COMPARISON OF THEORETICALLY DETERMINED OPERATIONAL CHARACTERISTICS OF EXISTING (DIESEL) AND COMPRESSED NATURAL GAS (CNG) TRACTOR ENGINES

BY

AKANDE FATAI BUKOLA

M. ENG/SEET/2000/0587

A THESIS SUBMITTED TO THE DEPARTMENT OF AGRICULTURAL ENGINEERING, FEDERAL UNIVERSITY OF TECHINOLOGY MINNA, IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE AWARD OF MASTERS OF ENGINEERING DEGREE (M.ENG) IN FARM POWER AND MACHINERY.

OCTOBER, 2004

DECLARATION

I, Akande, Fatai Bukola (M.ENG/SEET/2000/0587), declare that this thesis; Comparison of the Theoretically Determined Operational Characteristics of Existing Diesel and Compressed Natural Gas, CNG (Alternative fuel) for Tractor Engines, presented for the Award of Masters of Engineering in Agricultural Engineering (Farm Power and Machinery) has not been presented for any other degree elsewhere

---------------------------------- ----------------------------------

Akande F. B. **Date**

CERTIFICATION

This thesis titled "Comparison of theoretically determined operational characteristics of existing (Diesel) and Compressed Natural Gas (CNG) tractor engines" by Mr. Akande, Fatai Bukola (M.ENG/SEET/2000/0587) meets the regulations governing the degree of masters of Engineering (M.ENG.) of the Federal University of Technology, Minna and is approved for its contribution to scientific knowledge and literary presentation.

8.11.04	
Engr. Dr D. Adgidzi	**Date**
Supervisor	
8.11.04	
Engr. Dr. D. Adgidzi	**Date**
Head of Department /Chief Internal Examiner	
Dr. Y. D. Yiljep	23/7/2004
External Examiner	**Date**
6/12/04	6/12/04
Engr. Professor F.O. Akinbode	**Date**
Dean, SEET.	
10/12/04	
Professor J.A. Abalaka	**Date**
Dean, Postgraduate School, (PGS)	

DEDICATION

This project is dedicated to Allah (SWT) for giving me yet another opportunity to attain this level of my academic pursuit in life, Alhamdulillahi!!!

ACKNOWLEDGMENT

The successful completion of this research work is not by my efforts alone, but by the grace of God and the contribution of my able supervisor, lecturers, family members, friends, colleagues and well wishers: therefore, the contribution of these people need to be acknowledged.

I will like to express my profound gratitude to my supervisor and Head of Department, Engr. (Dr) D. Adgidzi for the provision of relevant materials, guidance and most importantly the creation of father to son relationship. "Are you working? Work oh! Work seriously and note that no master no marriage." This coupled with prompt attention encouraged me to work harder.

I am indebted to my senior colleagues in the Department for their contribution and support, they are, Engr (Prof.) E. S. A Ajisegiri, Engr. (Dr.) J.K.Adewumi, Engr. (Dr.) Z. D. Osunde, Engr. (Dr.) B. A. Alabadan, Engr. (Dr.) N. A. Egharevba, Engr. O Chukwu , Engr. P. A Idah, Engr. A. C. Onuachu, Engr. B. Mohammed and my Oga and friend Engr. Ayuba, Audu Balami. The same gratitude goes to the technical crew of the department under the leadership of Mr. A. O. Fabunmi. The following people and colleagues in the school of Engineering and Engineering technology are also acknowledged for their moral support and contributions to ensure the success of this work, they are: Engr. (Professor) F.O Akinbode, Dean School of Engr' g and Engr 'g Tech. Engr (Dr.) O.K. Abubakre, Dr. D . Aloko, Engr. Saka, Fatai, Engr Adeniyi Lekan the computer programmer of this work. Sister .R. Abdulsalam (SAAT), Engr Okegbile and my friend from Lapai, Engr. Abdullahi Mohammed. The assistance of Engr. N. Bale of Gas link in Lagos is also appreciated.

I am indeed grateful to my parent, Mr & Mrs Adesina Akande, my sisters for their spiritual support. The same gratitude goes to my wife, Amina Dasola Akande (Nee Dairo) for her moral and spiritual support and her understanding, my younger brother, Kamorudeen Adewale Akande and his friends, my sister, Yetunde Osowale are also acknowledged for their spiritual support.

To my colleagues most especially Engr. M. O. Owolabi (F. D.A.Abuja), Engr Mohammed Ndako, Mr Adache J. A. & Alhaji Mohammed Chiroma for their constructive criticism and support.

Finally, I am most grateful to Engr (Dr) Mohamed Gana Yisa former H.O.D Agric. Engineering Department, FUT. Minna for his assistance in my place of primary assignment and as my lecturer.

I am much grateful to the following people: Mr Gambari, Alfa Abdul Wahab and Abdul Akeem for their contributions in typing this work diligently. I am also grateful to Alfa Ganiyu Akinsumbo, Ola Ajiboye, sir T, Big Ben, & Alfa Ibrahim and Mesole Victor Yinka (UTB Plc, Kaduna branch.)

ABSTRACT

This work presents the theoretical determination of the operational characteristics of a tractor engine using compressed Natural Gas (CNG) as alternative fuel. Two engines were taken into consideration, the tractor diesel engine (existing) and the tractor CNG engine (design). The basic engine parameters of the existing engine were obtained from Steyr 8075a with effective engine power of 47kW. The basic indicated parameters such as the temperature, pressure and volume at intake, compression, combustion and expansion were determined and these values were used in construction of indicated diagrams for the two engines. From the indicated diagrams, the mean effective pressures were determined to be 0.783 mPa for the existing engine and 0.695 mPa for the design engine. The engine displacement capacities of the two engines were determined to be 3.00 x $10^{-3}m^3$ for the existing engine and for the CNG engine. The effective power for the CNG was determined to be 41.70 kW. The diameter of the cylinder of the existing engine was 0.095m and the radius of the crank is 0.052m. The indicated, specific and the hourly fuel consumption of the two engines were determined and tractor CNG engine was found to be more economical than the tractor diesel engine`. The hourly fuel consumption of the Tractor Diesel Engine at idle running, rated condition and under maximum torque were determined to be 4.61 kg/hr, 18.43 kg/hr and 18.71 kg/hr respectively while that of the Tractor CNG Engine were 2.75 kg/hr, 11.00 kg/hr and 11.17 kg/hr respectively. The specific effective fuel consumption of the two Tractor engines were determined under various condition as indicated to be 461 g/kWhr, 392.15 g/kW.hr and 433.50 g/kW.hr for Tractor Diesel engine and 275 g/kW.hr, 263.79 g/kWhr and 290.21 g/kW.hr for Tractor CNG engine. Fuel consumption in litres per kilometer

under maximum torque was 4.413 litres/km and 3.83 litres/km for Tractor Diesel engine and Tractor CNG engine respectively. The dynamic characteristics of the engine crank system such as the forces acting on the gudgeon pin and the gas pressure forces were determined. The forces of inertia, both first and second orders were equal for the two engines at specific angle of rotation of the crankshaft. The forces acting on the connecting rod were also determined and it is found to be greater in the diesel engine than the design (CNG) engine. Other forces acting on the crank mechanism such as centrifugal force of inertia, which is the same for the two engines, and the tangential forces acting on the crank pin were determined. All these forces were found to be greater in magnitude in the diesel engine than in the CNG engine, this may be due to the cumulative effect of the gas pressure forces. The moments of inertia of the flywheel of the two engines were also determined to be 0.5219 kg/m^2 and 0.4622 kg/m^2 respectively. The mass of the flywheel is determined to be 13.05 kg and 11.55 kg respectively. The CNG is more economical and has fewer tendencies to frequent maintenance/services. Based on the above analysis tractor CNG engine can successfully be used for farm operations. The statistical analysis show that the calculated parameters (CNG engine) are 99.98% correlated with the existing diesel engine parameter, with standard error and the standard deviation ranging within 0 –6%.

