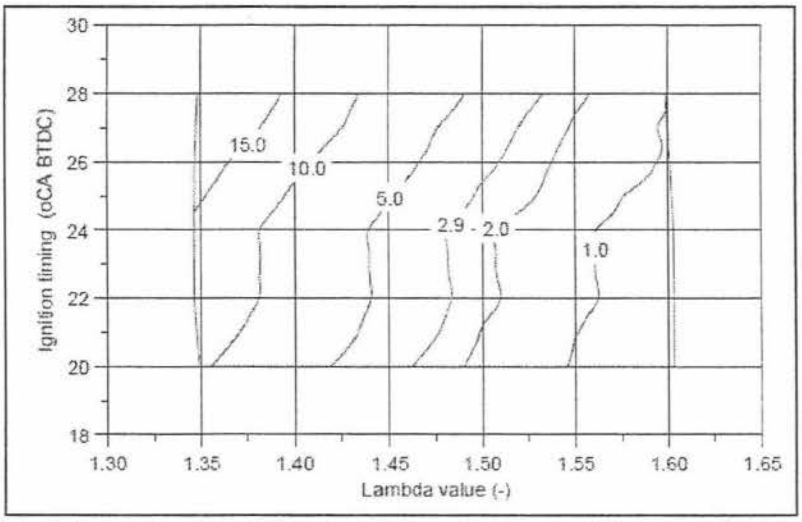


Figure 2.26: NO_x emissions as a function of lambda and ignition timing. [2]

As can be seen in the figure above, as the air/fuel ratio is leaned out the ignition timing has to be advanced to maintain low emissions of NO_x.

Benefits of the lean-burn concept are:

- Moderate Exhaust emissions
- NO_x formation controlled already during combustion
- high power output if turbocharging is used
- thermal loading close to diesel

The drawbacks are:

- turbocharging necessary to obtain sufficient power output
- transient engine response
- high requirements on the ignition system
- high cycle-cycle variations
- high methane emissions with natural gas (unburned methane is hard to oxidise)
- oxidation catalyst needed for CO and HC control

[2]

Engines optimised for low fuel consumption are tuned with an air/fuel ratio of $\lambda = 1.1\text{-}1.3$, at this air/fuel ratio high combustion temperatures are experienced. These high combustion temperatures cause the rapid increase in NO_x exhaust emissions. Also at this air/fuel ratio a TWC is not longer effective in reducing overall emission, it is therefore evident that with present and future emissions legislations this type of engine will become obsolete and be replaced by lean-burn engines.

2.9 Inductive discharge versus capacitive discharge ignition

2.9.1 Ignition overview

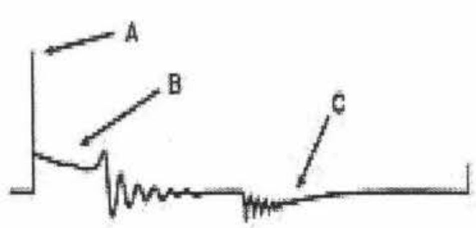
On a spark ignition engine, combustion is initialised by producing an electric spark between the electrodes of a spark plug in the combustion chamber. This spark is responsible for igniting the compressed air/fuel mixture at the right time. Consistent and reliable ignition under all conditions is essential to ensure fault free engine operation. Misfiring leads to:

- Combustion misses
- Damage and destruction of catalytic converter

- Poorer emission values
- High fuel consumption
- Lower engine performance

[14]

To create the spark between the electrodes of the spark plug the necessary ignition voltage has to be exceeded by the ignition system. The ignition voltage is dependant on the electrode gap, the density of the air/fuel ratio and the cylinder pressure at the moment of ignition, typical voltages are between 8000 and 14000 volts. The voltage at the spark plug electrode ionises the air in the plug gap, creating an electrical path across the gap. After flashover the voltage requirement is reduced and the current flows across the gap, this is known as the firing voltage. The firing voltage depends on the length of spark plasma (electrode gap and excursion by the air/fuel mixture flow). This current flow across the gap is what starts combustion, the duration of this ignition-spark combustion time is known as spark duration [14].



This shows the classic inductive-spark scope pattern using points. The voltage spike (A) is the high initial voltage required to ionize a current path across the spark-plug gap. The voltage required to maintain this current flow is comparatively low (B) but you can see that the spark duration is relatively long. After the spark breaks down, the voltage trails off back to zero (C).

Figure 2.27: Spark plug voltage characteristic [21]

It is important that the ignition must guarantee the process under all operating conditions. In order for the electric spark to ignite an air/fuel mixture in ideal conditions , an energy of about 0.2 mJ is required for each individual ignition event, provided the mixture composition is static, homogeneous, and stoichiometric. In real engine operation, however, much higher energy levels are required [14]. Lean mixtures or turbocharged engines need higher ignition voltages. At a given energy level the higher the ignition voltage the shorter the spark duration, this is unwanted as longer spark durations generally stabilise combustion and may also compensate inhomogeneous mixtures around the spark plug at the moment of ignition. Depending on the engine requirements the spark energy of ignition systems is in the range of approximately 30 to 100mJ [14].

Now that a basic understanding of the requirements of an ignition system is understood this section will go on to describe the two most common ignition type available today, inductive ignition and capacitive discharge ignition.

2.9.2 Inductive ignition

The ignition system of a coil-ignition system consists of:

- An ignition coild with a primary and secondary winding
- An ignition driver stage to control current in the primary winding (incorporated in the engine control unit or the ignition coil)
- A spark plug, which is connected to the high-voltage connection of the secondary winding

[14]

The figure below illustrates a distributor coil ignition system. Other coil ignition systems use one dedicated coil and ignition circuit per cylinder (static HT distribution system), these have become the usual choice of OEM vehicles.

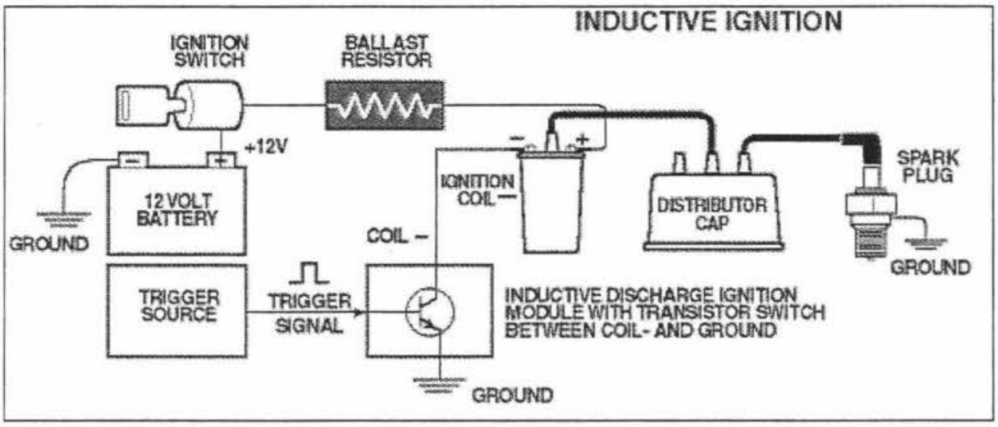


Figure 2.28: Inductive discharge ignition schematic [22]

There is generally approximately 100 times the number of secondary windings as there are primary windings, this is known as the turn ratio. This ratio is what enables the coil to generate such high voltage at the secondary windings. The primary windings are wound outside the large number of secondary windings wrapped around a soft iron core. The positive side of the primary is connected to the battery while the negative side is connected to the distributor points or the switching transistor for electronic ignition, and eventually to ground. The high tension (HT) tower leads the output of the coil to the distributor and eventually to the spark plugs.

The operation of an inductive ignition system is as follows:

1. Before the moment of ignition the ignition driver stage switches a current from the vehicles electrical system through to the primary winding of the coil. While this primary circuit it closed a magnetic field builds up in the primary winding, this is known as dwell time.

2. When ignition is required, the current through the primary winding is interrupted, the magnetic field energy collapses across the secondary winding (induction), creating a high voltage which in turn creates a spark at the spark plug.

Advantages of inductive ignition are that it is a simple, requiring only a few components. Inductive ignition also delivers a relatively long-duration spark. A disadvantage of inductive systems is that energy falls off at high RPM because insufficient time exists to charge the coil. OE Systems are generally good up to about 5,000 RPM. Performance Ignitions systems extend the RPM range upwards of 8,000 RPM [22].

2.9.3 Capacitive discharge ignition

The operating concept behind CDI differs from that of inductive ignition. CDI was developed to overcome the limitations of inductive ignition with respect to high speed, high output multi-cylinder reciprocating IC engines in high performance and competition applications, and for rotary piston engines.

The prominent operation characteristic of the CDI system is that it stores ignition energy in the electric field of a capacitor. The capacitance and charge voltage determine the amount of energy stored [14]. The storage capacitor is normally charged from 450 to 550 volts, the switching device (e.g. IGBT) then discharges this stored energy to the primary winding of the coils. The coil then amplifies this by about 100, due to the turn ratio, and the result is a high voltage (30,000 to 40,000 volts) that is then applied to the spark plug. The figure below illustrates the layout of a CDI system.

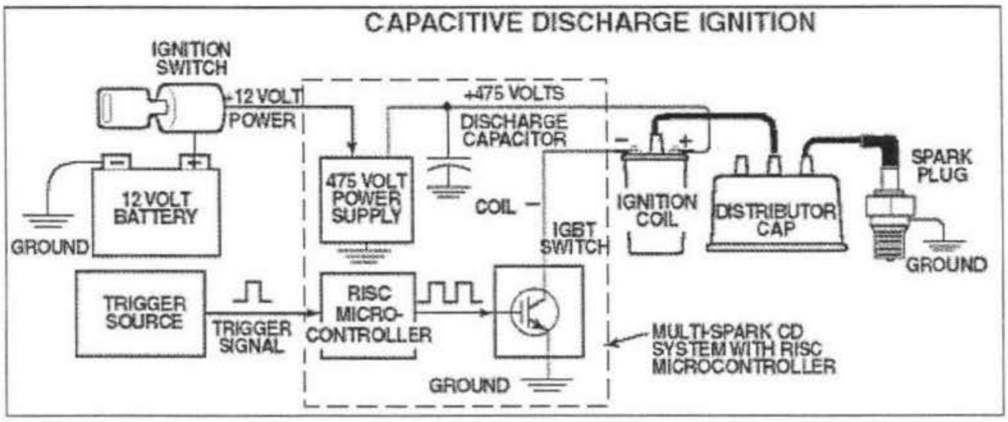


Figure 2.29: Capacitive discharge ignition schematic [23]

The main advantage of the CDI system is that it generally remains impervious to electrical shunts in the high ignition circuit. It is also able to produce strong spark energy up to 10,000

RPM. However the spark duration of 0.1 and 0.3 ms is to brief to ensure that a air/fuel mixture will ignite reliably, to overcome this CDI systems use multiple spark to effectively extend the spark duration. CDI systems are well suited to supercharged or turbocharged engines in the tremendous cylinder pressures in the combustion chamber increase the resistance that the ignition faces when lighting the spark, therefore more voltage is required to initiate the spark.

Which system is better? CDI systems offer the ability to fine tune ignition through to high RPM, however, it does not offer long spark duration such as inductive ignition. It is totally dependant on the application, if the environment requires high voltage spark due to lean mixtures and high cylinder pressures then CDI may be the choice. If the engine is a naturally aspirated and does not operate at high RPMs then inductive ignition will offer reliable ignition.

2.10 Advantages and disadvantages of natural gas

Advantages:

- Due to the simple composition of natural gas the exhaust emissions from complete combustion are far better than that of gasoline or diesel, having nearly zero sulphur and very low particulates, these emissions are mostly caused by the burning of lubricant oil that is found in the combustion chamber.
- Due to the low carbon-to-hydrogen ratio, less carbon dioxide is emitted after combustion per unit energy as compared to other carbon fuels.
- Natural gas easily forms a homogeneous mixture with the incoming air charge thereby eliminating the need for cold start
- Due to the high octane rating natural gas has superior anti-knock properties, thereby allowing for higher compression ratios that then relates to higher thermal efficiency as will be explained later.
- Natural gas has extended flammability limits 5-15%, thereby allowing for lean burn operation

- A lower adiabatic flame temperature, results in reduced NO_x emissions compared to diesel. NO_x can be controlled during lean burn combustion.
- Higher octane number, therefore less chance to detonate.
- Contains non-toxic components
- NG is lighter than air, therefore safer than LPG
- Methane is not a volatile compound
- Mostly heavy duty engines use natural gas, which results in a quieter operation than diesel.
- Lowers the chance of smog

Disadvantages:

- Transportation is complicated, gas stored at 200 bar requires 4 times the storage are of petrol with the same energy content. CNG can only be liquefied at -162degrees C at 2-3 bar pressure, in liquid form the volume is 600 times smaller than the non-compressed gas with the same energy content.
- CNG requires a dedicated catalyst with high loading of active catalytic components to maximise methane oxidation during engine operation.
- Due to the low energy content per unit volume, the driving range in decreased.
- CNG requires specialised refuelling stations.
- The added weight of fuel tank changes the vehicle dynamics and increases fuel consumption.
- Laminar flame speed of natural gas is lower than gasoline, therefore lower burning rate and an ignition modifier has to be used in dual fuel applications.
- Injection of low pressure natural gas into the ports or high pressure into the combustion chamber requires specialise injectors.
- Higher emission of methane, due to the stability of the methane molecule making it harder to oxidise
- Absorbs water vapour that may freeze under certain conditions
- Gives slightly inferior vehicle performance due the reduction in volumetric efficiency

- Possibility of backfire in inlet manifold if throttle body injection is used, thus a homogeneous mixture is present in the inlet manifold that may ignite during valve overlap

Chapter 3: Mechanical operation and development of valve

3.1. Valve Introduction

The Harrison Valve is a successful marriage of existing experience with simple on demand gas valves and modern electronic control units. By combining experience of diaphragm flow control valves with a mathematical analysis of flow patterns it has been possible to optimise the arrangement of diaphragms and various valve designs to give a linear flow rate through the Harrison valve based on an injector reference signal. This principle is common automotive practice throughout all manufacturers and therefore makes the valve capable of being controlled by any ECU (Electronic Control Unit). Because the unit responds to an ECU it exactly mimics liquid fuel flow to the engine that has been programmed by the OE vehicle manufacturer and can therefore be substituted for liquid fuel supply. This means that the valve can be applied universally to any vehicle controlled by an ECU and for any vehicles without an ECU a suitable low cost unit is available again with universal application.

3.1.1. Valve operation

As mentioned above the valve consist of an arrangement of diaphragms and various smaller valves designed to create a system that is linearly proportion to a frequency and pulse width modulated signal. The method in which the valve works will be explained in this section.

The valve has been designed using Bernoulli's equation for compressible flow:

$$\frac{1}{2}v^2 + H_1 + q - w = \frac{1}{2}v_2^2 + H_2 \qquad [3.1]$$

Where: H = enthalpy

q = heat added per unit mass

w = work done per unit mass

After some mathematical substitutions one is able to derive the following equation for mass flow rate per unit time:

$$m = A\rho_0\sqrt{\left[2\left(\frac{\gamma}{\gamma-1}\right)\frac{p_0}{\rho_0}r^{\frac{2}{\gamma}}(1-r^{(\gamma-1)/\gamma})\right]} \qquad [3.2]$$

As can be seen from the equation above if the quantity of $r^{\frac{2}{\gamma}}(1-r^{(\gamma-1)/\gamma})$ is at a maximum then the mass flow rate will be at its maximum. If we differentiate this equation and replace r = p_{throat}/p_0. We find that maximum flow occurs at a pressure differential of 0.528 for air. This means that if the downstream pressure is less than 0.528 times that of the upstream pressure then a condition known as "choked flow" occurs, this is the maximum mass flow rate. This makes it easy to control the flow past an orifice given the upstream pressure and orifice area.

Using this condition of chocked flow, the valve has been designed to maintain this condition throughout its operation.

The valve can be divided into four sections:

1. Second stage regulator with feedback
2. Main Control valve
3. Injector and reference chamber
4. Accelerator Pump and damper

The second stage regulator is fed via a first stage regulator that reduces the cylinder pressure to approximately 150 psi. The purpose of the second stage regulator is to reduce the input pressure to 40psi. One of the commonly found problems with pressure regulators is that when under high demand, i.e. at high flow rates, the regulator tends to reduce the output pressure. With this in mind the second stage regulator was designed with a pressure feedback system that maintains 40 psi in the valve though the full operating range. The valve will only drop below 40psi when the tank pressure is 60psi or less, the valve is then unable to flow the required volume at this low pressure due to the maximum area restriction of the various orifices.

The second stage feedback system works are follows:

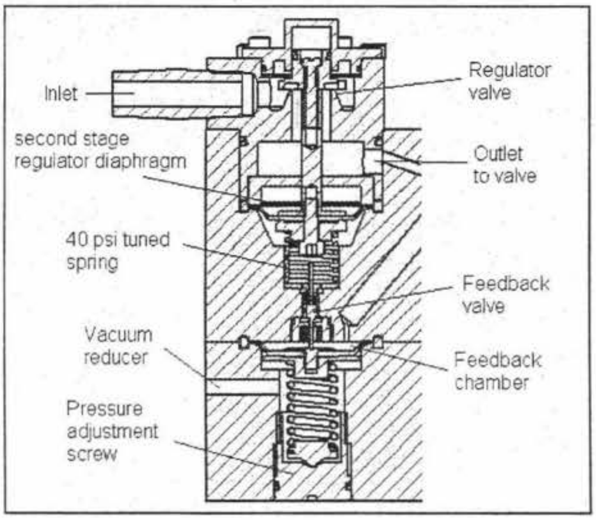


Figure 3.1: Second stage regulator and feedback assembly

The inlet pressure from the first stage regulator is 150psi, this is reduce by the 2nd stage regulator valve. The pressure drop across this orifice is set to result in a pressure of 40 psi passing through to the outlet. This initial pressure setting is achieved by the use of the second stage regulator diaphragm and tuned spring set to produce 40 psi of equivalent force. When valve receives no signal from the injector there is then no feedback required to maintain 40 psi. When the valve is required to flow, it will experience a pressure drop if no adjustment is made to the orifice area of the second stage regulator valve. The feedback chamber experiences this drop in pressure first, thereby causing a pressure differential across the feedback diaphragm. The resulting unbalanced forces require that the system return to equilibrium thus forcing the feedback needle off its seat, thereby creating a small orifice for gas flow. The gas flows past the feedback needle to the underside of the second stage regulator diaphragm thereby adding to the existing force, and thus resulting in lift of the second stage regulator valve and allowing for more flow. This mechanism creates a stable operating pressure under all flow requirement required by the engine.

The main control needle section of the valve has been the area of most development. This is the critical area and much time was spent doing flow analysis and design of needle and seating arrangements. Due to the wide range of fuel flow requirements of an engine, the valve had to be designed to create stable low and high flow rates with linear response. At idle condition the engine is rotating at approximately 780 RPM and requires only approximately 5 l/min of CNG, at 6000 RPM and at wide open throttle (WOT) the engine requires approximately 400 l/min of CNG. This indicates the wide range of metering required by the valve. It is therefore apparent that to accomplish this with one valve will be very difficult. It was therefore decided that the valve have two internal control valves, one to maintain idle and off idle conditions and the other to operate under high demand flow conditions.

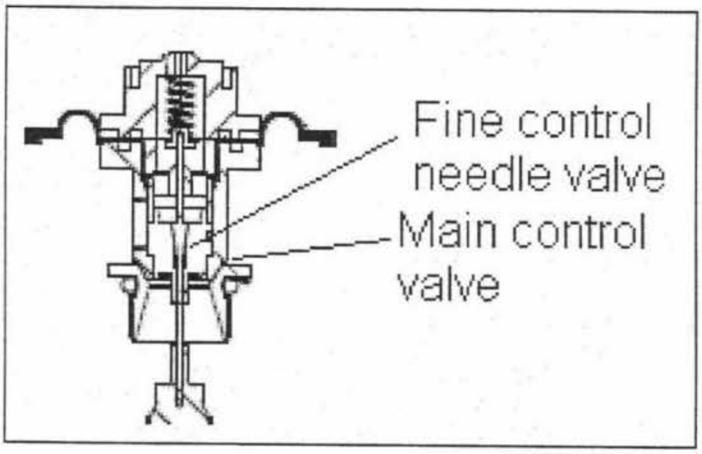


Figure 3.2: Main metering control valves

Above is the diagram of the flow control valves. As can be seen the fine control needle has been placed into the main control valve, to create one unit that is responsible for metering the flow. The fine control needle has been designed to flow a maximum of 45 l/min at 40 psi, at which stage the main control valve then meters any additional flow required.

The injector and reference chamber are the items used to create the reference signal to drive the metering valves. On this valve an standard petrol fuel injector is used to create the reference signal. The injector is rated at flowing 142.6 cc per min. This injector is commonly used on a Rover 1.4 MPI vehicle. A petrol injector is a pulse width and frequency modulated valve, used in electronically fuel injected vehicles. The petrol injector receives a signal from the ECU in the form of a square wave signal. When the signal goes high this charges the coil in the solenoid thereby lifting a needle valve, because there is a positively pressurised fuel within the injector, fluid will flow through the orifice created when the injector needle is lifted by the solenoid. In this way petrol is precisely measured to the engine. It is therefore logical that if this device is then used to create a gas reference signal, that the gas reference signal will then mimic the engine fuel map.

In the Harrison valve the injector is connected to the vehicle ECU and has a 40 psi gas pressure on the inlet. As the injector is activated the gas is passed through the injector into an accumulator called the reference chamber. As gas is passed to the reference chamber a pressure differential is created across the reference diaphragm. This pressure differential causes the diaphragm to lift thereby lifting the fine control needle, in this way the reference signal controls the flow of the valve. To alter the response of the valve to the reference signal, a fine adjustment valve is placed on the reference chamber to adjust the amount of "bleed-off" or pressure bleed in the chamber.

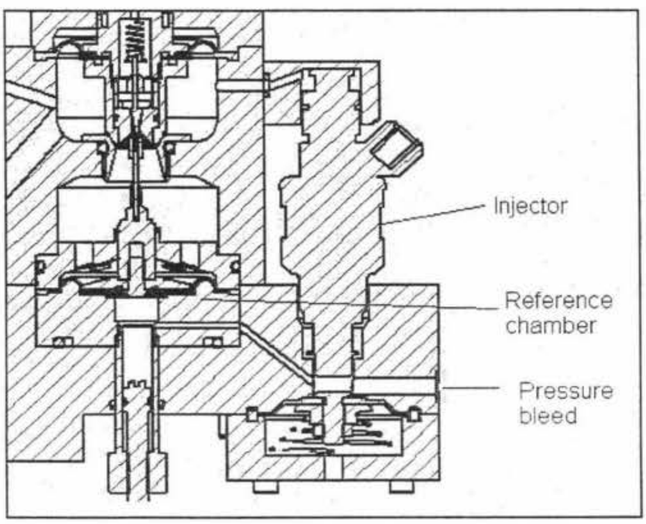


Figure 3.3: Injector and Reference chamber

The last portion of the valve is the accelerator pump and the damper chamber. Due to the operating nature of the injector, the reference chamber experiences pulses when the injector is operating at low frequencies and a high duty cycle. The resulting pulses cause the fine control needle to oscillate and therefore produce erratic flow, to minimise this, a damping chamber is installed to absorb these pressure pulsations. This damper chamber can be seen in the figure above, pictured below the injector.

The accelerator pump is used to give an instantaneous pressure rise in the reference chamber and thus increase flow when there is a sudden acceleration required from the engine. If this was not included the throttle response of the engine would be poor and result in "flat-spots" when attempting to accelerate. Because of the manner in which the valve works, there is a time delay in the response of the valve to the required output, it therefore requires an aid to minimise the response time when sudden acceleration is required. The accelerator pump is referenced to the intake plenum, normally in a vacuum state for naturally aspirated engines, when sudden acceleration occurs the throttle butterfly opens releasing the vacuum. With the sudden release of vacuum in the intake plenum the plunger of the accelerator pump is released and therefore reduces the volume and increases the pressure in the reference chamber. This forces the diaphragm to push the metering valve open and therefore results in an increase in flow. This allows the valve to respond to the injector response quickly.

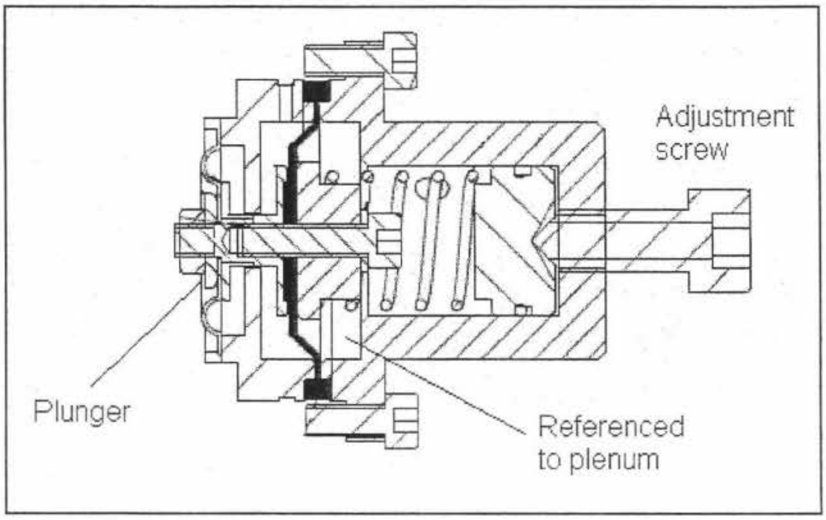


Figure 3.4: Accelerator Pump

Due to the valves ability to follow a OEM fuelling map and due to the fact the one is able to tailor the valve response to suit the driving requirements, the valve is a perfect answer to the need for a configurable, easily maintainable and easily installed product that results in little to no evidence of the use of an alternative fuel.

3.1.2. Fluid dynamics of valve

To aid in the development of the valve, CAD (Computer Aided Design) software was used to model and analyse the flow characteristics of the valve and its components. The package used for this was Solidworks with the Floworks fluid modelling add-on. Floworks allows one to model the fluid flow pattern through or around a 3-D designed object and analyse the turbulence and pressure concentrations within the system. This is a valuable tool saving development time and overall cost of the project.

The area that most development time was aimed at was that of the main control needle valve. When the vehicle is at idle it consumes approximately 5 litres/min for a 2 litre engine. This was found by experimentation on the Subaru test vehicle. To maintain this small amount of flow, regulation of a relatively small orifice had to be maintained. To do so the fine control needle was designed to flow from 0 to 40 litres/minute, this would control the flow for idle and off idle engine operation. The off idle situation is when the vehicle starts to accelerate from a stationary position, in this condition is difficult to control the precise fuelling and therefore the fine control needle was used to control this condition.

The following figures show two different valve designs, in these figures you are able to see the flow and pressure characteristics of each design.

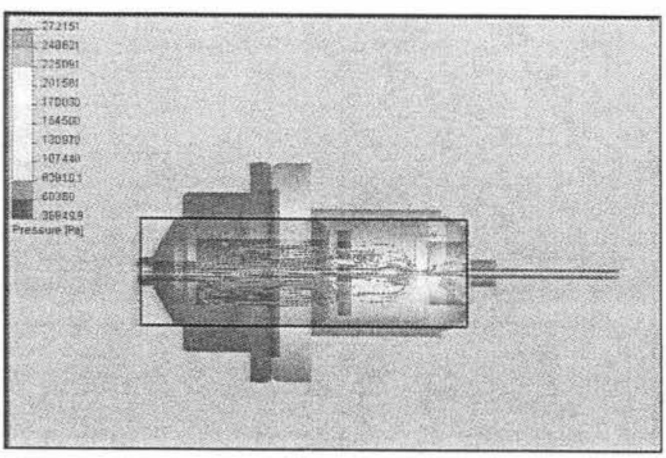


Figure 3.5: Floworks analysis of valve

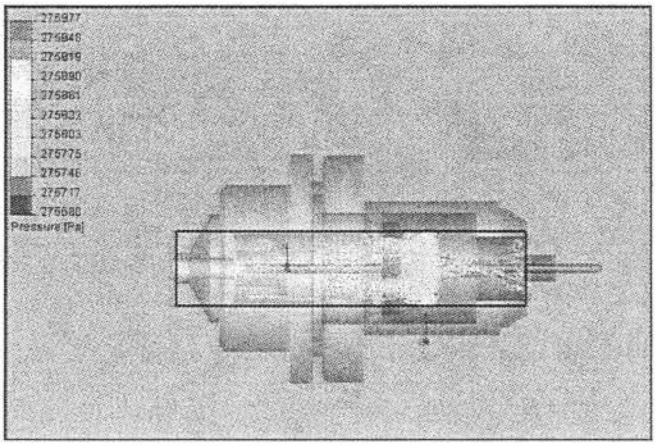


Figure 3.6: Floworks analysis of valve design

3.1.3. Numerical analysis of valve

To model the valve we assume that the gas is ideal and that it is an isentropic systems, such that the flow is adiabatic and frictionless. The analysis will only aim to simulate one-dimensional flow, meaning that the velocity and fluid properties only change in the streamwise direction.

The pressure drop across a valve orifice is important in the simulation of the system. The fluid flow has to be treated as compressible and turbulent. If the upstream to downstream pressure ratio is greater than that of the critical pressure ratio, P_{cr}, then the fluid velocity through the orifice will become sonic, this is known as choked flow. At this condition the mass flow through the orifice is linearly proportional to the upstream pressure. If the pressure ratio is less than P_{cr} then the mass flow depends nonlinearly on both the upstream and downstream pressures. The equation for mass flow through an orifice with area A_v is given by:

$$\dot{m}_v = \begin{cases} C_f A_v C_1 \dfrac{P_u}{\sqrt{T}} & if\ \dfrac{P_d}{P_u} \leq P_{cr} \\ C_f A_v C_2 \dfrac{P_u}{\sqrt{T}} \left(\dfrac{P_d}{P_u}\right)^{1/k} \sqrt{1-\left(\dfrac{P_d}{P_u}\right)^{(k-1)/k}} & if\ \dfrac{P_d}{P_d} > P_{cr} \end{cases} \qquad [3.3]$$

where $\dot{m}_v$ is the mass flow through the orifice, C_f is the nondimensional discharge coefficient, P_u is the upstream pressure and P_d is the downstream pressure.

$$C_1 = \sqrt{\frac{k}{R}\left(\frac{2}{k+1}\right)^{\frac{k+1}{K-1}}} \quad ; \quad C_2 = \sqrt{\frac{2k}{R(k-1)}} \quad ; \quad P_{cr} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad [3.4]$$

C_1, C_2 and P_{cr} are the constants for a given fluid. k is the ratio of specific heats and is equal to 1.4 for air. Substituting k we are able to calculate the values for C_1 = 0.040418, C_2 = 0.156174, and P_{cr} = 0.528.

Now that we have an equation to determine the flow through an orifice the next step is to determine an equation that will model the various chambers in the valve. From the equation of state for an ideal gas:

$$P = \rho RT \qquad [3.5]$$

If we multiply this by volume V , and rearrange it gives:

$$m = \frac{PV}{kRT} \qquad [3.6]$$

Where m is the mass in the chamber with volume V at pressure P and k is the ratio of specific heats.

Due to the chambers being formed by diaphragms the volume is not kept constant, therefore the equation above has to be differentiated with respect to volume and pressure to account for the varying pressure and volume.

$$\dot{m}_n = \frac{\partial m_n}{\partial V_n}\dot{V}_n + \frac{\partial m_n}{\partial P_n}\dot{P}_n \qquad [3.7]$$

$$\dot{m}_n = \frac{P_n}{kRT_n} A_n \dot{x} + \frac{V_n}{kRT_n} \dot{P}_n \qquad [3.8]$$

Since the volume of that chamber can alter, a governing equation needs to be determined, which is as follows:

$$V_n = V_{dv} + (xA_n) \qquad [3.9]$$

Substituting the above equation into the mass governing equation and rearranging we get:

$$\dot{P}_n = \left(\dot{m}_n - \frac{P_n A_n}{kRT} \dot{x} \right) \frac{kRT}{(V_{dv,n} + (xA_n))} \qquad [3.10]$$

The subscript n denotes the chamber number and $\dot{m}_n$ is the mass of gas in the chamber and is the difference between the mass flow into the chamber and the mass flow out of the chamber:

$$\dot{m} = \dot{m}_i - \dot{m}_o \qquad [3.11]$$

In order to determine the mass flow rate through the valve the supply pressure to the second stage regulator is assumed to be constant at 1 MPa and the outlet pressure to be 101 Kpa.

The next procedure is to develop governing equations to simulate the orifice areas of the various valves.

2^{nd} stage regulator valve:

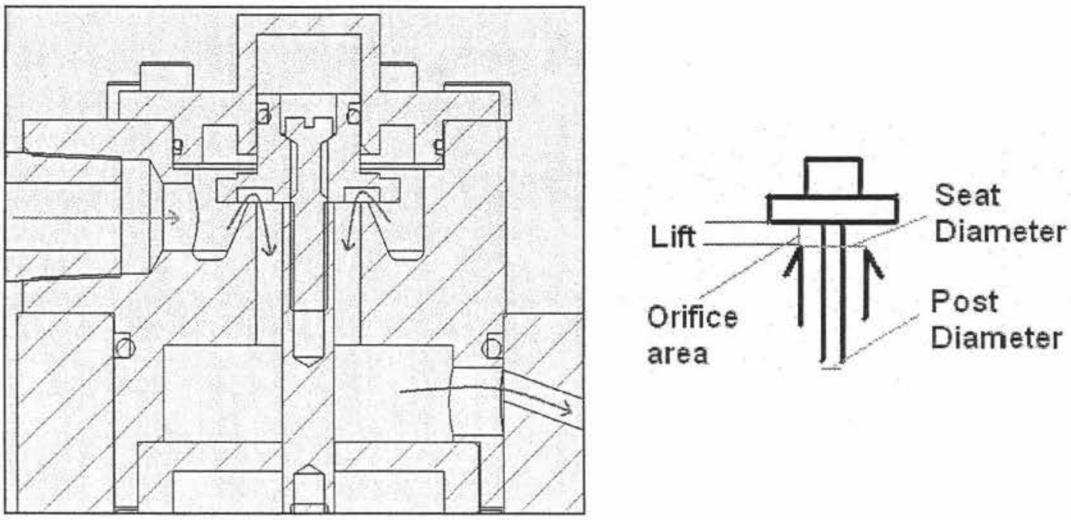


Figure 3.7: Second stage regulator valve model

$$\text{Orifice area: } A_{2nd} = \pi D_{seat} \cdot Lift \qquad [3.12]$$

$$\text{Maximum orifice area: } A_{2nd\,\max} = \pi \frac{\left(D_{seat}{}^{2} - d_{post}{}^{2}\right)}{4} \qquad [3.13]$$

where: $D_{seat} = 11\text{mm}$

$d_{post} = 5.5\text{mm}$

Therefore the orifice boundary conditions for the second stage regulator valve are:

$$A_{2nd} = 0 < \pi(11\times10^{-3})\cdot Lift \ \leq 7.128\text{x}10^{-5} \qquad [3.14]$$

The discharge coefficient for the second stage valve is $C_d = 0.9$.

Feedback valve:

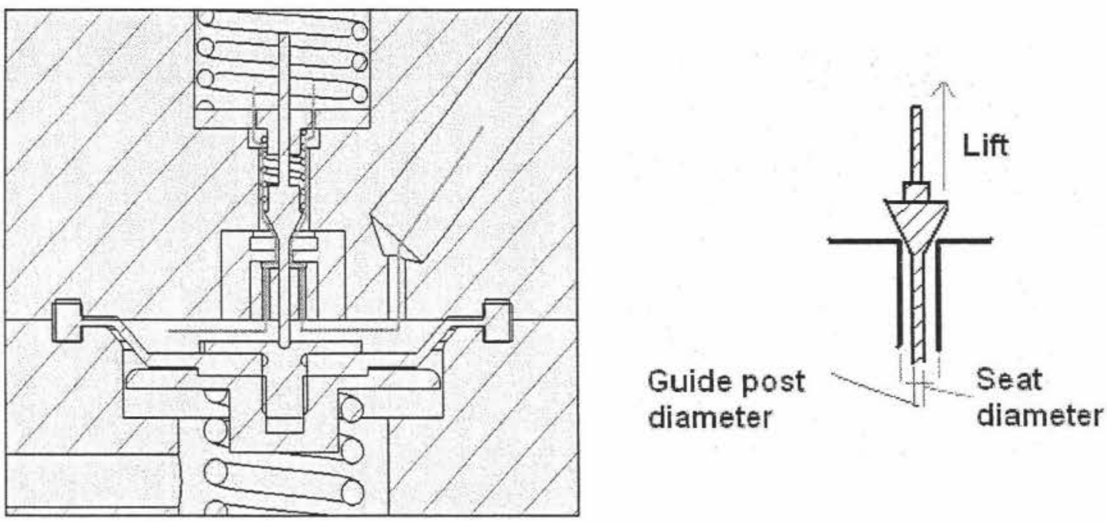


Figure 3.8: Second stage feedback regulator model

Orifice area: $$A_{feedback} = \pi\left(\frac{D_{seat}^{\;2} - (d_{post} + (2(Lift_{max} - Lift)\tan\theta_{seat}))^2}{4}\right) \quad [3.15]$$

where : $D_{seat} = 1.4$mm

$d_{post} = 1$mm

$Lift_{max} = 0.67$mm

$\Theta_{seat} = 40°$

Therefore the orifice boundary conditions are:

$$A_{feedback} = 0 < \pi\left(\frac{(1.4\times10^{-3})^2 - (2.1244\times10^{-3} - 2\cdot lift\cdot\tan 40)^2}{4}\right) \leq 7.54\times10^{-7} \quad [3.16]$$

The discharge coefficient for the feedback valve is $C_d = 0.6$

Main control valve:

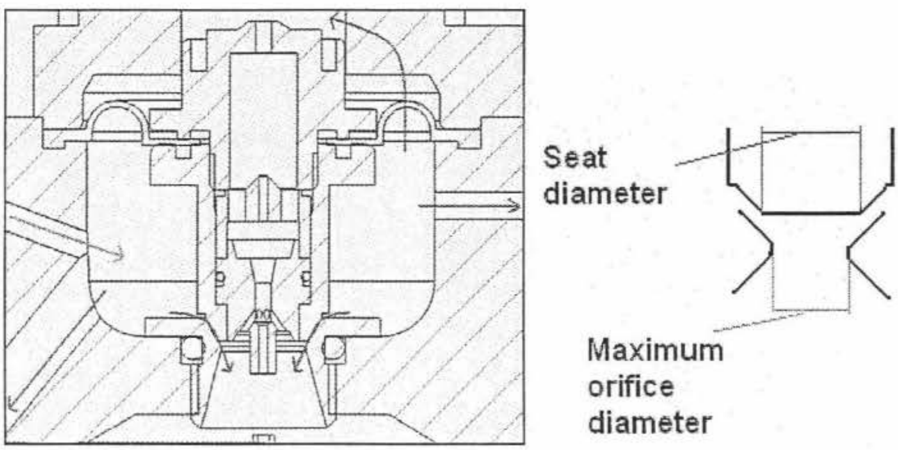


Figure 3.9: Main control valve model

$$\text{Orifice area: } A_{main} = \pi D_{seat} \sin \theta_{seat} \quad [3.17]$$

where : $D_{seat} = 10.11\text{mm}$

$\Theta_{seat} = 30°$

Therefore the main control valve boundary conditions are:

$$A_{main} = 0 < \pi 10.11 \times 10^{-3} \sin 30 \leq 7.088 \times 10^{-5} \quad [3.18]$$

The discharge coefficient for the main control valve is $C_d = 0.9$

Fine needle valve:

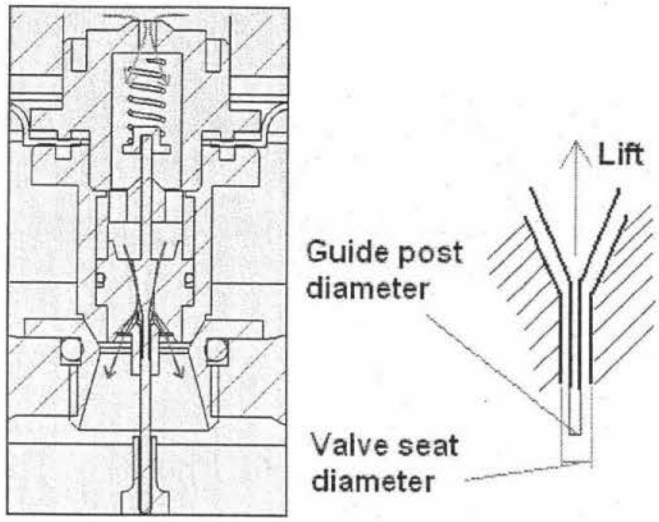


Figure 3.10: Fine needle valve model

$$\text{Orifice area: } A_{fine} = \pi\left(\frac{D_{seat}^{\ 2} - \left(d_{guide} + (2(Lift_{max} - Lift)\tan\theta_{seat})\right)^2}{4}\right) \quad [3.19]$$

Where: $D_{seat} = 1.85\text{mm}$

$d_{guide} = 0.7\text{mm}$

$Lift_{max} = 3.26\text{mm}$

$\Theta_{seat} = 10°$

Therefore the boundary conditions for the needle valve are:

$$A_{fine} = 0 < \pi\left(\frac{(1.85\times10^{-3})^2 - (1.85 - 2\cdot Lift\cdot\tan 10)^2}{4}\right) \leq 2.303\times10^{-6} \quad [3.20]$$

The discharge coefficient for the fine control needle was determined experimentally and is given in the graph below with respect to lift.