TABLE OF CONTENTS

CHAPTER ONE

CHAPTER TWO

CHAPTER THREE

CHAPTER FOUR

CHAPTER FIVE

LIST OF TABLES

LIST OF FIGURES

PARAMETERS FOR COMPUTER PROGRAMMING

	Parameters	Units	Computer/Natural Symbols
1.	Polytropic index for compression	—	n_1
2.	Compression ratio	—	ϵ
3.	Polytropic index for expansion	—	n_2
4.	Number of cylinders	—	i
5.	Ratio of piston stroke to diameter of cylinder	—	S/D
6.	Temperature of residual gases	^{o}K	T_r
7.	Coefficient of excess gases	—	α
8.	Level of pressure increase during combustion	—	λ_p
9.	Coefficient of heat utilised	—	ξ
10.	Rated power	kW	N_e^r
11.	Mechanical efficiency	—	η_m
12.	Rated speed of rotation	Min^{-1}	n_r
13.	Filling coefficient of cylinder	—	η_v
14.	Coefficient incompleteness of indicated diagram	—	γ
15.	Ratio of crank radius to length of connecting rod		λ
16.	Mass of piston assembly	kg	M_p
17.	Mass of connecting rod	kg	M_c

HEAT CALCULATION

Parameters

		Units	Computer/Natural Symbols
1.	Pressure at the end of induction (i.e. commencement of compression)	mPa	p_a
2.	Temperature at the end of induction	°K	T_a
3.	Pressure at the end of compression	mPa	p_c
4.	Temperature at the end of compression	°K	T_c
5.	Actual quantity of air	kmol /kg	L
6.	Quantity of residual gases	kmol /kg	M_r
7	Coefficient of residual gases	-	γ_r
8.	Total quantity of mixture at the end of compression	kmol/kg	M_1
9.	Quantity of gases at the end of combustion	kmol/kg	M_t
10.	Temperature at the end of combustion	°K	T_z
11.	Pressure at the end of combustion	mPa	P_z
12.	Level (index) of expansion of gases	-	ρ
13.	Pressure at the end of expansion	mPa	P_b
14.	Temperature at the end of expansion	°K	T_b
15.	X – coordinates polytropic index curve for compression and expansion	m^3	V_x
16.	Corresponding coordination polytropic index for pressure at compression	mPa	P_x
17.	Pressure co-ordinate for polytropic index for expansion	mpa	P_x
18.	Area of indicated diagram	mm^2	F

19	Mean theoretical indicated pressure	mPa	P^1_i
20.	Mean theoretical indicated pressure calculated by analytical method	mPa	P_i^{cal}
21.	Error in calculation of indicated diagram	-	δ
22.	Actual mean indicated pressure	mPa	P_i
23.	Mean effective pressure	mPa	P_e
24.	Speed of Piston	m/s	V_p
25.	Diameter of cylinder	mm	D
26.	Piston stroke	mm	S
27.	Displacement volume of cylinder	m^3	V_h
28.	Engine capacity	m^3	V_l
29.	Crank radius	mm	r
30.	Indicated fuel consumption	g/kwhr	g_i
31.	Effective fuel consumption	g/kw.hr	g_e
32.	Indicated efficiency	-	η_i
33.	Effective efficiency	-	η_e
34.	Surrounding pressure (atmospheric)	mPa	P_o
35.	Surrounding Temperature	^oK	T_o
36.	Mean motor heat capacity of air	kJ/kmol	μ_{cv}

DYNAMIC CALCULATION

Parameters		Units	Computer/Natural Symbols
1.	Crank angle	rad	α
2.	Force of inertia of the 1st order	N	P_{j1}
3.	Force of inertia of the 2nd order	N	P_{j2}
4.	Force of inertia acting on the gudgeon pin	N	P_j
5.	Pressure (force) of gases	N	P_r
6.	Resultant force acting on the gudgeon pin	N	P_{res}
7.	Components of the resultant force (pres.) acting along the Connecting rod	N	R^1
8.	Centrifugal force of inertia	N	p_c
9.	Tangential force	N	T
10.	Force acting along the crank radius	N	Z
11.	The resultant force acting on the crank pin	N	R
12.	Mean co-ordinate of tangential force diagram for single Cylinder	mm	P
13.	Related power for single cylinder engine	-	Ne^r_1
14.	Coefficient of irregular rotation of crankshaft	-	δ_{cr}
15.	Excess work	Nm	L_{ex}
16.	Moment of inertia of flywheel	kgm^2	J_{fw}
17.	Mass of flywheel	kg	M

CHAPTER ONE

1.0 INTRODUCTION

Energy is the foundation of industrialized world; without energy, life, as we know it would cease to exist. The yearly energy and fuel consumption rate has risen dramatically within the last years. This phenomenon is a direct result of globalization pressures, the international information network we call the Internet and a population that seems to be hitting the dangerous upswing of the Malthusian curve. Although, there is yet a current shortage of conventional fuels such as reserves of coal, oil and other fossil fuels[1] are limited and non-renewable. In addition, the common practice of burning oil, coal and other assorted hydrocarbons has resulted in hazardous environmental conditions such as the global warming, acid rain and dangerously high air pollution levels. This and other environmental disasters have brought about demands for alternative fuels and energy sources that are convenient, environmentally friendly and economically viable. **(Kris, 2002).**

The U.S. Department of energy defines alternative fuel, as fuel that is essentially non-petroleum and yields energy security and environmental benefits. Alternative fuels or alternatives can broadly be considered in two categories; those that serves as replacement for conventional fuels such as Liquefied Petroleum Gas (LPG), "Compressed Natural Gas", Liquefied Natural Gas (LNG), Dimethyl Ether (DME), Hydrogen, (H_2); and those that are blended with conventional fuels, examples being Alcohols (E85, M85) or bio fuels. **(Peter, 2001)**.

Projection of hydrocarbon supply and demand show that at current rates of demand growth, an estimated 3,700 billions barrels of renewable resources hydrocarbon use will not reach its peak until around 2035. In other words, if no breakthrough is made in highly-cost effective alternative fuels conventional or crude oil derived transportation fuels would continue to be readily available and more economic in a market-driven world **(Peter, 2001)**.

Through experimentation with alternative fuels, it soon became clear that alternative fuels had inherent environmental advantages as well. Each alternative fuel has some characteristics that give it an environmental advantage over petroleum fuels. Most are less damaging to the environment if spilled and generally the emission from alternative fuels are less reactive. Another reasons interest in alternative fuels has again centred on energy security is because emission control technology combined with cleaner petroleum fuels such as reformulated gasoline and "Clean diesel" has resulted in emission levels low enough to significantly depreciate the emission benefits of alternative fuels **(Bechtold, 1997)**.

The initial work on alternative fuels focused on which one was best from the viewpoint of technical feasibility, production capability and cost. That question was never answered with certainty and in the interim; development of alternative fuel vehicles technology has proceeded in parallel. Technical feasibility is no longer questioned, and the focus now has shifted more towards which alternative fuels can be produced at a competitive cost. Cost is calculated in terms not only of fuel price, but vehicle price and operating characteristics, and the expense of developing a national fuel distribution infrastructure. In addition, new issues such as public awareness and training of vehicle maintenance personnel have arisen as the use of alternative fuel vehicle spread. Professions only peripherally aware of vehicle technology, such as professional engineers that must design vehicle storage and maintenance facilities, will need to become familiar with the physical characteristics and safe handling practices of alternative fuels.