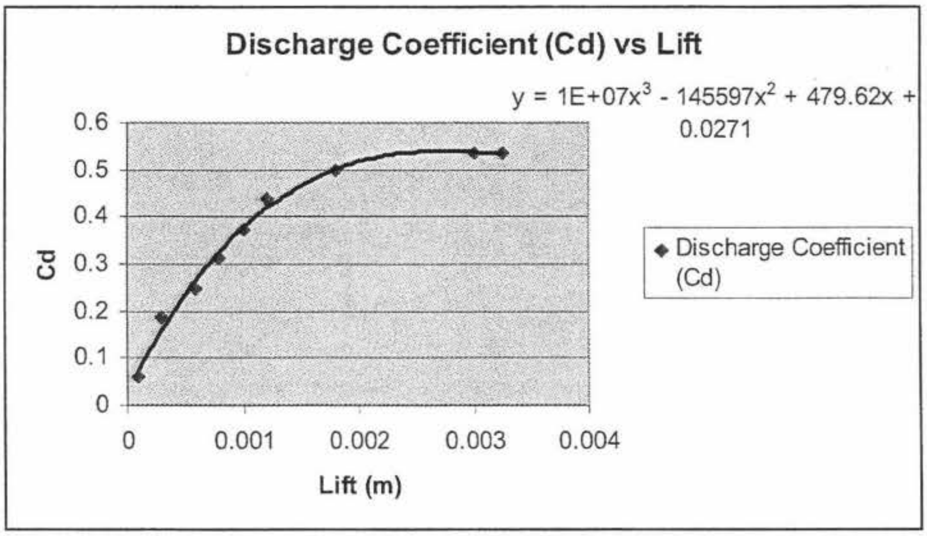


Figure 3.11: Discharge coefficient versus lift for needle valve

Discharge coefficients for all other orifices are considered to be between 0.59 and 0.62.

Finally the mass flow in each chamber can be calculated using the conservation of mass equation. Below is a schematic of the valve indicating the various chambers and orifices.

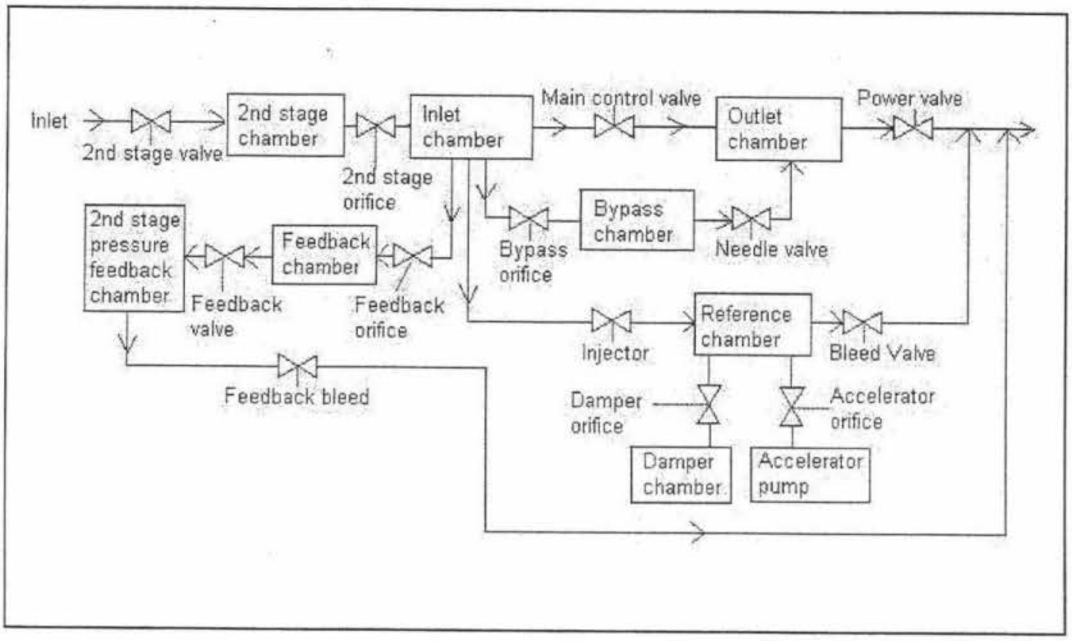


Figure 3.12: Flow chart for Harrison valve

The mass in each chamber is governed by the following equations:

2nd stage chamber:

$$\dot{m}_{2ndstage} = \dot{m}_{inlet} - \dot{m}_{2ndstageofifice} \quad [3.21]$$

$$V_{2ndstage} = V - lift_{2ndstage} A_{2ndstage} \quad [3.22]$$

$$A_{2ndstage} = 4.34 \times 10^{-4} m$$

$$lift_{max} = 6.56\text{mm} = 6.56\text{x}10\text{-}^{3}\text{m}$$

Inlet chamber:

$$\dot{m}_{inletchamber} = \dot{m}_{2ndstageorifice} - (\dot{m}_{feedbackorifice} + \dot{m}_{injector} + \dot{m}_{bypassorifice}) \quad [3.24]$$

$$V_{inletchamber} = V + lift_{main} A_{main} \quad [3.25]$$

$$A_{main} = 5.433 \times 10^{-4} m$$

$\text{lift}_{max} = 6.48\text{mm} = 6.48\text{x}10\text{-}^{3}\text{m}$

Outlet chamber:

$$\dot{m}_{outletchamber} = \dot{m}_{maincontvalve} + \dot{m}_{needlevalve} - \dot{m}_{powervalve} \quad [3.26]$$

$$V_{outletchamber} = V - lift_{ref} A_{ref} \quad [3.27]$$

$$A_{ref} = 5.309 \times 10^{-4} m$$

$\text{lift}_{max} = 6.25\text{mm} = 6.25\text{x}10\text{-}^{3}\text{m}$

2nd stage feedback chamber:

$$\dot{m}_{2ndstgfdback} = \dot{m}_{feedbackvalve} - \dot{m}_{feedbackbleed} \quad [3.28]$$

$$V_{2ndstagfeedback} = V + lift_{2ndstage} A_{2ndstage} \quad [3.29]$$

$$A_{2ndstage} = 4.34 \times 10^{-4} m$$

$lift_{max} = 6.56\text{mm} = 6.56\text{x}10\text{-}^{3}\text{m}$

Feedback chamber:

$$\dot{m}_{feedbackchamber} = \dot{m}_{feedbackorifice} - \dot{m}_{feedbackvalve} \quad [3.30]$$

$$V_{feedbackchamber} = V - lift_{feedback} A_{feedback} \quad [3.31]$$

$$A_{feedback} = 4.34105 \times 10^{-4} m$$

$lift_{max} = 2.68\text{mm} = 2.68\text{x}10\text{-}^{3}\text{m}$

Bypass chamber:

$$\dot{m}_{bypasschmber} = \dot{m}_{bypassorifice} - \dot{m}_{needle} \qquad [3.32]$$

$$V_{bypasschamber} = V - lift_{main} A_{main} \qquad [3.33]$$

$$A_{main} = 5.433 \times 10^{-4} m$$

$lift_{max}$ = 6.48mm = 6.48x10-^{3}m

Reference chamber:

$$\dot{m}_{referencechamber} = \dot{m}_{injector} + \dot{m}_{accelorifice} + \dot{m}_{damperorifice} - \dot{m}_{bleedvalve} \qquad [3.34]$$

$$V_{referencechamber} = V + lift_{ref} A_{ref} \qquad [3.35]$$

$$A_{ref} = 5.309 \times 10^{-4} m$$

$lift_{max}$ = 6.25mm = 6.25x10-^{3}m

Damper chamber:

$$\dot{m}_{damperchamber} = \dot{m}_{damperorifice} \qquad [3.36]$$

$$V_{damperchamber} = V + lift_{damper} A_{damper} \qquad [3.37]$$

$$A_{2ndstage} = 4.34 \times 10^{-4} m$$

$lift_{max}$= 6.56mm = 6.56x10-^{3}m

Accelerator chamber:

$$\dot{m}_{accelpump} = \dot{m}_{accelorifice} \qquad [3.38]$$

$$V_{accelchamber} = V + lift_{accel} A_{accel} \qquad [3.39]$$

$$A_{ref} = 1.56 \times 10^{-4} m$$

$$lift_{max} = 4.84mm = 4.84x10^{-3}m$$

Finally we have to develop equations to govern the valve movement. It was decided that the best method to do this would be to model the lift in the diaphragms as all variable orifice valves are controlled by diaphragms.

2nd stage diaphragm:

$$F_{mass2ndstage} = m_{2ndstage}\ddot{x}_{2ndstage} \qquad [3.40]$$

$$F_{s2ndstage} = k(x_{02ndstage} + x_{2ndstage}) \qquad [3.41]$$

$$F_{Phighpressure} = P_{highpressure} \times A_{2ndstagevalve}) \qquad [3.42]$$

$$Diameter_{2ndstagevalve} = 19mm = 19 \times 10^{-3} m$$

$$A_{2ndstagevalve} = 2.835 \times 10^{-4} m^2$$

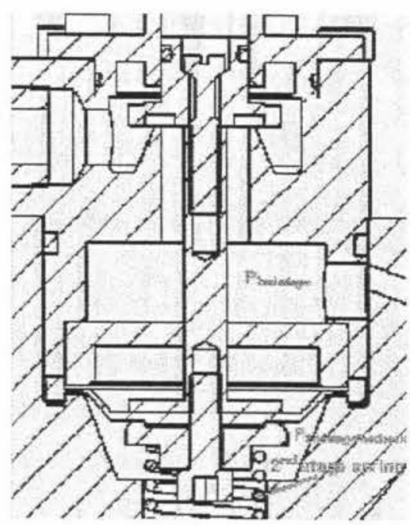
$$F_{2ndstage} = P_{2ndstage} \times A_{2ndstagediaphragm} \qquad [3.43]$$

$$F_{p2ndstagefeedback} = P_{2ndstagefeedback} \times A_{2ndstagediaphragm} \qquad [3.44]$$

$$Diameter_{p2ndstagediaphragm} = 29mm = 29 \times 10^{-3} m$$

$$A_{feedbackvalve} = 6.605 \times 10^{-6} m^2$$

$$F_{flow2ndstage} = \rho q v \cos\alpha \qquad [3.45]$$



$$\sum F = 0 \qquad [3.46]$$

Solving for x:

$$x_{2ndstage} = \frac{F_{Phighpressure} + F_{2ndstage} - (F_{p2ndstagefeedback} + F_{Flow2ndstaqge})}{k_{2ndstage}} - x_{02ndstage} \qquad [3.47]$$

2^{nd} stage feedback diaphragm:

$$F_{massfeedback} = m\ddot{x} \quad [3.48]$$

$$F_{PADJSpring} = k(x_{0Adj} + x_{PADT}) \quad [3.49]$$

$$F_{feedbackvalve} = k(x_{0Feedback} + x_{feedback}) \quad [3.50]$$

$$F_{feedback} = P_{feedback} \times A_{feedbackdiaphragm} \quad [3.51]$$

$$Diameter_{feedbackdiaphragm} = 29mm = 19 \times 10^{-3} m$$

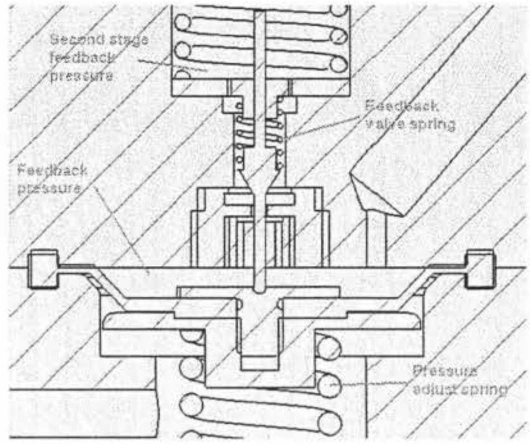
$$A_{feedbackdiaphragm} = 6.605 \times 10^{-4} m^2$$

$$F_{2ndstagefeedback} = P_{2ndstagefeedback} \times A_{feedbackvalve} \quad [3.52]$$

$$Diameter_{feedbackvalve} = 3.5mm = 3.5 \times 10^{-3} m$$

$$A_{feedbackvalve} = 9.62 \times 10^{-6} m^2$$

$$F_{flowfeedback} = \rho q v \cos\alpha \quad [3.53]$$



The summation of all forces shout equal zero.

$$\sum F = 0 \quad [3.46]$$

The mass is small in this case and may be neglected. Solving for x we get:

$$x_{feedback} = \frac{F_{Pfeedback} + F_{2ndstage} + k_{feedbackvalve} x_{0feedbackvalve} - (F_{flow} + k_{PADJ} x_{0PADJ})}{(k_{PADJ} - k_{feedbackvalve})} \quad [3.54]$$

Reference diaphragm:

$$F_{massref} = m_{ref}\ddot{x} \qquad [3.54]$$

$$F_{Finespring} = k(x_{0Fine} + x_{Fine}) \qquad [3.55]$$

$$F_{\mathrm{Re}\,fspring} = k(x_{0refspring} + x_{refspring}) \qquad [3.56]$$

$$F_{P\mathrm{Re}\,f} = P_{\mathrm{Re}\,f} \times A_{\mathrm{Re}\,fdiaphragm} \qquad [3.57]$$

$$Diameter_{\mathrm{Re}\,fdiaphragm} = 36mm = 36 \times 10^{-3}\,m$$

$$A_{feedbackdiaphragm} = 1.018 \times 10^{-3}\,m^2$$

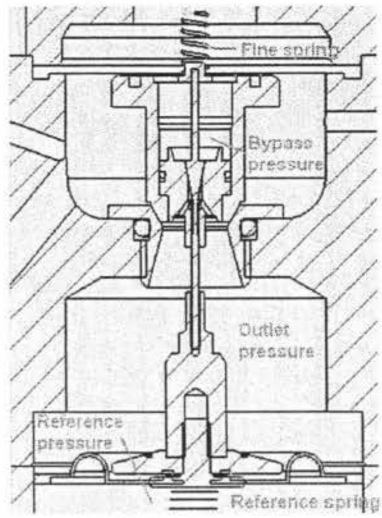
$$F_{Poutlet} = P_{outlet} \times A_{\mathrm{Re}\,fdiaphragm} \qquad [3.58]$$

$$F_{PBypass} = P_{Bypass} \times A_{FineNeedle}$$

$$Diameter_{FineNeedle} = 2.8mm = 2.8 \times 10^{-3}\,m$$

$$A_{feedbackvalve} = 6.158 \times 10^{-6}\,m^2$$

$$F_{flowfeedback} = \rho q v \cos\alpha \qquad [3.58]$$



The summation of all forces shout equal zero.

$$\sum F = 0 \qquad [3.46]$$

Solving for x:

$$x_{reference} = \frac{F_{massref} + P_{Potlet} + F_{bypass} + k_{fine}x_{0\,fine} - (F_{pref} + F_{flow} + k_{ref}x_{0ref})}{(k_{ref} - k_{fine})} \qquad [3.59]$$

Main diaphragm:

$$F_{massmain} = m_{main}\ddot{x}_{main} \quad [3.60]$$

$$F_{mainspring} = k(x_{0main} + x_{main}) \quad [3.61]$$

$$F_{Pinlet} = P_{inlet} \times A_{maindiaphragm} \quad [3.62]$$

$$Diameter_{maindiaphragmlower} = 37mm = 37 \times 10^{-3} m$$

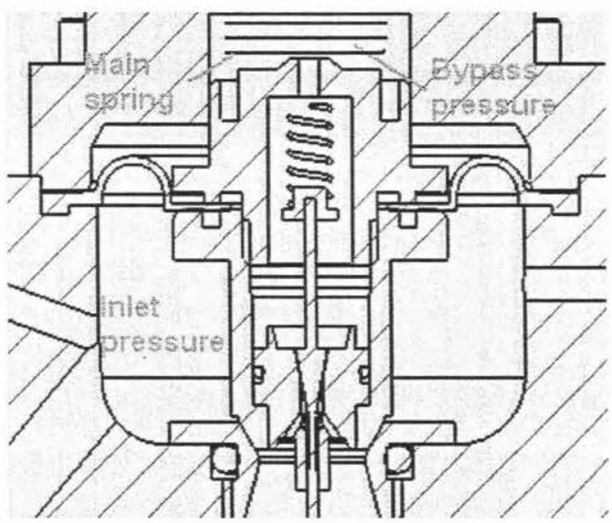
$$A_{maindiaphragmlower} = 1.075 \times 10^{-3} m^2$$

$$F_{pbypass} = P_{bypass} \times A_{maindiaphragmupper} \quad [3.63]$$

$$Diameter_{maindiaphragmupper} = 38.38mm = 38.38 \times 10^{-3} m$$

$$A_{maindiaphragmupper} = 1.157 \times 10^{-4} m^2$$

$$F_{flowmain} = \rho q v \cos\alpha \quad [3.64]$$



Summing all forces and solving for x:

$$x_{main} = \frac{F_{pinlet} + F_{flowmain} - (F_{massmain} + F_{pbypass})}{k_{main}} - x_{0main} \quad [3.65]$$

Using the equations developed above one can use them to determine the valve response due to pressure changes in the various chambers. This enables one to size diaphragms and alter orifice areas for desired outcomes. The volume of the chambers is determined from the 3-D model and limits placed on the maximum and minimum volume possible for each chamber.

3.2. Integration of Valve into an existing petrol vehicle platform

3.2.1. Vehicle platform and conversion requirements

The test vehicle used was a Subaru Legacy sedan; the engine model is EJ20, this engine is a 2 litre, naturally aspirated, four cylinder boxer configuration. The engine is fuel injected and features wasted spark ignition, meaning that the spark plug fires on both the power stroke and in exhaust stroke. Control of the fuel injection and ignition timing is done via an electronic control unit mounted in the driver compartment. The engine did not use a closed loop fuel measurement system, meaning that the engine did not use a lambda sensor to monitor engine fuelling; rather the fuel map was programmed by the OEM according to sensor inputs denoting engine operating conditions. No emissions control hardware such as an exhaust gas recirculation device was implemented by the OEM. The engine compression ratio was 9.5:1 and because the vehicle was to be used with both petrol and natural gas the compression ratio was not changed for testing. This would be to the detriment of efficiency for CNG operation, but was required due to the fact that the petrol fuelling system did not incorporate a feedback system to alter the fuel map.

Considerations that had to be taken into account when installing the CNG conversion equipment was: mounting, placement to minimise the evidence of conversion and also to incorporate the conversion equipment into the existing system. The equipment required for the conversion to a duel fuel vehicle was as follows:

- CNG storage tank
- High pressure hose
- First stage regulator
- Harrison Valve
- Mixer unit

Electronic interface was also required to ensure smooth transition between fuels and for engine adjustments; this topic will be covered in a later section.

The vehicle was fitted with a 70 litre spun steel gas cylinder that was mounted in the rear storage area. Gas lines were installed and routed to the engine bay and connected to the first stage regulator. The installation met with the ISO standards required for natural gas vehicles and passed regulatory requirements. The 70 litre tank has a storage capacity of 17.5 m^3 at 200 bar which equates to 21.8 litres of petrol. The Subaru averaged 10 km/litre when operated with petrol, it is therefore noted that when the cylinder is full the range of the vehicle should be approximately 220 km. This was a benchmark for the fuel economy testing.

The first stage regulator used was sourced from a local provider. The regulator is typically similar to that used by all other manufacturers to reduce the cylinder pressure to typically 150 psi. The figure below shows the first stage regulator and the filler valve assembly.

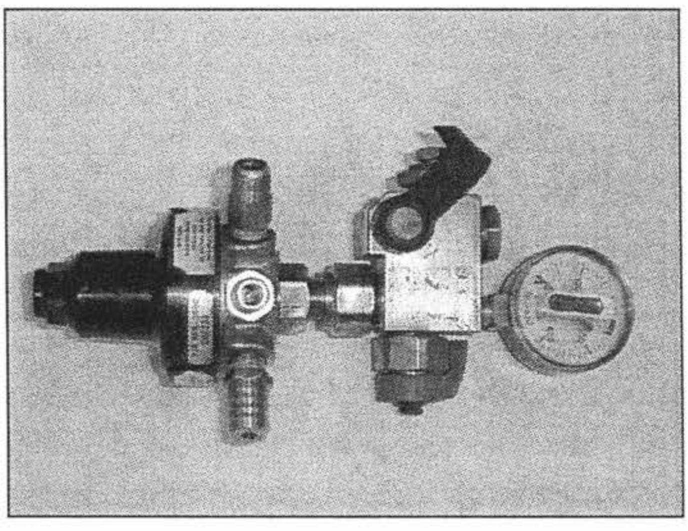


Figure 3.13: First stage regulator and filler point

The regulator is heated by the vehicles coolant system so as to stop it freezing when under high flow conditions due to the thermal expansion of natural gas. The outlet of the first stage regulator is then connected to the inlet of the Harrison valve.

The Harrison valve and mixer unit were both mounted on the engine and further discussion on the location of the valve and mixing unit will be explained in the following sections.

3.2.2. Valve location

The mounting position of the valve is important so as to maintain reasonable throttle response and increase vehicle performance. During initial testing the valve was placed in the front of the engine bay, this position resulted in a long outlet tube to the mixer unit. Below is a figure taken showing the positioning of the valve during first testing.

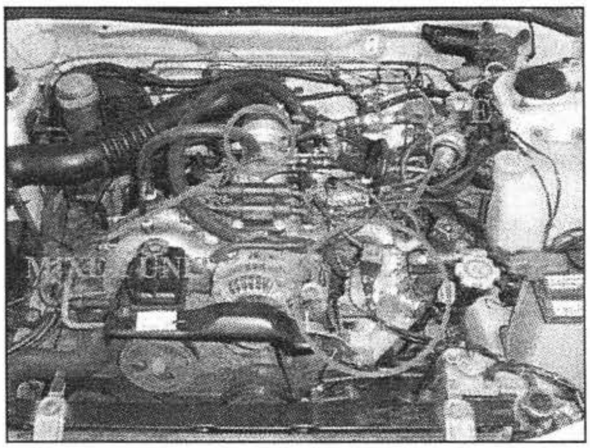


Figure 3.14: Harrison Valve installation on Subaru

As can be seen the long outlet tube has to cross over the entire engine before it reaches the mixer. This long outlet tube resulted in slow throttle response and also in bad ride quality. Due to the engine using wasted spark ignition system, any excess gas in the inlet manifold would be ignited during valve overlap. During rapid deceleration the residual gas in the outlet tube would over fuel the engine and result in a backfire. This is an undesirable condition in that it damages engine components such as the throttle body, petrol injector and airflow meter. To minimise these backfires the unit was removed from the front of the engine and mounted on the fire wall which greatly reduced the outlet tube length and improved the response of the vehicle. Due to the precise measuring there is no longer a large amount of residual gas in the inlet manifold. The ideal location of the valve would be to have the outlet of the valve attached to the mixer unit, this would result in the best response. However there is little space in this area of the engine bay and it is not possible to locate the valve in this location.

3.2.3. Natural gas injection device

There are a large variety of mixer units available to introduce natural gas into the inlet air stream. The figure below depicts the mixer used for this project.

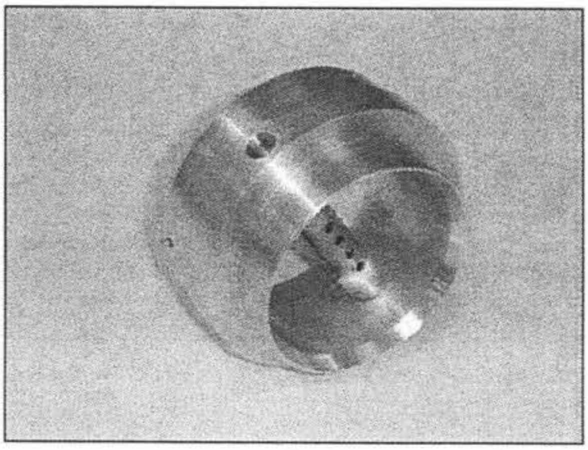


Figure 3.15: CNG mixer

The mixer did not have much development that has gone into the design, the focus was however to prove the functionality of the valve. The mixer was placed on the inlet of the throttle body; this was done so that the throttle body itself functioned as a tumble flap, causing the air and natural gas to form a homogenous mixture before being drawn into the cylinder. The mixer introduces the natural gas into the centre of the inlet air stream where the airflow is at its maximum thereby ensuring the good mixing occurs.

Development is still underway with the design of four injectors. These injectors will be placed in the inlet manifold, one on each runner, approximately 150mm from the inlet valves. The aim of this development is to determine whether sufficient mixing is performed through the inlet valve and also during the compression stage of the cycle to form a homogenous mixture that will result in little cycle-to-cycle variations in combustion pressures. This will increase the vehicle performance and also reduce the chance of backfire due to less gas being present in the inlet manifold.

3.3. Diesel conversion system

3.3.1 Diesel conversion requirements

To illustrate the versatility of the valve, it was decided that the valve be placed on a heavy duty diesel engine. The engine chosen was a CY6102BZLQ used in busses in China. The engine was a 5.785 litre, inline six cylinder, turbo charged, direct injection diesel.

To convert the engine to natural gas operation a number of engine components had to be modified. The compression ratio implemented for diesel was 17:1, for natural gas operation this had to be reduced to avoid detonation. It is possible to have a compression ratio of approximately 14 – 15:1 for a naturally aspirated engine due to the high octane rating of CNG, about 120. Due to the test engine being turbocharged a compression ratio of 13:1 was chosen for a number of reasons, firstly for base operation with very little boost a compression ratio of 13:1 allows for good thermal efficiency and thus good engine performance with respect to power output and torque. Also a higher compression ratio results in a higher expansion ratio therefore more work is done during the expansion stroke resulting in lower exhaust temperatures. For the purpose of demonstrating the lean burn capabilities of the valve a high compression ratio may be utilised due to the reduced tendency of knock at leaner air/fuel mixtures. However one must then realise that less boost may be used so that the engine does not encounter detonation. To lower the compression ratio the piston bowl was increased, this also gave clearance for the spark plug to protrude into the chamber at TDC. The difference in the diesel and CNG pistons is illustrated below.

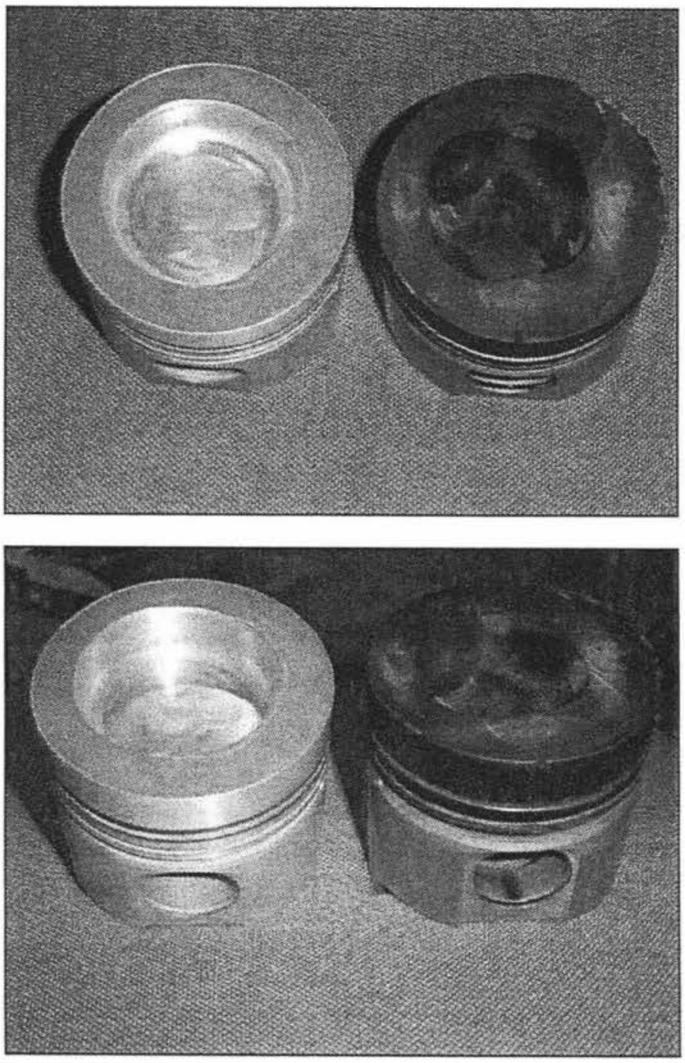


Figure 3.16: CNG and Diesel pistons

The piston on the left is the modified CNG piston, as can be seen the bowl has been increased and the protrusion in the centre of the diesel piston used to generate turbulence has been removed.

Other than changing the bowl shape of the diesel pistons to reduce the compression ratio, an ignition source was required to initiate combustion. The diesel head was removed from the engine so that spark plugs could be fitted, the spark plugs would be located in the same recesses used for the diesel injectors. This gave the spark plug a central location within the combustion chamber thereby resulting in an equal length path for flame propagation and increasing possibility of complete combustion.

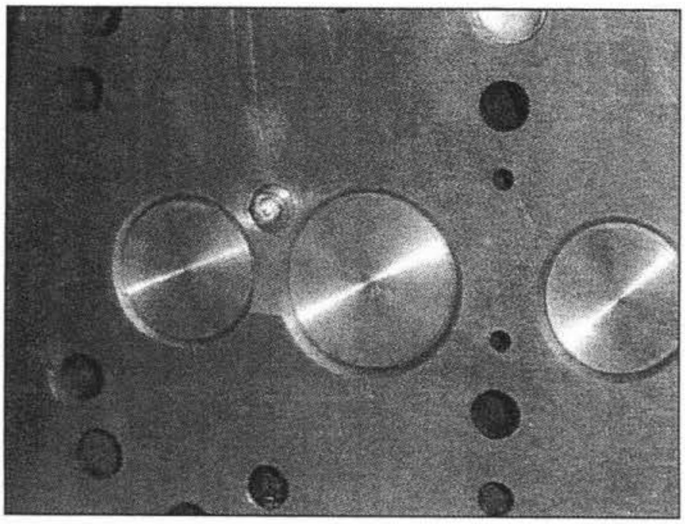


Figure 3.17: Spark plug location in cylinder head

The spark plugs used were NGK DH8S, which is in the lower heat range and has a spark gap of 0.8mm. The spark gap was not altered for the initial testing, but research has shown that a spark gap of approximately 0.5mm is best suited to natural gas operation.

This conversion was a simple first stage conversion that addressed only the vital components that are required to convert a diesel to CNG operation. No effort was made to optimise the engine with regard to inlet manifold, valve alterations or cam profile alterations, all of which would have a significant impact on the engine if optimised for CNG usage. For testing purpose the engine was mapped to have a lambda value of 1.3 and where possible a higher value, such as during lean cruise conditions. The reason for tuning the engine to operate at this air/fuel ratio was that lambda 1.3 was found to be the best compromise for power output, torque and emissions. When the mixture was made slightly richer the torque and power increased however so did the NOx, if the fuelling was increase to stoichiometric the power and torque output exceeded that of diesel by such a large value that it was unsafe to maintain that condition in case of engine failure. We were unable to lean the mixture any further than lambda 1.3 at a large percentage of throttle as the engine experienced misfire, only at light throttle was it possible to approach lambda 1.4. The data presented in the testing and validation section shows the torque, power and emissions for the engine mapped to the criteria mentioned above.

3.3.2. Placement of valve

To ensure that good mixing takes place it was decided to place the mixer unit at the inlet to the turbo. Using the turbo intake spool as a mixing device aided in the mixing of air and natural gas to form a homogeneous mixture. This mixture then passed through the intercooler and then to the inlet manifold. Due to the diesel engine inlet system only being responsible for the transport of air and also due to diesel operating with a large excess air factor the inlet manifold is not designed for optimum air flow. As can be seen in the figure below the inlet manifold consists of numerous right angles, thus reducing flow. This was another reason for the mixer being place at the inlet in the turbo, to allow for the best chance of getting an even distribution of air/fuel mixture to all cylinders.

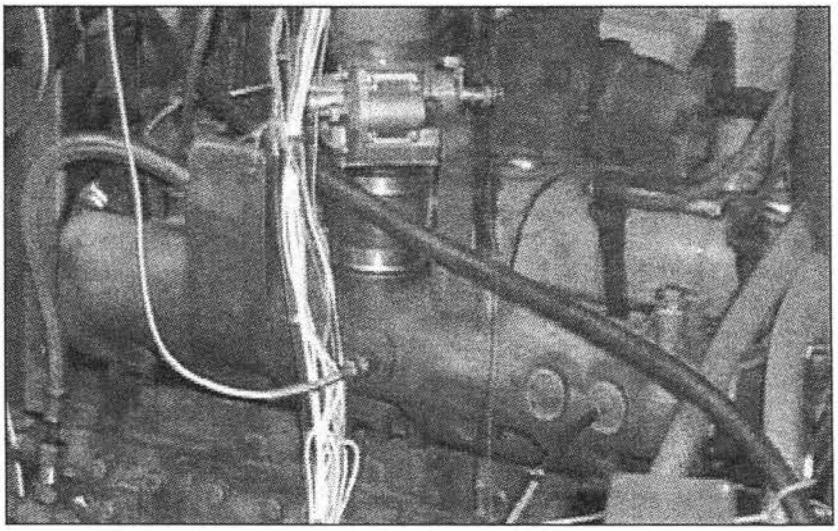


Figure 3.18: Intake plenum for Isuzu engine

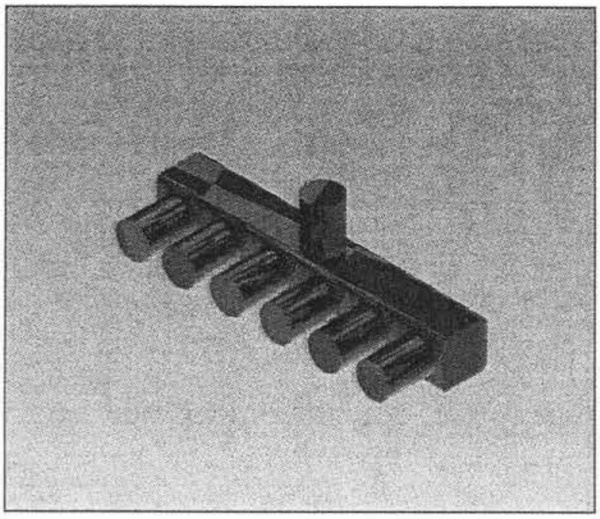


Figure 3.19: Flow path of intake plenum

The figure above depicts the path the inlet charge has to follow to the inlet ports. As can be seen the central inlet to the plenum is at 90 degrees. This is not conducive to good mixture distribution to all cylinders. It was therefore decided that the best possible way to minimise cylinder-to-cylinder variations was to create a homogeneous mixture prior to the intercooler and using high boost pressure to ensure that the plenum was charged with an evenly distributed mixture.

3.3.3. Intercooler

To increase the performance of a turbocharged engine an intercooler can be fitted to cool the intake charge resulting in a denser intake charge and reducing the chance of detonation. Two types of intercoolers are available: air-to-air and water-to-air intercoolers. Both types of intercoolers have their own advantages and disadvantages. The air-to-air intercooler offers greater simplicity, greater thermal efficiency at high speed, greater reliability, low maintenance, and lower cost. The water-to-air intercooler offers better thermal efficiency at low speeds, better throttle response, lower boost-pressure loss and less compressor surge [23].

After considering the facts mentioned above the decision was made to use a water-to-air intercooler because of the fact that the volume of the water-to-air intercooler is much less than that of an air-to-air intercooler to offer the same amount of heat exchange thereby resulting in a smaller intercooler. Also due to the location of an engine in a bus (mostly in the rear) and due to town driving conditions (low speed, stop-start traffic), the use of an air-to-air intercooler would require separate ducting and a fan to ensure that the intercooler has sufficient air flow to scrub the heat from the intake charge. If the water-to-air intercooler were to be used in a bus the coolant may be routed from the bottom of the radiator or a separate radiator may be used to cool the coolant for the intercooler.

The intercooler used was sourced from Cummings. Below is a picture of the intercooler indicating the size relative to the engine.

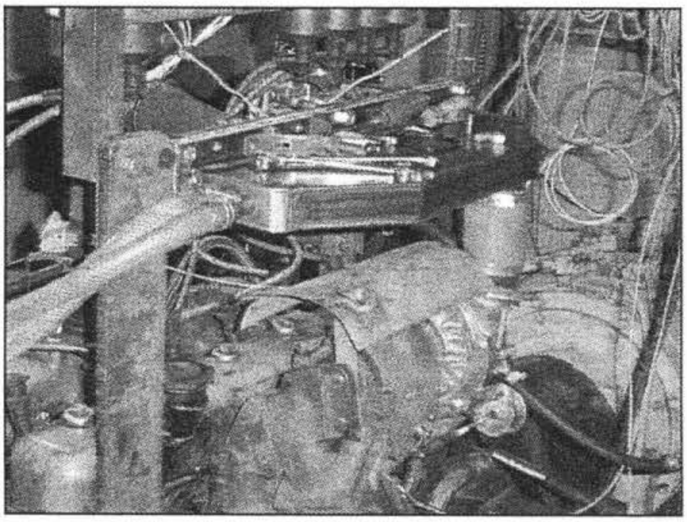


Figure 3.20: Intercooler mounted on engine

The intercooler is relatively small and measures: 540mmx180mmx40mm. The intercooler efficiency can be seen in the table below.

RPM	Ambient air temperature (°C)	Inlet air temperature (°C)	Boost Pressure (mbar)	Exhuast gas temperature (°C)
800	22.2	30	159	751.1
1000	22	30.5	164	749.4
1200	22.4	31.1	217	715
1400	22.6	32.4	348	752.1
1600	23.1	35.5	589	790.6
1800	22.9	37.5	751	826.5
2000	22.6	37.8	729	842.7

Table 3.1: Intercooler efficiency

As can be seen in the table above the heat exchange capability of the intercooler was severely stretched at WOT above 1600 RPM. Considering the exhaust gas temperature and the heat exchange experienced in the turbo, the efficiency of the intercooler is well demonstrated in the table above.

If the intercooler were to have its own radiator and coolant system then the efficiency of the intercooler can be increased. This has been proposed to be the best solution for an intercooler system on this particular application.

3.3.4. Timing discs

For the ECU and ignition system to operate correctly they required signals that referenced TDC and the number of cylinders. The ECU only required a tooth disc with a patter that represented the number of cylinder, i.e. 6 teeth for six cylinders. The ignition system also required the same amount of holes drilled in the disc as the number of cylinder, i.e. 6 holes for 6 cylinders, and one extra for indexing. Both these discs had to be mounted on the engine and rotate at half engine speed, i.e. camshaft speed. Due to the engine being converted to run solely on natural gas the diesel pump was no longer required; this pump drive was driven via a 2:1 reduction gear from the crank shaft. It therefore rotated at half engine speed and was utilised for the timing disks drive. To mount the timing disks on the drive a boss was designed that mated with the drive output to which the discs could be bolted to.

The next items that had to be designed were the sensor mounts. The sensors used were inductive sensors and therefore required very tight tolerances, the air gas between the sensor and timing disc bad to be 0.5mm±0.12mm. To mount the sensor brackets the existing bolt locations for the diesel fuel pump were utilised. Below is a SolidWorks rendering of the sensor mounting bracket.

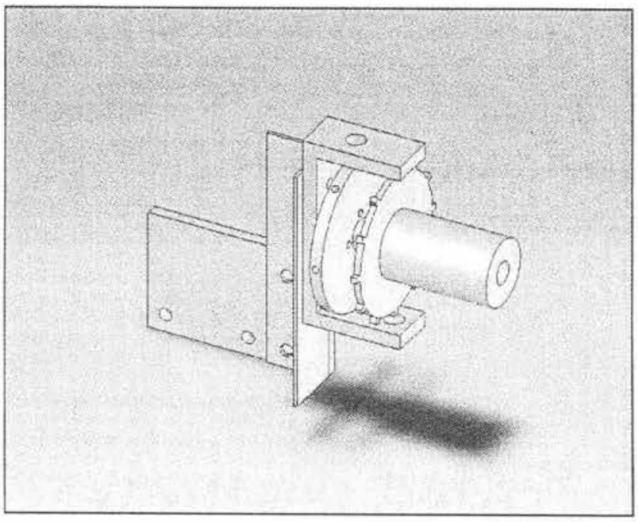


Figure 3.21: SolidWorks model of timing assembly

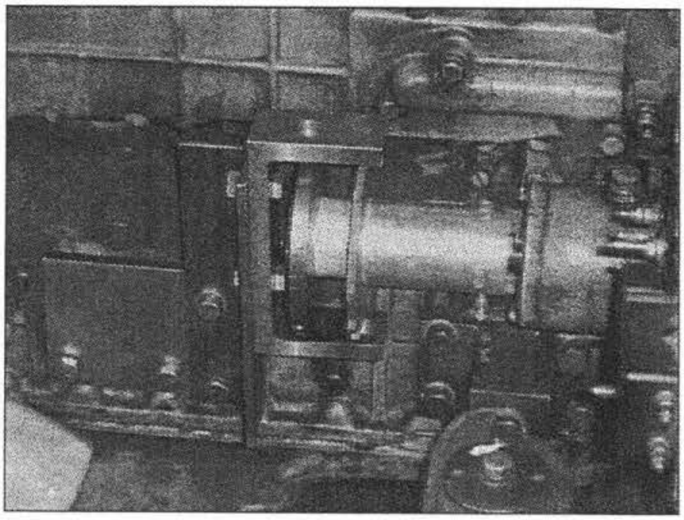


Figure3.22: Timing discs on Isuzu engine

The design of the mounting system was crude and can be refined. It served the purpose of the experiment in proving that the Harrison Valve was capable of fuelling a diesel engine conversion. The design did however prove to be a hindrance in the testing, during operation at an RPM of approximately 2200 RPM the boss and disc assemble started to oscillate. It was found that due to the wear in the engine the reduction gears and diesel pump drive bearings caused timing variations during operation. This will be discussed in a later section.

Chapter 4: Electronic control systems

The electronics required during this project varied from simple injector control for bench testing to complex ignition modifier used to alter the ignition timing for engine operation on natural gas. This section will describe the electronics required for conversions to dual-fuel operation and for the diesel engine conversion to natural gas.

4.1. Bench testing system

During testing and development of the Harrison valve, simulation electronics was required to drive the petrol injector so as to simulate the various driving conditions that the vehicle will encounter during everyday use. The testing system would consist of a computer program for easy user interface and a injector control unit to drive the petrol injector.

The first step was to find a suitable driver for the injector, the LM1949 was chosen for this job. This chip is specifically designed to control a NPN Darlington transistor that drives the high current injector solenoid coil, featuring peak and hold current control for operating the coil. The current required to open a solenoid is several times greater than the current necessary to merely hold it open; therefore, the LM1949, by directly sensing the actual solenoid current, initially saturates the driver until the "peak" injector current is four times that of the idle or "holding" current. This guarantees opening of the injector. The current is then automatically reduced to the sufficient holding level for the duration of the input pulse. The operation of the LM1949 is illustrated below. Appendix 3 contains the datasheet for this device.

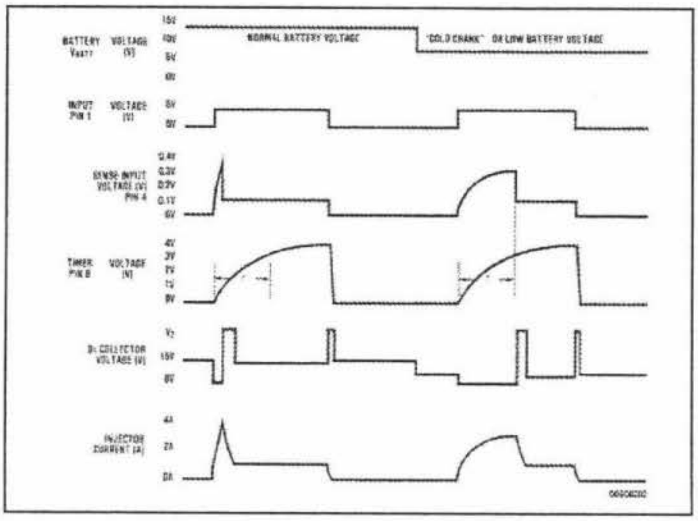


Figure 4.1: LM1949 Peak and hold current control output

The BDX33C NPN Darlington transistor was chosen to drive the high current injector. Appendix 3 contains the datasheet for this device.

Next a microcontroller was chosen to generate the pulse with modulated signal for input to the injector driver. The PIC18F242 was chosen for this task. The reason for choosing this device was mostly due to the ease of programming and also that it has been used before in previous projects and its operation is well known. The PIC18F242 has many features not used for this controller, and may be considered as overboard, but it was readily available and it had been used before by the author. Appendix 3 contains the datasheet for this device.

For communication between the computer and controller, communication via the RS232 serial port was the easiest; the MAX232CPE chip was chosen to control the communication between the computer and microcontroller. Appendix 3 contains the datasheet for this device.

The program to simulate the ECU was written in VisualBasic. This is a simple programming language therefore making it easy to write small programs such as that used for this simulation. The figure below is a screen shot of the program.

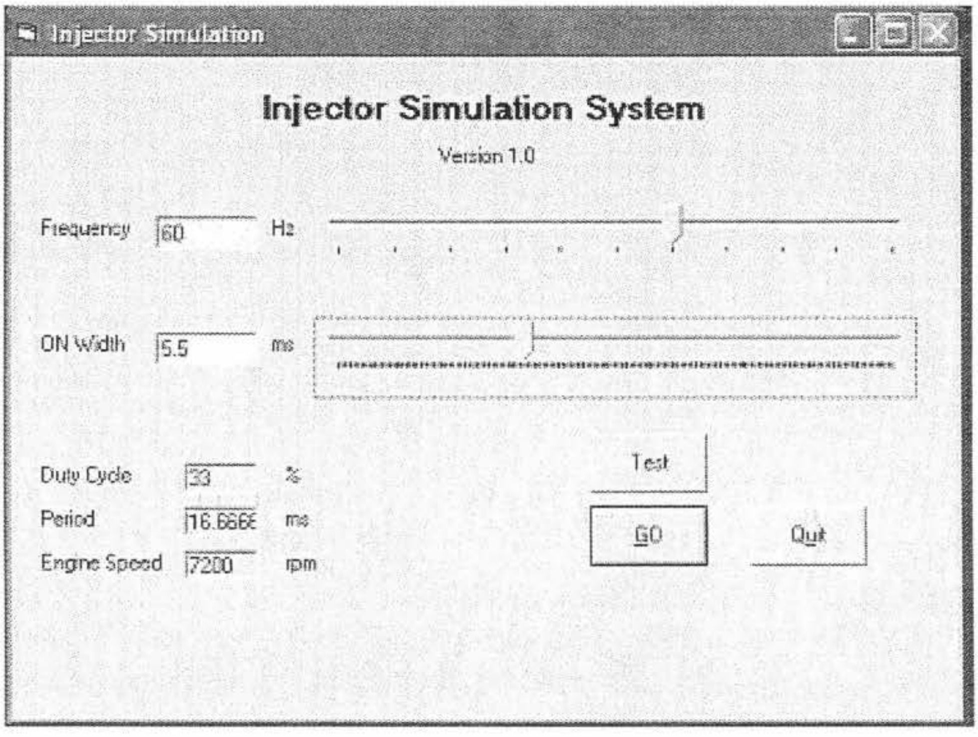


Figure 4.2: Injector simulation System

As can be seen on in the picture above the injector simulator program allows one to alter the frequency and the pulse width. The window also gives a display of the duty cycle percentage, the period of the square wave and the engine speed for the given frequency.

Injectors normally require approximately 2.0 ms to open and 1.5 ms to close, therefore the minimum pulse width to activate and injector is 3.0 ms. Measuring the signal with an oscilloscope on the Subaru test vehicle this was confirmed, at idle the injector pulse width was 3.0 ms. At 6000 RPM and WOT the pulse width is 9.9 ms, the pulse width doesn't exceed that of 9.9 ms for the Subaru. Some vehicles may have a larger pulse width dependant on the injection system and the control strategy utilised.

Using this package it is simple to simulate all injector requirements to test the Harrison Valve. In the section testing and validation, the bench test results, or flow test results are printed with respect to pulse width and frequency, thereby indicating a speed and throttle percentage open.

4.2. Subaru electronic control strategy

For testing on the Subaru a number of electronic interfaces were required to operate the vehicle on natural gas, the following electronics was required:

- Ignition modifier
- Injectors emulator
- Petrol pump cut-out
- Transition electronics

The ignition modifier unit is responsible for modifying the ignition timing. For spark ignition (SI) engines ignition timing varies depending on engine load and RPM.

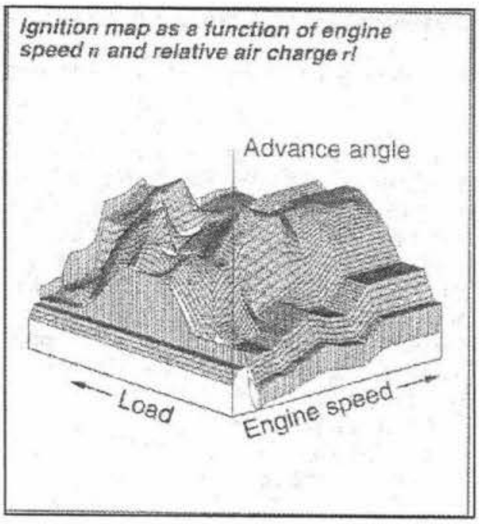


Figure 4.3: Ignition map [14]

The timing of the ignition point is programmed into the ECU to produce peak cylinder pressure approximately 20° ATDC (After Top Dead Centre). Due to the difference in flame propagation of petrol and natural gas, natural gas burning slower, it is therefore obvious that to maintain peak cylinder pressure at 20° ATDC that the ignition has to be advanced with respect to that of the petrol ignition point. If the ignition is not modified when the vehicle is operating on natural gas then the result would be less power due to peak cylinder pressure occurring at the bottom of the piston stroke and also the possibility of increased exhaust temperatures due to combustion taking place outside of the cylinder. The result of this late combustion would result in an increase in fuel consumption and harmful exhaust emissions. If prolonged operation continued under these circumstances damage may be caused to the exhaust valves and the valve seats.

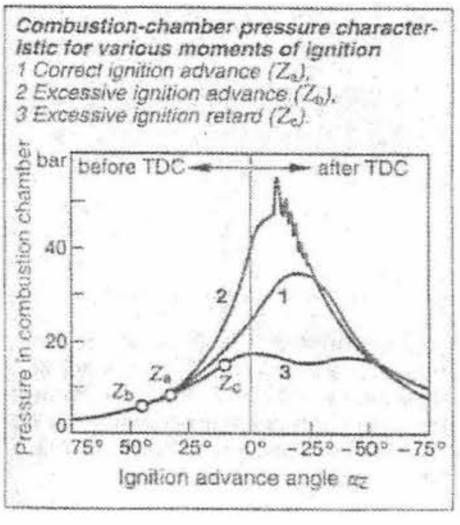


Figure 4.4: Ignition timing effect on combustion pressure [14]

4.2.1. Commonly used ignition systems in light duty vehicles

The ignition modifier unit is responsible for intercepting the ignition signal sent by the ECU. There are four common types of ignition configurations:

1. Distributor
2. Transistorised ignition (TI)
3. Electronic ignition (EI and DLE)
4. Capacitor discharge ignition (CDI)

Most modern motor vehicles have moved away from the old form of mechanical spark distribution, using points and distributors. Modern distributors are a combination of electronic control and mechanical spark distribution. These systems are known as electronic ignition

(EI) with rotating high-voltage distribution, and are commonly present of Honda engines. The figure below illustrates a basic schematic of the system.

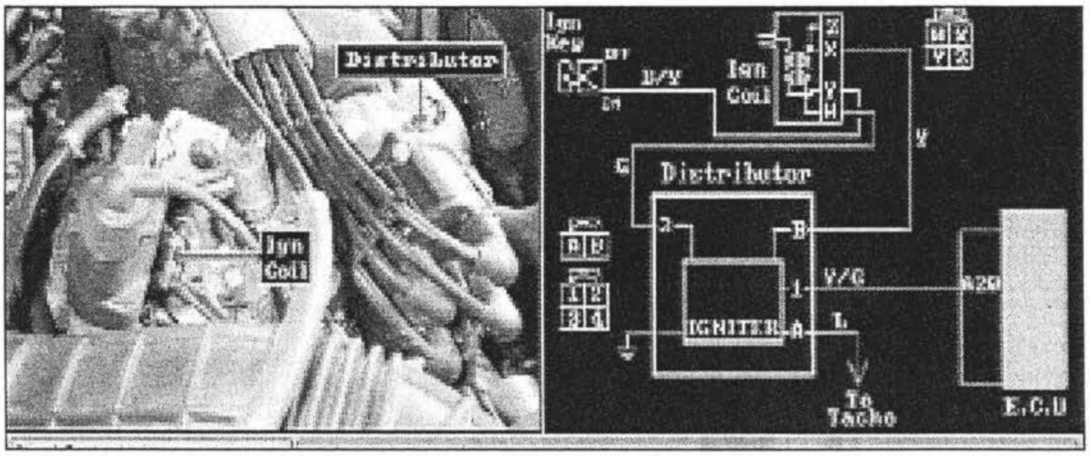


Figure 4.5: Electronic Distributor Ignition [24].

The ECU controls the ignition timing by sending a square wave signal to the igniter on the distributor, the distributor is then responsible for transferring the high voltage output from the secondary coil to the high tension leads which are connected to the spark plugs. In this application the output from the ECU would be routed into the ignition modifier via a relay.

The distributorless electronic ignition systems, or commonly known as transistorised ignition (TI) systems use ignitors that contain triggering electronics and driver stages with primary-voltage and primary current limiting for protection [14]. These systems do away with mechanical distribution of high secondary voltage from the coil to the spark plug. In this system the coil is situated directly on top of the spark plug, therefore each spark plug has its own coil or in some case one coil to two spark plugs. The figure below illustrates the basic schematic for the single-spark ignition coil configuration.

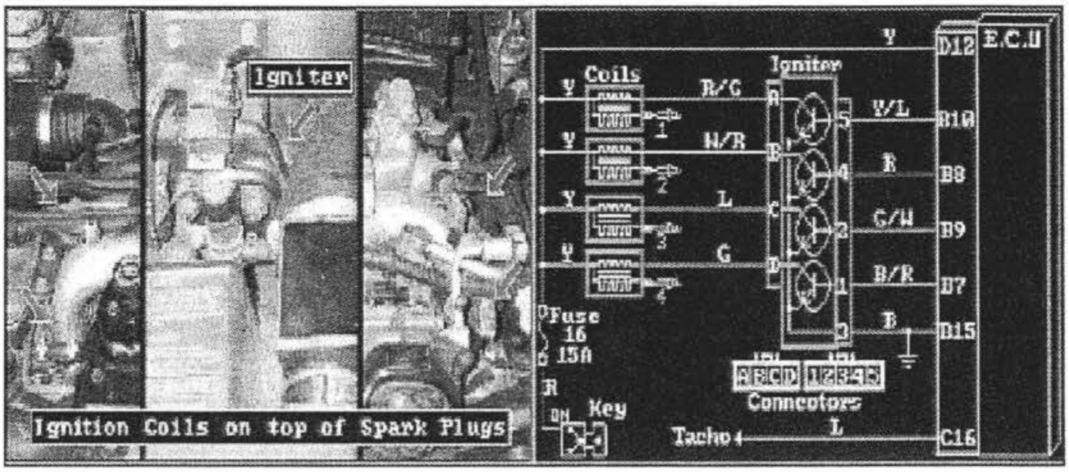


Figure 4.6: Transistorised ignition, single fire [24].

As can be seen in the figure above the ECU has four outputs to the ignitor, thereby controlling each driver stage. This type of configuration allows for the greatest scope for adjustment. To interface this type of ignition system, requires that all four control wire be intercepted by the ignition modifier so as to adjust the entire ignition map. The ignition modifier would then have four outputs that would be wired to the ignitor.

The following ignition system is similar to the DLE mentioned previously. The only difference in this configuration is that two spark plugs share one coil. The high voltage output of the coil is connected to two spark plugs belonging to cylinders that 360° out of phase of each other. As illustrated in the figure below, for a four cylinder with the firing order of 1423, the two coils will be connected as follows:

Coil 1 : Cylinder 1 & 2

Coil 2 : Cylinder 3 & 4

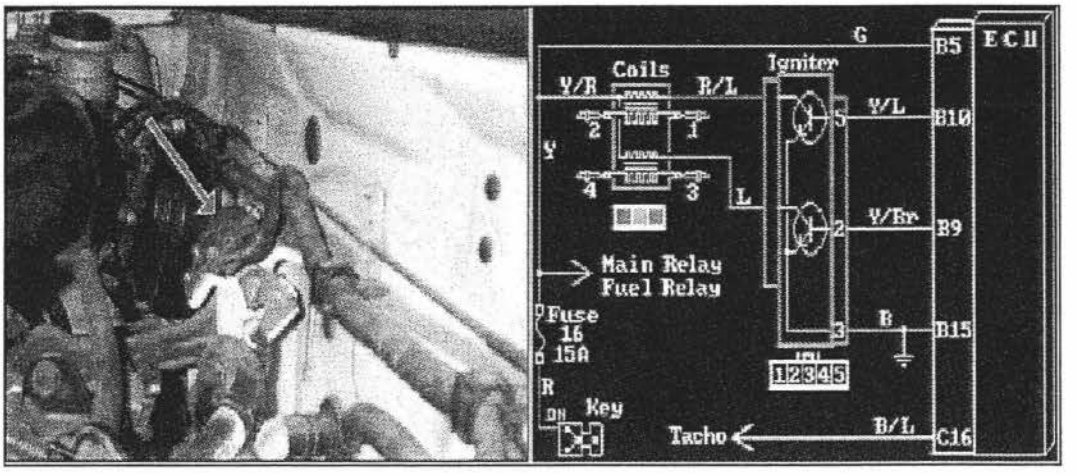


Figure 4.7: Transistorised ignition, dual spark [24].

As can be seen in the figure above, the ECU has only two control signals that are sent to the ignitor, due to two spark plugs firing at the same time. To modify this ignition system only the two ignition control wires from the ECU have to be interfaced with and then modified for natural gas operation.

The next system illustrated here operates with the same principles as the single fire TI ignition system, however this system incorporates the ignitor into the ECU. The output from the ECU is the primary driving voltage and current rather than a low voltage activation signal. The figure below illustrates the schematic for this configuration.

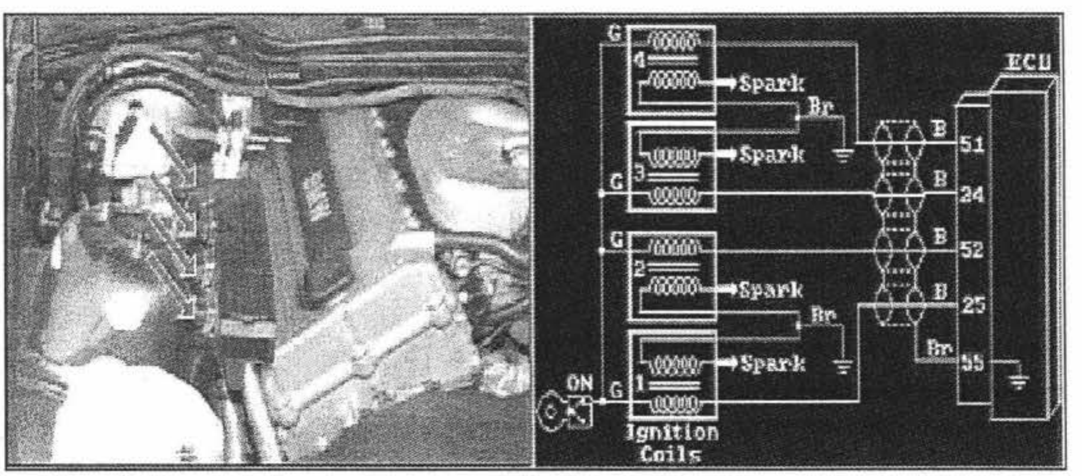


Figure 4.8: Electronic ignition with integrated ignitor [24].

To interface with this type of ignition system the ignition modifier requires its own driver circuitry. The ignition system would operate by intercepting the primary signal from the ECU and modify it and send its own primary signal to the coils. This type of ignition system requires added circuitry to the basic ignition modifier unit.

4.2.2. Ignition modifier

The ignition modifier consists of a microprocessor, passive and active interface electronics and some switching relays.

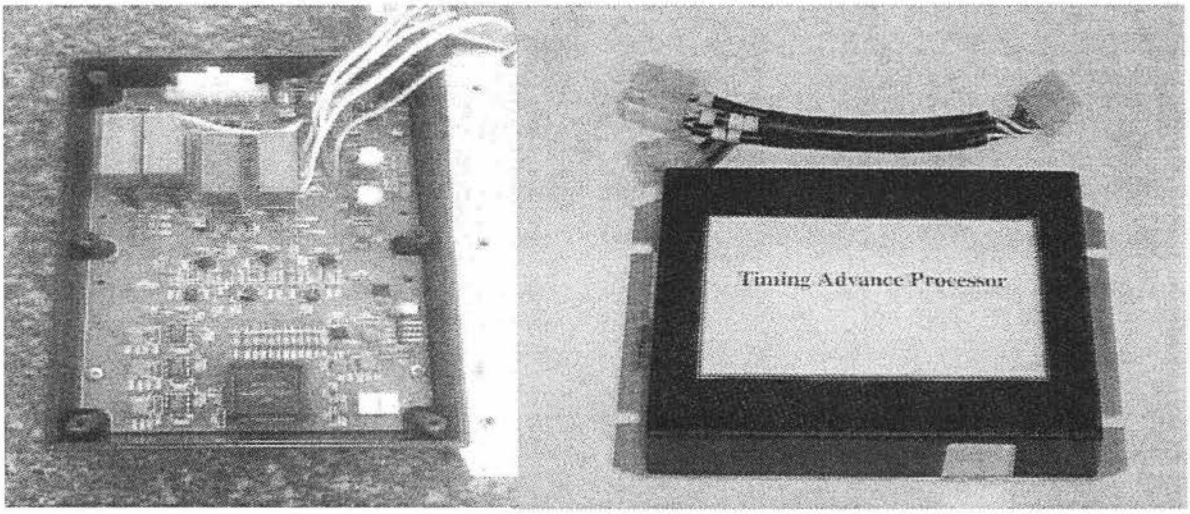


Figure 4.9: Ignition modifier

The modifier works by sampling the input from the ECU and using a lookup table to determine the corrected ignition timing, this corrected output is then sent to the ignitor.

The ignition modifier has to be programmed for each type of vehicle to determine the best ignition timing for the various engine operating conditions. This has only to be done once for each make and model of vehicle and can then be applied to other vehicles of the same make and model. The programming is done by interfacing a computer with the ignition modifier while the vehicle is run on a dynamometer. The vehicle is run through a setup program that simulates various driving conditions so that the ignition modifier may be programmed. During the setup phase the vehicle is set to various operating conditions and the ignition timing is altered at each condition until optimum ignition is achieved, i.e. best possible torque and power without any misfires. Once the setup phase has been completed and the entire engine ignition map has been programmed, the vehicle is then run though another test cycle to check that all ignition changes do not have any undesired results such as misfire or high exhaust emissions output.

Not only does is the ignition modifier responsible for altering the timing, it also controls all the relays needed to switch off the petrol fuel pump, cut-out the injectors and switch to the injector emulator.

When switching from petrol to natural gas certain conditions have to be met to ensure that the transition is smooth and seamless. It has been found from experience that it is best to make the transition between fuels on a trailing throttle, i.e. when the driver has taken their foot off the throttle and the vehicle is decelerating. During this point the engine is under no load and the transition is not felt. When the vehicle driver activates the switch to operate on natural gas the ignition modifier monitors the engine RPM until it finds that the vehicle is decelerating, the modifier then activates the valve and there is a small overlap where both the natural gas valve and petrol injectors operate. After this small time delay the injectors are cut out and the petrol fuel pump is disabled.

The ignition modifier used on the Subaru was developed by Auckland University in conjunction with Gas Association of New Zealand in the early 1990's. This product functions correctly but due to technological progress, the microprocessor and switching transistors have all been superseded and are no longer state of the art. The processor used in the ignition modifier is a Motorola MC68SEC311E2FN that operates at 8 megahertz, this processor however working well for this application is now out dated making it more expensive and harder to come by than its better successors. Development is currently under way to develop a new ignition modifier at Massey University that will feature updated electronics and more user friendly programming software.

4.2.3. Injector emulator

The injector emulator is used during gas operation to simulate the petrol injectors. During gas operation the petrol injectors are disabled using a relay, the ECU however does diagnostic checks to determine whether any devices are not operating correctly, this was described in the OBD section earlier. Due to these checks the ECU has to be "tricked" that the injectors are operating as per normal. Below is a picture of the injector emulator.

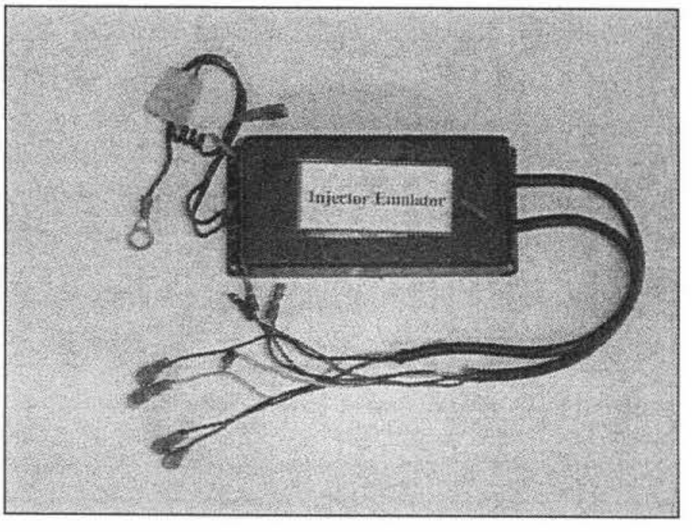


Figure 4.10: Injector emulator

The emulator consists of resistors and inductive coils that imitate the petrol injector. When the vehicle is operated on natural gas the petrol injector ground return wires that are driven by the ECU are routed to the injector emulator via a relay. Therefore the combination of inductive coils and resistors are able to imitate the signal response of the injector solenoid thereby fooling the ECU that the vehicle is still operating on petrol. When the vehicle operates on petrol the relay switches the ground return wires from the injectors back to the ECU.

If this device was not used the vehicle the OBD system would recognise a fault had occurred and turned on the MIL light, the vehicle would then operate in "limp home mode". In this condition the engine is operated at low RPM and low throttle so as to minimise any consequential damage that may result from this malfunction. The vehicle would continue to operate in this manner until the ECU was reset and the error code removed from the memory.

4.3. Diesel engine conversion

To successfully convert a diesel engine to operate on natural gas not only does the mechanical nature of the engine have to be changed, but control electronics has to be added to monitor and drive the engine.

The previous statement refers to an ECU and ignition system. The particular diesel engine that was converted for this project was controlled completely by a mechanical system. The fuel injection system was mechanically driven off the crank shaft and being a diesel, no ignition system was required. As diesel engines operate by compression ignition, whereby the air/fuel mixture is heated to its autoignition point during the compression stroke and no spark plugs are required to ignite the mixture. It was therefore required for this project that a suitable ECU and ignition system be chosen to operate the engine.

There are a variety of commercially available ECU systems available, varying from complex systems used for race vehicles and high performance aftermarket OEM replacements to simple low end basic engine management systems. For the diesel conversion it was not necessary to have an ECU system that offered surplus operations such as sequential injection, variable cam control, ABS, traction control etc. It was therefore decided that the ECU only have the basic engine management capability however still maintaining a wide range of adjustability.

A majority of ECU systems are able to control both fuel injection and ignition timing, with most ECU packages having control electronics for inductive ignition and have "add-on" packages for CDI ignition. For this diesel conversion it was decided that a separate ignition system be used that is based on capacitive discharge ignition. The system chose has been successfully used on other natural gas engines around the world and has been designed for natural gas engines.

The ECU package and the ignition package will be describe in the following sections, detailing the requirements, the operation and the control strategy used.

4.3.1. ECU Control system

The ECU system chosen for the diesel conversion was the Link Engine Management LEMV5 unit. This unit is the baseline product produced by Link ElectroSystems Ltd. that features the basic engine management required for this project.

The Link ECU features hardware:

Microprocessor specification:

- Make: Motorola E20FN4
- Clock speed: 16 Megahertz
- Program memory: 20K ROM
- Volatile memory: 768 bytes SRAM
- Non-volatile memory: 512 bytes EEPROM
- Analogue to digital converter: 8 channel, 8 bit
- Main timer: 16 bit, 250nS resolution
- Asynchronous serial port: NRZ format, full duplex
- Synchronous serial port: Full duplex (SPI)

MAP Sensor (Internal)

- Make: Motorola MPX series silicon strain bridge transducer
- Pressure Range: 0 kPa (absolute vacuum) to 255 kPa (23 PSI boost)

[25]

Control features of the Link LEMV5 are:

Inputs	
Cylinder Pulse	Cylinder trigger input (see below).
Sync Pulse	Multi-coil and sequential fuel applications (see below).
MAP	Internal Manifold Air Pressure Sensor. Reads up to 250kPa (1.5 Bar or 22psi of boost).
Engine Temperature	Engine coolant temperature sensor.
Narrow Band O2 Sensor	Exhaust gas narrow band oxygen sensor input.
Wide Band O2 Sensor	0-5V input from a wide band oxygen sensor controller.
TPS	Throttle position sensor.
Battery Voltage	For injector open time correction.
Inlet air temp	For inlet air temperature mixture correction.
Boost Adjust	0-5V input from a dash mounted knob for boost adjustment.

Outputs	
Two Injector Drives	Group or staged injection. Up to six injectors per drive.
Three Ignition Drives	Either distributor or wasted spark.
Inverted Ignition Drive	Compatible with Honda and Ford ignition systems.
Fuel Pump Relay Control	Automatic fuel pump priming and control.
Radiator Fan	Radiator cooling fan relay control.
Voltage Regulated Output	5V regulated output to power input sensors.
Auxiliary Outputs	Selectable fully configurable output functions as follows:
· Aux1	Tachometer Drive - Low level output for factory tacho control.

	Boost Control - Directly drives a wastegate control solenoid.
	RPM Switch - Adjustable on and off RPM settings.
	Idle Speed - Drives two wire idle solenoid.
· Aux2	Auxiliary output that may be switched using either MAP, RPM, injector duty cycle or throttle position.
· IG3	Tachometer - Low level output for factory tacho control.
	Shift Light - Directly drives a dash mounted LED.

[25]

The ECU programming may be done either via a PC or a tuning module that can be purchased from Link ElectroSystems Ltd. For this project the tuning was done via the PC making it easier to monitor the engine reactions to altering timing and fuelling. A screen shot of the interface software is shown below.

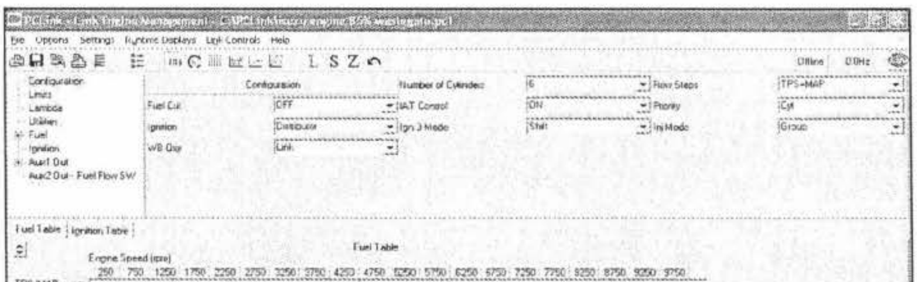


Figure 4.11: PCLink software

As can be seen in the figure above the user interface is kept simple and easy to understand. In the centre of the screen is the fuel zone adjustment, this allows the user to adjust fuelling for a given operating condition. The fuel table consists of 8 rows labelled TPS/MAP, thus representing the load axis, and 20 columns labelled engine speed increasing in 500 RPM steps. Therefore each "zone" represents an individual engine operating condition. When the engine uses forced induction the load axis is split into two, when the MAP is below 110kPa the TPS signal input is used to determine the row, when the MAP is above 110kPa the rows are determine according to the MAP signal input.

Below the fuel zone table are the engine runtime values. These values indicate the current engine operating condition, giving readouts for engine speed, manifold air pressure, engine temperature and injector pulse width to name a few.

It is possible to use narrow-band or wide-band oxygen sensors to create a closed-loop feedback ECU program using the Link LEMV5, however this option was not utilised for the experiments due to ignition problems faced during testing, which will be explained in a later section. The CNG converted engine was operated with an open-loop fuelling map to demonstrate the versatility of the Harrison Valve, enabling the programmer to alter the fuel map to achieve the required emissions and power output for the given operating condition. The tuning procedure and results are discussed in a later chapter.

4.3.2 Ignition system

The ignition system chosen for the diesel conversion was that of a CDI system due to the reasons stated in section 2.9. The product purchased was a CD200 unit from Altronic Inc. The CD200 is a high energy digital capacitor discharge system that has been designed for use on 1 to 8 cylinder small industrial natural gas engines. It features a microprocessor design that processes angular position input signals from a magnetic pick-up sensor that senses holes or protrusions on a timing disk.