1.10 What Is Natural Gas?

Natural gas is one of the most abundant fossil fuels and currently supplies about 30% of the energy demand in the United States as shown in Fig. 1.1.

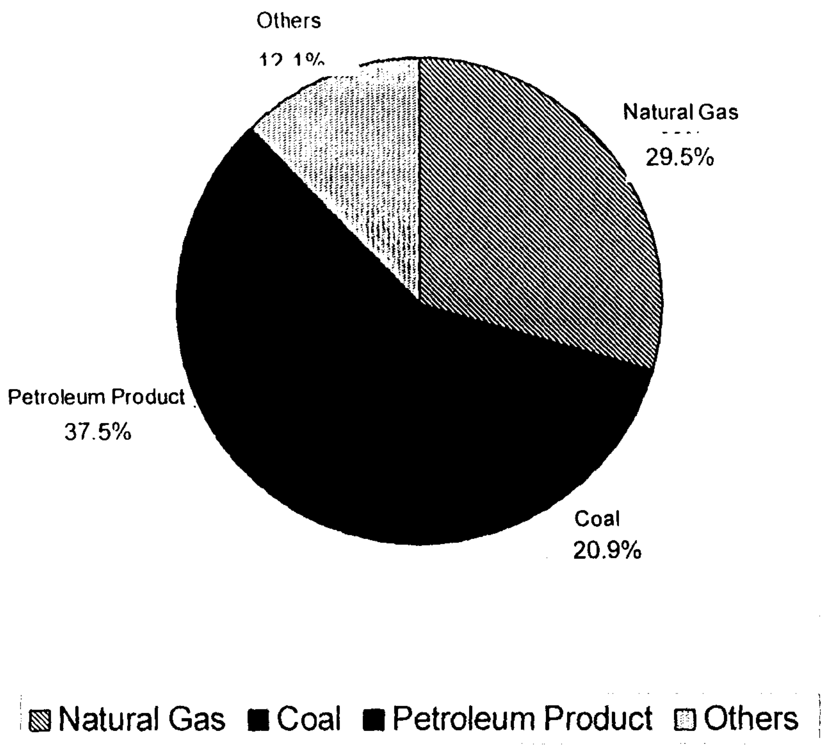


Fig. 1.1: U.S. Energy Consumption, %.

(Source: Energy Information Administration, "Annual Energy Outlook, 1996")

The interest for natural gas as an alternative fuel stems mainly from its clean burning qualities, its domestic resources base and its commercial availability to end-users. The primary uses of natural gas in the U.S. are for space heating, electricity-generation and industrial processes. Natural gas is a very good spark ignition internal combustion engine fuel and it was used as a fuel in the very early days of engine development. However, relative to petroleum fuel, the ability to store sufficient amount of natural gas for on – board vehicle has presented a significant barrier to its broad use as a transportation fuel. Significant advances have been made in high-pressure cylinders that can store natural gas at high pressure (up to 3600 psi about 24.82 mPa) that are made up of light weighted materials including aluminium and carbon fibre. **(Bechtold, 1997)**

Natural gas is the cleanest burning alternative fuel. Exhaust emission from Natural gas vehicles (NGVs) are much lower than those from gasoline powered vehicles. For instance, NGV_S emission or carbon monoxide (CO) is approximately 70% lower, non-methane organic gas emission is 89% lower and oxides of nitrogen emission is 87% lower, in addition to these reduction in pollutants, NGVs also emit significantly lower amount of green house gases and toxins than do petrol vehicles. **(Indiamart, 2002.)**.

Natural gas is available in two forms; compressed Natural gas (CNG) and liquefied Natural Gas (LNG).

1.1.1 Compressed Natural Gas (CNG).

This form of natural gas is compressed into high-pressure fuel cylinder to power a car or truck. It comes from special CNG fuel station. It is odourless, odorant are added to detect leak and spills. It has a heating value that ranges between 49543.38kJ/kg to 54893.6kJ/kg (Kris, 2001), CNG is a zero-sulphur fuel and produces low particulate emission. It produces less carbon mono-oxide (CO) and carbon (iv) oxide (CO_2) emission than the standard hydrocarbon because of its higher energy content. It also has a superior antiknock performance, thus allowing higher compression ratio engines leading to greater efficiency. It also leads to lower NO_X (oxides of Nitrogen) emission **(Peter, 2001).**

1.1.2 Liquefied Natural Gas (LNG)

This is Natural gas that is very cold. It is made by refrigerating natural gas to minus 260°F(-162°C) to condense it into a liquid. This is called liquefaction. The liquefaction process removes most of the water vapour, butane, propane and other trace gases that are usually included in the ordinary natural gas. The resulting LNG is usually more than 98% methane. As at the time of this report (1999), caterpillar, Cummins, Detroit vessel, Mack, and Navistar sell heavy-duty natural gas engines that can operate on LNG (http: www.goggle.com/serarch, 2002).

LNG must be used in conjunction with gas detector because odorants cannot be added. CNG is the preferred method of natural gas storage on vehicles. (LNG) is gaining favour for use in heavy-duty vehicles where use of CNG would still entail space and tracking carrying capacity penalties.

Storing natural gas as LNG enables heavy-duty vehicles have the same operating range as when using liquid petroleum fuels. Typically, storing natural gas as LNG instead of CNG results in a fuel storage system that is less than half the weight and volume of a CNG system. Regardless of the method of storage, the cost and emissions advantages of natural gas make it a very popular alternative fuel (Bechtold, 1997). In other words, the liquid form of natural gas (LNG) has more energy for the amount of space it takes up. It is denser than natural gas or CNG. So, much more energy can be stored in the same amount of space on a car or truck. That means LNG is good for large trucks that need to go a long distance before they stop for more fuel.

Heavy-duty engine such as caterpillar 3126 (tractor) and C –12 engines will run on CNG or LNG. Increasingly stringent standards on engines and vehicle emissions necessitate the development of more sophisticated control approaches for internal combustion engines. Natural gas has significant potentials to reduce vehicle emissions but most dedicated or dual fuel (CNG – petrol) engines currently in operation have fairly primitive systems for controlling air fuel ratio and spark timing. **(:/ Enhancement of Natural gas heavy-duty engine technology. htm, 4//04/02.)**.

1.2.0 Objectives Of The Study

The main objective of this project is to compare the theoretically determined operational characteristics of existing (diesel) and CNG (Alternative Fuel) tractor engines. The specific objectives of this work are as stated below:

(i) To design an internal combustion engine that will be powered by Compressed Natural Gas (CNG) and to compare this design with an existing diesel tractor engine with respect to the mean effective pressure, effective power and the diameter of the piston bore (cylinder).

(ii) To determine the fuel economy indices of the two engines and make necessary comparison to enhance the use of CNG as alternative fuel for tractor engines.

(iii) To determine the Kinematics, Dynamics and flywheel parameters of the tractor CNG engine to know the forces and moments acting on its crank mechanism and the reciprocating components.

1.3 Justification Of The Study

Petroleum is found between Natural Gas and water beneath the earth surface to a depth of 4600 – 9200 meters. Natural gas is found in the highest proportion compared to others to the extent that currently several countries are flaring off vast quantities of natural gas, a wastage that could easily be converted into a power source for generating set and to drive tractors and automobiles.