The specifications of the CD200 are as follows:

- Number of cylinders: 8
- Input power: 12VDC, 1.0 Amp
- 24VDC, 0.5 Amp
- Max. Voltage output: 40 KV
- Spark Duration: 300-600 microsec.
- Timing Adjustment:
- Manual (8-position switch) User-Selectable Increments
- RPM Range 25 to 2500 RPM
- Analog input range 4-20mA or 0-5VDC
- Overall max. timing range 25° of retard
- Over speed set point range: 25 to 2500 RPM
- Output switch rating: 0.5 Amp, 32 VDC max.
- Communications: Modbus RTU (RS-485)

The CD200 features a Windows based terminal program that allows the user to alter the features available on the CD200 such as timing adjustment curves based on RPM. The terminal program also displays the system primary and secondary discharge diagnostics. Below is a screen shot of the CD200 Terminal Program.

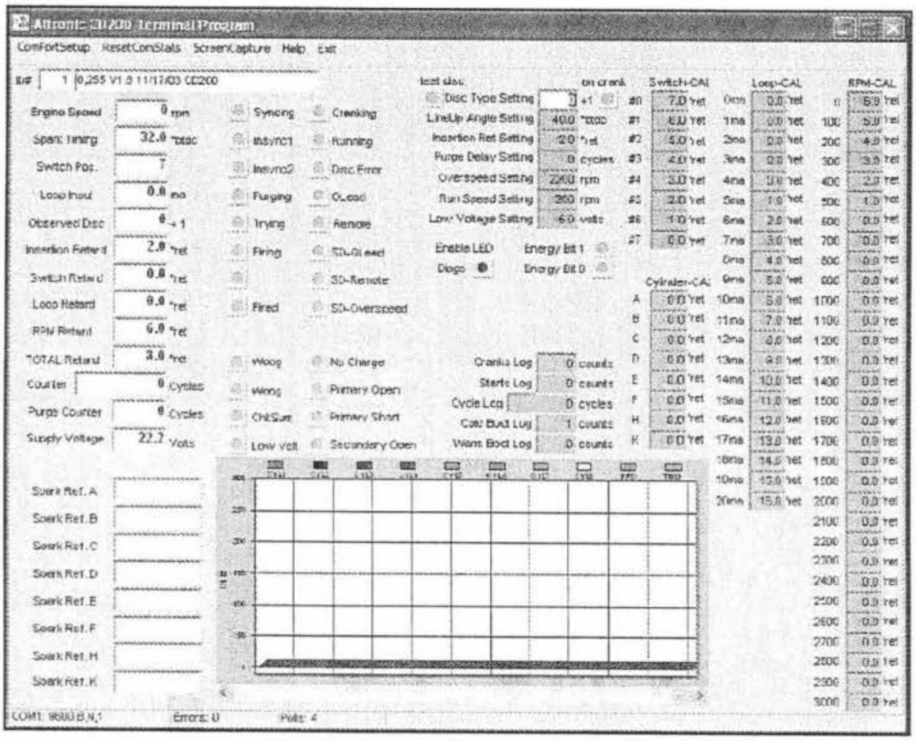


Figure 4.12: CD200 Terminal program

The CD200 requires that for single fire operation one coil per cylinder is required. The coils used for ignition were sourced from the same company, part number 501061. The primary resistance of the coil is 0.1 – 0.2Ω and the secondry resistance of 4400 - 6900Ω. Each coil is controlled individually by the CD200 unit with the wiring diagram shown in appendix 3. The figure below illustrates the mounting position of the CD200 unit and the coils. The high tension (HT) leads have to be kept at least two inches from any other form of wiring to reduce electromagnetic interference.

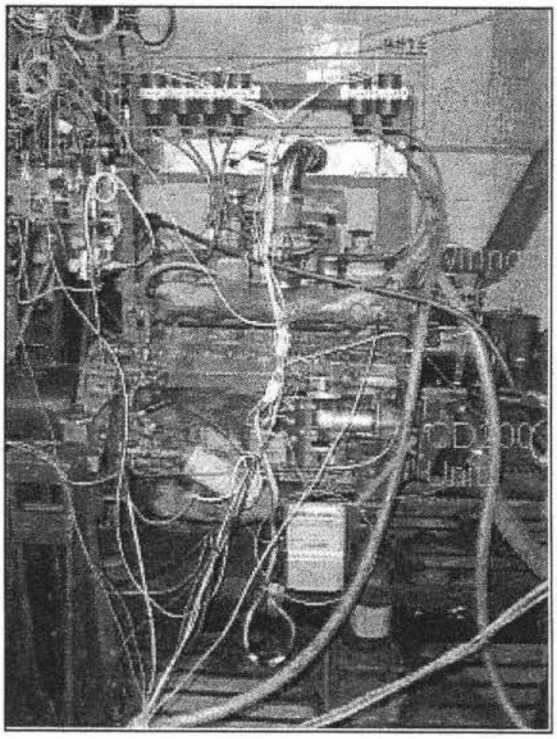


Figure 4.13: CD200 placement on test engine

The control principle of the ignition used during testing of the natural gas converted engine will be explained in a later section

Chapter 5: Testing and Validation

Various tests were conducted to demonstrate the linearity and versatility of the valve. The tests conducted varied from a simple bench test to demonstrate the flow characteristics of the valve to exploring the lean burn limits in a 5.8 litre, six cylinder diesel engine that was converted to operate on CNG. The tests and testing procedures will be discussed in this section

5.1. Bench Tests

The aim of the bench test was to determine whether the valve behaived in a linear fashion during various simulated operating conditions. The bench tests were carried out using compressed dry air as the gas medium and a computer program to simulate the ECU signal from an ECU found onboard the vehicle. The program on the computer allowed one to change both the frequency and the pulse width so as to simulate a certain engine operating condition. For example if a car was idling at a set of traffic lights the engine crank shaft would be rotating at approximately 780 RPM and the pulse width to the injector would be approximately 3ms. If the car used sequential injection, thereby meaning that the injector is only activated on the inlet stroke, the frequency of the signal would only be 6.5Hz as for a four stoke engine each cylinder only has an intake stroke every two crank shaft rotations. At maximum speed of 6000RPM and WOT (Wide Open Throttle), this is when the engine is experiencing maximum load, the injector signal frequency would be 50 Hz and the pulse width 9.9ms. Using the above mentioned information it is then easy to simulate any of the driving conditions that the engine may experience. To then determine the mass flow rate required at the various operating conditions some simple mathematics can be use to calculate the air consumption of any engine given the capacity of the engine, the RPM and the volumetric efficiency. The volumetric efficiency of an engine is determined by the inlet manifold, the inlet valves and also by the camshaft design to mention only a few items, therefore every engine will have a different volumetric efficiency. For simplification it is generally accepted to use a volumetric efficiency value of 80% for this simple type of equation. The equation for air consumption of an engine is as follows:

$$AirConsumption = \frac{RPM \times Displacement \times VolumetricEffieciency}{2} \quad [5.1]$$

This equation is used to then determine the air consumption of the engine for the RPM operating range. Once all the calculations have been completed the air consumption value is then used to calculate the theoretical mass flow rate of CNG using the air/fuel ratio at stoichiometric operation. The stoichiometric air/fuel ratio for natural gas is approximately 16.9:1 depending on the composition of the gas, the more methane present in the composition the higher the air/fuel ratio. For pure methane the air/fuel ratio is 17.2:1, for our purposes we will use 16.9:1. The following equation is then used to determine the natural gas consumption given the air consumption:

$$NaturalGasConsumption = \frac{AirConsumption}{Air / FuelRatio} \quad [5.2]$$

Following these calculations one has reference flow figures for WOT operation throughout the engine operating RPM range. Using the computer program and connecting the output of the valve through a set of rotameters it is then easy to tune the valve to produce the required flow rates at WOT. When the vehicle operates at part throttle the air flow is then a fraction of that at WOT and therefore the gas flow rate a fraction of that required at WOT and is a linear response, therefore the valve only has to be tuned to the flow rates at WOT due to the linear response characteristics of the valve. Finer tuning can then be done once the valve has been installed on the vehicle if required. Below is a picture of the test bench setup used to tune the valve.

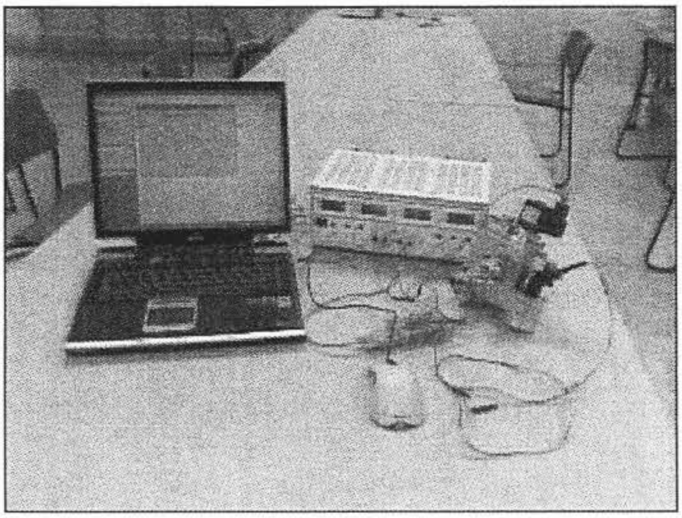


Figure 5.1: Valve test bench excluding gas cylinder and rotameters

The following graph illustrates the flow characteristic required of the valve for various engine operating conditions.

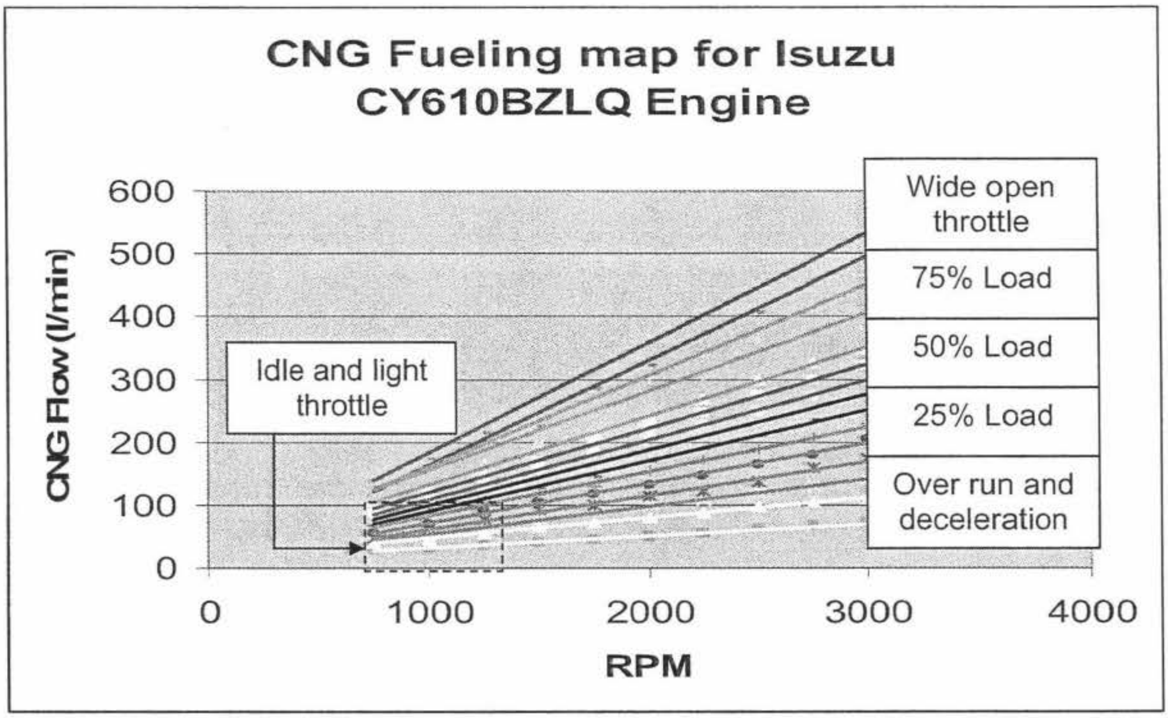


Figure 5.2: Valve flow requirements

5.2. Retrofit Test

Following the bench test to determine linearity, tests were required to demonstrate the versatility of the valve and to demonstrate the ability to retrofit the valve to a vehicle that has an onboard ECU that is normally used to operate the vehicle on petrol. One of these tests involved fitting the valve to a standard everyday saloon vehicle, the vehicle was a 1990, 2.0 litre, 4 cylinder boxer configuration, naturally aspirated Subaru Legacy.

The engine remained unmodified from the form used on petrol operation, having a compression ration of 9.5:1; no modifications were made to the inlet valves, the valve timing or to the inlet manifold. The reason for not modifying any of the intake components was to demonstrate the retrofit application of the valve, that it is simple and easy to install requiring little work to the engine.

Some simple electronics was used to shut off the petrol injectors and intercept the signal from the ECU to emulate the petrol injectors when the vehicle operated on natural gas so as to deceive the ECU into thinking it was still operating on petrol. The electronics was described in section 4.2. Due to the slow burning rate of natural gas some intelligent electronics was required to alter the advance of the spark timing to ensure that complete combustion occurs and that the engine produces the greatest amount of torque and power for the given mixture. The electronic ignition advance circuit intercepted the signal from the ECU to the igniters and altered it to a value at the appropriate location in a lookup table.

Once all the electronics and valve had been installed on the vehicle a simple centre point injection unit was designed that attached to the inlet of the throttle body. Throttle body injection was chosen not only for simplicity but also due to the fact that natural gas is less dense than air and if not properly mixed will remain in laminar layers resulting in poor mixture formation in the combustion chamber. This poor mixture formation would result in poor combustion due to the variations of air/fuel ratio concentrations in the combustion chamber not allowing the flame to propagate across the entire combustion chamber and as a result the engine would produce poor torque, power, emissions and ultimately high fuel consumption. However if a homogeneous mixture is created prior to the combustion chamber and enters the combustion chamber as a homogeneous mixture then at the onset of the spark the flame will propagate across the chamber resulting in complete combustion and therefore more torque, power, lower emissions and ultimately lower fuel consumption.

At the completion of all the installation the valve was finally tuned using a lambda sensor with a digital readout to the cockpit of the vehicle. The valve was tuned to a setting of lambda 1, i.e. stoichiometric mixture. The vehicle was then taken to Auckland University to use the dynamometer and emissions measurement equipment. The test procedure was as follows:

1. Power run on petrol (WOT through RPM Range)
2. Part load runs on petrol, 25%,50% and 75%
3. Power run on CNG
4. Part load run on CNG, 25%,50% and 75%

Throughout the testing the vehicle emissions were monitored and recorded. Below are the results for the vehicle showing the difference in power output of petrol and CNG.

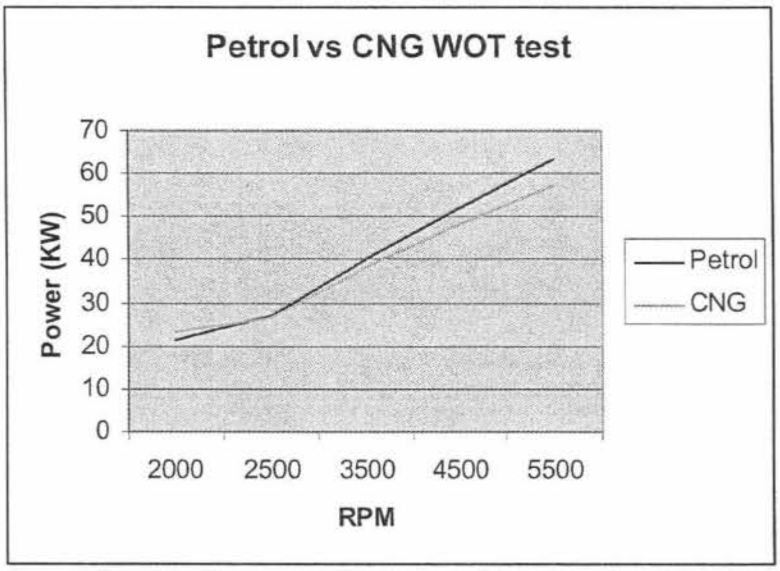


Figure 5.3: Petrol and CNG power output at wide open throttle

In the figure above it is clear to see that at 2000 RPM CNG offers a greater power output than that of petrol, however, as the RPM increases we see a steady increase in the difference of power output between petrol and natural gas. At 2000 RPM there is a 8.43% power increase and finally at 5500 RPM there is a 10.14% power loss during CNG operation. The increasing difference in power output can be largely attributed to the volumetric efficiency of the engine, and the spark timing. The spark timing during this test run was not altered to achieve maximum power and therefore will result in a power loss if the existing ignition map had been programmed for a different air/fuel ratio. This is an obvious disadvantage of having an open-loop ignition system, if information could be gathered as to the actual air/fuel ratio then the ignition modifier will have the ability to optimise the ignition timing when air/fuel ratios change during vehicle operation. This will be an area of further investigation later on.

The next load range tested was that of 50% load. At this load point a similar result was found as that in the wide open throttle test. The 50 % load test however indicated a power loss during CNG operation throughout the RPM range. At 2000 RPM the power difference was 13.5% and at 5500 RPM the final power loss was 5.4%. This is a true indication that volumetric efficiency plays a large role in the performance during natural gas operation. Due to the throttle only being open 50% and natural gas displacing 10% of air the effect of the throttle body on the volumetric efficiency is evident.

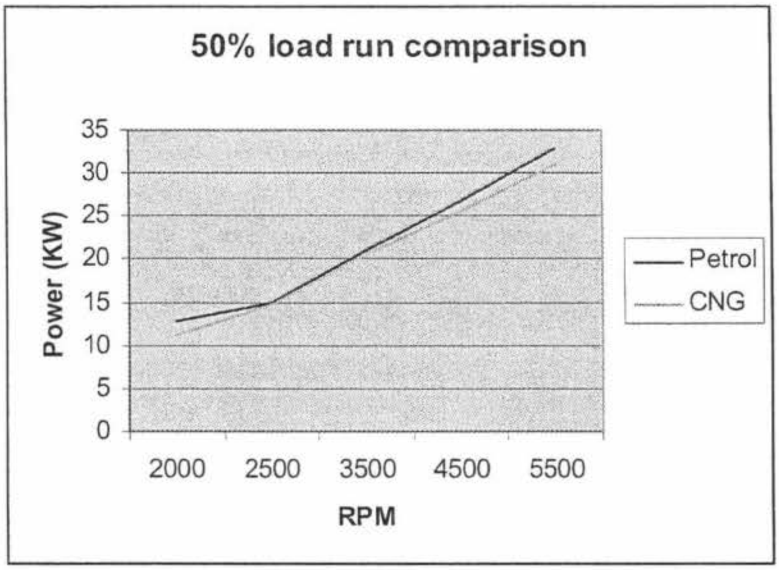


Figure 5.4: 50% load run comparison

The power output of the vehicle is obviously hindered by the volumetric efficiency and ignition timing present during testing. Below is an indication of the power output during the dynamometer testing for load runs of 25%, 50%, 75% and wide open throttle.

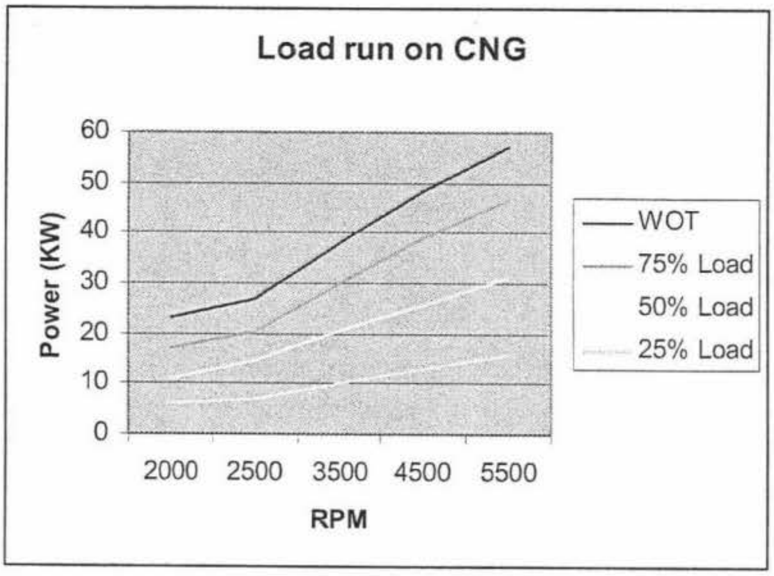


Figure 5.5: Power output from dynamometer testing through load range

Not only does the vehicle performance have to be compared with respect to power but it also has to reflect the emissions performance. The following data was recorded during the power runs and compares the exhaust emissions for petrol and CNG operation.

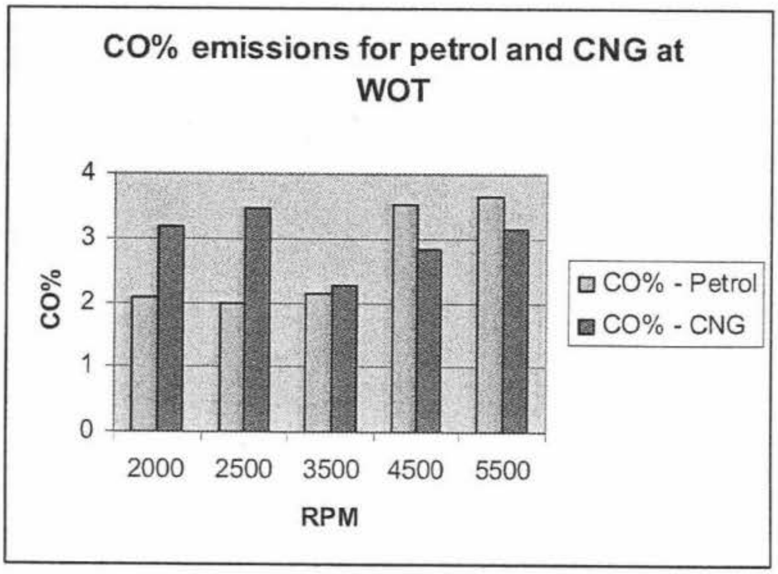


Figure 5.6: CO% emissions for petrol and CNG at WOT

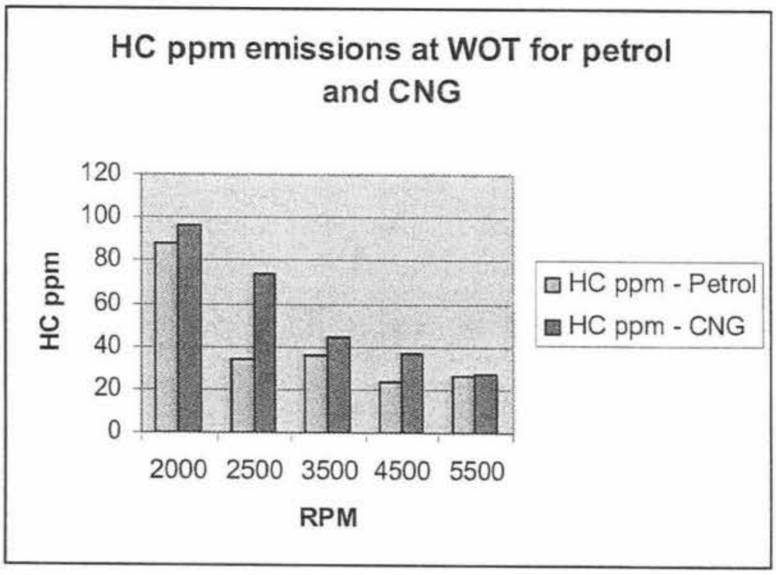


Figure 5.7: HC ppm emissions at WOT for petrol and CNG

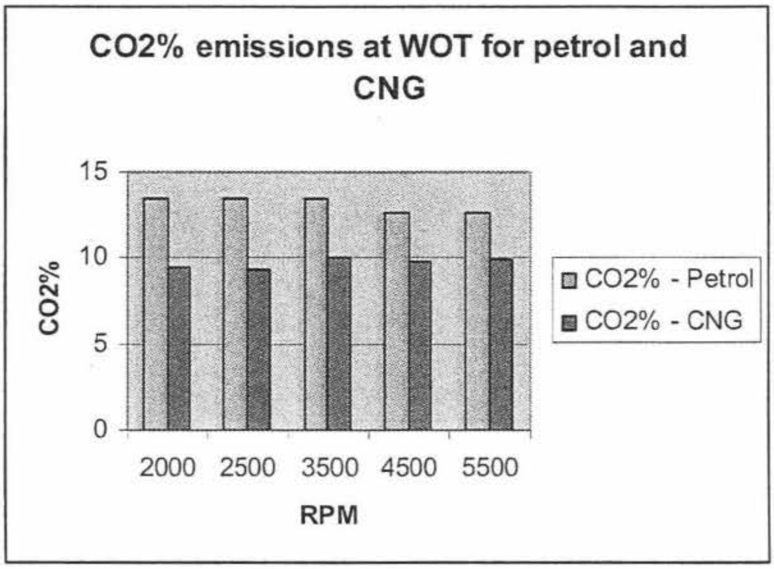


Figure 5.8: CO2% emissions at WOT for petrol and CNG

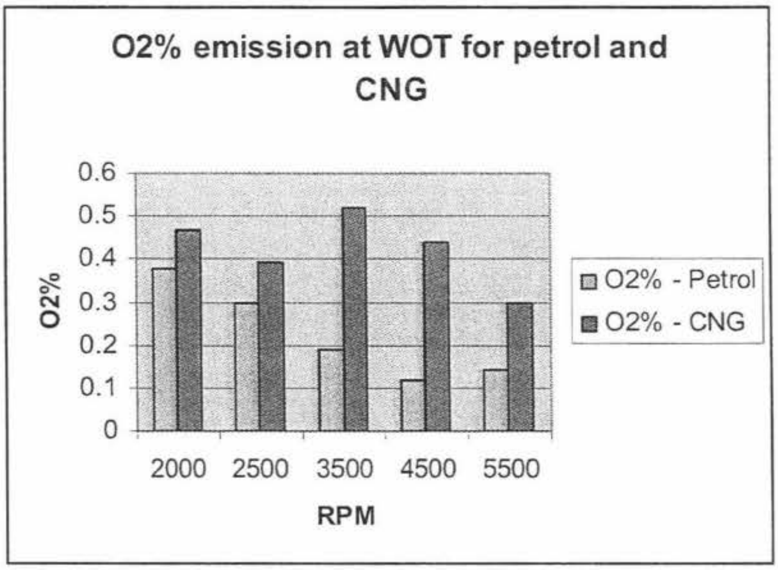


Figure 5.9: 02% emissions at WOT for petrol and CNG

As can be seen from the emissions results the vehicle was not running at stoichiometric but was rather running "rich", i.e. with a shortage of oxygen. As a result the hydrocarbons are high due to the incomplete combustion of the air/fuel and unburnt hydrocarbons being present in the exhaust, the oxygen percentage in the exhaust is low and the CO emissions are high at low throttle. Unfortunately we did not have the opportunity to optimise the fuelling during the dynamometer testing to improve the vehicles performance. At this fuelling setting the valve produced a fuel consumption of 10.94 m^3/100Km for town and highway driving.

After the results from the dynamometer testing illustrated that the vehicle was running on a rich mixture of CNG a mobile 5 gas analyser was borrowed from a company by the name of Petro-Ject and setup in the vehicle, the valve was then tuned to produce an air/fuel ratio of 19:1. The vehicle was unfortunately never taken back to the dynamometer facility; however fuel consumption tests found that the vehicle would do 250Km per fill this equates to a fuel consumption of 7 m^3/100Km. Driving of the vehicle with the fuelling optimised for emissions found that more power was available and resulted in smother driving. It was evident that the oxygen sensor used for the lambda meter was faulty and could no longer be used as a reference.

5.3. Diesel engine testing

To demonstrate the true versatility of the valve the final proof was to convert a 5.8 litre diesel engine to run on CNG. The conversion was by no means an optimisation for the engine to operate on CNG, but rather a first stage conversion to demonstrate the capabilities of the valve.

The engine that was used was a 5.8 litre, in-line 6 cylinder, turbocharged Isuzu engine. Before the engine was converted to CNG operation baseline tests were performed on diesel to determine the engine condition and performance, these can be seen in appendix 2. The tests included load runs of:

- 25%, 50%, 75% and 100%
- And also a variable load run at the 1600RPM (the indicated peak torque RPM.)

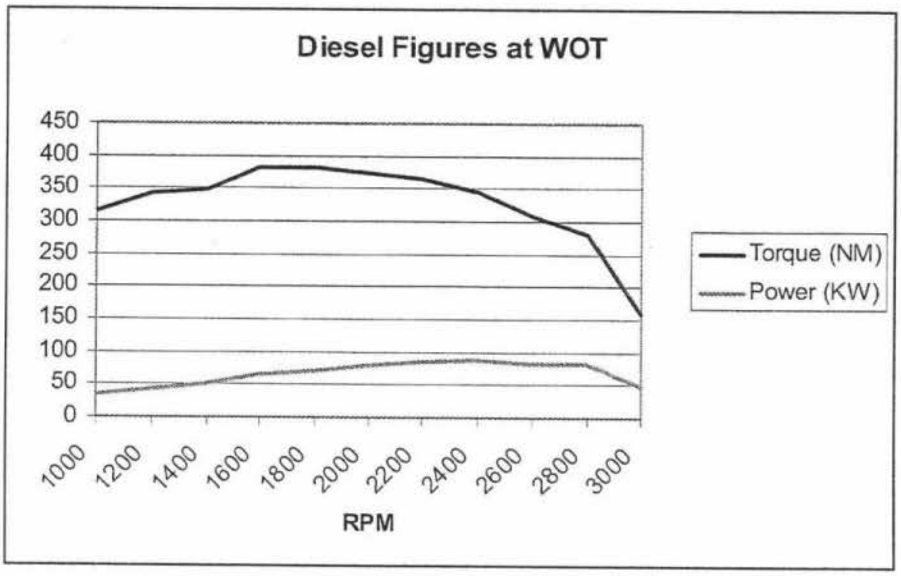


Figure 5.10: Diesel engine output at WOT

The peak torque of 381.5 Nm was achieved at 1800 RPM and a peak power output of 78.08 KW at 2000 RPM. Unfortunately no emissions results were measured during these baseline tests for comparison to CNG. On the completion of the tests the engine was removed from the engine dynamometer and prepared for conversion to natural gas operation.

5.4. Comparison of diesel and CNG performance

As mentioned before no data was taken or available for emissions when the diesel tests were done, therefore it is unfortunately impossible to make a comparison as to the emissions, it is however known that when converting a diesel to natural gas the particulate emissions is reduce by approximately 80%. The comparison between CNG and diesel operation will then have to be with relation to power and torque output only and relative fuel consumption. Below is a graph illustrating the torque and fuel consumption for both CNG and diesel.

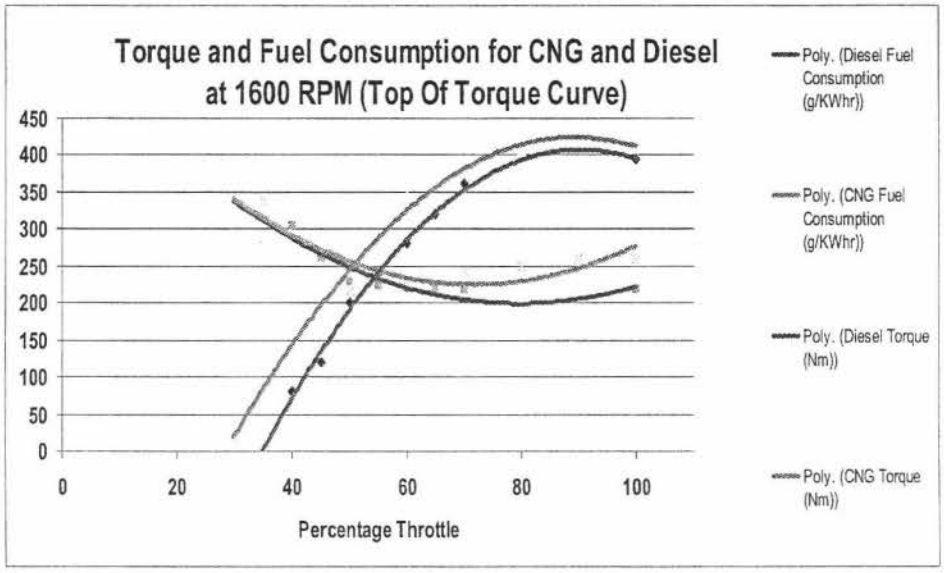


Figure 5.11: Torque and fuel consumption for CNG and diesel at 1600 RPM

As can be seen, even with an air/fuel ratio of 24:1 CNG offers more torque throughout the throttle range. It is however shown that as the throttle is opened the natural gas fuel consumption starts to increase up to a 15% difference at 100% throttle. This is due to the fact that the air/fuel ratio was kept constant during testing and the diesel altered as to the manufacturers setting. Another influence that may have had an effect on the fuel consumption and power output was the fact that the ignition could not be advanced further than 38° BTDC and thus may have resulted in complete combustion only occurring late in the piston power stroke and continuing to burn during the exhaust stroke. If further testing could be performed the difference at 100% throttle could be possibly closed by increasing the fuelling fractionally thereby dropping the specific fuel consumption or by altering the timing disks that restricted the amount of ignition advance. Unfortunately this was not possible due to time constraints.

If one then compares the 100% load data it can be seen that CNG offers more torque at approximately 1800RPM, however CNG has a steep turque curve starting much lower than diesel. This is because of the air/fuel ratio once again being kept constant through the RPM range. The power output of CNG also shows a final value greater than that of diesel however it also starts at a lower value than that of diesel at the lower RPM. The reason the CNG test data only reaches 2000 RPM is due to the fact that the timing disc used for the ignition module experienced severe oscillations at 2200 RPM thus resulting in the ECU receiving erroneous signals from the pick-up sensor, thereby resulting in misfire. Had there been more time for testing this problem would have been addressed and corrected.

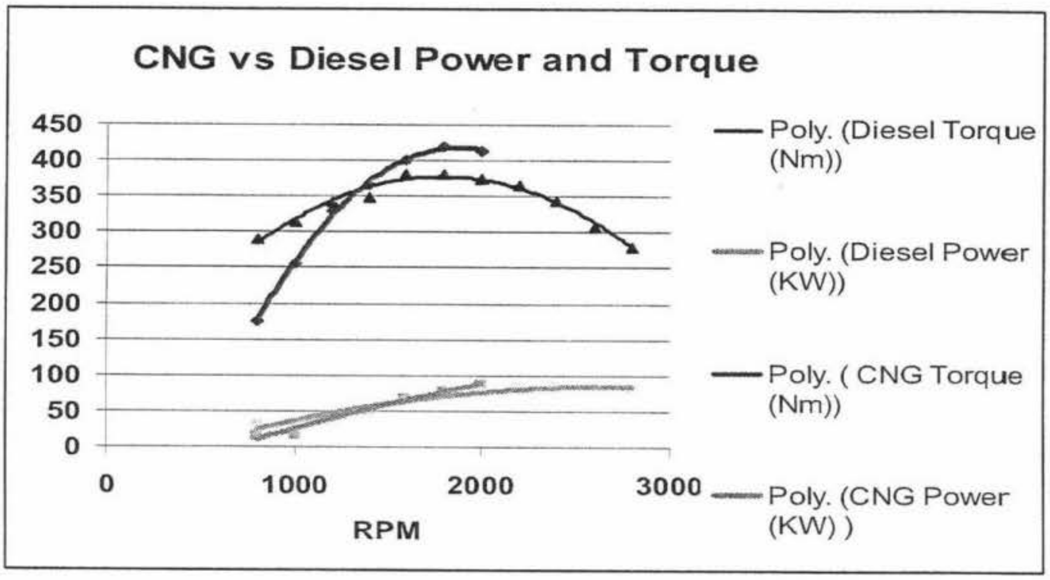


Figure 5.12: CNG versus Diesel power and torque

From the graph above it is evident that CNG operation can offer better engine performance with respect to power and torque output.

Considering the fuel consumption, the following graph shows the fuel consumption and power output comparison of diesel and CNG operation at WOT. As can be seen from the graph at low RPM the fuel consumption is incredibly high, this can be attributed to the low charge density due to low turbo pressure. The turbo pressure data is as follows:

RPM	Intake Pressure (mbar)
2000	729
1800	751
1600	589
1400	348
1200	217
1000	164
800	159

Table 5.1: Turbo pressure with respect to RPM

As can be seen from the table above and the graph below, the intake pressure below 200 mbar requires a richer air/fuel ratio. This is a clear indication that the fuelling at lower RPM from idle (780 RPM) to approximately 1200 RPM the fuelling has to be increased to approach almost stoichiometric. This fuel consumption is consistent through all load ranges tested, which also reflects the point made above.

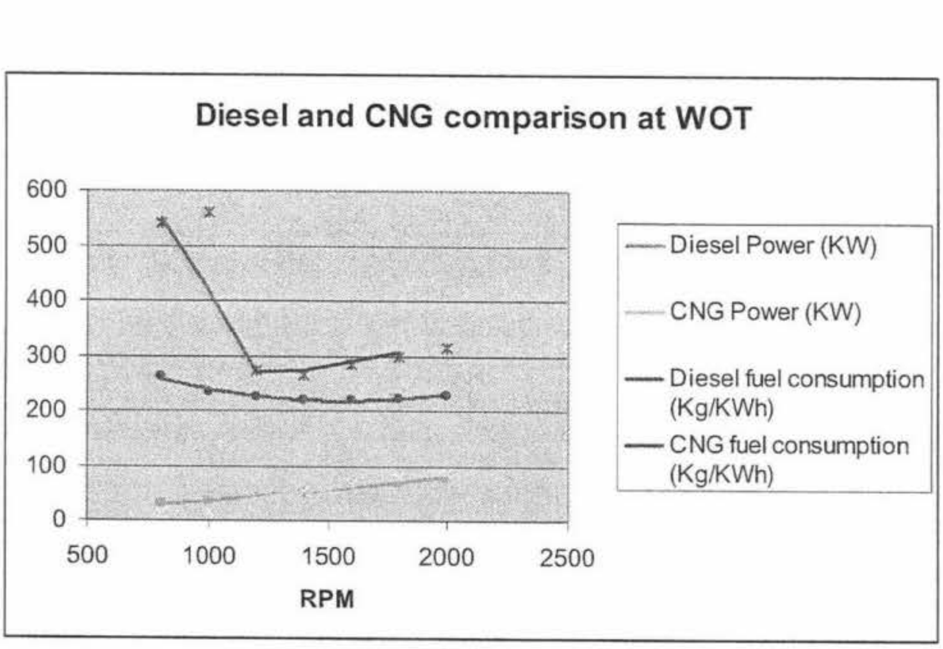


Figure 5.13: Diesel and CNG comparison at WOT

Unfortunately the goal of achieving 1:1 fuel consumption as compared to diesel was not achieved. However at 1600 RPM and approximately 50% throttle opening the specific fuel consumption of CNG was lower than that of diesel, at this point the timing would have been at the optimum at 38° BTDC therefore creating more torque and power. The timing at the point is optimum and can be illustrated by the emissions output at this point relative to the rest of the data gathered for the 1600 RPM peak toque run. The following table gives the emissions data for this graph.