Countries such as Nigeria are seen as ideal for gas-powered generating sets, as good quality natural gas is readily available, cheaper than diesel power. The gas engines is readily available, cheaper than diesel power, the gas engines also emit less emissions consequently complying with all environmental regulations **(African Review, 2002)**.

Fuel formulation on the other hand is a huge area that is as yet largely unexplored. Recent research shows that alternative fuels can significantly reduce soot and NO_X (Oxides of Nitrogen) emission simultaneously, while preserving the high efficiency and power density of diesel-cycle engine. In addition, many alternative fuels are renewable, so they have the potential to reduce petroleum imports as well as green house gas emission.

CHAPTER TWO

2.0 LITERATURE REVIEW

2.1.0 Sources Of Energy (Fuel)

One of the characteristics of technically advancing society, at least before 1978 is that the rate of growth of energy use is considerably higher than the rate of population growth and closely parallel to the growth of economic output **(Sporn, 1957 in Liljedahl et al, 1989)**. Possession of surplus energy is a requisite for any kind of advanced civilization. If human beings must depend strictly on their own muscle, it will take all of their mental and physical strength to supply the basic necessities of life. The difference in per capital energy consumption is closely related to differences in standard of living **(Liljedahl et al, 1989)**. The important sources of energy are as follows:

(i) Solar Energy (direct)

(ii) Solar energy (indirect); Fossil fuel – oil, "Natural Gas", Coal, Peat and shale and tar sands (b) Biomass (wood, corn cobs etc), (c) Wind, (d) Tiles, (e) water

(iii) Nuclear energy, and

(iv) Geothermal energy.

Most of our present power use is coming from stored fossil fuel, oil, gas, and coal. Hence, we are in the fossil-fuel age. How long those supplies will last is debatable, but they are non-renewable. Modern farming in the world today depends on petroleum fuels, so it is of interest to know when the supply will begin to diminish. The date is uncertain, but in the United States, for example, production of petroleum has not met demand since 1947 **(Scarlott, 1957 in Liljedahl et al 1989)**. In 1973, United States imported about 40% of its oil needs **(Ray, 1973)** and by 1978 over 50% was imported.

2.1.1 Oil Shale

From oil shale, it is possible to obtain nearly a barrel of oil per tonne are available in considerable but indeterminate amounts. For power plant use, some attention is being given to recovery of the oil by methods other than mining. **(Liljedahl et al, 1989)**

2.1.2 Solar Energy

Solar energy is our one inexhaustible energy source. A rough measure of energy falling on the earth is a maximum of approximately $940W/m^2$ and on average of $630W/m^2$. Although, this energy source will likely become very useful for space heating and for agricultural operations such as crop drying. Presently, it is doubtful that it will be useful for operating tractors because of low concentration of energy and lack of suitable means of collecting and concentrating it. Another problem is the large difference in the amount of available solar energy at different times of year and day and at different locations. **(Liljedahl et al, 1989)**

2.1.3 Nuclear Energy

This energy will ease the load on fossil fuels, as it is well adapted to large power plant operation, where radiation shielding can be employed. It is a concentrated, clean and easily controlled energy source, the energy is 4 ml of Uranium 235 being equivalent to energy stored in $140m^3$ of coal **(Nuclear Energy, 1961)**. Controlled fusion may offer some hope for future power plants, even those as small as a farm tractor, since fusion atomic energy does not require heavy shielding because it is not radioactive.

2.2.0 Alternative Fuel And Their Origin

In this part of the write-up, the various alternative fuels would be explained and the reasons for considering them for use in transportation vehicles. The processes of production of each of the alternative fuels would be presented and the recent production volumes in the developed nations where the use of Alternative fuel vehicle is utmost. The typical impact on vehicle performance in terms of power, drivability, cold-start capability and others would be presented to include the vehicle emission characteristics.

The Alternative fuels under consideration are; Alcohols (Methanol & Ethanol), Natural Gas {Compressed Natural Gas (CNG), Liquefied Natural Gas (LNG) & Liquefied Petroleum Gas (LPG)}, Vegetable oils, hydrogen, Electric fuel.

2.2.1 The Alcohol

Methanol and ethanol are the alcohols (primary alcohols) considered to be potential transportation alternative fuels. Other higher alcohols have not been seriously considered as alternative fuels for use unmixed with other fuels in engines. Tertiary Butyl Alcohol (TBA) has been used as a gasoline extender and co-solvent when mixing methanol with gasoline but not as a fuel by itself. Recently, Di Methyl Ether (DME) made using methanol; has been proposed for use as a diesel engine alternative fuel because of its favourable emissions characteristics relative to using diesel fuel.

Methanol and ethanol are both liquid and have several physical and combustion properties similar to gasoline and diesel fuels hence they make good substituted as alternative fuels. These properties are similar enough so that the same basic engine and fuel system technologies can be used for methanol and ethanol as for gasoline and diesel fuel. Both methanol and ethanol have higher octane rating than typical petrol-which allows alcohol engines to have much higher compression ratios, increasing thermal efficiency. However, they have lower energy density, compared to the conventional fuels. **(Bechtold, 1997).**

Methanol and ethanol have inherent advantages compared to conventional petrol and diesel fuels in that their emissions are less reactive in the atmosphere producing smaller amount of ozone, the harmful component of smog. Methanol and ethanol have the disadvantage in that they produce formaldehyde and acetaldehyde as combustion by-products in larger quantity than the toxic compounds from the petroleum products they replace.

Methanol and ethanol were long considered as a good spark-ignition engine alternative fuel, it has also been proved that they can be used as diesel engine alternative fuels. A significant advantage of alcohol fuels is that when they are combusted in diesel engines, they do not produce any soot or particulate and can be tuned to also produce very low levels of oxides of nitrogen.

2.2.1.1 Methanol

Methanol (CH_3OH) is an alcohol fuel. As engine fuels, ethanol and methanol have similar chemical and physical characteristics. Methanol is methane with one hydrogen molecule replaced by a hydroxyl radical.

```
      H
      |
H — C — OH
      |
      H
```

(Ababio, 2001)

It is produced from natural gas in production plants with 60% total energy efficiency. The currently proffered process of producing methanol is steam reformation of natural gas. In this process, any sulphur present in natural gas is first removed. Next, the natural gas is reacted with steam in the presence of a catalyst under high heat and pressure to form carbon monoxide (CO) and hydrogen. These elements are then put through the methanol production catalyst to make methanol. Methanol can be made with any renewable resources containing carbon such as seaweed, waste wood and garbage. In 1995, methanol production in the U.S. totaled 6.5 billion litres ($6.5x10^6$ m^3 i.e. 1.7 billion gallons), which made it the 21st, ranked chemical in term of use **(Chemical & Engineering News, 1996)**.

This is a promising alternative, with a diversity of fuel applications with proven environmental, economic and consumer benefits. It is widely used today to produce the oxygenate of MTBE (Methyl, tertiary butyl ether) added to cleaner burning gasoline. Cars, trucks and buses running millions of miles on methanol have proven its use as a total replacement for gasoline and diesel fuels in conventional engines. In 1995, 10.0 billion litres (10^7m^3), (2.64 billion gallon) of MTBE was produced in the U. S making it the 12^{th} most used chemical **(Chemical & Engineering News, 1996)**.

2.2.1.1.1 Vehicle Emission Characteristics

Methanol burns without a visible flame, which is a safety concern, this also demonstrates that methanol does not produce soot or smoke when combusted. Methanol exposure studies have shown that methanol does not cause any harm in the quantities that would accumulate in the body from exposure from refueling vapour or from unburned methanol in the exhaust **(HEI, 1994)**. Methanol has a high latent of vaporization, hence, peak combustion temperature can be reduced with correspondingly low emissions of oxides of nitrogen (NOx).

Methanol is less photo- chemically reactive than gasoline. Its evaporative emission contributes less to smog formation; and because of it oxygen content, it facilitates leaner combustion resulting in low CO emissions.

Emission of "M85" in spark-ignition light-duty vehicles have been shown to have significant reduction in specific Ozone – reforming potentials **(SAE, 1994)**. Methanol contains no sulphur, so it does not contribute to atmospheric sulphur (iv) oxide (SO_2), SOx and NOx emissions lead to acidic deposition, use of methanol would make a minor contribution to reducing acidic rain **(DOE/PE – 0100P, 1991)**. The CO_2 emissions of methanol vehicles are theoretically about 94% those of similar petroleum fuelled vehicles, assuming they have the same fuel efficiency **(U.S. Dept. of Energy, 96)**.

2.2.1.1.2 Vehicle Performance Impacts

Methanol spark-ignition engines have the capability to be 15 – 20% more efficient than their petrol counterparts. This is achieved through lean-burn technology, made possible by methanol's wide flammability limits. Beside, having superior thermal efficiency, lean-burn engines have lower exhaust emissions with simpler oxidation-catalyst technology.

Ford built 630 dedicated methanol Escorts from 1981 to 1983. The engine in these vehicles had 20% higher torque than the original petrol versions. Acceleration from 0 to 96km/hr was 15.8 seconds on M85 compared to 16.7 seconds on petrol. Subsequent Crown Victorias had an acceleration time of 12.2 seconds on M85 compared to 12.6 seconds on petrol **(Clean Fuels Report, 1991)**.

Flexible Fuel Vehicles (FFVs) are able to operate on petrol, M85 or any mixture of two fuels. This is achieved through the use of a fuel composition sensor in the fuel line from the vehicle tank to the engine. The engine control system automatically adjusts the Air-Fuel (A/F) ratio and the spark timing for the blend of methanol and gasoline travelling to the engine.

FFVs do leave room for improvement as General motors demonstrated by optimising 3.1L Lumina to dedicated M85 operation. The engine "M85" – a fuel blend of 85% methanol and 15% petrol – added to enhance cold start engine compression ratio was increased from 8-9 (used in FFV version) to 11.0. This was obtained by using smaller piston – bowl, and crevice volume. Performance results as compared to the FFV predecessor are shown in table 2.1 **(Wang, 1996)**.

Table 2.1 Performance Results

Characteristics	**FFV**	**Dedicated**
Acceleration, 0 – 96km/h (sec)	1.04	9.4
Quarter mile (sec)	17.7	17.2
Fuel Economy (mpg[a])	36.6	37.9
Emissions (g/ml)		
OMHCE	0.15	0.06
CO	2.06	0.80
NOx	0.16	0.22
Formaldehyde (mg/ml)	14.4	5.7

(Source: Wang, M, Q, 1996)

a = gasoline equivalent

2.2.1.2 Ethanol

Ethanol is the second member in the group of Alcohols after methanol. It is also known as ethyl alcohol, grain alcohol or ETOH, and is a clear colourless liquid with a characteristic agreeable odour. Two higher blends of ethanol, E85 and E95 are being explored as alternative fuels in demonstration programs. Ethanol is also made into ether. Ethyl Tertiary-Butyl Ether (ETBE) that has 10% ethanol blends reduces CO better than any reformulated gasoline blend. Ethanol is a safe replacement for toxic octane enhancers in gasoline such as benzene, toluene and xylene.

Ethanol has long been considered as a good spark-ignition engine fuel, and engines were run on ethanol very early in engine development **(Bechtold, 1997)**. It has a very good combustion properties and it is self sufficient i.e. it can be produced by the agricultural sector, which would satisfy their needs and sell the excess to others.

Ethanol produced for use as fuel (fuel ethanol) in the U.S. is produced almost exclusively using fermentation technology. The preferred feedstock is corn, though other grains and crops such as potatoes and beets can be used. Agricultural wastes such as cheese are also considered good feedstock for ethanol production. Almost any source of starch or sugar is a potential feedstock for ethanol production.

U.S. currently has a production capacity of fuel ethanol of 1.1 billion gallons ($4.125x10^6$ m^3) per year: In 1991, the U.S. produced 875 million gallons ($3.28x10^6$ m^3) of fuel ethanol that was exported to Brazil. U.S. imports about 25million gallons ($9.375x10^4$ m^3) of ethanol per year from Caribbean countries **(Bechtold, 1997)**.

2.2.1.2.1 Vehicle Emission Characteristics

Ethanol by itself has a very low vapour pressure, but when blended in small amounts with petrol, it causes the resulting blend to have a disproportionate increase in vapour pressure. The primary emission advantage of using ethanol blends is that CO emissions are reduced through the "blend-leaning" effect that is caused by the oxygen content of ethanol. The oxygen in the fuel contributes to combustion much the same as adding additional air.

Test carried out by Ford on their 1996 Model Taurus FFV using E85 on tail pipe emission compared to using the standard emissions testing gasoline (Indolene) showed that the engine-out emissions of HC and NOx were lower than for petrol. E85 produces acetaldehyde instead of formaldehyde when methanol or M85 is combusted. Ford found that the level of acetaldehyde was the same as for M85 **(Cowart, 1995)**. An advantage of acetaldehyde over formaldehyde is that it is less reactive in the atmosphere, which contributes less to ground level ozone formation.

Low sulphur content of E85 should be a benefit in reducing catalyst deterioration compared to vehicles using petrol. The vehicle technology to use E85 is virtually the same as that to use M85.

2.2.1.2.2 Vehicle Performance Impacts

Flexible Fuel Vehicles (FFVs) built to use E85 should experience very similar driveability as when using gasoline (petrol). Though performances should be improved by about 5% because of the intake charge cooling effects (high latent heat of vaporization) and high Octane number of ethanol. In a FFV using E85 fuel efficiency in energy term is equal or slightly better since the combustion properties of ethanol should favour a more aggressive is used to optimize an FFV for use of E85 lead to higher performance and greater fuel efficiency.

2.2.2 Natural Gas

Natural gas is a mixture of hydrocarbon mainly methane (CH_4) and is produced either from gas wells or in conjunction with crude oil production. The interest of natural gas as an alternate fuel stems mainly from its clean burning qualities, its domestic resources base, and its commercial availability to end - users.

Ethane is typically the only other hydrocarbon found in significant amounts in natural gas, though often less than 10 percent volume. Natural gas may also include carbon (iv) oxide, nitrogen, and very small amounts of hydrogen and helium. The composition of natural gas is important because of its heating value, and physical properties may change which can affect combustion.

The properties of natural gas are dominated by "methane". Methane is widely acknowledged to be formed from four sources; (a) organic matter that is decomposed in the presence of heat; (b) organic matter that is converted through the action of micro-organisms; (c) Oil and other heavy hydrocarbons that produce methane in the presence of heat; (d) Coal which releases methane overtime **(Oppenhelmer, 1981)**.

Very little processing needs to be done on natural gas to make it suitable for use as fuel. Water vapour, sulphur, and heavy hydrocarbons are removed from natural gas before it is sent to its destination, usually via a pipeline.