Percentage Throttle	Exhaust Temp	CO	CO2	HC	O2	NO	Lambda
100	779.4	0.02	9.7	420	4.79	500	1.32
90	793.4	0.02	9.8	595	4.79	610	1.32
80	796.3	0.02	9.8	630	4.6	549	1.32
70	789.4	0.02	9.6	270	5.07	384	1.36
60	784.5	0.02	9.8	245	6.7	447	1.33
50	767.2	0.03	9.6	130	5.16	191	1.37
40	750	0.03	9.7	65	5.05	120	1.36
35	737.6	0.02	9.8	315	4.6	102	1.32
30	711.3	0.04	9.7	490	4.7	24	1.33

Table 5.2: Emissions for 1600RPM varying load run

If we look at the data presented in the table above and pay particular attention to the 40% and 50% percentage throttle rows. We see that the HC are particularly low at 40% and double at 50%, also interesting is the relatively low NOx and the attainable lambda value of 1.36 and 1.37. If one then looks at the data for 60% throttle to 90% throttle we notice a sharp increase in exhaust temperature, this can be related to combustion duration taking longer and therefore combustion taking place late in the power stroke thereby resulting in hotter exhaust gas during the exhaust stroke.

Further emissions data can be found in appendix 2.

5.5. Changes for diesel conversion

For the diesel conversion to be more successful and to approach the goal of achieving a 1:1 comparison of diesel fuel consumption to CNG consumption some areas need to be addressed. As can be seen on the graph comparing diesel and CNG torque and fuel consumption at 1600 RPM with varying throttle, CNG did in fact better diesel at the lower throttle opening percentage. This indicates that the fuelling and ignition timing were correct at that point, however as the throttle is opened the CNG consumption steadily increases and a drop in the difference of torque between the two is observed. This is an indication that there is still scope to possibly reduce or increase the fuelling, or that there is a possibility for the spark advance to be increased to ensure that complete combustion is present before the exhaust valve is opened. If the mixture is made leaner then the spark has to be advanced to compensate for the lower flame speed propagation. However due to the spark ignition system that was being used, this would require a new timing disc to be made.

During testing on the dynamometer the converted diesel engine was only able to rev up to 2000 RPM, this was because of misfire occurring at this point. Initially it was thought that the misfire occurred due to the high ignition requirement to ignite the air/fuel mixture, however it was found that the misfire occurred regularly independent of load and air/fuel ratio. After investigation it was found that the ignition timing disks collided with the pick-up sensor thereby creating a false signal and thus creating a misfire. During the conversion of the diesel engine the timing disks were mounted on the diesel-pump drive shaft because it rotated at half engine speed and also because it was easily accessible due to the diesel-pump not required during CNG operation. This was explained in section 3.3.4.

Due to the age and wear of the engine the bearings used to support the diesel-pump shaft had worn and therefore had play, i.e. the shaft had vertical and horizontal movement, approximately 1 mm of play was found in the bearings. Added to this movement was the fact that the timing boss that had the timing disk bolted to was not 100% symmetrical with the pump shaft. Therefore as the engine was slowly rotated the gap between the timing disks would vary from a minimum of 0.4mm to a maximum of 0.6mm. The sensor from Altronic Inc. required that the sensor be mounted 0.5 mm from the timing disk with a tolerance of 0.12mm, therefore the sensor was at it most extreme limits of sensing. It was finally found that when the engine approached 2200RPM the vertical oscillations in the shaft due to the worn bearings caused the timing disks to collide with the sensor. When this collision occurred the ignition ECU calculated that the engine speed was approximately 11,000 RPM and therefore resulted in a system fault due to multiple ignition requirements during a single event. This was an unfortunate occurance that hindered testing. To prevent this from occurring again a new timing assembly was developed for future testing. This new timing assembly features a flexable coupling that allows for shaft oscillations while the timing disc are housed in a fixed location within a sensor housing, the housing feature the same mounting points as that used for the sensor brackets used before.. The figure below shows the new design for the timing disc assembly.

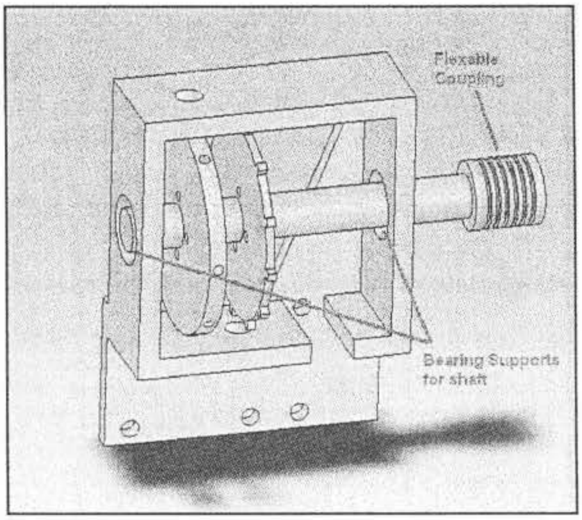


Figure 5.14: New Timing disk assembly with flexible coupling and bearings

Another area that could be considered for improvement is the intake manifold. The manifold used for the diesel engine is not conducive to smooth flow for air; the manifold consists of right angles that restrict flow. If the manifold were to be removed, scope lies in the implementation of the water-to-air intercooler to be used as the intake plenum and having mandrel bent intake runners built to form the intake manifold. This will then allow one to tailor the length of the intake runner so that the volumetric efficiency may be optimised at a certain RPM due to the ramming effect caused by the intake pulsations present in the intake manifold. Using mandrel bent intake runners will allow for smooth laminar flow within the intake system and thus increasing the volumetric efficiency. This method would not only increase the volumetric efficiency but also allow for a better distribution of air/fuel mixture to all the cylinders, unlike the manifold used for diesel operation. This new configuration will also help to reduce the overall size of the intake system, thereby reducing the volume of combustible air/fuel mixture present in the intake system if in the occasion of a back fire.The reduction of the inlet chamber volume would aid in the response of the engine when the demand requirements alter.

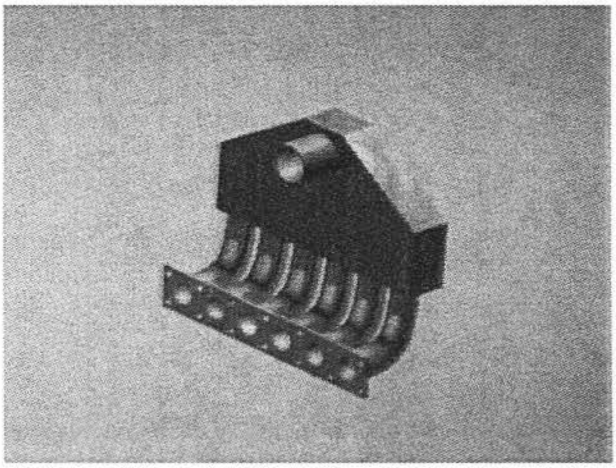


Figure 5.15: Integrated intercooler and plenum

To help increase the volumetric efficiency of the engine attention may be aimed at the intake valves. The valves diameter may be increase thereby increasing the orifice area for air/fuel mixture to flow through, also the valve timing may be changed to induce more scavenging thereby aiding to increase the volumetric efficiency.

Chapter 6: Future developments

Due to the mechanical nature of the valve some problematic issues arise. Throughout the valve components need to be balanced to ensure proper operation of the valve, e.g. spring coefficients have to be calculated to ensure the valve responds to pressure changes accurately, proportionally and with a good response time. This poses problems in the repeatability and the ability of the valve to maintain precise control of gas to the engine in that if a minor mistake is made during the construction of the valve and components are not correctly seated, lined up or located, the performance of the valve will be dramatically compromised thus requiring disassembly and a check to be done to determine the cause of the problem and what actions have to be taken to rectify the problem. During operation of the valve any minor particles entering with the inlet gas may be lodged in a crucial orifice therefore causing a dramatic change in the performance of the valve in terms of the response and also the operating pressures. These are some of the problematic areas of the valve that have to be addressed.

As mentioned before the valve is a coming together of proven mechanical measurement devises and modern day electronic control equipment to offer a system that can be tailored to one's requirements. The method used to reference the electronic signal is an electrical to mechanical conversion using a pulse width and frequency modulated injector solenoid that is driven by the vehicles onboard computer. This converts a pulse width and frequency modulated signal from the ECU to a pressure signal used to measure the precise amount of gas required for the given engine operating condition. This pressure reference signal is then amplified using a two stage valve assembly that meters the required fuel for the engine operating condition. This method has one major area of concern in that due to the variation of frequency and amplitude of the signal being supplied to the injector solenoid an equivalent pressure signal in both frequency and amplitude is experienced in the reference chamber thus causing sporadic flow when high duty cycles are present with low operating frequencies. The method used to relieve these sporadic pressure rises is to use a damping chamber to absorb the pressure spikes present in the reference chamber. The damping system consists of a diaphragm with one side connected to the reference signal and the other connected to a spring. This only allows for a small range of frequencies to be damped due to the natural frequency response of the damper that is determined by the spring coefficient and the area of the diaphragm. An improvement in the configuration of the damper system would be to develop a system that alters the time response as the frequency of the reference signal changes thus to ensure the precise measurement of the gas to the engine.

Another area of concern is the use of adjustable bypass valves to relieve the pressure build-up in the reference chamber and the feedback chamber on the second stage regulator. If any dirt or other particles find their way into these fine valves and block the orifices the valve may experience a dramatic change in the operating pressure and the valve output flow. An example of this would be that if the bleed-off valve for the feedback on the second stage regulator was to get blocked the valve would experience a increase in operating pressure equal to the pressure build up under the feedback diaphragm, this would result in a increased flow rate as the valve operates under a chocked flow condition where the mass flow rate is only dependant on the upstream pressure. The valve would then oversupply gas to the engine and the engine will be running in a rich condition resulting in increased fuel consumption and also possibly higher exhaust emissions. If the feedback becomes fully blocked the valve operating pressure would continue to increase until the operating pressure in the valve reaches the input pressure to the second stage regulator of approximately 1 MPa resulting in damage to the internal diaphragms and ultimately the failure of the valve. The possible use of a fine particle filter will lessen the chances of this occurring. The filter may be place on the inlet to the second stage regulator or further upstream on the first stage regulator. These fine bleed valve are sourced from an outside manufacturer, this is not desirable if the valve is to be put into production. The focus is now on the development of a valve that will have the fine bleed off valve incorporated in the body of the valve, this will not only result in a cleaner looking product but will also allow one to setup the valve in the final stage of manufacturing and seal the access to the fine adjustment valves so that the public are not able to gain access to alter the valves operation and consequently damage the valve. If an adjustment is required a special tool may be used to make the adjustment, the valve will have to be taken to an authorised dealer or installation establishment to have the adjustments made and thus maintain the warranty.

Due to the mechanical nature of the valve all the component characteristics have to be matched to each other to allow for an accurate response from the valve. If however there are any discrepancies between any components due to either manufacturing faults or due to bad assembly, tuning the valve response accurately becomes very difficult and the valve may experience hysteresis of fluctuations during operation. Unfortunately there is little one can do to avoid these situations from occurring; however there are simple steps that can be taken to minimise the risks of these situations occurring.

To produce this valve so that it may be sold to the general public steps need to be taken to lesson the complexity and also decrease the number of parts involved in the assembly. One of the first issues to address is the design and method of production of the valve body; the general idea is to have to body injection moulded from a high grade automotive plastic such as that found in modern day vehicle intake manifolds, e.g. Rynite from the company DuPont. This body would then incorporate the fine bleed-off valve seats, main valve control seat and also the output power valve body. The valve consists of three main body components:

1) The main body
2) The Top (CNG or LPG)
3) The Bottom

The top would require two designs; one for CNG and one for LPG. The LPG unit requires an evaporator unit on the inlet to evaporate the LPG into it's gaseous form so that It may be metered through the valve. The CNG valve only requires a shut off solenoid valve on the inlet as the fuel is already in a gaseous state and does not need to change state.

The bottom is universal to both LPG and CNG, housing the damping unit, the reference chamber and adjustment mechanism and lastly the second stage regulator pressure setting adjustment mechanism.

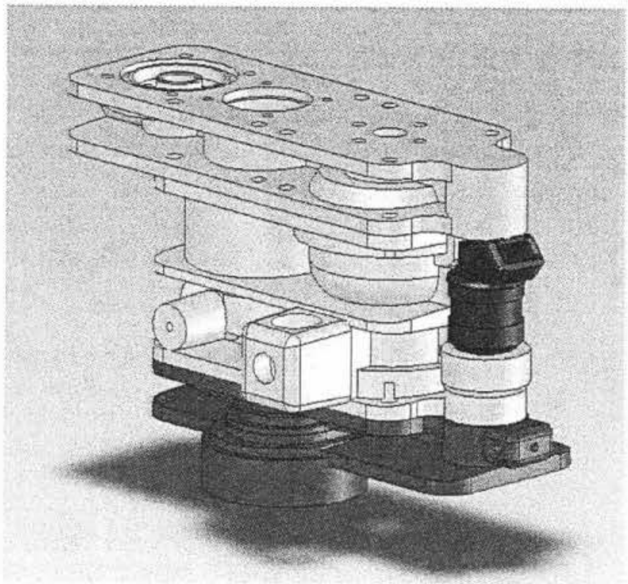


Figure 6.1: Valve prototype

The second step would be to redesign the internal components into subassemblies that are simple to replace in event of a malfunction in the valve. The valve currently consists of approximately 50 individual components that have to be machined to tolerances of ± 0.05mm, this makes it crucial that all components are made with the greatest amount of care and are best made using numerically controlled machines such as CNC and 4 or 5 axis milling machines that product components from software programs. This is not conducive to high production runs and is very time consuming and therefore results in high production costs. It is therefore an essential step to reduce the complexity of the valve if it is to be put into production. The main areas that will be concentrated on will be the:

1) Main control valve
2) Second stage regulator (for CNG only)
3) Reference chamber assembly

In the design of these subassemblies it enables the manufacturer to not only reduce the complexity of the valve but also enables the manufacturer to make sealed units, i.e. the subassemblies once manufactured are sealed and cannot be opened by the user so as to determine the how the valve operates. This area of design will have to be carefully considered so that the valve still maintains its precise control of gas but reduces the amount of time required to produce all of the components and also in creating subassemblies that if they are substituted that the valve will still operate correctly. It is therefore important that the construction of the subassemblies is designed so that the precision and quality of the components can be easily monitored to ensure that there is little if no variation between individual subassemblies. The design of the subassemblies should be such that if a failure occurs in a valve, it is then a simple matter of removing the failed component and replacing it with a new subassembly. This will not only reduce the turnaround time for valve maintenance but will also reduce the manufacturing time and lesson the possibility of variations between different valves.

Both the CNG and LPG valves are still in the development phase and are at present still being refined to make them more responsive, less critical and more user friendly. The future developments discussed above are being considered and are currently in the design phase with manufacturers of injection moulded products advice being sought so as to develop a design that will satisfy all the conditions described above. Once all the subassemblies and valve bodies have been constructed in the manner suggested above, the valve will become a serious competitor in the market for aftermarket retrofit alternative fuel devices and also be very popular in the use of diesel engine conversions.

Chapter 7: Conclusion

CNG is seen as the most promising alternative fuel at present, containing between 80% and 95% methane and some other heavier hydrocarbons. Methane has a simple structure and therefore better emissions after combustion, particulate matter emissions are reduced by approximately 80-95% and NO_x emissions are reduced by 50-80%. Natural gas also has a very high anti-knock value which allows one to increase the thermal efficiency of the engine. Recent surveys have found that the world resources of natural gas are estimated to be 6358.575 trillion cubic feet. At today's current level of consumption this is estimated to last for the next 70 years.

Due to methane having a high octane a rating of approximately 120 and also the extended flammability limits of 5-15%, natural gas is well suited to lean burn operation. The successful conversion of a diesel engine using the Harrison CNG control valve enabled the engine to operate at $\lambda = 1.4$, with the ignition system limiting further leaning. At $\lambda = 1.4$ NO_x emission are reduced as compared to when the engine is operated at $\lambda = 1$. The use of after treatment equipment such as an oxidation catalyst will aid in the reduction of NO_x emissions. The conversion of the sedan to operate as a bi-fuel vehicle had the desired results in achieving vehicle operation with minimal user input and minimal effects on engine performance with respect to engine power and drivability of the vehicle. Increased range of the vehicle was also realised with the precise measurement of the air/fuel ratio enabling the vehicle to increase the drivable range on a fill of natural gas by approximately 60km.

One of the major disadvantages of natural gas is its low energy density, therefore requiring large heavy storage cylinders. This can be overcome by liquefying natural gas, however natural gas only liquefies at -162°C thereby requiring cryogenic freezing which increases complexity of the system and also the cost. The addition of heavy steel storage tanks to a petrol vehicle alters the dynamics of the vehicle requiring modifications to the suspension. Due to the limited free space in a vehicle the maximum cylinder storage volume was limited to approximately 60 litres. This reduced the driving range of the vehicle when operating on natural gas to approximately 250km depending on driving conditions. This limited range requires that a well established network of refuelling stations be in place to ensure that natural gas will become a viable alternative fuel. However due to the vehicle retaining bi-fuel operation the vehicle user may switch from CNG to petrol operation when the need arises.

Unfortunately natural gas has no future in New Zealand as natural gas supplies have been depleted and no future exploration for natural gas fields is being undertaken. However natural gas has seen a resurgence in Europe, Asia and the U.S.A. With ever stringent emissions laws natural gas offers the best emissions performance when compared to other hydrocarbon fuels. With the use of oxidation catalysts designed to operate on natural gas engines and the design of other after-treatment systems to minimise CH_4 emissions, natural gas may become a viable alternative fuel for heavy-duty vehicles that operate on a circuit, i.e. on a route that will allow for refuelling stations to be placed in the route at appropriate points.

After market natural gas conversions are becoming less prominent as OBD systems become more complex and with emissions governing regulations requiring that OBD systems remain operational even during alternative fuel operation. Alternative fuel equipment manufacturers will therefore have to work closely with OEM vehicle manufacturers to develop vehicles that can offer bi-fuel operation.

The thesis detailed the mathematical analysis done to develop the valve and to improve performance. It also detailed the use of simulation software to analyse the flow patterns within certain components of the valve enabling design considerations to be easily made and implemented and thus reducing the development time of the valve.

The successful conversion of both a 2 litre sedan and a 5.6 litre diesel engine illustrates the true versatility of the valve. The valve is a true modular system, enabling the conversion of almost any size engine, with large capacity engines being fuelled by two or more valves to cater for the required fuel flow rate.

The major disadvantage of the valve is due to its mechanical complexity, consisting of approximately 50 individual parts. This area results in high manufacturing tolerances and high manufacturing costs. Future development of the valve must be aimed at the reduction of components and simplification of the valve operation. This will enable the valve to become more reliable and also reduce manufacturing costs when the valve enters production. The ability of the valve to electronically control the air/fuel ratio according to the OEM programmed map so that it mimics petrol operation or for the valve to be controlled by a relatively inexpensive ECU sets the valve apart from other currently available on the aftermarket retrofit equipment market. The ability of the valve being able to control such a large variety of engines and the ease of installation also sets it apart from the competition. With further development the valve may become a viable alternative fuel conversion solution for pre 2004 model electronic controlled petrol vehicles.

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Appendix A

Fuel properties data sheet.

Fuel Property	Gasoline	No. 2 Diesel fuel	Methanol	Ethanol	M85	E85	Natural Gas	Propane	Soybean Methyl Ester	Hydrogen
Formula	C_4 to C_{12}	C_8 to C_{25}	CH_3OH	C_2H_5OH			CH_4	C_3H_8	C_{18} to C_{19}	H2
Molecular weight	100 – 1-5	200	32.04	46.07			16	44.09	300 approx.	2.02
Composition:										
Carbon	85-88	84-87	37.5	52.2	43-45	56-58	75	82	78	0
Hydrogen	12-15	13-16	12.6	13.1	12-13	13-14	25	18	11	100
Oxygen	0-4	0	49.9	34.7	43-44	29-30	0	0	11	0
Density (kg/l)	0.69-0.79	0	0.796	0.79	0.79-0.8	0.78-0.79		0.50	0.87	0.0013(g), 0.007(l)
Specific gravity (relative density) 5/15°C	0.69-0.79	0.81-0.89	0.796	0.794	0.79-0.8	0.79-0.8		0.5	0.87	0.07(g), 0.07(l)
Freezing point °C	-40	-40 to -1	-97.5	-114	-	-	-182	-187		-275
Boiling point °C	27-225	188-343	65	78	49-66	49-80-	-162	-42	-	-253
Vapour pressure, kPa @ 38°C	48-103	<1	32	15.9	48-103	38-83	Not applicable	1303	<1	Not applicable
Specific heat, KJ/(kg-K)	2.0	1.8	2.5	2.4	2.4	2.3	-	2.48	-	14.2
Viscosity, mPa-s @ 20°C	0.37-0.44	2.6-4.1	0.59	1.19	0.55-0.57	1.07-1.08	-	0.102	3-6	0.009
Water solubility, 21 °C	Negligible	Negligible	100	100	100	100	Negligible	0.065	-	Negligible

Water in fuel, Vol%										
Electrical conductivity, mhos/cm	$1x10^{-14}$	$1x10^{-12}$	4.4x10-7	$1.35x10^{-9}$	-	-	-	-	-	-
Latent heat of vaporation, KJ/kg	349	233	1178	923	1055	836	510	426	-	448
Lower heating value, 1000KJ/L	30-33	35-37	15.8	21.1	17.9-18.3	22.4-22.9	6.6 @16,600 kPa	23	32	8.4
Flash point °C	-43	74	11	13	Slightly warmer than gasoline	Slightly warmer than gasoline	-188	-104	-	-
Autoignition Temperature, °C	257	316	464	423	>gasoline	>gasoline	540	457	-	-
Flamability limits, Vol %					Wider than gasoline	Wider than gasoline				
Lower	1.4	1.0	7.3	4.3			5	2.1	-	49
Higher	7.6	6.0	36.0	19.0			15	9.5	-	75
StoichiometriAir/fuel ratio, weight	14.7	14.7	6.45	9.00	7.7	9.9	17.2	15.7	-	34.3
Flame spread rate, m/s	4-6	-	2-4	-	Slower than gasoline	Slower than gasoline	Not applicable	Not applicable	-	-
Flame visibility	Visible in	Visible in	Invisible in	Difficult to	Initially	Initially	Visible in	Visible in	Visible in	Invisible in

	all conditions	all conditions	daylight	see in Daylight	good, decreases	good, decreases	all conditions	all conditions	all conditions	direct sunlight
Octane Number										
Research	88-100	-	108.7	108.6	108	107	120 (est.)	112	-	-
Motor	80-90	-	88.6	89.7	89	89	120 (est.)	97	-	-
Cetane number	-	40-55	-				-	-	52	-

Appendix B

Diesel engine conversion results.

Load runs

CNG Load run test results

25% load

RPM	Torque	Power	Blowby	Intake Pressure	Intake Temp	Exhaust Temp	CO	CO2	HC	O2	NO	Lambda	Fuel Consumption (kg/Hr)	Relative Fuel Consumtion (g/KWHr)
2200	119	27.4	42.3	260	30.2	804.4	0.03	9.6	380	4.8	86	1.33	13.5	492.7007
2000	139.8	29.15	44.1	232	30.2	794.7	0.03	9.8	345	4.73	106	1.33	12.6	432.247
1800	160.4	30.11	43.3	205	29.9	777.6	0.03	9.7	325	4.6	133	1.33	11.3	375.2906
1600	186.2	31.03	43.8	164	29.7	745.8	0.03	9.8	310	4.58	260	1.32	10.5	338.3822
1400	200.9	29.36	45.8	130	29.4	720.4	0.02	9.6	310	4.8	280	1.33	9.5	323.5695
1200	194	24.31	45.5	89	29.1	675.2	0.02	8.9	740	6.07	99	1.43	7.6	312.6285
1000	133.3	13.95	45	88	29	718.8	0.03	8.9	505	6.21	68	1.46	7	501.7921
800	107.7	8.88	46	106	28.9	733.8	0.03	9.3	415	5.52	285	1.4	6.2	698.1982

50% load run

RPM	Torque	Power	Blowby	Intake Pressure	Intake Temp	Exhaust Temp	CO	CO2	HC	O2	NO	Lambda	Fuel Consumption (kg/Hr)	Relative Fuel Consumtion (g/KWHr)
2000	388.8	81.39	80.8	721	36.7	844.6	0.03	10	250	4.3	567	1.3	23.8	292.4192
1800	375	70.36	79	709	36	824	0.03	9.6	280	5	371	1.35	24.2	343.9454
1600	353.6	58.91	69.5	491	33.4	783.2	0.03	9.6	350	5	400	1.35	21	356.476
1400	320	46.34	62.1	299	31.4	743.6	0.03	9.5	400	5.28	443	1.36	16.5	356.0639
1200	310.5	38.76	57.3	190	30.4	706.7	0.03	9.5	520	5.35	575	1.35	10.5	270.8978
1000	218.9	22.8	55	150	29.9	743.1	0.04	9.6	530	5.12	580	1.33	8.9	390.3509
800	181.3	15.25	53.1	156	29.9	757.9	0.05	9.9	470	4.6	1000	1.3	7.7	504.918

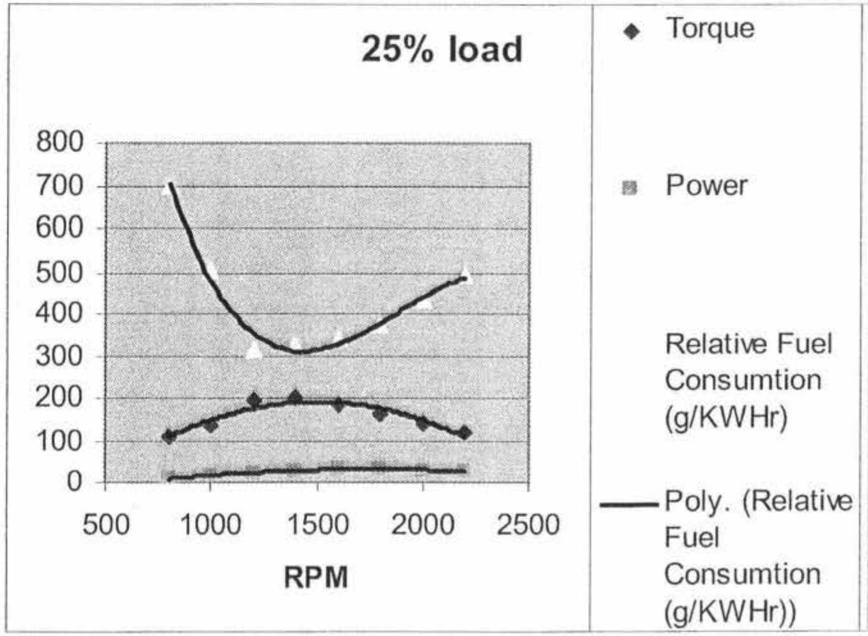
75% load run

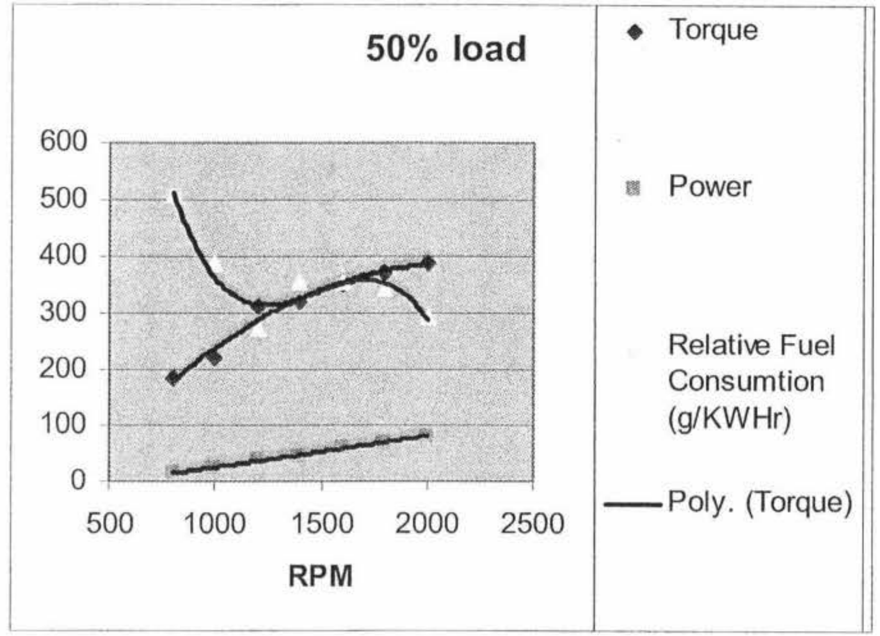
RPM	Torque	Power	Blowby	Intake Pressure	Intake Temp	Exhaust Temp	CO	CO2	HC	O2	NO	Lambda	Fuel Consumption (kg/Hr)	Relative Fuel Consumtion (g/KWHr)
2000	410.2	85.14	77.9	734	36.6	837.6	0.03	9.7	750	4.9	558	1.32	25.6	300.6812
1800	409.9	76.41	82.7	759	36.9	821.9	0.03	9.5	1300	5.14	370	1.33	23	301.0077
1600	381.8	63.21	76	573	34.7	788.9	0.03	9.4	460	5.4	318	1.38	18.5	292.6752
1400	356.9	51.17	67.4	342	32.5	755.5	0.03	9.6	510	5.15	525	2.35	14.6	285.3234
1200	330.8	41.39	57.3	214	31.3	723	0.04	9.6	430	5.09	860	1.35	11.2	270.5968
1000	253.6	26.46	56.9	165	30.4	754.6	0.05	9.9	420	4.6	850	1.3	9	340.1361
800	206.3	17.23	54.8	160	30.1	770.3	0.06	10.5	400	3.5	2000	1.2	7.8	452.6988

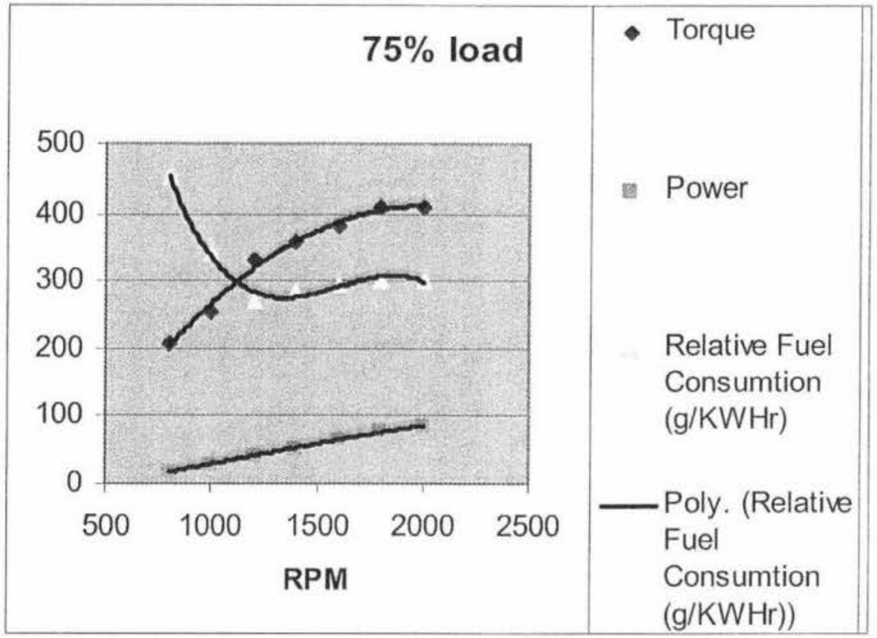
100% load run

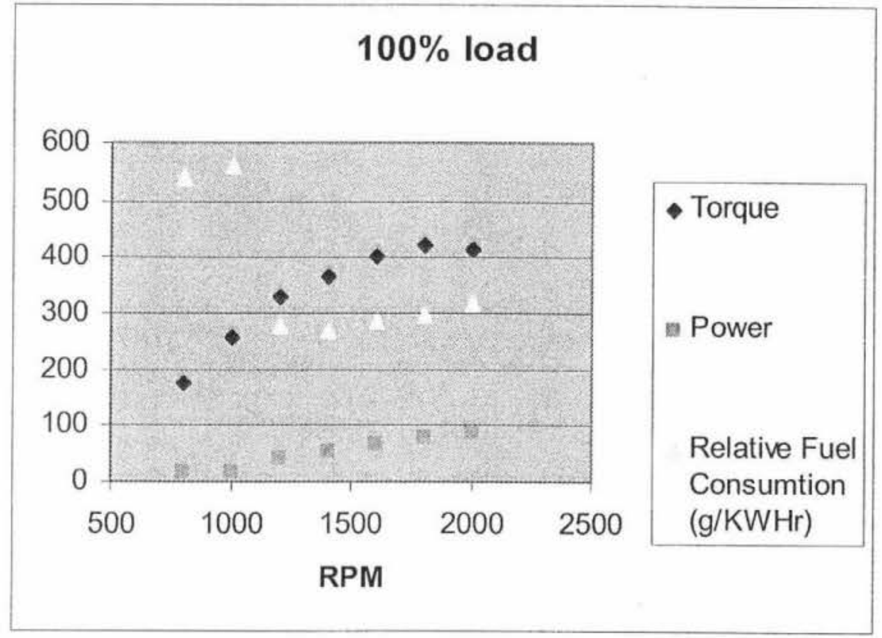
RPM	Torque	Power	Blowby	Intake Pressure	Intake Temp	Exhaust Temp	CO	CO2	HC	O2	NO	Lambda	Fuel Consumption (kg/Hr)	Relative Fuel Consumtion (g/KWHr)
2000	414.2	86.42	79.4	729	37.8	842.7	0.04	10	740	4.45	630	1.28	27.2	314.742
1800	420.6	78.72	82.8	751	37.5	826.5	0.04	9.8	460	4.8	460	1.32	23.4	297.2561
1600	399.8	66.4	77.6	589	35.5	790.6	0.04	9.6	410	5.28	483	1.34	18.8	283.1325
1400	365.1	52.6	67.2	348	32.4	752.1	0.04	9.7	430	4.94	750	1.33	14	266.1597
1200	328	41.97	58	217	31.3	715	0.04	9.6	500	5.05	837	1.33	11.5	274.0052
1000	255.7	16.94	54	164	30.5	714.2	0.05	9.9	365	4.61	880	1.3	9.5	560.8028
800	175.4	14.81	54	159	30	751.1	0.06	9.8	425	4.88	860	1.32	8	540.1756

1600 RPM Varying Load														
Percentage Throttle	Torque	Power	Blowby	Intake Pressure	Intake Temp	Exhaust Temp	CO	CO2	HC	O2	NO	Lambda	Fuel Consumption (kg/Hr)	Relative Fuel Consumtion (g/KWHr)
100	420.1	69.98	78.1	582	31.7	779.4	0.02	9.7	420	4.79	500	1.32	19.5	278.651
90	415.7	68.7	72	609	33.4	793.4	0.02	9.8	595	4.79	610	1.32	19	276.5648
80	406.8	67.75	74.1	609	44.6	796.3	0.02	9.8	630	4.6	549	1.32	18.2	268.6347
70	377.4	63.02	71.9	561	33.3	789.4	0.02	9.6	270	5.07	384	1.36	16.5	261.8216
60	345.2	57.67	64.7	459	32.8	784.5	0.02	9.8	245	6.7	447	1.33	12.9	223.6865
50	242.9	40.73	52.4	294	31.4	767.2	0.03	9.6	130	5.16	191	1.37	8.8	216.057
40	137.4	23.3	36.8	121	30.3	750	0.03	9.7	65	5.05	120	1.36	6.8	291.8455
35	84.7	14.21	29.4	55	29.4	737.6	0.02	9.8	315	4.6	102	1.32	4.8	337.7903
30	20.1	3.31	22	14	29.7	711.3	0.04	9.7	490	4.7	24	1.33	4.2	1268.882


25% load
800
700
600
500
400
300
200
100
0
500
1000
1500
2000
2500
RPM
Torque
Power
Relative Fuel Consumtion (g/KWHr)
Poly. (Relative Fuel Consumtion (g/KWHr))


50% load
600
500
400
300
200
100
0
500
1000
1500
2000
2500
RPM
Torque
Power
Relative Fuel Consumtion (g/KWHr)
Poly. (Torque)


75% load
500
400
300
200
100
0
500
1000
1500
2000
2500
RPM
Torque
Power
Relative Fuel Consumtion (g/KWHr)
Poly. (Relative Fuel Consumtion (g/KWHr))


100% load
600
500
400
300
200
100
0
500
1000
1500
2000
2500
RPM
Torque
Power
Relative Fuel Consumtion (g/KWHr)

Diesel load tests

100% load

TBTMP	SPEED	TORQUE	P	ALPHA	BH	FUELCOS	BLOWBY	P_BARO	P_EC	P_EXH	P_INTAKE	P_OIL
	rpm	Nm	kW	%	kg/h	g/kWh	l/min	mbar	kW	mbar	mbar	Bar
0903732	2991	158	49.49	100	17.76	358.04	91.4	1008.9	55.87	41	864	3.89
0903838	2789	279.9	81.75	100	23.25	280.81	83.3	1009	92.73	43	961	3.57
0903937	2591	307.4	83.4	100	22.12	265.69	84.8	1009	94.51	37	955	3.3
0904034	2392	345	86.43	100	21.94	253.12	78.1	1009	97.64	30	924	3.1
0904131	2195	366	84.14	100	20.09	238.06	78.3	1009	94.13	22	784	2.88
0904239	1996	373.5	78.08	100	17.89	228.51	72.4	1009	86.2	13	613	2.6
0904333	1797	381.5	71.79	100	15.98	223.28	65.3	1009	78.34	12	467	2.42
0904430	1597	381	63.73	100	14	220.72	61.3	1009.1	68.71	5	345	2.18
0904538	1397	348.8	51.04	100	11.23	220.31	62.1	1009.1	54.35	8	227	1.95
0904633	1199	342.6	43.02	100	9.74	226.38	59	1009.1	45.41	6	155	1.76
0904730	998	314.1	32.77	100	7.64	232.59	55.7	1009.1	34.35	-1	92	1.52
0904816	801	375.5	31.52	100	8.26	262	59.7	1009.1	32.99	2	95	1.21