Compared to liquid hydrocarbon, and other alternative fuels natural gas contain less energy per unit volume. For this reason, transport over long distances and across oceans is not practical except when liquefied. As discussed earlier, natural gas used for transportation fuels can either be CNG or LNG.

2.2.2.1 Vehicle Emission Characteristics

Natural gas is made up of methane mainly and this dominates its emissions characteristics. CH_4 mixes readily with air and has a high octane rating which makes it a very good "spark-ignition engine fuel". It has a high ignition temperature that makes it unsuitable for use in compression-ignition engines. Methane barely participates in the atmospheric reactions that produce ozone though it does contribute to global warning when released to the atmosphere. Because of its high hydrogen to carbon ratio (4:1), methane produces about 10% less CO_2 than combustion of equivalent petrol or diesel fuel (CO_2 is the primary contributor to global warning). Natural gas can be used for both light and heavy-duty vehicles and have varying emission characteristic.

(a) **Light-duty vehicles (CNG-Fuelled):** - are capable of low gaseous exhaust emissions. In the CNG vehicles developed by the auto manufacturers to date, individual port fuel injection with three-way catalyst system has been used to simultaneously oxidize exhaust CO and hydrocarbon while reducing the oxides of nitrogen, NOx **(Bechtold, 1997)**.

Dedicated Natural Gas vehicles produce little or no evaporative emissions during fuelling and use. For gasoline engine, evaporation and fuelling emissions account for at least 50 percent of a vehicle's total hydrocarbon emissions **(Indiamart, 2002)**.

(b) **Heavy-duty vehicles: -** Heavy-duty natural gas engines to date have been spark-ignition adaptation of diesel engines. To improve engine efficiency to be closer to that of diesel engines, the heavy-duty natural gas engines use lean-burn combustion.

For emission control, they use an oxidation catalyst to control methane and carbon monoxide emissions. NOx are kept low through lean combustion and particulate emissions are of no concern. Heavy-duty vehicles using natural gas favour storing natural gas as LNG instead of CNG. LNG vehicles have the same exhaust emission as CNG vehicles except that LNG vehicles might occasionally vent methane from the fuel storage system.

2.2.2.2 Vehicle Performance Characteristics

(a) **Light-Duty vehicle: -** can increase their power and efficiency by increasing compression ratio. Compared to typical petrol, natural gas has a high octane rating, which supports higher compression ratios. However, there are two detriments to light-duty natural gas vehicle performance: weight of the fuel system and decrease engine specific power output. The weight of the natural gas fuel system will always be more than a liquid fuel system carrying the amount of gasoline energy. The weight naturally hurts vehicle acceleration and will degrade adequate fuel economy proportionately.

Natural gas light-duty vehicle should have the same good durability characteristics that gasoline vehicles have obtained.

(b) **Heavy-duty vehicle: -** Have the same or more power than the equivalent displacement diesel version. Limiting factors to power output includes oxides of nitrogen emissions (increased power means, richer operation and generation of more

NOx for a given displacement engine) and exhaust value life (Spark ignition engines experience higher exhaust value temperature than diesel engines) rather than acceleration, heavy duty engine performance is more a function of maximum horsepower (hp) and torque size.

2.2.3 LP Gas (Propane)

LP gas or LPG means Liquefied Petroleum Gas. LPG includes several light hydrocarbons whose main distinguishing characteristics are becoming a liquid when put under modest pressure (less than 300 psi about 2.07 mPa). Propane and butane are the most common LP gases and, for vehicle use, LPG is essentially all propane and is most often called propane.

Propane is produced as a by-product of natural gas processing and in association with crude oil refining. It is undesirable to leave LP gases in natural gas as they come out of the ground because of their tendency to liquefy under modest pressure.

The major use of LP gas is for space heating in homes and for commercial purpose, though it is an important feedstock for petrochemicals. Propane is a popular barbecue grill fuel and many forklift trucks operating inside warehouses use propane. LP gases other than propane (the most important being butane) could be used also in vehicle fuel with appropriate engine modification.

2.2.3.1 Vehicle Emission Characteristics

Propane shares several emissions advantages with natural gas and has some additional ones to its own. Like natural gas, propane vehicles do not have any evaporative or running loss emissions associated with the fuel. Unburned hydrocarbons are easier to oxidize in oxidation catalysts than methane, which results in low unburned hydrocarbon emission. Unburned hydrocarbon emission from propane is also less reactive by about 2/3 compared to unburned hydrocarbon from gasoline **(Bassey et al 1993)**. Propane being a gas mixes very well with the air before entering the engine, resulting in low carbon-monoxide emissions. Oxide of nitrogen emissions can also be reduced to a very low level if the air-fuel ratio is kept at the stoichiometric value, using a three-way catalyst.

2.2.3.2 Vehicle Performance Characteristics

(a) Light-Duty Vehicle: - For vapourized propane fuel systems propane enters the engine as a gas instead of part liquid and part gas as gasoline does by entering the engine fully vapourized, some air that could otherwise be used for combustion is displaced. Therefore, theoretically, propane vehicles should have lower power and slower acceleration than their gasoline counterparts, especially on bi-fuel configuration. Acceleration in terms of 0-80 km/h, time can be up to 10% slower. Driveability should be acceptable in systems that have been set up and maintained properly. Cold-start can be problematic since the mechanical control systems rely on engine airflow to meter propane.

2.2.4.0 Vegetable Oils

The interest in using vegetable oil as alternative fuels originated within the agricultural community as a fuel for agricultural tractors and equipment. In 1982, the American Society of Agric. Engineers held the International Conference on plant and vegetables as fuels **(Duke and Bagby, 1982)**. This conference made the point that vegetable oils could be a viable alternative fuels for use in diesel engines. In Europe, vegetable oils were tested in engine dynamometer, and field tests were conducted in tractors, trucks and diesel engine passenger cars.

The most popular types of crops from which vegetable oils can be extracted include Soybeans, Sunflower, Peanuts, rapeseed, and Chinese fallow trees. Dozen of candidate plants can yield significant oil yield for acre **(Duke and Bagby, 1982).**

Initially, it was believed that vegetable oils could be used directly with minimal processing and preparation. However, engine testing proved that while diesel engines operated satisfactorily in "raw" vegetable oils, combustion residues and deposits would quickly cause problems with fuel injectors, piston rings and oil stability. The esterified version of the vegetable oils have been given the generic label of "Biodiesel" which have much improved characteristics of fuels. The favourable properties of biodiesel reduce smoke, particulates and gaseous emissions when used in a typical transit bus. The major impediments to use of such blends of Biodiesel "(B20)" with diesel fuel, is the cost of biodiesel compared to some other alternative fuels that could be used.

2.2.4.1 Vehicle Emissions Characteristics

Straight biodiesel (Soymethyl ester) has a cetane rating significantly higher than typical No 2 diesel fuels, slightly lower heating values, slightly higher viscousity and contain approximately 10% mass of oxygen. The lower heating value will cause a small loss in maximum power if the engine is not recalibrated. In a pre-chamber diesel engine using a transient eight-mode test, straight Soymethyl ester showed a significant reduction in hydrocarbon emissions, no significant change in carbon monoxide (CO) emissions, a slight reduction in oxides of nitrogen emissions, reduced particulate emissions and lower mutagenicity of the particulate matter formed. Other benefits of using Soymethyl ester include reduced toxic emissions, very low sulphate emissions and much more pleasant odours.

2.2.4.2 Vehicle Performance Characteristics

On a mass basis, neat biodiesel has approximately 13% less energy than typical diesel fuel (this is as a result of approximately 10% oxygen content in biodiesel) Biodiesels have higher specific gravity of approximately 0.88 compared to 0.82 for diesel fuels regain some of the loss energy on a mass basis for an overall impact of approximately 7% loss in energy per unit volume.