	SPEED	REDFAC	T_AIR	T_EC	T_EXH	T_EXH_BT	T_INTAKE	T_OIL	TWO
	rpm		Cel	Nm	Cel	Cel	Cel	Cel	Cel
	2991	1.129	34.2	178.4	349.2	478.4	97.4	95.6	78.9
	2789	1.134	34.7	317.53	410.7	549.5	101	101.7	82.2
	2591	1.133	34.8	348.27	409	533.3	100.2	108.3	80.9
	2392	1.13	35.2	389.8	424.4	535.3	98	108.3	82.4
	2195	1.119	35.4	409.42	428.3	533.9	90.9	109.1	80.5
	1996	1.104	35.9	412.31	429.9	536.8	81.3	109.7	80.9
1800	1797	1.091	36	416.28	433.8	550.2	73.1	107.5	78.9
	1597	1.076	36	410.82	433.3	545.8	65	107.8	82.4
	1397	1.065	35.5	371.47	411	510.6	56.8	106.7	85.4
	1199	1.056	35.5	361.61	399.4	496.3	51	105.1	83.2
	998	1.046	35	329.26	377.6	455.1	46.5	103.6	80.6
	801	1.047	34.8	393.04	398.4	518	45.5	101.3	82.6

25% load

$TIMESTMP	SPEED	ALPHA	TORQUE	P	BH	BLOWBY	FUELCOSP	P_BARO	P_EC	P_EXH	P_INTAKE
	rpm	%	Nm	kW	kg/h	l/min	g/kWh	mbar	kW	mbar	mbar
50908202415	2797	77.6	78.8	23.07	11.55	78.8	519.36	1009.6	25.39	22	605
50908202459	2595	77.6	211.1	57.37	16.11	85.9	281.33	1009.6	63.65	23	733
50908202550	2391	77.6	319.7	80.07	19.94	80.4	250.27	1009.6	89.59	24	860
50908202636	2196	77.6	358.2	82.38	19.79	79	239.2	1009.6	91.99	19	813
50908202729	1992	77.6	370.4	77.25	17.61	72.9	228.83	1009.5	85.38	16	646
50908202810	1798	77.6	378.8	71.31	15.88	63.4	222.52	1009.6	77.99	9	502
50908202907	1599	77.6	384.7	64.41	14.2	63.7	219.91	1009.5	69.54	4	375
50908202958	1404	77.6	343.6	50.51	11.15	57.4	219.75	1009.6	53.89	4	241
50908203133	1199	77.6	339.4	42.61	9.68	57.6	226.36	1009.6	44.99	0	164
50908203227	1001	77.6	314.4	32.95	7.64	54	233.13	1009.6	34.54	-5	100
50908203321	793	77.6	274.5	22.8	5.47	50.3	237.79	1009.5	23.79	-2	62
	SPEED	P_OIL	REDFAC	T_AIR	T_EC	T_EXH	T_EXH_BT	T_INTAKE	T_OIL	TWO	
	rpm	Bar		Cel	Nm	Cel	Cel	Cel	Cel	Cel	
	2797	3.4	1.101	33.5	86.68	270.5	357.4	79.5	108.9	84	
	2595	3.17	1.109	33.5	234.18	313.4	443.3	85.1	110.1	84.3	
	2391	3.03	1.119	33.9	357.79	371.4	501.9	91.4	110.6	80	
	2196	2.8	1.117	34.5	400.06	403	517.8	89.9	110.6	84.7	
	1992	2.56	1.105	35.1	409.37	413.1	527	82.5	110.9	84.7	
	1798	2.38	1.094	35.4	414.21	422.8	541.7	75	109.6	84.3	
	1599	2.17	1.08	35.6	415.31	428.3	545	66.2	109.4	83.4	
	1404	1.96	1.067	35.3	366.67	407.3	504.9	58.3	109	83.9	
	1199	1.73	1.056	34.8	358.32	394.8	492.7	51.3	106.2	84.9	
	1001	1.5	1.048	34.8	329.48	374.5	453.9	46.7	104.8	81.2	
	793	1.2	1.043	34.7	286.42	335.4	404.9	43.8	101.2	84.6	

50% load

$TIMESTMP	SPEED	ALPHA	TORQUE	P	BH	BLOWBY	FUELCOS	P_BARO	P_EC	P_EXH	P_INTAKE
	rpm	%	Nm	kW	kg/h	l/min	g/kWh	mbar	kW	mbar	mbar
50908201239	2797	81.6	135.7	39.75	14.52	81.9	358.62	1009.5	44.21	29	722
50908201330	2595	81.6	245.5	66.71	18.18	90.5	271.15	1009.5	74.64	29	822
50908201413	2392	81.6	332.3	83.25	21.1	89.1	253.57	1009.5	93.66	27	913
50908201519	2199	81.6	353.5	81.4	19.64	77.3	239.92	1009.5	91.03	22	804
50908201602	1989	81.6	368.4	76.72	17.62	74.2	229.58	1009.5	84.88	13	636
50908201645	1801	81.6	375.4	70.82	15.8	65.5	222.84	1009.5	77.43	11	493
50908201735	1604	81.6	380.9	63.99	14.16	62.2	221.52	1009.5	69.17	6	372
50908201822	1405	81.6	342.7	50.44	11.13	59.5	221.31	1009.6	53.84	4	238
50908201903	1204	81.6	338.5	42.68	9.74	59.3	227.94	1009.5	45.18	3	164
50908201950	1005	81.6	312.3	32.85	7.69	54.5	233.28	1009.6	34.51	-3	101
50908202056	799	81.6	275.1	23.01	5.36	49.7	236.64	1009.5	24.03	-3	62
	SPEED	P_OIL	REDFAC	T_AIR	T_EC	T_EXH	T_EXH_BT	T_INTAKE	T_OIL	TWO	
	rpm	Bar		Cel	Nm	Cel	Cel	Cel	Cel	Cel	
	2797	3.41	1.112	33.9	150.94	296.3	417.1	86.8	111.4	73	
	2595	3.15	1.119	34.2	274.71	336.4	475.9	91.2	112.1	81.3	
	2392	2.96	1.125	34.3	373.88	390.1	515.4	95.3	112.6	85.8	
	2199	2.74	1.118	34.4	395.33	410.9	520.6	90.9	113.4	86	
	1989	2.5	1.106	35.1	407.56	418.2	528.6	83.2	112.9	85.6	
	1801	2.33	1.093	35.5	410.46	423.2	539.4	74.9	111.5	84.1	
	1604	2.14	1.081	35.7	411.7	427.2	542.1	67	110.5	83	
	1405	1.92	1.067	35.5	365.82	406.7	502.9	58.5	110	84.2	
	1204	1.69	1.059	35.2	358.27	398.9	495.4	53.1	108.5	84.2	
	1005	1.44	1.051	34.8	328.11	378.9	456.3	48.2	106.6	82	
	799	1.15	1.044	34.6	287.28	332.6	402.3	44.3	103.2	85.5	

75% load

$TIMESTMP	SPEED	ALPHA	TORQUE	P	BH	BLOWBY	FUELCOSP	P_BARO	P_EC	P_EXH	P_INTAKE
	rpm	% Nm	~~Nm~~ kW	~~kW~~ %	kg/h	l/min	g/kWh	mbar	kW	mbar	mbar
50908195943	2795	210	61.46	87.6	18.35	92.9	298.42	1009.4	69.04	33	862
50908200035	2594	295	80.13	87.6	21.32	90.7	265.99	1009.4	90.47	34	938
50908200139	2398	342.2	85.93	87.6	21.54	82.4	251.29	1009.4	96.92	31	919
50908200229	2200	361.6	83.3	87.6	19.88	82.8	237.76	1009.4	93.22	20	801
50908200312	1996	368.8	77.11	87.6	17.84	77.1	233.17	1009.4	85.3	14	634
50908200357	1801	381.6	71.96	87.6	16.09	67.9	224.24	1009.4	78.67	10	491
50908200442	1601	384	64.4	87.6	14.27	62.9	221.4	1009.4	69.61	5	365
50908200523	1401	351.1	51.5	87.6	11.33	58.6	219.74	1009.4	55.03	2	239
50908200606	1199	343.5	43.13	87.6	9.88	58.9	227.39	1009.4	45.67	1	163
50908200651	1000	316.2	33.13	87.6	7.73	56.9	234.4	1009.5	34.83	-12	99
50908200742	790	274.8	22.73	87.6	5.39	53.3	241.94	1009.4	23.76	-5	61
	SPEED	P_OIL	REDFAC	T_AIR	T_EC	T_EXH	T_EXH_BT	T_INTAKE	T_OIL	TWO	
	rpm	Bar		Cel	Nm	Cel	Cel	Cel	Cel	Cel	
	2795	3.39	1.123	34.4	235.89	331.7	472.9	94	109	79.5	
	2594	3.14	1.129	34.8	333.03	381.4	519.5	97.8	111.3	84.2	
	2398	3.06	1.128	35.4	385.94	409.5	528.1	97.1	111.3	76.3	
	2200	2.83	1.119	35.6	404.63	418.9	527.8	91.3	111	83.4	
	1996	2.6	1.106	35.7	407.98	423.4	534.4	83	110.4	82.1	
	1801	2.42	1.093	35.9	417.18	430.5	547.8	74.7	109.3	83	
	1601	2.22	1.081	36.2	415.06	432.1	547.2	66.9	108.9	79.8	
	1401	2	1.069	36.1	375.22	413.6	514.3	59.3	107.9	82.9	
	1199	1.75	1.059	35.7	363.76	403.6	501	53.2	106	83.7	
	1000	1.52	1.051	35.3	332.42	383.7	462	48.6	104.8	78.7	
	790	1.19	1.046	34.6	287.28	337.2	404.5	45.1	101.8	81.9	

1600 RPM varying load

$TIMESTMP	SPEED	ALPHA	TORQUE	P	BH	BLOWBY	FUELCOSP	P_BARO	P_EC	P_EXH
	rpm	%	Nm	kW	kg/h	l/min	g/kWh	mbar	kW	mbar
50908203555	1599	68	360.6	60.4	13.15	68.9	218.43	1009.5	64.58	9
50908203636	1596	64.6	320.2	53.52	11.68	64.4	218.52	1009.6	57.08	3
50908203741	1595	60.4	280.4	46.83	10.27	63.7	219.14	1009.5	49.7	4
50908203849	1594	56.4	239.9	40.05	8.98	61.2	224.43	1009.6	42.33	3
50908203939	1593	52	200.6	33.46	7.64	59	229.68	1009.6	35.25	4
50908204058	1592	47.8	159.9	26.66	6.44	58.4	240.84	1009.6	27.98	2
50908204155	1592	45.2	120.4	20.07	5.24	56.4	261.76	1009.6	21.01	1
50908204302	1591	43.3	80.7	13.46	4.11	55.4	305.02	1009.6	14.04	2
	SPEED	P_OIL	REDFAC	T_AIR	T_EC	T_EXH	T_EXH_BT	T_INTAKE	T_OIL	TWO
	rpm	Bar		Cel	Nm	Cel	Cel	Cel	Cel	Cel
	1599	2.37	1.069	34.4	385.55	398.6	508.9	59.6	102.5	83.6
	1596	2.36	1.066	34.4	341.53	378.5	476.1	58	102.6	84
	1595	2.37	1.061	34.8	297.62	345.5	433.5	54.8	102.6	83.7
	1594	2.35	1.057	34.7	253.61	316	394.7	52.1	103.1	86.4
	1593	2.33	1.053	34.7	211.27	289	360.5	49.9	103.4	88.8
	1592	2.32	1.049	34.4	167.79	253.8	314.2	47.4	103.9	88.8
	1592	2.34	1.046	33.9	126	228	274.5	45.7	103.9	88.3
	1591	2.35	1.043	33.7	84.25	200.7	236.6	43.8	103.8	89.2

Appendix C

Data sheets for electronics

MICROCHIP

PIC18FXX2

28/40-pin High Performance, Enhanced FLASH Microcontrollers with 10-Bit A/D

High Performance RISC CPU:

- C compiler optimized architecture/instruction set
  - Source code compatible with the PIC16 and PIC17 instruction sets
- Linear program memory addressing to 32 Kbytes
- Linear data memory addressing to 1.5 Kbytes

Device	On-Chip Program Memory		On-Chip RAM (bytes)	Data EEPROM (bytes)
	FLASH (bytes)	# Single Word Instructions		
PIC18F242	16K	8192	768	256
PIC18F252	32K	16384	1536	256
PIC18F442	16K	8192	768	256
PIC18F452	32K	16384	1536	256

- Up to 10 MIPs operation:
  - DC - 40 MHz osc./clock input
  - 4 MHz - 10 MHz osc./clock input with PLL active
- 16-bit wide instructions, 8-bit wide data path
- Priority levels for interrupts
- 8 x 8 Single Cycle Hardware Multiplier

Peripheral Features:

- High current sink/source 25 mA/25 mA
- Three external interrupt pins
- Timer0 module: 8-bit/16-bit timer/counter with 8-bit programmable prescaler
- Timer1 module: 16-bit timer/counter
- Timer2 module: 8-bit timer/counter with 8-bit period register (time-base for PWM)
- Timer3 module: 16-bit timer/counter
- Secondary oscillator clock option - Timer1/Timer3
- Two Capture/Compare/PWM (CCP) modules. CCP pins that can be configured as:
  - Capture input: capture is 16-bit, max. resolution 6.25 ns (TCY/16)
  - Compare is 16-bit, max. resolution 100 ns (TCY)
  - PWM output: PWM resolution is 1- to 10-bit, max. PWM freq. @: 8-bit resolution = 156 kHz
10-bit resolution = 39 kHz
- Master Synchronous Serial Port (MSSP) module, Two modes of operation:
  - 3-wire SPI™ (supports all 4 SPI modes)
  - I²C™ Master and Slave mode

Peripheral Features (Continued):

- Addressable USART module:
  - Supports RS-485 and RS-232
- Parallel Slave Port (PSP) module

Analog Features:

- Compatible 10-bit Analog-to-Digital Converter module (A/D) with:
  - Fast sampling rate
  - Conversion available during SLEEP
  - Linearity ≤ 1 LSb
- Programmable Low Voltage Detection (PLVD)
  - Supports interrupt on-Low Voltage Detection
- Programmable Brown-out Reset (BOR)

Special Microcontroller Features:

- 100,000 erase/write cycle Enhanced FLASH program memory typical
- 1,000,000 erase/write cycle Data EEPROM memory
- FLASH/Data EEPROM Retention: > 40 years
- Self-reprogrammable under software control
- Power-on Reset (POR), Power-up Timer (PWRT) and Oscillator Start-up Timer (OST)
- Watchdog Timer (WDT) with its own On-Chip RC Oscillator for reliable operation
- Programmable code protection
- Power saving SLEEP mode
- Selectable oscillator options including:
  - 4X Phase Lock Loop (of primary oscillator)
  - Secondary Oscillator (32 kHz) clock input
- Single supply 5V In-Circuit Serial Programming™ (ICSP™) via two pins
- In-Circuit Debug (ICD) via two pins

CMOS Technology:

- Low power, high speed FLASH/EEPROM technology
- Fully static design
- Wide operating voltage range (2.0V to 5.5V)
- Industrial and Extended temperature ranges
- Low power consumption:
  - < 1.6 mA typical @ 5V, 4 MHz
  - 25 µA typical @ 3V, 32 kHz
  - < 0.2 µA typical standby current

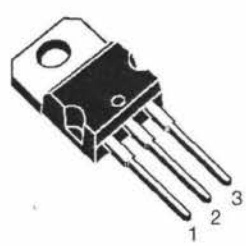
BDX33B BDX33C
BDX34B BDX34C

COMPLEMENTARY SILICON POWER DARLINGTON TRANSISTORS

DESCRIPTION

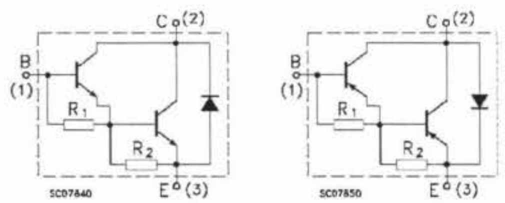
The BDX33B and BDX33C are silicon Epitaxial-Base NPN power transistors in monolithic Darlington configuration mounted in Jedec TO-220 plastic package. They are intented for use in power linear and switching applications.

The complementary PNP types are BDX34B and BDX34C respectively.



TO-220

INTERNAL SCHEMATIC DIAGRAM



R_1 Typ. = 10 KΩ R_2 Typ. = 150 Ω

ABSOLUTE MAXIMUM RATINGS

Symbol	Parameter				Unit
		NPN	BDX33B	BDX33C	
		PNP	BDX34B	BDX34C	
V_{CBO}	Collector-Base Voltage ($I_E = 0$)		80	100	V
V_{CEO}	Collector-Emitter Voltage ($I_B = 0$)		80	100	V
I_C	Collector Current		10		A
I_{CM}	Collector Peak Current		15		A
I_B	Base Current		0.25		A
P_{tot}	Total Dissipation at $T_c \leq 25$ °C		70		W
T_{stg}	Storage Temperature		-65 to 150		°C
T_j	Max. Operating Junction Temperature		150		°C

For PNP types voltage and current values are negative.

THERMAL DATA

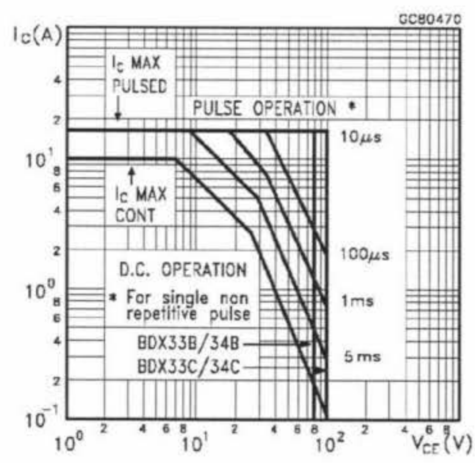
$R_{thj-case}$	Thermal Resistance Junction-case	1.78	°C/W

ELECTRICAL CHARACTERISTICS (T_{case} = 25 °C unless otherwise specified)

Symbol	Parameter	Test Conditions	Min.	Typ.	Max.	Unit
I_{CBO}	Collector Cut-off Current (I_E = 0)	for **BDX33B/34B** V_{CB} = 80 V for **BDX33C/34C** V_{CB} = 100V T_{case} = 100 °C for **BDX33B/34B** V_{CB} = 80 V for **BDX33C/34C** V_{CB} = 100 V			0.2 0.2 5 5	mA mA mA mA
I_{CEO}	Collector Cut-off Current (I_B = 0)	for **BDX33B/34B** V_{CE} = 40 V for **BDX33C/34C** V_{CE} = 50V T_{case} = 100 °C for **BDX33B/34B** V_{CE} = 40 V for **BDX33C/34C** V_{CE} = 50 V			0.5 0.5 10 10	mA mA mA mA
I_{EBO}	Emitter Cut-off Current (I_C = 0)	V_{EB} = 5 V			5	mA
$V_{CEO(sus)}$*	Collector-Emitter Sustaining Voltage (I_B = 0)	I_C =100 mA for **BDX33B/34B** for **BDX33C/34C**	80 100			V V
$V_{CER(sus)}$*	Collector-emitter Sustaining Voltage (R_{BE} =100 Ω)	I_C = 100 mA for **BDX33B/34B** for **BDX33C/34C**	80 100			V V
$V_{CEV(sus)}$*	Collector-emitter Sustaining Voltage (V_{BE} =-1.5 V)	I_C = 100 mA for **BDX33B/34B** for **BDX33C/34C**	80 100			V V
$V_{CE(sat)}$*	Collector-emitter Saturation Voltage	I_C = 3 A I_B = 6 mA			2.5	V
V_{BE}*	Base-emitter Voltage	I_C = 3 A V_{CE} = 3 V			2.5	V
h_{FE}*	DC Current Gain	I_C = 3 A V_{CE} = 3 V	750			V
V_F*	Parallel-Diode Forward Voltage	I_F = 8 A			4	V
h_{fe}	Small Signal Current Gain	I_C = 1 A V_{CE} = 5 V f = 1MHz	100			

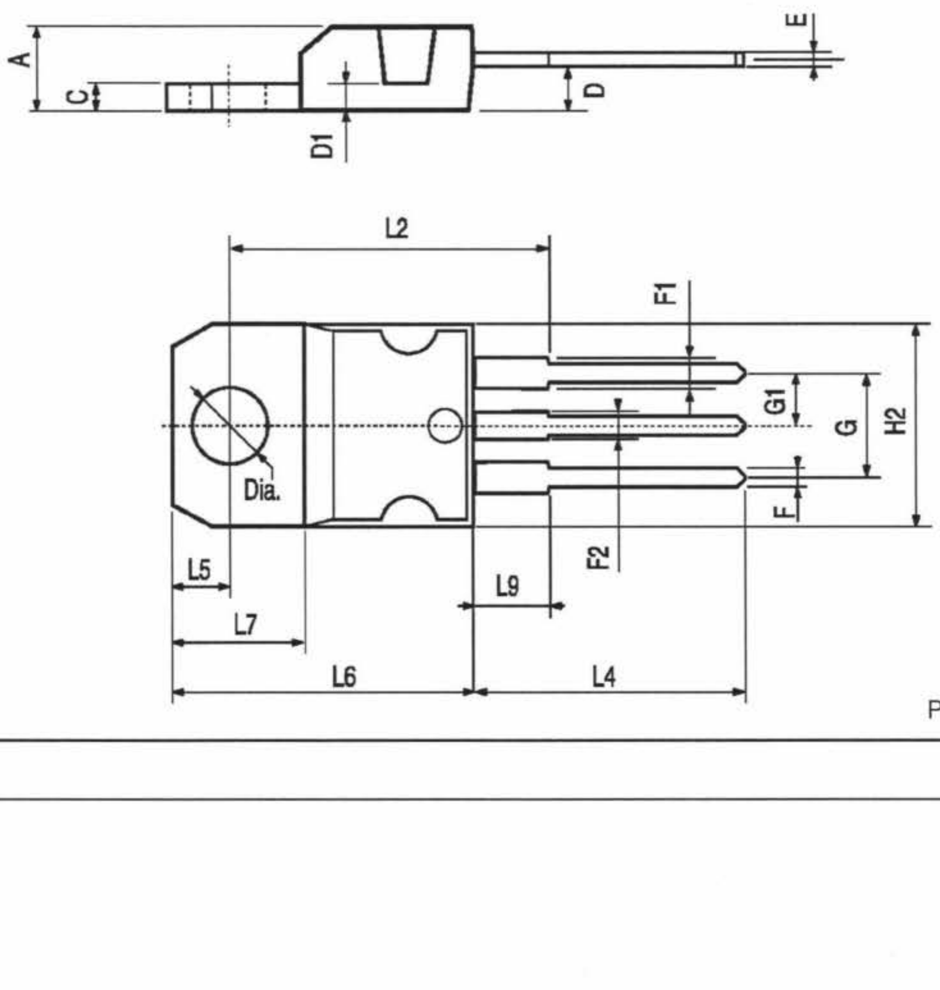
* Pulsed: Pulse duration = 300 µs, duty cycle 1.5 %
For PNP types voltage and current values are negative.

Safe Operating Area



TO-220 MECHANICAL DATA

DIM.	mm			inch		
	MIN.	TYP.	MAX.	MIN.	TYP.	MAX.
A	4.40		4.60	0.173		0.181
C	1.23		1.32	0.048		0.051
D	2.40		2.72	0.094		0.107
D1		1.27			0.050	
E	0.49		0.70	0.019		0.027
F	0.61		0.88	0.024		0.034
F1	1.14		1.70	0.044		0.067
F2	1.14		1.70	0.044		0.067
G	4.95		5.15	0.194		0.203
G1	2.4		2.7	0.094		0.106
H2	10.0		10.40	0.393		0.409
L2		16.4			0.645	
L4	13.0		14.0	0.511		0.551
L5	2.65		2.95	0.104		0.116
L6	15.25		15.75	0.600		0.620
L7	6.2		6.6	0.244		0.260
L9	3.5		3.93	0.137		0.154
DIA.	3.75		3.85	0.147		0.151



P011C

Order this document by MC7800/D

MOTOROLA

MC7800, MC7800A, LM340, LM340A Series

Three-Terminal Positive Voltage Regulators

THREE–TERMINAL POSITIVE FIXED VOLTAGE REGULATORS

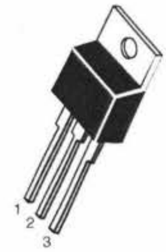
SEMICONDUCTOR TECHNICAL DATA

These voltage regulators are monolithic integrated circuits designed as fixed–voltage regulators for a wide variety of applications including local, on–card regulation. These regulators employ internal current limiting, thermal shutdown, and safe–area compensation. With adequate heatsinking they can deliver output currents in excess of 1.0 A. Although designed primarily as a fixed voltage regulator, these devices can be used with external components to obtain adjustable voltages and currents.

- Output Current in Excess of 1.0 A
- No External Components Required
- Internal Thermal Overload Protection
- Internal Short Circuit Current Limiting
- Output Transistor Safe–Area Compensation
- Output Voltage Offered in 2% and 4% Tolerance
- Available in Surface Mount D2PAK and Standard 3–Lead Transistor Packages
- Previous Commercial Temperature Range has been Extended to a Junction Temperature Range of –40°C to +125°C

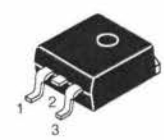
T SUFFIX
PLASTIC PACKAGE
CASE 221A

Heatsink surface connected to Pin 2.



Pin 1. Input
2. Ground
3. Output

D2T SUFFIX
PLASTIC PACKAGE
CASE 936
(D2PAK)

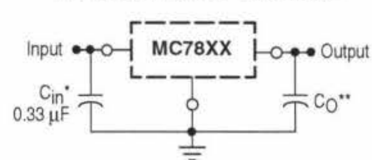


Heatsink surface (shown as terminal 4 in case outline drawing) is connected to Pin 2.

DEVICE TYPE/NOMINAL OUTPUT VOLTAGE

MC7805AC LM340AT–5 MC7805C LM340T–5	5.0 V	MC7812C LM340T–12	12 V
		MC7815AC LM340AT–15 MC7815C LM340T–15	15 V
MC7806AC MC7806C	6.0 V		
MC7808AC MC7808C	8.0 V	MC7818AC MC7818C	18 V
MC7809C	9.0 V	MC7824AC MC7824C	24 V
MC7812AC LM340AT–12	12 V		

STANDARD APPLICATION



A common ground is required between the input and the output voltages. The input voltage must remain typically 2.0 V above the output voltage even during the low point on the input ripple voltage.

XX, These two digits of the type number indicate nominal voltage.

* C_{in} is required if regulator is located an appreciable distance from power supply filter.

** C_O is not needed for stability; however, it does improve transient response. Values of less than 0.1 µF could cause instability.

ORDERING INFORMATION

Device	Output Voltage Tolerance	Operating Temperature Range	Package
MC78XXACT	2%	T_J = –40° to +125°C	Insertion Mount
LM340AT–XX			
MC78XXACD2T			Surface Mount
MC78XXCT	4%		Insertion Mount
LM340T–XX			
MC78XXCD2T			Surface Mount

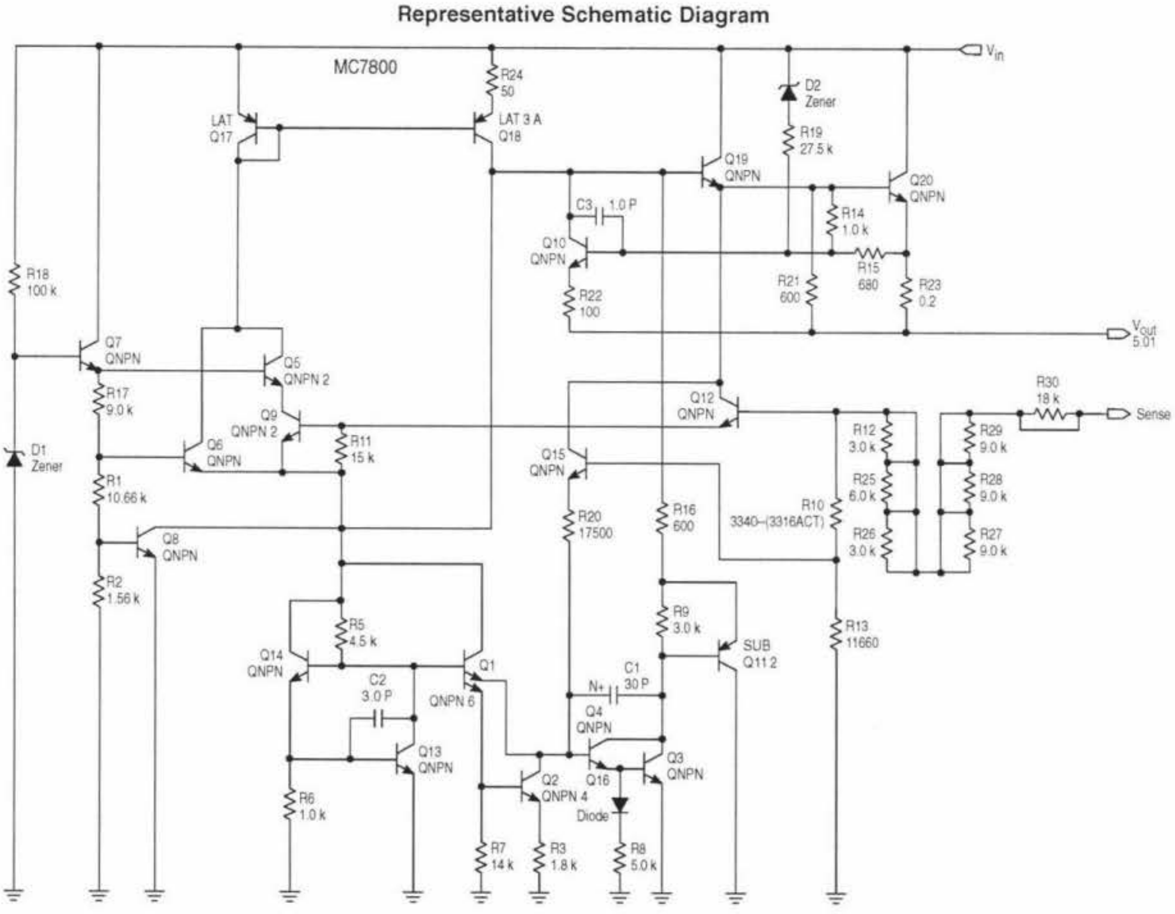
XX indicates nominal voltage.

MAXIMUM RATINGS (T_A = 25°C, unless otherwise noted.)

Rating	Symbol	Value	Unit
Input Voltage (5.0 – 18 V)	V_I	35	Vdc
(24 V)		40	
Power Dissipation			
Case 221A			
T_A = 25°C	P_D	Internally Limited	W
Thermal Resistance, Junction–to–Ambient	$R_{\theta JA}$	65	°C/W
Thermal Resistance, Junction–to–Case	$R_{\theta JC}$	5.0	°C/W
Case 936 (D²PAK)			
T_A = 25°C	P_D	Internally Limited	W
Thermal Resistance, Junction–to–Ambient	$R_{\theta JA}$	See Figure 13	°C/W
Thermal Resistance, Junction–to–Case	$R_{\theta JA}$	5.0	°C/W
Storage Junction Temperature Range	T_{stg}	–65 to +150	°C
Operating Junction Temperature	T_J	+150	°C

NOTE: ESD data available upon request.

Representative Schematic Diagram



This device contains 22 active transistors.

ELECTRICAL CHARACTERISTICS (V_{in} = 10 V, I_O = 500 mA, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7805C/LM340T–5			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	4.8	5.0	5.2	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 7.0 Vdc ≤ V_{in} ≤ 20 Vdc 8.0 Vdc ≤ V_{in} ≤ 20 Vdc	V_O	 4.75 –	 5.0 –	 5.25 –	Vdc
Line Regulation (Note 2) 7.5 Vdc ≤ V_{in} ≤ 20 Vdc, 1.0 A 8.0 Vdc ≤ V_{in} ≤ 12 Vdc	Reg_{line}	 – –	 0.5 0.8	 20 10	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.0 A 5.0 mA ≤ I_O ≤ 1.5 A (T_A = 25°C)	Reg_{load}	 – -	 1.3 1.3	 25 25	mV
Quiescent Current	I_B	–	3.2	6.5	mA
Quiescent Current Change 7.0 Vdc ≤ V_{in} ≤ 25 Vdc 5.0 mA ≤ I_O ≤ 1.0 A (T_A = 25°C)	ΔI_B	 – –	 0.3 0.08	 1.0 0.8	mA
Ripple Rejection 8.0 Vdc ≤ V_{in} ≤ 18 Vdc, f = 120 Hz	RR	62	83	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	0.9	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.6	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.3	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 10 V, I_O = 1.0 A, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7805AC/LM340AT–5			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	4.9	5.0	5.1	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 7.5 Vdc ≤ V_{in} ≤ 20 Vdc	V_O	4.8	5.0	5.2	Vdc
Line Regulation (Note 2) 7.5 Vdc ≤ V_{in} ≤ 25 Vdc, I_O = 500 mA 8.0 Vdc ≤ V_{in} ≤ 12 Vdc, I_O = 1.0 A 8.0 Vdc ≤ V_{in} ≤ 12 Vdc, I_O = 1.0 A, T_J = 25°C 7.3 Vdc ≤ V_{in} ≤ 20 Vdc, I_O = 1.0 A, T_J = 25°C	Reg_{line}	 – – – –	 0.5 0.8 1.3 4.5	 10 12 4.0 10	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A 250 mA ≤ I_O ≤ 750 mA	Reg_{load}	 – – –	 1.3 0.8 0.53	 25 25 15	mV
Quiescent Current	I_B	–	3.2	6.0	mA
Quiescent Current Change 8.0 Vdc ≤ V_{in} ≤ 25 Vdc, I_O = 500 mA 7.5 Vdc ≤ V_{in} ≤ 20 Vdc, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 0.3 – 0.08	 0.8 0.8 0.5	mA
Ripple Rejection 8.0 Vdc ≤ V_{in} ≤ 18 Vdc, f = 120 Hz, I_O = 500 mA	RR	68	83	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc

NOTES: 1. T_{low} = –40°C for MC78XXAC, C, LM340AT–XX, LM340T–XX T_{high} = +125°C for MC78XXAC, C, LM340AT–XX, LM340T–XX

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (continued) (V_{in} = 10 V, I_O = 1.0 A, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7805AC/LM340AT–5			Unit
		Min	Typ	Max	
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance (f = 1.0 kHz)	r_O	–	0.9	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.3	–	mV/°C

NOTES: 1. T_{low} = –40°C for MC78XXAC, C, LM340AT–XX, LM340T–XX T_{high} = +125°C for MC78XXAC, C, LM340AT–XX, LM340T–XX

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (V_{in} = 11 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7806C			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	5.75	6.0	6.25	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 8.0 Vdc ≤ V_{in} ≤ 21 Vdc 9.0 Vdc ≤ V_{in} ≤ 21 Vdc	V_O	 5.7 –	 6.0 –	 6.3 –	Vdc
Line Regulation, T_J = 25°C (Note 2) 8.0 Vdc ≤ V_{in} ≤ 25 Vdc 9.0 Vdc ≤ V_{in} ≤ 13 Vdc	Reg_{line}	 – –	 0.5 0.8	 24 12	mV
Load Regulation, T_J = 25°C (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	1.3	30	mV
Quiescent Current (T_J = 25°C)	I_B	–	3.3	8.0	mA
Quiescent Current Change 8.0 Vdc ≤ V_{in} ≤ 25 Vdc 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – –	 0.3 0.08	 1.3 0.5	mA
Ripple Rejection 9.0 Vdc ≤ V_{in} ≤ 19 Vdc, f = 120 Hz	RR	58	65	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	0.9	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.3	–	mV/°C

NOTES: 1. T_{low} = –40°C for MC78XXAC, C T_{high} = +125°C for MC78XXAC, C

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (V_{in} = 11 V, I_O = 1.0 A, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7806AC Min	MC7806AC Typ	MC7806AC Max	Unit
Output Voltage (T_J = 25°C)	V_O	5.88	6.0	6.12	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 8.6 Vdc ≤ V_{in} ≤ 21 Vdc	V_O	5.76	6.0	6.24	Vdc
Line Regulation (Note 2) 8.6 Vdc ≤ V_{in} ≤ 25 Vdc, I_O = 500 mA 9.0 Vdc ≤ V_{in} ≤ 13 Vdc, I_O = 1.0 A	Reg_{line}	 – –	 5.0 1.4	 12 15	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A 250 mA ≤ I_O ≤ 750 mA	Reg_{load}	 – – –	 1.3 0.9 0.2	 25 25 15	mV
Quiescent Current	I_B	–	3.3	6.0	mA
Quiescent Current Change 9.0 Vdc ≤ V_{in} ≤ 25 Vdc, I_O = 500 mA 9.0 Vdc ≤ V_{in} ≤ 21 Vdc, I_O = 1.0 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.8 0.8 0.5	mA
Ripple Rejection 9.0 Vdc ≤ V_{in} ≤ 19 Vdc, f = 120 Hz, I_O = 500 mA	RR	58	65	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance (f = 1.0 kHz)	r_O	–	0.9	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.3	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 14 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7808C Min	MC7808C Typ	MC7808C Max	Unit
Output Voltage (T_J = 25°C)	V_O	7.7	8.0	8.3	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 10.5 Vdc ≤ V_{in} ≤ 23 Vdc	V_O	7.6	8.0	8.4	Vdc
Line Regulation, T_J = 25°C, (Note 2) 10.5 Vdc ≤ V_{in} ≤ 25 Vdc 11 Vdc ≤ V_{in} ≤ 17 Vdc	Reg_{line}	 – –	 6.0 1.7	 32 16	mV
Load Regulation, T_J = 25°C (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	1.4	35	mV
Quiescent Current	I_B	–	3.3	8.0	mA
Quiescent Current Change 10.5 Vdc ≤ V_{in} ≤ 25 Vdc 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – –	 – –	 1.0 0.5	mA
Ripple Rejection 11.5 Vdc ≤ V_{in} ≤ 18 Vdc, f = 120 Hz	RR	56	62	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O

NOTES: 1. T_{low} = –40°C for MC78XXAC, C T_{high} = +125°C for MC78XXAC, C

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (continued) (V_{in} = 14 V, I_O = 500 mA, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7808C Min	MC7808C Typ	MC7808C Max	Unit
Output Resistance f = 1.0 kHz	r_O	–	0.9	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.4	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 14 V, I_O = 1.0 A, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7808AC Min	MC7808AC Typ	MC7808AC Max	Unit
Output Voltage (T_J = 25°C)	V_O	7.84	8.0	8.16	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 10.6 Vdc ≤ V_{in} ≤ 23 Vdc	V_O	7.7	8.0	8.3	Vdc
Line Regulation (Note 2) 10.6 Vdc ≤ V_{in} ≤ 25 Vdc, I_O = 500 mA 11 Vdc ≤ V_{in} ≤ 17 Vdc, I_O = 1.0 A 10.4 Vdc ≤ V_{in} ≤ 23 Vdc, T_J = 25°C	Reg_{line}	 – – –	 6.0 1.7 5.0	 15 18 15	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A 250 mA ≤ I_O ≤ 750 mA	Reg_{load}	 – – –	 1.4 1.0 0.22	 25 25 15	mV
Quiescent Current	I_B	–	3.3	6.0	mA
Quiescent Current Change 11 Vdc ≤ V_{in} ≤ 25 Vdc, I_O = 500 mA 10.6 Vdc ≤ V_{in} ≤ 23 Vdc, I_O = 1.0 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.8 0.8 0.5	mA
Ripple Rejection 11.5 Vdc ≤ V_{in} ≤ 21.5 Vdc, f = 120 Hz, I_O = 500 mA	RR	56	62	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	0.9	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.4	–	mV/°C

NOTES: 1. T_{low} = –40°C for MC78XXAC, C T_{high} = +125°C for MC78XXAC, C

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (V_{in} = 15 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7809CT			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	8.65	9.0	9.35	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 11.5 Vdc ≤ V_{in} ≤ 24 Vdc	V_O	8.55	9.0	9.45	Vdc
Line Regulation, T_J = 25°C (Note 2) 11 Vdc ≤ V_{in} ≤ 26 Vdc 11.5 Vdc ≤ V_{in} ≤ 17 Vdc	Reg_{line}	 – –	 6.2 1.8	 32 16	mV
Load Regulation, T_J = 25°C (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	1.5	35	mV
Quiescent Current	I_B	–	3.4	8.0	mA
Quiescent Current Change 11.5 Vdc ≤ V_{in} ≤ 26 Vdc 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – –	 – –	 1.0 0.5	mA
Ripple Rejection 11.5 Vdc ≤ V_{in} ≤ 21.5 Vdc, f = 120 Hz	RR	56	61	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	1.0	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.5	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 19 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7812C/LM340T–12			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	11.5	12	12.5	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 14.5 Vdc ≤ V_{in} ≤ 27 Vdc	V_O	11.4	12	12.6	Vdc
Line Regulation, T_J = 25°C (Note 2) 14.5 Vdc ≤ V_{in} ≤ 30 Vdc 16 Vdc ≤ V_{in} ≤ 22 Vdc 14.8 Vdc ≤ V_{in} ≤ 27 Vdc, I_O = 1.0 A	Reg_{line}	 – – –	 3.8 0.3 –	 24 24 48	mV
Load Regulation, T_J = 25°C (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	8.1	60	mV
Quiescent Current	I_B	–	3.4	6.5	mA
Quiescent Current Change 14.5 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 1.0 A, T_J = 25°C 15 Vdc ≤ V_{in} ≤ 30 Vdc 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.7 0.8 0.5	mA
Ripple Rejection 15 Vdc ≤ V_{in} ≤ 25 Vdc, f = 120 Hz	RR	55	60	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc

NOTES: 1. T_{low} = –40°C for MC78XXAC, C, LM340AT–XX, LM340T–XX T_{high} = +125°C for MC78XXAC, C, LM340AT–XX, LM340T–XX

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (continued) (V_{in} = 19 V, I_O = 500 mA, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7812C/LM340T–12 Min	Typ	Max	Unit
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	1.1	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.8	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 19 V, I_O = 1.0 A, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7812AC/LM340AT–12 Min	Typ	Max	Unit
Output Voltage (T_J = 25°C)	V_O	11.75	12	12.25	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 14.8 Vdc ≤ V_{in} ≤ 27 Vdc	V_O	11.5	12	12.5	Vdc
Line Regulation (Note 2)	Reg_{line}				mV
14.8 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 500 mA		–	3.8	18	
16 Vdc ≤ V_{in} ≤ 22 Vdc, I_O = 1.0 A		–	2.2	20	
14.5 Vdc ≤ V_{in} ≤ 27 Vdc, T_J = 25°C		–	6.0	120	
Load Regulation (Note 2)	Reg_{load}				mV
5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C		–	–	25	
5.0 mA ≤ I_O ≤ 1.0 A		–	–	25	
Quiescent Current	I_B	–	3.4	6.0	mA
Quiescent Current Change	ΔI_B				mA
15 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 500 mA		–	–	0.8	
14.8 Vdc ≤ V_{in} ≤ 27 Vdc, T_J = 25°C		–	–	0.8	
5.0 mA ≤ I_O ≤ 1.0 A, T_J = 25°C		–	–	0.5	
Ripple Rejection 15 Vdc ≤ V_{in} ≤ 25 Vdc, f = 120 Hz, I_O = 500 mA	RR	55	60	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance (f = 1.0 kHz)	r_O	–	1.1	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–0.8	–	mV/°C

NOTES: 1. T_{low} = –40°C for MC78XXAC, C, LM340AT–XX, LM340T–XX T_{high} = +125°C for MC78XXAC, C, LM340AT–XX, LM340T–XX

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (V_{in} = 23 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7815C/LM340T–15 Min	Typ	Max	Unit
Output Voltage (T_J = 25°C)	V_O	14.4	15	15.6	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 17.5 Vdc ≤ V_{in} ≤ 30 Vdc	V_O	14.25	15	15.75	Vdc
Line Regulation, T_J = 25°C (Note 2) 17.9 Vdc ≤ V_{in} ≤ 30 Vdc 20 Vdc ≤ V_{in} ≤ 26 Vdc	Reg_{line}	 – –	 8.5 3.0	 30 28	mV
Load Regulation, T_J = 25°C (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	1.8	55	mV
Quiescent Current	I_B	–	3.5	6.5	mA
Quiescent Current Change 17.5 Vdc ≤ V_{in} ≤ 30 Vdc 17.5 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 1.0 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.8 0.7 0.5	mA
Ripple Rejection 18.5 Vdc ≤ V_{in} ≤ 28.5 Vdc, f = 120 Hz	RR	54	58	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	1.2	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–1.0	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 23 V, I_O = 1.0 A, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7815AC/LM340AT–15 Min	Typ	Max	Unit
Output Voltage (T_J = 25°C)	V_O	14.7	15	15.3	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 17.9 Vdc ≤ V_{in} ≤ 30 Vdc	V_O	14.4	15	15.6	Vdc
Line Regulation (Note 2) 17.9 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 500 mA 20 Vdc ≤ V_{in} ≤ 26 Vdc 17.5 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 1.0 A, T_J = 25°C	Reg_{line}	 – – –	 8.5 3.0 7.0	 20 22 20	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A 250 mA ≤ I_O ≤ 750 mA	Reg_{load}	 – – –	 1.8 1.5 1.2	 25 25 15	mV
Quiescent Current	I_B	–	3.5	6.0	mA
Quiescent Current Change 17.5 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 500 mA 17.5 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 1.0 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.8 0.8 0.5	mA

NOTES: 1. T_{low} = –40°C for MC78XXAC, C, LM340AT–XX, LM340T–XX T_{high} = +125°C for MC78XXAC, C, LM340AT–XX, LM340T–XX

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (continued) (V_{in} = 23 V, I_O = 1.0 A, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7815AC/LM340AT–15			Unit
		Min	Typ	Max	
Ripple Rejection 18.5 Vdc ≤ V_{in} ≤ 28.5 Vdc, f = 120 Hz, I_O = 500 mA	RR	60	80	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	1.2	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–1.0	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 27 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7818C			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	17.3	18	18.7	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 21 Vdc ≤ V_{in} ≤ 33 Vdc	V_O	17.1	18	18.9	Vdc
Line Regulation, (Note 2) 21 Vdc ≤ V_{in} ≤ 33 Vdc 24 Vdc ≤ V_{in} ≤ 30 Vdc	Reg_{line}	 – –	 9.5 3.2	 50 25	mV
Load Regulation, (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	2.0	55	mV
Quiescent Current	I_B	–	3.5	6.5	mA
Quiescent Current Change 21 Vdc ≤ V_{in} ≤ 33 Vdc 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – –	 – –	 1.0 0.5	mA
Ripple Rejection 22 Vdc ≤ V_{in} ≤ 33 Vdc, f = 120 Hz	RR	53	57	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_{il} - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	1.3	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–1.5	–	mV/°C

NOTES: 1. T_{low} = –40°C for MC78XXAC, C T_{high} = +125°C for MC78XXAC, C

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (V_{in} = 27 V, I_O = 1.0 A, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7818AC Min	MC7818AC Typ	MC7818AC Max	Unit
Output Voltage (T_J = 25°C)	V_O	17.64	18	18.36	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 21 Vdc ≤ V_{in} ≤ 33 Vdc	V_O	17.3	18	18.7	Vdc
Line Regulation (Note 2) 21 Vdc ≤ V_{in} ≤ 33 Vdc, I_O = 500 mA 24 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 1.0 A 24 Vdc ≤ V_{in} ≤ 30 Vdc, I_O = 1.0 A, T_J = 25°C 20.6 Vdc ≤ V_{in} ≤ 33 Vdc, I_O = 1.0 A, T_J = 25°C	Reg_{line}	 – – – –	 9.5 3.2 3.2 8.0	 22 25 10.5 22	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A 250 mA ≤ I_O ≤ 750 mA	Reg_{load}	 – – –	 2.0 1.8 1.5	 25 25 15	mV
Quiescent Current	I_B	–	3.5	6.0	mA
Quiescent Current Change 21 Vdc ≤ V_{in} ≤ 33 Vdc, I_O = 500 mA 21.5 Vdc ≤ V_{in} ≤ 30 Vdc, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.8 0.8 0.5	mA
Ripple Rejection 22 Vdc ≤ V_{in} ≤ 32 Vdc, f = 120 Hz, I_O = 500 mA	RR	53	57	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	$\mu V/V_O$
Output Resistance f = 1.0 kHz	r_O	–	1.3	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–1.5	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 33 V, I_O = 500 mA, $T_J = T_{low}$ to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7824C Min	MC7824C Typ	MC7824C Max	Unit
Output Voltage (T_J = 25°C)	V_O	23	24	25	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 27 Vdc ≤ V_{in} ≤ 38 Vdc	V_O	22.8	24	25.2	Vdc
Line Regulation, (Note 2) 27 Vdc ≤ V_{in} ≤ 38 Vdc 30 Vdc ≤ V_{in} ≤ 36 Vdc	Reg_{line}	 – –	 2.7 2.7	 60 48	mV
Load Regulation, (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A	Reg_{load}	–	4.4	65	mV
Quiescent Current	I_B	–	3.6	6.5	mA
Quiescent Current Change 27 Vdc ≤ V_{in} ≤ 38 Vdc 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – –	 – –	 1.0 0.5	mA

NOTES: 1. T_{low} = –40°C for MC78XXAC, C T_{high} = +125°C for MC78XXAC, C

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

ELECTRICAL CHARACTERISTICS (continued) (V_{in} = 33 V, I_O = 500 mA, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7824C			Unit
		Min	Typ	Max	
Ripple Rejection 28 Vdc ≤ V_{in} ≤ 38 Vdc, f = 120 Hz	RR	50	54	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance f = 1.0 kHz	r_O	–	1.4	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–2.0	–	mV/°C

ELECTRICAL CHARACTERISTICS (V_{in} = 33 V, I_O = 1.0 A, T_J = T_{low} to T_{high} [Note 1], unless otherwise noted.)

Characteristic	Symbol	MC7824AC			Unit
		Min	Typ	Max	
Output Voltage (T_J = 25°C)	V_O	23.5	24	24.5	Vdc
Output Voltage (5.0 mA ≤ I_O ≤ 1.0 A, P_D ≤ 15 W) 27.3 Vdc ≤ V_{in} ≤ 38 Vdc	V_O	23.2	24	25.8	Vdc
Line Regulation (Note 2) 27 Vdc ≤ V_{in} ≤ 38 Vdc, I_O = 500 mA 30 Vdc ≤ V_{in} ≤ 36 Vdc, I_O = 1.0 A 30 Vdc ≤ V_{in} ≤ 36 Vdc, T_J = 25°C 26.7 Vdc ≤ V_{in} ≤ 38 Vdc, I_O = 1.0 A, T_J = 25°C	Reg_{line}	 – – – –	 11.5 3.8 3.8 10	 25 28 12 25	mV
Load Regulation (Note 2) 5.0 mA ≤ I_O ≤ 1.5 A, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A 250 mA ≤ I_O ≤ 750 mA	Reg_{load}	 – – –	 2.1 2.0 1.8	 15 25 15	mV
Quiescent Current	I_B	–	3.6	6.0	mA
Quiescent Current Change 27.3 Vdc ≤ V_{in} ≤ 38 Vdc, I_O = 500 mA 27 Vdc ≤ V_{in} ≤ 38 Vdc, T_J = 25°C 5.0 mA ≤ I_O ≤ 1.0 A	ΔI_B	 – – –	 – – –	 0.8 0.8 0.5	mA
Ripple Rejection 28 Vdc ≤ V_{in} ≤ 38 Vdc, f = 120 Hz, I_O = 500 mA	RR	45	54	–	dB
Dropout Voltage (I_O = 1.0 A, T_J = 25°C)	$V_I - V_O$	–	2.0	–	Vdc
Output Noise Voltage (T_A = 25°C) 10 Hz ≤ f ≤ 100 kHz	V_n	–	10	–	µV/V_O
Output Resistance (f = 1.0 kHz)	r_O	–	1.4	–	mΩ
Short Circuit Current Limit (T_A = 25°C) V_{in} = 35 Vdc	I_{SC}	–	0.2	–	A
Peak Output Current (T_J = 25°C)	I_{max}	–	2.2	–	A
Average Temperature Coefficient of Output Voltage	TCV_O	–	–2.0	–	mV/°C

NOTES: 1. T_{low} = –40°C for MC78XXAC, C T_{high} = +125°C for MC78XXAC, C

2. Load and line regulation are specified at constant junction temperature. Changes in V_O due to heating effects must be taken into account separately. Pulse testing with low duty cycle is used.

Figure 1. Peak Output Current as a Function of Input/Output Differential Voltage (MC78XXC, AC)

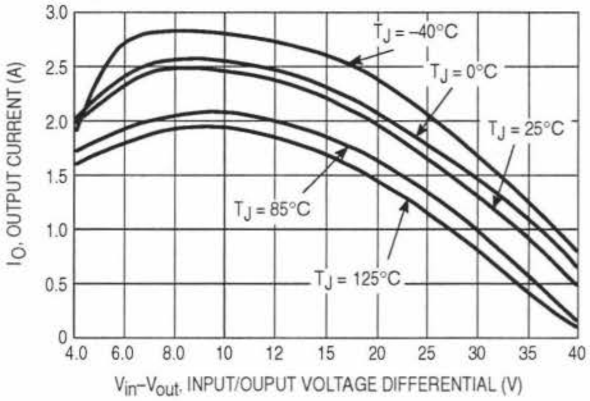


Figure 2. Ripple Rejection as a Function of Output Voltages (MC78XXC, AC)

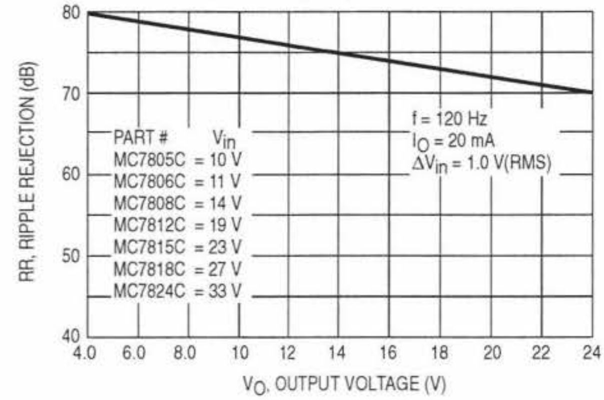


Figure 3. Ripple Rejection as a Function of Frequency (MC78XXC, AC)

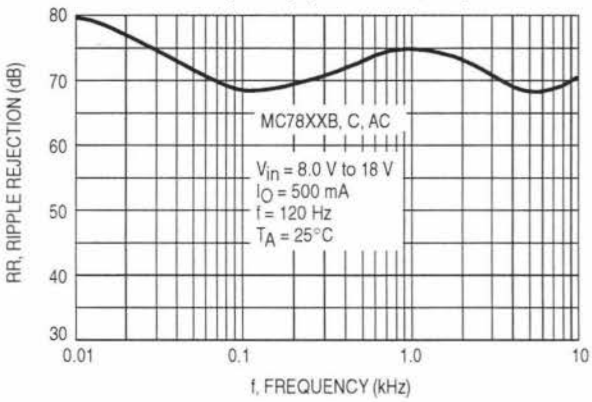


Figure 4. Output Voltage as a Function of Junction Temperature (MC7805C, AC)

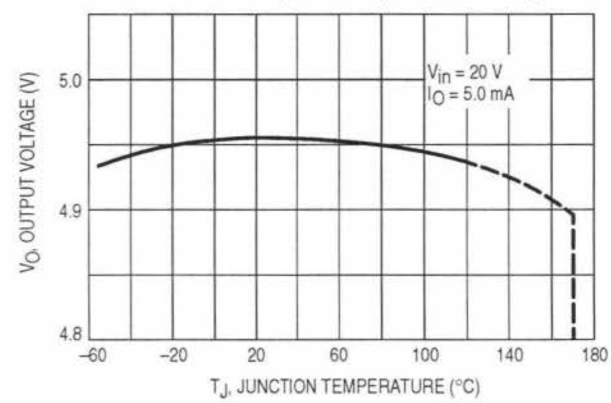


Figure 5. Output Impedance as a Function of Output Voltage (MC78XXC, AC)

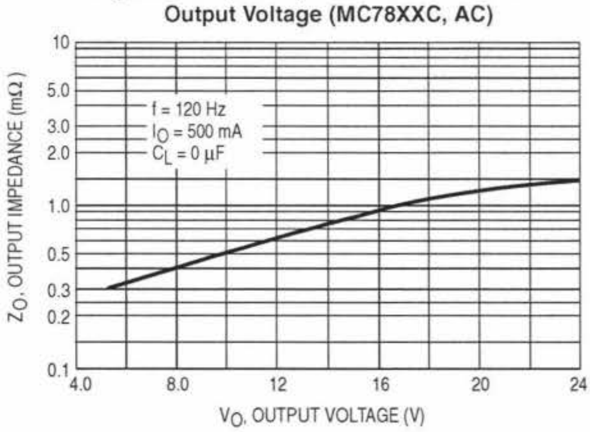
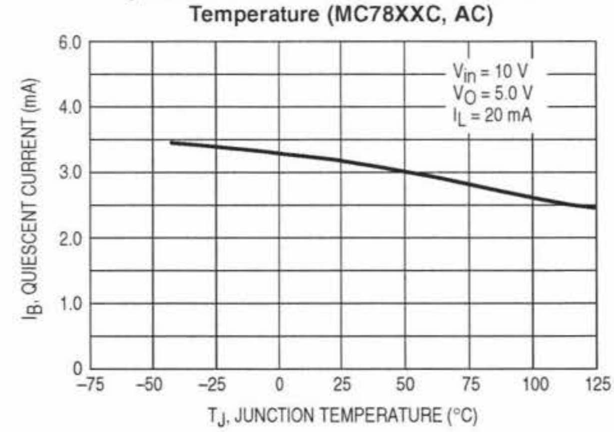


Figure 6. Quiescent Current as a Function of Temperature (MC78XXC, AC)



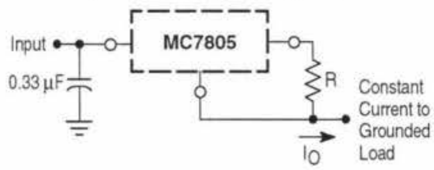
APPLICATIONS INFORMATION

Design Considerations

The MC7800 Series of fixed voltage regulators are designed with Thermal Overload Protection that shuts down the circuit when subjected to an excessive power overload condition, Internal Short Circuit Protection that limits the maximum current the circuit will pass, and Output Transistor Safe–Area Compensation that reduces the output short circuit current as the voltage across the pass transistor is increased.

In many low current applications, compensation capacitors are not required. However, it is recommended that the regulator input be bypassed with a capacitor if the regulator is connected to the power supply filter with long wire lengths, or if the output load capacitance is large. An input bypass capacitor should be selected to provide good high–frequency characteristics to insure stable operation under all load conditions. A 0.33 μF or larger tantalum, mylar, or other capacitor having low internal impedance at high frequencies should be chosen. The bypass capacitor should be mounted with the shortest possible leads directly across the regulators input terminals. Normally good construction techniques should be used to minimize ground loops and lead resistance drops since the regulator has no external sense lead.

Figure 7. Current Regulator



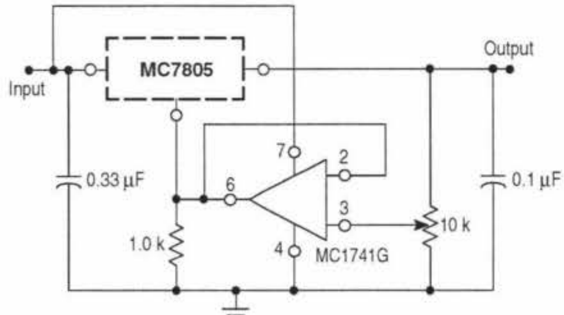
The MC7800 regulators can also be used as a current source when connected as above. In order to minimize dissipation the MC7805C is chosen in this application. Resistor R determines the current as follows:

$$I_O = \frac{5.0\ V}{R} + I_B$$

$I_B \cong 3.2$ mA over line and load changes.

For example, a 1.0 A current source would require R to be a 5.0 Ω, 10 W resistor and the output voltage compliance would be the input voltage less 7.0 V.

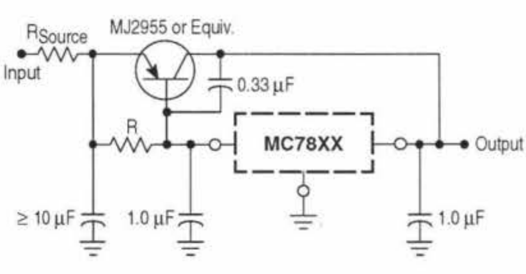
Figure 8. Adjustable Output Regulator



$V_O = 7.0$ V to 20 V
$V_{IN} - V_O \geq 2.0$ V

The addition of an operational amplifier allows adjustment to higher or intermediate values while retaining regulation characteristics. The minimum voltage obtainable with this arrangement is 2.0 V greater than the regulator voltage.

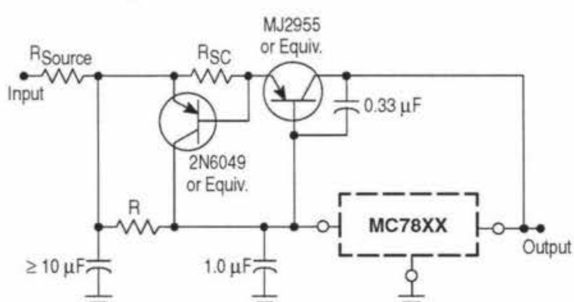
Figure 9. Current Boost Regulator



XX = 2 digits of type number indicating voltage.

The MC7800 series can be current boosted with a PNP transistor. The MJ2955 provides current to 5.0 A. Resistor R in conjunction with the V_{BE} of the PNP determines when the pass transistor begins conducting; this circuit is not short circuit proof. Input/output differential voltage minimum is increased by V_{BE} of the pass transistor.

Figure 10. Short Circuit Protection



XX = 2 digits of type number indicating voltage.

The circuit of Figure 9 can be modified to provide supply protection against short circuits by adding a short circuit sense resistor, R_{SC}, and an additional PNP transistor. The current sensing PNP must be able to handle the short circuit current of the three–terminal regulator. Therefore, a four–ampere plastic power transistor is specified.

Figure 11. Worst Case Power Dissipation versus Ambient Temperature (Case 221A)

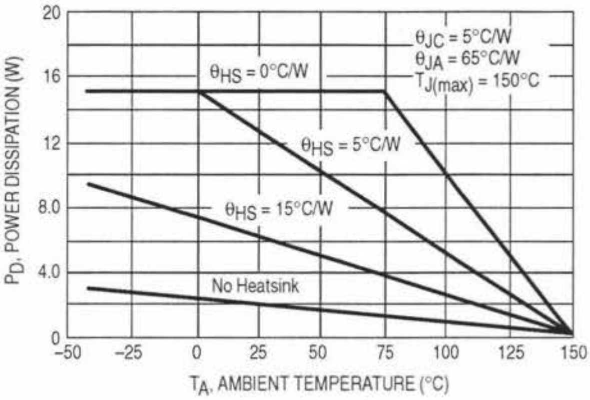


Figure 12. Input Output Differential as a Function of Junction Temperature (MC78XXC, AC)

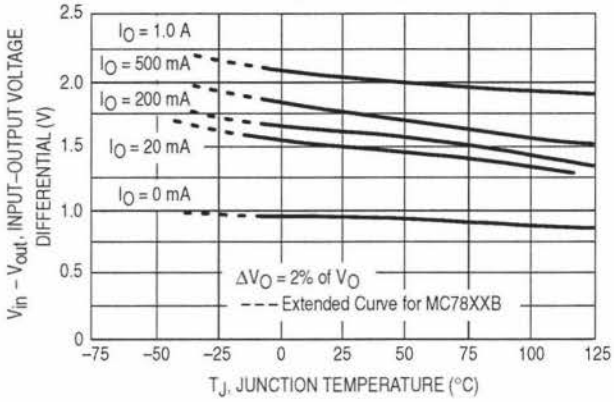
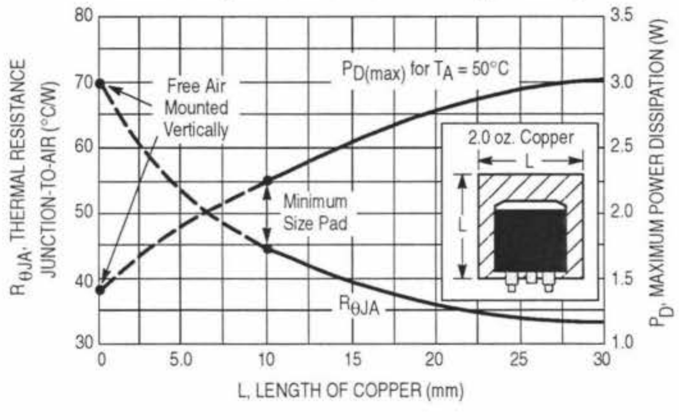


Figure 13. D²PAK Thermal Resistance and Maximum Power Dissipation versus P.C.B. Copper Length



DEFINITIONS

Line Regulation – The change in output voltage for a change in the input voltage. The measurement is made under conditions of low dissipation or by using pulse techniques such that the average chip temperature is not significantly affected.

Load Regulation – The change in output voltage for a change in load current at constant chip temperature.

Maximum Power Dissipation – The maximum total device dissipation for which the regulator will operate within specifications.

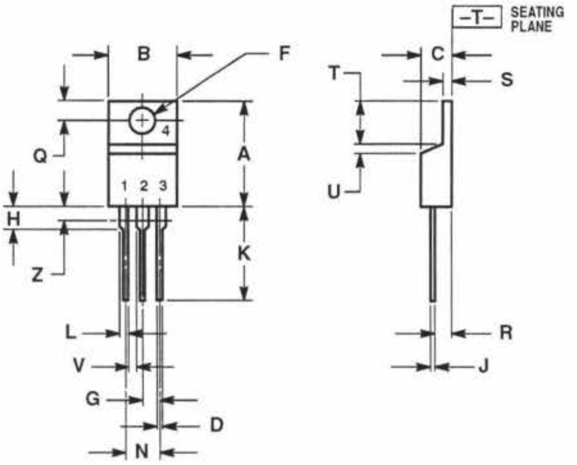
Quiescent Current – That part of the input current that is not delivered to the load.

Output Noise Voltage – The rms ac voltage at the output, with constant load and no input ripple, measured over a specified frequency range.

Long Term Stability – Output voltage stability under accelerated life test conditions with the maximum rated voltage listed in the devices' electrical characteristics and maximum power dissipation.

OUTLINE DIMENSIONS

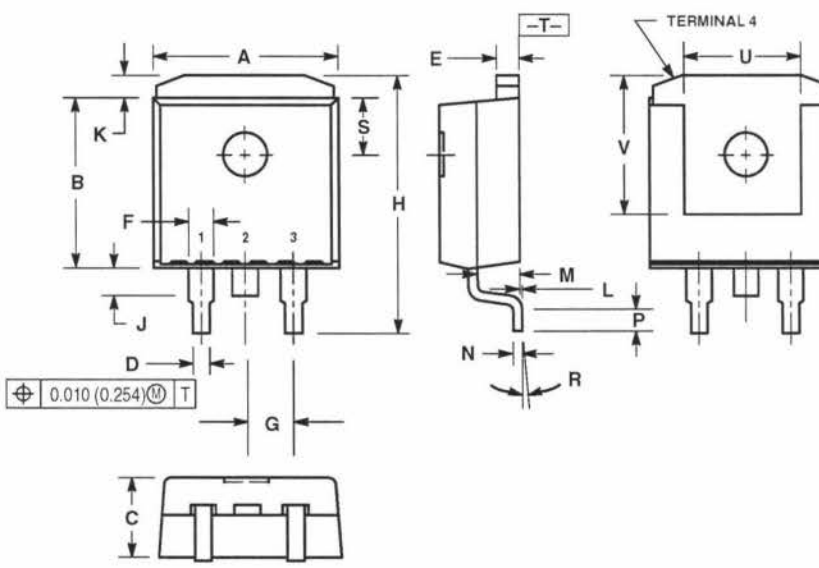
T SUFFIX
PLASTIC PACKAGE
CASE 221A–06
ISSUE Y



NOTES:
1. DIMENSIONING AND TOLERANCING PER ANSI Y14.5M, 1982.
2. CONTROLLING DIMENSION: INCH.
3. DIM Z DEFINES A ZONE WHERE ALL BODY AND LEAD IRREGULARITIES ARE ALLOWED.

DIM	INCHES MIN	INCHES MAX	MILLIMETERS MIN	MILLIMETERS MAX
A	0.570	0.620	14.48	15.75
B	0.380	0.405	9.66	10.28
C	0.160	0.190	4.07	4.82
D	0.025	0.035	0.64	0.88
F	0.142	0.147	3.61	3.73
G	0.095	0.105	2.42	2.66
H	0.110	0.155	2.80	3.93
J	0.018	0.025	0.46	0.64
K	0.500	0.562	12.70	14.27
L	0.045	0.060	1.15	1.52
N	0.190	0.210	4.83	5.33
Q	0.100	0.120	2.54	3.04
R	0.080	0.110	2.04	2.79
S	0.045	0.055	1.15	1.39
T	0.235	0.255	5.97	6.47
U	0.000	0.050	0.00	1.27
V	0.045	–	1.15	–
Z	–	0.080	–	2.04

D2T SUFFIX
PLASTIC PACKAGE
CASE 936–03
(D^2PAK)
ISSUE B



NOTES:
1. DIMENSIONING AND TOLERANCING PER ANSI Y14.5M, 1982.
2. CONTROLLING DIMENSION: INCH.
3. TAB CONTOUR OPTIONAL WITHIN DIMENSIONS A AND K.
4. DIMENSIONS U AND V ESTABLISH A MINIMUM MOUNTING SURFACE FOR TERMINAL 4.
5. DIMENSIONS A AND B DO NOT INCLUDE MOLD FLASH OR GATE PROTRUSIONS. MOLD FLASH AND GATE PROTRUSIONS NOT TO EXCEED 0.025 (0.635) MAXIMUM.

DIM	INCHES MIN	INCHES MAX	MILLIMETERS MIN	MILLIMETERS MAX
A	0.386	0.403	9.804	10.236
B	0.356	0.368	9.042	9.347
C	0.170	0.180	4.318	4.572
D	0.026	0.036	0.660	0.914
E	0.045	0.055	1.143	1.397
F	0.051 REF		1.295 REF	
G	0.100 BSC		2.540 BSC	
H	0.539	0.579	13.691	14.707
J	0.125 MAX		3.175 MAX	
K	0.050 REF		1.270 REF	
L	0.000	0.010	0.000	0.254
M	0.088	0.102	2.235	2.591
N	0.018	0.026	0.457	0.660
P	0.058	0.078	1.473	1.981
R	5° REF		5° REF	
S	0.116 REF		2.946 REF	
U	0.200 MIN		5.080 MIN	
V	0.250 MIN		6.350 MIN	

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JAPAN: Nippon Motorola Ltd.: SPD, Strategic Planning Office, 4–32–1, Nishi–Gotanda, Shinagawa–ku, Tokyo 141, Japan. 81–3–5487–8488

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– http://sps.motorola.com/mfax/

ASIA/PACIFIC: Motorola Semiconductors H.K. Ltd.; 8B Tai Ping Industrial Park, 51 Ting Kok Road, Tai Po, N.T., Hong Kong. 852–26629298

HOME PAGE: http://motorola.com/sps/

19-4323; Rev 9; 4/00

MAXIM

+5V-Powered, Multichannel RS-232 Drivers/Receivers

General Description

The MAX220–MAX249 family of line drivers/receivers is intended for all EIA/TIA-232E and V.28/V.24 communications interfaces, particularly applications where ±12V is not available.

These parts are especially useful in battery-powered systems, since their low-power shutdown mode reduces power dissipation to less than 5μW. The MAX225, MAX233, MAX235, and MAX245/MAX246/MAX247 use no external components and are recommended for applications where printed circuit board space is critical.

Applications

Portable Computers

Low-Power Modems

Interface Translation

Battery-Powered RS-232 Systems

Multidrop RS-232 Networks

Features

Superior to Bipolar

- ♦ Operate from Single +5V Power Supply (+5V and +12V—MAX231/MAX239)
- ♦ Low-Power Receive Mode in Shutdown (MAX223/MAX242)
- ♦ Meet All EIA/TIA-232E and V.28 Specifications
- ♦ Multiple Drivers and Receivers
- ♦ 3-State Driver and Receiver Outputs
- ♦ Open-Line Detection (MAX243)

Ordering Information

PART	TEMP. RANGE	PIN-PACKAGE
MAX220CPE	0°C to +70°C	16 Plastic DIP
MAX220CSE	0°C to +70°C	16 Narrow SO
MAX220CWE	0°C to +70°C	16 Wide SO
MAX220C/D	0°C to +70°C	Dice*
MAX220EPE	-40°C to +85°C	16 Plastic DIP
MAX220ESE	-40°C to +85°C	16 Narrow SO
MAX220EWE	-40°C to +85°C	16 Wide SO
MAX220EJE	-40°C to +85°C	16 CERDIP
MAX220MJE	-55°C to +125°C	16 CERDIP

Ordering Information continued at end of data sheet.

**Contact factory for dice specifications.*

Selection Table

Part Number	Power Supply (V)	No. of RS-232 Drivers/Rx	No. of Ext. Caps	Nominal Cap. Value (μF)	SHDN & Three-State	Rx Active in SHDN	Data Rate (kbps)	Features
MAX220	+5	2/2	4	4.7/10	No	—	120	Ultra-low-power, industry-standard pinout
MAX222	+5	2/2	4	0.1	Yes	—	200	Low-power shutdown
MAX223 (MAX213)	+5	4/5	4	1.0 (0.1)	Yes	✔	120	MAX241 and receivers active in shutdown
MAX225	+5	5/5	0	—	Yes	✔	120	Available in SO
MAX230 (MAX200)	+5	5/0	4	1.0 (0.1)	Yes	—	120	5 drivers with shutdown
MAX231 (MAX201)	+5 and +7.5 to +13.2	2/2	2	1.0 (0.1)	No	—	120	Standard +5/+12V or battery supplies; same functions as MAX232
MAX232 (MAX202)	+5	2/2	4	1.0 (0.1)	No	—	120 (64)	Industry standard
MAX232A	+5	2/2	4	0.1	No	—	200	Higher slew rate, small caps
MAX233 (MAX203)	+5	2/2	0	—	No	—	120	No external caps
MAX233A	+5	2/2	0	—	No	—	200	No external caps, high slew rate
MAX234 (MAX204)	+5	4/0	4	1.0 (0.1)	No	—	120	Replaces 1488
MAX235 (MAX205)	+5	5/5	0	—	Yes	—	120	No external caps
MAX236 (MAX206)	+5	4/3	4	1.0 (0.1)	Yes	—	120	Shutdown, three state
MAX237 (MAX207)	+5	5/3	4	1.0 (0.1)	No	—	120	Complements IBM PC serial port
MAX238 (MAX208)	+5	4/4	4	1.0 (0.1)	No	—	120	Replaces 1488 and 1489
MAX239 (MAX209)	+5 and +7.5 to +13.2	3/5	2	1.0 (0.1)	No	—	120	Standard +5/+12V or battery supplies; single-package solution for IBM PC serial port
MAX240	+5	5/5	4	1.0	Yes	—	120	DIP or flatpack package
MAX241 (MAX211)	+5	4/5	4	1.0 (0.1)	Yes	—	120	Complete IBM PC serial port
MAX242	+5	2/2	4	0.1	Yes	✔	200	Separate shutdown and enable
MAX243	+5	2/2	4	0.1	No	—	200	Open-line detection simplifies cabling
MAX244	+5	8/10	4	1.0	No	—	120	High slew rate
MAX245	+5	8/10	0	—	Yes	✔	120	High slew rate, int. caps, two shutdown modes
MAX246	+5	8/10	0	—	Yes	✔	120	High slew rate, int. caps, three shutdown modes
MAX247	+5	8/9	0	—	Yes	✔	120	High slew rate, int. caps, nine operating modes
MAX248	+5	8/8	4	1.0	Yes	✔	120	High slew rate, selective half-chip enables
MAX249	+5	6/10	4	1.0	Yes	✔	120	Available in quad flatpack package

+5V-Powered, Multichannel RS-232 Drivers/Receivers

ABSOLUTE MAXIMUM RATINGS—MAX220/222/232A/233A/242/243

Supply Voltage (V_{CC}) .... -0.3V to +6V
Input Voltages
T_{IN} .... -0.3V to (V_{CC} - 0.3V)
R_{IN} (Except MAX220) .... ±30V
R_{IN} (MAX220) .... ±25V
T_{OUT} (Except MAX220) (Note 1) .... ±15V
T_{OUT} (MAX220) .... ±13.2V
Output Voltages
T_{OUT} .... ±15V
R_{OUT} .... -0.3V to (V_{CC} + 0.3V)
Driver/Receiver Output Short Circuited to GND .... Continuous
Continuous Power Dissipation (T_A = +70°C)
16-Pin Plastic DIP (derate 10.53mW/°C above +70°C) 842mW
18-Pin Plastic DIP (derate 11.11mW/°C above +70°C) 889mW
20-Pin Plastic DIP (derate 8.00mW/°C above +70°C) .. 440mW
16-Pin Narrow SO (derate 8.70mW/°C above +70°C) ... 696mW
16-Pin Wide SO (derate 9.52mW/°C above +70°C) 762mW
18-Pin Wide SO (derate 9.52mW/°C above +70°C) 762mW
20-Pin Wide SO (derate 10.00mW/°C above +70°C) 800mW
20-Pin SSOP (derate 8.00mW/°C above +70°C) 640mW
16-Pin CERDIP (derate 10.00mW/°C above +70°C) 800mW
18-Pin CERDIP (derate 10.53mW/°C above +70°C) 842mW
Operating Temperature Ranges
MAX2_ _AC_ _, MAX2_ _C_ _ .... 0°C to +70°C
MAX2_ _AE_ _, MAX2_ _E_ _ .... -40°C to +85°C
MAX2_ _AM_ _, MAX2_ _M_ _ .... -55°C to +125°C
Storage Temperature Range .... -65°C to +160°C
Lead Temperature (soldering, 10sec) .... +300°C

Note 1: Input voltage measured with T_{OUT} in high-impedance state, $\overline{SHDN}$ or V_{CC} = 0V.
Note 2: For the MAX220, V+ and V- can have a maximum magnitude of 7V, but their absolute difference cannot exceed 13V.