Because of the lower energy per unit volume, vehicles using neat biodiesel should experience a loss in fuel economy of about 7% on average. Biodiesel has higher "viscousity" and higher "pour points" compared to typical diesel fuel, which could affect operation in very cold temperature. Like diesel fuels, pour point additives are effective at decreasing pour point.

Engine oil dilution is a potential problem with biodiesel, since it is more prone to operation and polymerization than diesel fuel. The presence of biodiesel in engine oil could cause thick sludge to occur with the consequence that the oil becomes too thick to pump.

2.2.5.0 Hydrogen

Hydrogen has many characteristics that make it the "ultimate" alternative fuel to fossil energy fuels. Hydrogen can be combusted directly in internal combustion engines or it can be used in fuel cells to produce electricity with high efficiency (30-50% over the typical load range). When hydrogen is oxidized in fuel cells, the only emission is water vapour. When combusted in internal combustion engine (ICE), water vapour is the major emission, though, some oxides of nitrogen may be formed if combustion temperatures are high enough. Therefore, the use of hydrogen as a transportation vehicle fuel would result in few or no emission that would contribute to Ozone formation.

Since hydrogen molecule do not occur in nature, it is typically produced by "reforming" a hydrocarbon or alcohol fuel or by using electricity to split water into hydrogen and oxygen. The size of the contribution of hydrogen fuel to carbon (iv) oxide (CO_2) emissions depends on the source of the hydrocarbon fuel that was reformed or the source of the electricity used to split water. Research is underway to develop novel, non-polluting means of hydrogen production such as from Algae that makes use of sunlight or other biological methods **(Bechtold, 1997).**

The major drawback to using hydrogen as a fuel is the storage medium compared to all other fuels, hydrogen has the lowest energy storage density. Hydrogen can be stored as a compressed gas at pressure of 20.68 mPa similar to CNG, liquefied or store in metal hydride (which absorbs hydrogen when cooled and release it when heated, and at a temperature of – 252.9^{0}C). In 1995, the United State produced 6.7 billion cubic meter of hydrogen, though this does not include hydrogen produced in refineries used for hydro treating petroleum products **(Chemical and Eng'g News, 1996).**

2.2.5.1 Vehicle Emission Characteristics

When hydrogen is validated in fuel cell, the only significant emission is water vapour, when combusted in I.C.E. (Spark or diesel) some oxides of nitrogen and peroxides may be produced. None of the toxic emissions typical of petroleum fuels are present **(Swain et al, 1983)**.

Like CNG or Propane vehicles, hydrogen vehicles should not produce evaporative emission since fuel system would be closed.

2.2.5.2 Vehicle Performance Characteristics

Only experimental hydrogen vehicles have been built to date, and it is not possible to derive meaningful conclusion about vehicle performance characteristics from them. Based on the I.C.E work conducted to date, hydrogen engines should be able to produce the same amount of power that petroleum fuel engines do with superior efficiency since the lean limit of hydrogen is much lower than petroleum or other alternative fuels. A major concern is the operating range, and present research reveals that some models can reach a speed of 144 km/hr and can travel up to 450 km before they need refueling **(:/Fuel cell vehicle htm, 1980)**.

2.2.5.3 Fuel Cell Vehicle – The Zero Emission Vehicles Of The Future

Another zero-emission vehicles (ZEV_S) is the fuel cell powered by vehicles when the fuel cells are fuelled with pure hydrogen, they are considered to be zero emission vehicles. Fuel cells have been used on spacecraft for many years to power electric equipment. These are fuelled with liquid hydrogen from the spacecraft rocket fuel tanks.

Fuel cell vehicles turn hydrogen fuel and oxygen into electricity. The electricity then powers the electric motor; just like electricity from batteries power the motor of an electric vehicle. Fuel cells combine oxygen from air with hydrogen from the vehicle fuel tank to produce electricity. When oxygen and hydrogen are combined they give off energy and water. In a fuel cell, this is done without any burning (combustion).

There are number of ways that hydrogen can be produced to the fuel cell. One way is to put hydrogen gas into the fuel cells along with air. Hydrogen gas can come from gaseous or liquid hydrogen stored in the vehicle.

The other way to produce hydrogen to the fuel cell is to store it on vehicle in liquid form. To make hydrogen liquid, it is chilled and compressed. Liquid hydrogen is very cold more than -252.9^0C.

Another way to get hydrogen into fuel cell is to use a reformer. A reformer is a device that removes hydrogen from a hydrogen fuel like methanol or gasoline. There is another type of fuel cell that can be fuelled with methanol directly. This is called a direct-methanol fuel cell. This type of cell does not need a reformer to separate the hydrogen from methanol. The fuel cell removes the hydrogen from the liquid methanol inside the fuel cell **(Fuel cell vehicle.htm, 1980).**

2.2.6 P. Series Fuel

P-series is a new fuel that is classified as an alternate fuel. It is the latest to be taken under the branch of alternative fuels. The U.S. government has just recently added certain blends of Methyl tetra hydro furan (MTHF), ethanol and hydrocarbon known as the P-series fuel to the definition of "alternative fuel".

P-series fuels are blends of ethanol, methyl tetra hydro furan (MTHF) and pentane plus, with butane added for blends that would be used in severe cold weather conditions to meet cold-start requirements.

These contain at least 60% of non-petroleum energy content derived from MTHF (Manufactured Solely from biomass feed stocks) and ethanol are substantially not petroleum and may yield substantial energy security and substantial environmental benefits **(Indiamart, 2002).**

2.2.7 Solar Fuel

Solar energy technologies use sunlight to warm and light homes, heat water and generate electricity. Some research has gone into evaluating how solar energy may be used to power vehicles. However, the long-term possibility of operating a vehicle on solar power alone is slim. Solar power, may however, be used to run certain auxiliary systems in the vehicle. Solar energy is derived from the sun in order to collect this energy and use it to fuel a vehicle; photovoltaic cells are used. Pure solar energy is 100% renewable and a vehicle run on this emits no emissions **(Indiamart, 2002)**.

2.2.8 Biomass Fuel

Biomass fuel is an energy source derived from living organisms. Most commonly, it is the plant residue, harvested dried and burned, or further processed into liquid or gaseous fuels. Biomass is a known renewable source of energy since green plants are essentially solar collectors that capture and store sunlight in the form of chemical energy. The most familiar and widely used is wood, but cereal straws, seed hulls, corn stalks and even animal waste and garbage are used. Dry wood has a heating value of 8500 Btu/lb (about 19.77 mJ/kg). Wood accounted for 25% of all energy used at the beginning of this century, but with increase in use of fossil fuels, its significance rapidly declined in 1976, only 1 to 2% of the U.S. energy is supplied by wood.

Biomass is still a primary source of energy for developing societies. In a few instances, it is also a major source of power as in Brazil, where sugarcane is converted to ethanol and in China where fuel gas is obtained from dung and in Western Europe there are over 200 power plants that burns rubbish to produce electricity.

A drawback of using biomass is that it is a solid fuel and solid fuels are not as convenient or versatile as liquid or gases. However, techniques can convert the biomass into liquid or gaseous form. These techniques include partial combustion, anaerobic digestion, or fermentation. Growing plants removes CO_2 from the atmosphere that is released back to the atmosphere when biomass fuels are used. Thus, the overall concentration of atmospheric CO_2 should not change and global warning should not result.