Stresses beyond those listed under "Absolute Maximum Ratings" may cause permanent damage to the device. These are stress ratings only, and functional operation of the device at these or any other conditions beyond those indicated in the operational sections of the specifications is not implied. Exposure to absolute maximum rating conditions for extended periods may affect device reliability.

ELECTRICAL CHARACTERISTICS—MAX220/222/232A/233A/242/243

(V_{CC} = +5V ±10%, C1–C4 = 0.1µF, MAX220, C1 = 0.047µF, C2–C4 = 0.33µF, T_A = T_{MIN} to T_{MAX}, unless otherwise noted.)

PARAMETER	CONDITIONS		MIN	TYP	MAX	UNITS
RS-232 TRANSMITTERS						
Output Voltage Swing	All transmitter outputs loaded with 3kΩ to GND		±5	±8		V
Input Logic Threshold Low				1.4	0.8	V
Input Logic Threshold High	All except MAX220		2	1.4		V
	MAX220: V_{CC} = 5.0V		2.4			
Logic Pull-Up/Input Current	All except MAX220, normal operation			5	40	µA
	$\overline{SHDN}$ = 0V, MAX222/242, shutdown, MAX220			±0.01	±1	
Output Leakage Current	V_{CC} = 5.5V, $\overline{SHDN}$ = 0V, V_{OUT} = ±15V, MAX222/242			±0.01	±10	µA
	V_{CC} = $\overline{SHDN}$ = 0V, V_{OUT} = ±15V			±0.01	±10	
Data Rate	All except MAX220, normal operation			200	116	kb/s
Transmitter Output Resistance	V_{CC} = V+ = V- = 0V, V_{OUT} = ±2V		300	10M		Ω
Output Short-Circuit Current	V_{OUT} = 0V		±7	±22		mA
RS-232 RECEIVERS						
RS-232 Input Voltage Operating Range					±30	V
RS-232 Input Threshold Low	V_{CC} = 5V	All except MAX243 $R2_{IN}$	0.8	1.3		V
		MAX243 $R2_{IN}$ (Note 2)	-3			
RS-232 Input Threshold High	V_{CC} = 5V	All except MAX243 $R2_{IN}$		1.8	2.4	V
		MAX243 $R2_{IN}$ (Note 2)		-0.5	-0.1	
RS-232 Input Hysteresis	All except MAX243, V_{CC} = 5V, no hysteresis in shdn.		0.2	0.5	1	V
	MAX243			1		
RS-232 Input Resistance			3	5	7	kΩ
TTL/CMOS Output Voltage Low	I_{OUT} = 3.2mA			0.2	0.4	V
TTL/CMOS Output Voltage High	I_{OUT} = -1.0mA		3.5	V_{CC} - 0.2		V
TTL/CMOS Output Short-Circuit Current	Sourcing V_{OUT} = GND		-2	-10		mA
	Shrinking V_{OUT} = V_{CC}		10	30		

ELECTRICAL CHARACTERISTICS—MAX220/222/232A/233A/242/243 (continued)

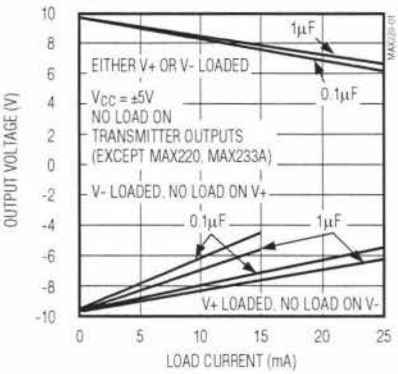
(V_{CC} = +5V ±10%, C1–C4 = 0.1µF, MAX220, C1 = 0.047µF, C2–C4 = 0.33µF, $T_A = T_{MIN}$ to T_{MAX}, unless otherwise noted.)

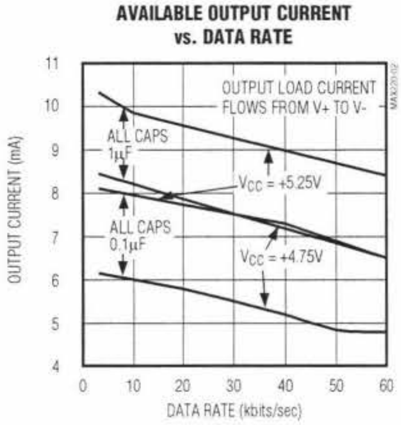
PARAMETER	CONDITIONS		MIN	TYP	MAX	UNITS
TTL/CMOS Output Leakage Current	$\overline{SHDN} = V_{CC}$ or $\overline{EN} = V_{CC}$ ($\overline{SHDN}$ = 0V for MAX222), $0V \le V_{OUT} \le V_{CC}$			±0.05	±10	µA
$\overline{EN}$ Input Threshold Low	MAX242			1.4	0.8	V
$\overline{EN}$ Input Threshold High	MAX242		2.0	1.4		V
Operating Supply Voltage			4.5		5.5	V
V_{CC} Supply Current ($\overline{SHDN} = V_{CC}$), Figures 5, 6, 11, 19	No load	MAX220		0.5	2	mA
		MAX222/232A/233A/242/243		4	10	
	3kΩ load both inputs	MAX220		12		
		MAX222/232A/233A/242/243		15		
Shutdown Supply Current	MAX222/242	T_A = +25°C		0.1	10	µA
		T_A = 0°C to +70°C		2	50	
		T_A = -40°C to +85°C		2	50	
		T_A = -55°C to +125°C		35	100	
$\overline{SHDN}$ Input Leakage Current	MAX222/242				±1	µA
$\overline{SHDN}$ Threshold Low	MAX222/242			1.4	0.8	V
$\overline{SHDN}$ Threshold High	MAX222/242		2.0	1.4		V
Transition Slew Rate	C_L = 50pF to 2500pF, R_L = 3kΩ to 7kΩ, V_{CC} = 5V, T_A = +25°C, measured from +3V to -3V or -3V to +3V	MAX222/232A/233A/242/243	6	12	30	V/µs
		MAX220	1.5	3	30	
Transmitter Propagation Delay TLL to RS-232 (normal operation), Figure 1	t_{PHLT}	MAX222/232A/233A/242/243		1.3	3.5	µs
		MAX220		4	10	
	t_{PLHT}	MAX222/232A/233A/242/243		1.5	3.5	
		MAX220		5	10	
Receiver Propagation Delay RS-232 to TLL (normal operation), Figure 2	t_{PHLR}	MAX222/232A/233A/242/243		0.5	1	µs
		MAX220		0.6	3	
	t_{PLHR}	MAX222/232A/233A/242/243		0.6	1	
		MAX220		0.8	3	
Receiver Propagation Delay RS-232 to TLL (shutdown), Figure 2	t_{PHLS}	MAX242		0.5	10	µs
	t_{PLHS}	MAX242		2.5	10	
Receiver-Output Enable Time, Figure 3	t_{ER}	MAX242		125	500	ns
Receiver-Output Disable Time, Figure 3	t_{DR}	MAX242		160	500	ns
Transmitter-Output Enable Time ($\overline{SHDN}$ goes high), Figure 4	t_{ET}	MAX222/242, 0.1µF caps (includes charge-pump start-up)		250		µs
Transmitter-Output Disable Time ($\overline{SHDN}$ goes low), Figure 4	t_{DT}	MAX222/242, 0.1µF caps		600		ns
Transmitter + to - Propagation Delay Difference (normal operation)	$t_{PHLT} - t_{PLHT}$	MAX222/232A/233A/242/243		300		ns
		MAX220		2000		
Receiver + to - Propagation Delay Difference (normal operation)	$t_{PHLR} - t_{PLHR}$	MAX222/232A/233A/242/243		100		ns
		MAX220		225		

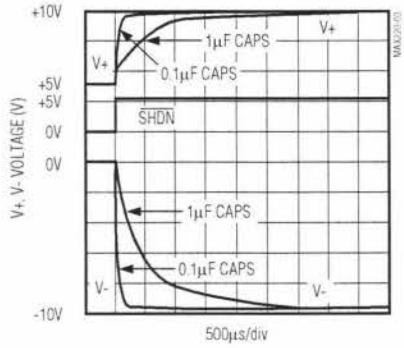
Note 3: MAX243 $R2_{OUT}$ is guaranteed to be low when $R2_{IN}$ is ≥ 0V or is floating.

Typical Operating Characteristics

MAX220/MAX222/MAX232A/MAX233A/MAX242/MAX243







ABSOLUTE MAXIMUM RATINGS—MAX223/MAX230–MAX241

V_{CC} .... -0.3V to +6V
V+ .... (V_{CC} - 0.3V) to +14V
V- .... +0.3V to -14V
Input Voltages
T_{IN} .... -0.3V to (V_{CC} + 0.3V)
R_{IN} .... ±30V
Output Voltages
T_{OUT} .... (V+ + 0.3V) to (V- - 0.3V)
R_{OUT} .... -0.3V to (V_{CC} + 0.3V)
Short-Circuit Duration, T_{OUT} .... Continuous
Continuous Power Dissipation (T_A = +70°C)
14-Pin Plastic DIP (derate 10.00mW/°C above +70°C) 800mW
16-Pin Plastic DIP (derate 10.53mW/°C above +70°C) 842mW
20-Pin Plastic DIP (derate 11.11mW/°C above +70°C) 889mW
24-Pin Narrow Plastic DIP (derate 13.33mW/°C above +70°C) 1.07W
24-Pin Plastic DIP (derate 9.09mW/°C above +70°C) 500mW
16-Pin Wide SO (derate 9.52mW/°C above +70°C) 762mW
20-Pin Wide SO (derate 10 00mW/°C above +70°C) 800mW
24-Pin Wide SO (derate 11.76mW/°C above +70°C) 941mW
28-Pin Wide SO (derate 12.50mW/°C above +70°C) 1W
44-Pin Plastic FP (derate 11.11mW/°C above +70°C) 889mW
14-Pin CERDIP (derate 9.09mW/°C above +70°C) 727mW
16-Pin CERDIP (derate 10.00mW/°C above +70°C) 800mW
20-Pin CERDIP (derate 11.11mW/°C above +70°C) 889mW
24-Pin Narrow CERDIP (derate 12.50mW/°C above +70°C) 1W
24-Pin Sidebraze (derate 20.0mW/°C above +70°C) 1.6W
28-Pin SSOP (derate 9.52mW/°C above +70°C) 762mW
Operating Temperature Ranges
MAX2 _ _ C _ _ .... 0°C to +70°C
MAX2 _ _ E _ _ .... -40°C to +85°C
MAX2 _ _ M _ _ .... -55°C to +125°C
Storage Temperature Range .... -65°C to +160°C
Lead Temperature (soldering, 10sec) .... +300°C

Stresses beyond those listed under "Absolute Maximum Ratings" may cause permanent damage to the device. These are stress ratings only, and functional operation of the device at these or any other conditions beyond those indicated in the operational sections of the specifications is not implied. Exposure to absolute maximum rating conditions for extended periods may affect device reliability.

ELECTRICAL CHARACTERISTICS—MAX223/MAX230–MAX241

(MAX223/230/232/234/236/237/238/240/241, V_{CC} = +5V ±10; MAX233/MAX235, V_{CC} = 5V ±5%, C1–C4 = 1.0µF; MAX231/MAX239, V_{CC} = 5V ±10%; V+ = 7.5V to 13.2V; T_A = T_{MIN} to T_{MAX}; unless otherwise noted.)

PARAMETER	CONDITIONS		MIN	TYP	MAX	UNITS
Output Voltage Swing	All transmitter outputs loaded with 3kΩ to ground		±5.0	±7.3		V
V_{CC} Power-Supply Current	No load, T_A = +25°C	MAX232/233		5	10	mA
		MAX223/230/234–238/240/241		7	15	
		MAX231/239		0.4	1	
V+ Power-Supply Current		MAX231		1.8	5	mA
		MAX239		5	15	
Shutdown Supply Current	T_A = +25°C	MAX223		15	50	µA
		MAX230/235/236/240/241		1	10	
Input Logic Threshold Low	T_{IN}; EN, $\overline{SHDN}$ (MAX233); $\overline{EN}$, SHDN (MAX230/235–241)				0.8	V
Input Logic Threshold High	T_{IN}		2.0			V
	EN, $\overline{SHDN}$ (MAX223); $\overline{EN}$, SHDN (MAX230/235/236/240/241)		2.4			
Logic Pull-Up Current	T_{IN} = 0V			1.5	200	µA
Receiver Input Voltage Operating Range			-30		30	V

+5V-Powered, Multichannel RS-232 Drivers/Receivers

ELECTRICAL CHARACTERISTICS—MAX223/MAX230–MAX241 (continued)

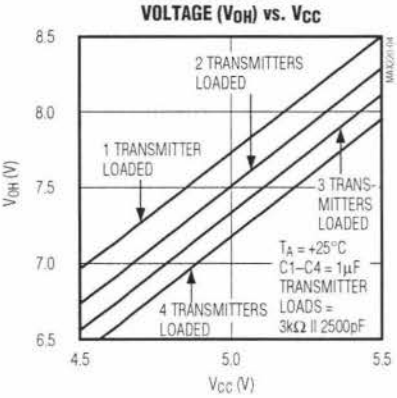
(MAX223/230/232/234/236/237/238/240/241, V_{CC} = +5V ±10; MAX233/MAX235, V_{CC} = 5V ±5%, C1–C4 = 1.0µF; MAX231/MAX239, V_{CC} = 5V ±10%; V+ = 7.5V to 13.2V; T_A = T_{MIN} to T_{MAX}; unless otherwise noted.)

<table>
<tr><th>PARAMETER</th><th colspan="3">CONDITIONS</th><th>MIN</th><th>TYP</th><th>MAX</th><th>UNITS</th></tr>
<tr><td rowspan="2">RS-232 Input Threshold Low</td><td rowspan="2">T_A = +25°C, V_{CC} = 5V</td><td colspan="2">Normal operation
$\overline{SHDN}$ = 5V (MAX223)
SHDN = 0V (MAX235/236/240/241)</td><td>0.8</td><td>1.2</td><td></td><td rowspan="2">V</td></tr>
<tr><td colspan="2">Shutdown (MAX223)
$\overline{SHDN}$ = 0V,
EN = 5V ($R4_{IN}$, $R5_{IN}$)</td><td>0.6</td><td>1.5</td><td></td></tr>
<tr><td rowspan="2">RS-232 Input Threshold High</td><td rowspan="2">T_A = +25°C, V_{CC} = 5V</td><td colspan="2">Normal operation
$\overline{SHDN}$ = 5V (MAX223)
SHDN = 0V (MAX235/236/240/241)</td><td></td><td>1.7</td><td>2.4</td><td rowspan="2">V</td></tr>
<tr><td colspan="2">Shutdown (MAX223)
$\overline{SHDN}$ = 0V,
EN = 5V ($R4_{IN}$, $R5_{IN}$)</td><td></td><td>1.5</td><td>2.4</td></tr>
<tr><td>RS-232 Input Hysteresis</td><td colspan="3">V_{CC} = 5V, no hysteresis in shutdown</td><td>0.2</td><td>0.5</td><td>1.0</td><td>V</td></tr>
<tr><td>RS-232 Input Resistance</td><td colspan="3">T_A = +25°C, V_{CC} = 5V</td><td>3</td><td>5</td><td>7</td><td>kΩ</td></tr>
<tr><td>TTL/CMOS Output Voltage Low</td><td colspan="3">I_{OUT} = 1.6mA (MAX231/232/233, I_{OUT} = 3.2mA)</td><td></td><td></td><td>0.4</td><td>V</td></tr>
<tr><td>TTL/CMOS Output Voltage High</td><td colspan="3">I_{OUT} = -1mA</td><td>3.5</td><td>V_{CC} - 0.4</td><td></td><td>V</td></tr>
<tr><td>TTL/CMOS Output Leakage Current</td><td colspan="3">0V ≤ R_{OUT} ≤ V_{CC}; EN = 0V (MAX223);
$\overline{EN}$ = V_{CC} (MAX235–241)</td><td></td><td>0.05</td><td>±10</td><td>µA</td></tr>
<tr><td rowspan="2">Receiver Output Enable Time</td><td rowspan="2">Normal operation</td><td colspan="2">MAX223</td><td></td><td>600</td><td></td><td rowspan="2">ns</td></tr>
<tr><td colspan="2">MAX235/236/239/240/241</td><td></td><td>400</td><td></td></tr>
<tr><td rowspan="2">Receiver Output Disable Time</td><td rowspan="2">Normal operation</td><td colspan="2">MAX223</td><td></td><td>900</td><td></td><td rowspan="2">ns</td></tr>
<tr><td colspan="2">MAX235/236/239/240/241</td><td></td><td>250</td><td></td></tr>
<tr><td rowspan="3">Propagation Delay</td><td rowspan="3">RS-232 IN to TTL/CMOS OUT, C_L = 150pF</td><td colspan="2">Normal operation</td><td></td><td>0.5</td><td>10</td><td rowspan="3">µs</td></tr>
<tr><td rowspan="2">$\overline{SHDN}$ = 0V (MAX223)</td><td>t_{PHLS}</td><td></td><td>4</td><td>40</td></tr>
<tr><td>t_{PLHS}</td><td></td><td>6</td><td>40</td></tr>
<tr><td rowspan="2">Transition Region Slew Rate</td><td colspan="3">MAX223/MAX230/MAX234–241, T_A = +25°C, V_{CC} = 5V, R_L = 3kΩ to 7kΩ, C_L = 50pF to 2500pF, measured from +3V to -3V or -3V to +3V</td><td>3</td><td>5.1</td><td>30</td><td rowspan="2">V/µs</td></tr>
<tr><td colspan="3">MAX231/MAX232/MAX233, T_A = +25°C, V_{CC} = 5V, R_L = 3kΩ to 7kΩ, C_L = 50pF to 2500pF, measured from +3V to -3V or -3V to +3V</td><td></td><td>4</td><td>30</td></tr>
<tr><td>Transmitter Output Resistance</td><td colspan="3">V_{CC} = V+ = V- = 0V, V_{OUT} = ±2V</td><td>300</td><td></td><td></td><td>Ω</td></tr>
<tr><td>Transmitter Output Short-Circuit Current</td><td colspan="3"></td><td></td><td>±10</td><td></td><td>mA</td></tr>
</table>

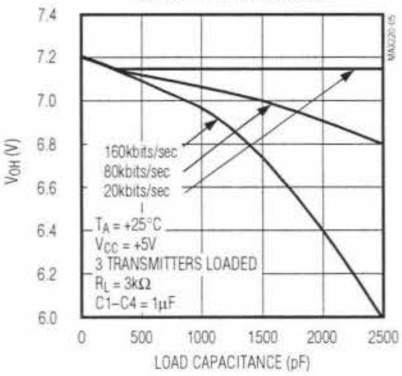
Typical Operating Characteristics

MAX223/MAX230–MAX241

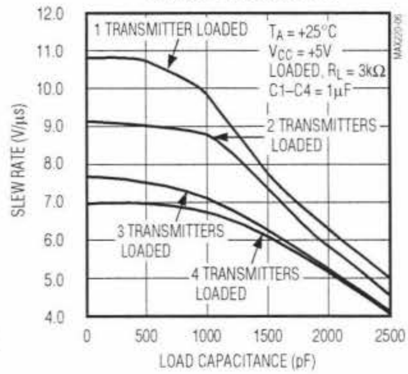
TRANSMITTER OUTPUT VOLTAGE (V_{OH}) vs. V_{CC}



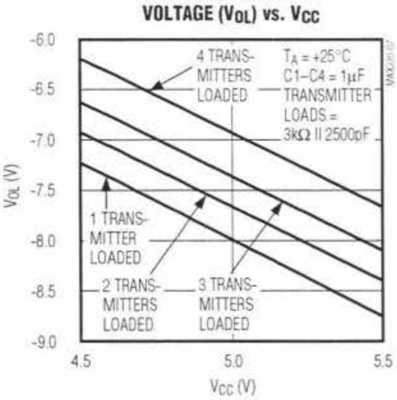
TRANSMITTER OUTPUT VOLTAGE (V_{OH}) vs. LOAD CAPACITANCE AT DIFFERENT DATA RATES



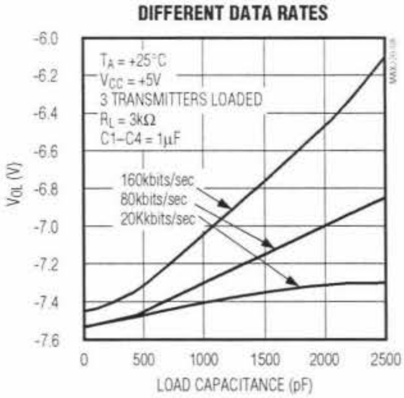
TRANSMITTER SLEW RATE vs. LOAD CAPACITANCE



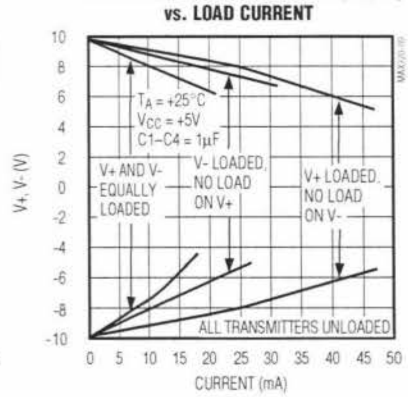
TRANSMITTER OUTPUT VOLTAGE (V_{OL}) vs. V_{CC}



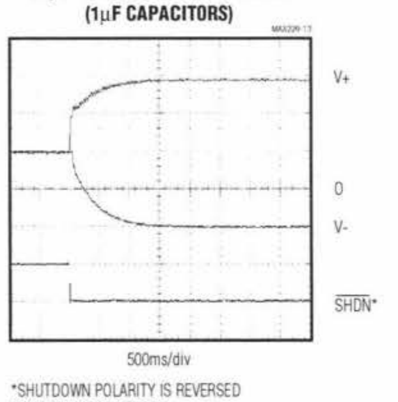
TRANSMITTER OUTPUT VOLTAGE (V_{OL}) vs. LOAD CAPACITANCE AT DIFFERENT DATA RATES



TRANSMITTER OUTPUT VOLTAGE (V+, V-) vs. LOAD CURRENT



V+, V- WHEN EXITING SHUTDOWN (1µF CAPACITORS)



*SHUTDOWN POLARITY IS REVERSED FOR NON MAX241 PARTS

+5V-Powered, Multichannel RS-232 Drivers/Receivers

ABSOLUTE MAXIMUM RATINGS—MAX225/MAX244–MAX249

Supply Voltage (V_{CC}) .... -0.3V to +6V
Input Voltages
$\overline{T_{IN}}$, $\overline{ENA}$, $\overline{ENB}$, $\overline{ENR}$, $\overline{ENT}$, $\overline{ENRA}$,
$\overline{ENRB}$, $\overline{ENTA}$, $\overline{ENTB}$ .... -0.3V to (V_{CC} + 0.3V)
R_{IN} .... ±25V
T_{OUT} (Note 3) .... ±15V
R_{OUT} .... -0.3V to (V_{CC} + 0.3V)
Short Circuit (one output at a time)
T_{OUT} to GND .... Continuous
R_{OUT} to GND .... Continuous

Continuous Power Dissipation (T_A = +70°C)
28-Pin Wide SO (derate 12.50mW/°C above +70°C) 1W
40-Pin Plastic DIP (derate 11.11mW/°C above +70°C) ... 611mW
44-Pin PLCC (derate 13.33mW/°C above +70°C) 1.07W
Operating Temperature Ranges
MAX225C_ _, MAX24_C_ _ .... 0°C to +70°C
MAX225E_ _, MAX24_E_ _ .... -40°C to +85°C
Storage Temperature Range .... -65°C to +160°C
Lead Temperature (soldering, 10sec) .... +300°C

Note 4: Input voltage measured with transmitter output in a high-impedance state, shutdown, or V_{CC} = 0V.

Stresses beyond those listed under "Absolute Maximum Ratings" may cause permanent damage to the device. These are stress ratings only, and functional operation of the device at these or any other conditions beyond those indicated in the operational sections of the specifications is not implied. Exposure to absolute maximum rating conditions for extended periods may affect device reliability.

ELECTRICAL CHARACTERISTICS—MAX225/MAX244–MAX249

(MAX225, V_{CC} = 5.0V ±5%; MAX244–MAX249, V_{CC} = +5.0V ±10%, external capacitors C1–C4 = 1µF; T_A = T_{MIN} to T_{MAX}; unless otherwise noted.)

PARAMETER	CONDITIONS		MIN	TYP	MAX	UNITS
RS-232 TRANSMITTERS						
Input Logic Threshold Low				1.4	0.8	V
Input Logic Threshold High			2	1.4		V
Logic Pull-Up/Input Current	Tables 1a–1d	Normal operation		10	50	µA
		Shutdown		±0.01	±1	
Data Rate	Tables 1a–1d, normal operation			120	64	kbits/sec
Output Voltage Swing	All transmitter outputs loaded with 3kΩ to GND		±5	±7.5		V
Output Leakage Current (shutdown)	Tables 1a–1d	$\overline{ENA}$, $\overline{ENB}$, $\overline{ENT}$, $\overline{ENTA}$, $\overline{ENTB}$ = V_{CC}, V_{OUT} = ±15V		±0.01	±25	µA
		V_{CC} = 0V, V_{OUT} = ±15V		±0.01	±25	
Transmitter Output Resistance	V_{CC} = V+ = V- = 0V, V_{OUT} = ±2V (Note 4)		300	10M		Ω
Output Short-Circuit Current	V_{OUT} = 0V		±7	±30		mA
RS-232 RECEIVERS						
RS-232 Input Voltage Operating Range					±25	V
RS-232 Input Threshold Low	V_{CC} = 5V		0.8	1.3		V
RS-232 Input Threshold High	V_{CC} = 5V			1.8	2.4	V
RS-232 Input Hysteresis	V_{CC} = 5V		0.2	0.5	1.0	V
RS-232 Input Resistance			3	5	7	kΩ
TTL/CMOS Output Voltage Low	I_{OUT} = 3.2mA			0.2	0.4	V
TTL/CMOS Output Voltage High	I_{OUT} = -1.0mA		3.5	V_{CC} - 0.2		V
TTL/CMOS Output Short-Circuit Current	Sourcing V_{OUT} = GND		-2	-10		mA
	Shrinking V_{OUT} = V_{CC}		10	30		
TTL/CMOS Output Leakage Current	Normal operation, outputs disabled, Tables 1a–1d, 0V ≤ V_{OUT} ≤ V_{CC}, $\overline{ENR\_}$ = V_{CC}			±0.05	±0.10	µA

ELECTRICAL CHARACTERISTICS—MAX225/MAX244–MAX249 (continued)

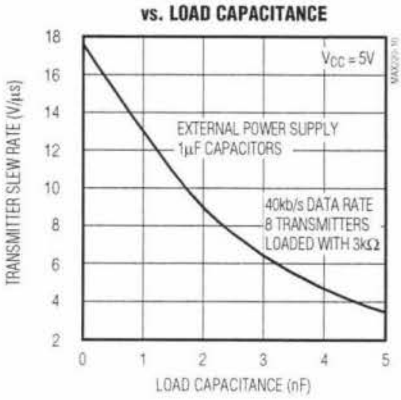
(MAX225, V_{CC} = 5.0V ±5%; MAX244–MAX249, V_{CC} = +5.0V ±10%, external capacitors C1–C4 = 1µF; T_A = T_{MIN} to T_{MAX}; unless otherwise noted.)

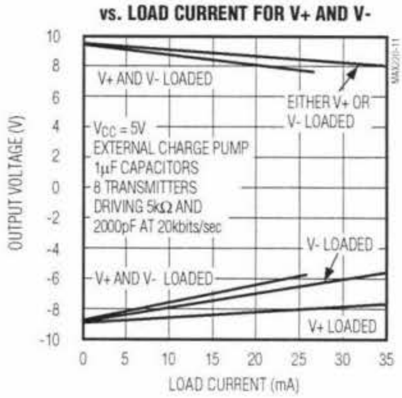
PARAMETER	CONDITIONS		MIN	TYP	MAX	UNITS
POWER SUPPLY AND CONTROL LOGIC						
Operating Supply Voltage		MAX225	4.75		5.25	V
		MAX244–MAX249	4.5		5.5	
V_{CC} Supply Current (normal operation)	No load	MAX225		10	20	mA
		MAX244–MAX249		11	30	
	3kΩ loads on all outputs	MAX225		40		
		MAX244–MAX249		57		
Shutdown Supply Current	T_A = +25°C			8	25	µA
	T_A = T_{MIN} to T_{MAX}				50	
Control Input	Leakage current				±1	µA
	Threshold low			1.4	0.8	V
	Threshold high		2.4	1.4		
AC CHARACTERISTICS						
Transition Slew Rate	C_L = 50pF to 2500pF, R_L = 3kΩ to 7kΩ, V_{CC} = 5V, T_A = +25°C, measured from +3V to -3V or -3V to +3V		5	10	30	V/µs
Transmitter Propagation Delay TLL to RS-232 (normal operation), Figure 1	t_{PHLT}			1.3	3.5	µs
	t_{PLHT}			1.5	3.5	
Receiver Propagation Delay TLL to RS-232 (normal operation), Figure 2	t_{PHLR}			0.6	1.5	µs
	t_{PLHR}			0.6	1.5	
Receiver Propagation Delay TLL to RS-232 (low-power mode), Figure 2	t_{PHLS}			0.6	10	µs
	t_{PLHS}			3.0	10	
Transmitter + to - Propagation Delay Difference (normal operation)	t_{PHLT} - t_{PLHT}			350		ns
Receiver + to - Propagation Delay Difference (normal operation)	t_{PHLR} - t_{PLHR}			350		ns
Receiver-Output Enable Time, Figure 3	t_{ER}			100	500	ns
Receiver-Output Disable Time, Figure 3	t_{DR}			100	500	ns
Transmitter Enable Time	t_{ET}	MAX246–MAX249 (excludes charge-pump start-up)		5		µs
		MAX225/MAX245–MAX249 (includes charge-pump start-up)		10		ms
Transmitter Disable Time, Figure 4	t_{DT}			100		ns

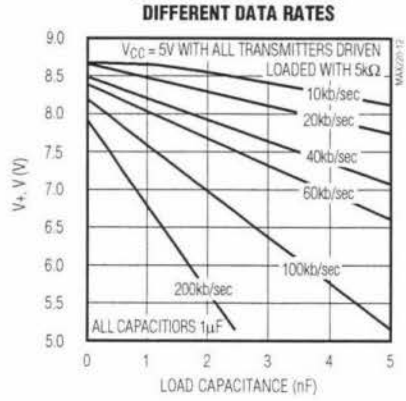
Note 5: The 300Ω minimum specification complies with EIA/TIA-232E, but the actual resistance when in shutdown mode or V_{CC} = 0V is 10MΩ as is implied by the leakage specification.

Typical Operating Characteristics

MAX225/MAX244–MAX249







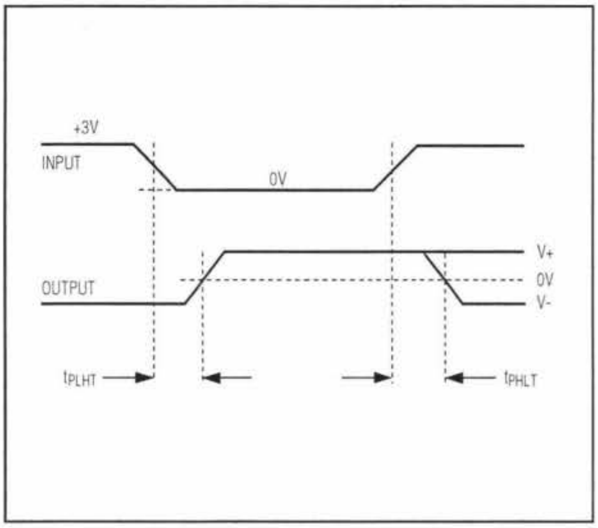


Figure 1. Transmitter Propagation-Delay Timing

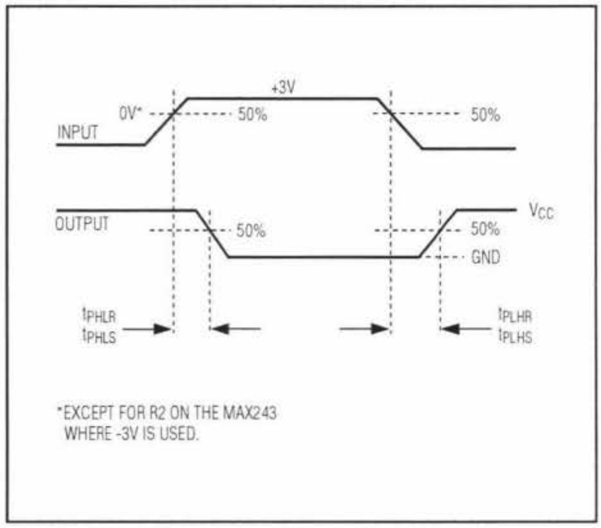


Figure 2. Receiver Propagation-Delay Timing

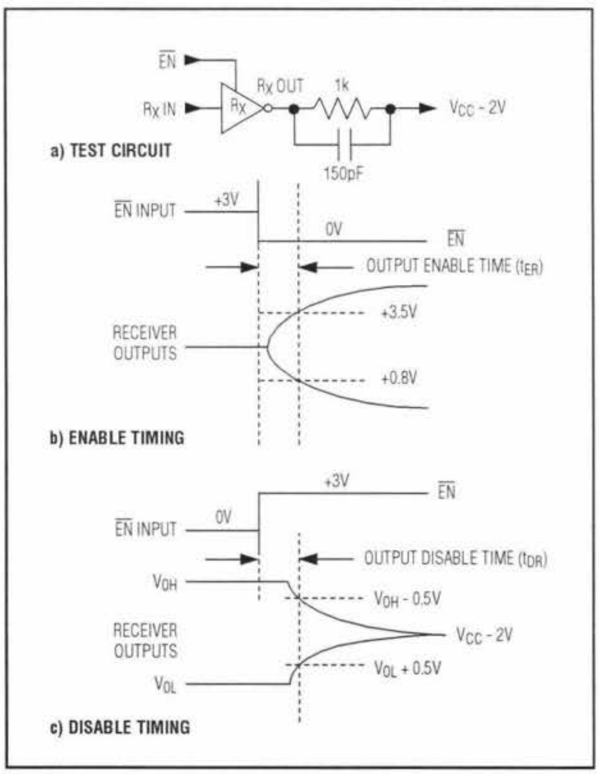


Figure 3. Receiver-Output Enable and Disable Timing

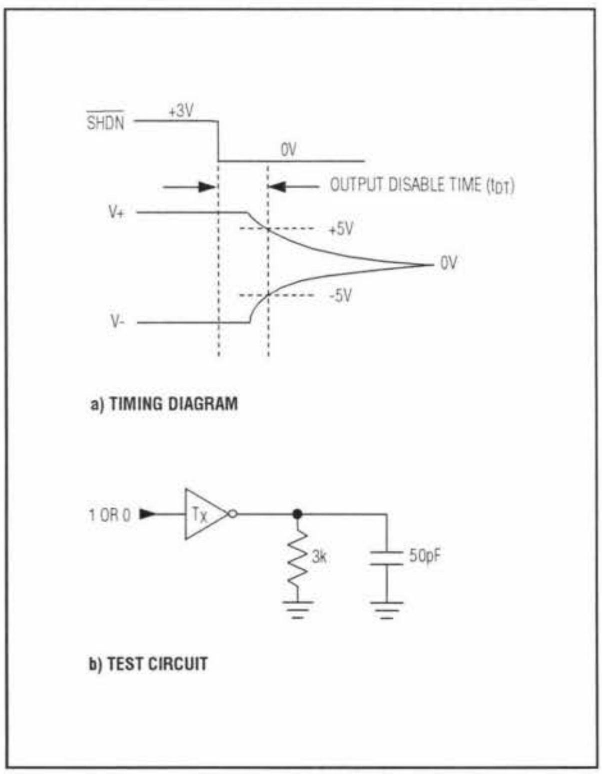


Figure 4. Transmitter-Output Disable Timing