Biomass contains less sulphur than most fossil fuels, therefore, biomass fuels should reduce the impact of acid rain.

A total net production of biomass energy has been estimated at 100 million mega watts per year. Forest and woodlands account for about 40% of the total and oceans about 35%. Considering all the constraints on biomass harvesting as estimated that about 6 million mW/year of biomass is available for energy use **(Kris, 2002)**.

2.3.0 Fuel Chemistry/Combustion Process Of Alternative Fuels

Source of fuels: - Almost all of the fuels commonly used in farm tractors and other automobiles are products of crude petroleum. There are so many factors to be considered in the selection of an alternative fuel. Some of these factors are: - cost per unit work done, availability, compatibility with the engine, safety, storage, management, and convenience.

2.3.1 Chemical Composition Of Petroleum

Crude petroleum is made up of combined carbon and hydrogen in approximately the proportion of 86% carbon to 14% hydrogen. The atoms of carbon and hydrogen combine in many ways to form many different hydrocarbon compounds in crude oil. The fuels made from crude oil are never one single hydrocarbon. The engineers are concerned mainly with the physical properties and operating characteristics of a fuel.

The hydrocarbon making up most of the refined fuels belong to the paraffin (C_nH_{2n+2}) Olefin (C_nH_{2n}), diolefin (C_nH_{2n-2}), naphthalene (C_nH_{2n}) and their aromatic ($C_nH_{2n-6}(n=6,12)$) families. The first member of the homologous series of the paraffin (Alkane) is methane (CH_4, n = 1) which is the major constituent of the natural gas (CNG or LNG) and n =3, C_3H_8, the major constituent of liquefied petroleum gas – Propane. When n = 1 to 4, the hydrocarbon are gases at room temperature and pressures. Butane (C_4H_{10}) boils at 0^oCat atmospheric pressure. Alkanes (Paraffin) and a few other hydrocarbons exhibit Isomerism while the normal compounds of the paraffin series have straight – chain molecular structures.

The paraffin is saturated and quite stable and low in gun-forming properties. The straight-chain (normal) hydrocarbons generally detonate in an engine while the isomers are highly knock-resistant. For example, n-heptane is the low anti-knock fuel (octane no = 0) used as a references fuel in knock testing. Triptane, an isomer of heptane, is one of the most knock-free hydrocarbons known. These fuels have the same chemical formula but they are structurally different and vary in widely burning characteristics.

The Olefins (Alkenes) are important because they occur in fuels made by the cracking process. They are more resistant to detonation and are unsaturated, hence, they undergo additional reaction with hydrogen becoming saturated (paraffin).

The diolefin are unsaturated but have straight chain structures. This family of hydrocarbon is the principal cause of instability of cracked gasoline in storage, causing gum and off-colour and lower anti-knock properties.

Naphthenes have same chemical formula as the olefins, but they are saturated compound and have high anti knock values. They have molecules made up of carbon and hydrogen with a ring structure, and the prefix cycles is used in their name, e.g. Cyclo hexane (C_6H_{12}).

The aromatics are ringed structured. They are unsaturated but stable chemically e.g. benzene (C_6H_6), Alpha methyl naphthalene ($C_{11}H_{16}$) is another aromatics of importance because it is used as a reference fuel in testing diesel fuel. It has a high anti-knock value as a spark ignition engine, but poor ignition quality as a diesel fuel.

2.3.2 Combustion Process For Hydrocarbon

The combustion process consists of chemically combining oxygen from the air with carbon and hydrogen in the fuel. Heat is liberated in the process and a pressure increase results. The final or desired result occurs during the power and expansion stroke when a volume change takes place and work is done.

Air is the source of oxygen. By weight, air is 23.1% oxygen and 76.9%nitrogen. But air by volume consists of 20.8% oxygen, and 79.2% nitrogen, thus, for 1kg of oxygen (O_2) there is 3.33kg of Nitrogen (N_2) and for each cubic meters of oxygen in air, there is 3.8m^3 of Nitrogen **(Lijedahl et al, 1989)**.

Hydrocarbon fuels combustion under ideal conditions would produce CO_2 and water (H_2O) as the only products according to the equation below:

$C_nH_m + PO_2 + 3.76PN_2 \Rightarrow nCO_2 + (^m/_2)\, H_2O + 3.76PN_2 + \Delta H^oC$, -----------------(2.1)

Where n and m are positive integers, P is a positive number, and delta H^oC is the heat of combustion, or "enthalpy" under ideal conditions. The specific reactions for four alternative fuels under ideal conditions are given below:

(i) Compressed Natural Gas (CNG)

$CH_4\,(g) + 2O_2 + 7.52\ N_2 \Rightarrow CO_2\ 2H_2O + 7.52\ N_2 + 797570$ kJ/kmol

(ii) Liquefied Petroleum Gas (Propane)

$C_3H_8 + 5O_2 + 18.8\ N_2 \Rightarrow 3\ CO_2 + 4\ H_2O + 18.8N_2 + 2{,}032{,}800$ kJ/kmol

(iii) Methanol

$CH_3OH + 1.5O_2 + 5.64N_2 \Rightarrow CO_2 + 2H2O + 5.6N_2 + 851{,}840$ kJ/kmol

(iv) Ethanol

$C_2H_5OH + 3O_2 + 11.38H_2 \Rightarrow 2\ CO_2 + 3\ H_2O + 11.38N_2 + 1{,}541{,}000$ kJ/kmol.

However, hydrocarbon fuels combustion under actual (non-ideal) combustion conditions produces several intermediate products in addition to CO_2 and H_2O. This is the general combustion reaction for any hydrocarbon fuel under non-ideal conditions:

$$C_nH_{2n} + O_2 + N_2 \Rightarrow C_kH_y + aCO_2 + bCO + CH_2O + dNO_x + eN_2O + fH^+ + gOH^* + hO^* + jN_2 + \Delta Hc. \text{ -----------(2 .2)}$$

Where C_kH_y = unburned kg or partially burned hydrocarbon fuel, CO_2-Carbon (iv) Oxide, H_2O – water, CO – Carbon (ii) Oxide, NOx = Oxides of Nitrogen (x = 1 - 3), N_2O – Nitrous oxide, OH^* – Hydroxyl radical, O^* – Oxygen radical and or single oxygen, H^+ = hydrogen ion, ΔHc = heat of combustion (enthalpy) under actual (non-ideal) conditions and co-efficient a, b, c, d, e, f, g, h, j vary with fuels and operating condition. The heat of combustion is also referred to either as the "lower heating value" (with "gaseous" water as one of the products) or as the higher "heating value" (with liquid water as one of the products).

Combustion in internal combustion engines is a very complex chain reaction system that leads to hundred of intermediate products, by - products and end-products (Alternatives to Tradition transportation fuel, 1994). In I.C.E., the product in the exhaust are determined by a large amount of parameters including the fuel to air ratio, compression ratio, fuel consumption, internal design, operating condition of the cylinder and the combustion chamber, and exhaust inflation. For methane, propane and methanol fuels, the numbers of identified reaction products in the exhaust are 25, 41 and 84 respectively **(Alternative to traditional transportation fuel, 1994).**

2.3.3 Incomplete Combustion

When a hydrocarbon fuel burns completely the oxygen in the air combining with the hydrogen to form water and with carbon to form carbon (iv) oxide. If the burning is not complete, then some of the carbon atoms only combine with one oxygen rather than two, to form carbon (iv) oxide, (CO), a highly poisonous gas. Some of the carbon atoms may remain stuck together with each other and with some of the hydrogen atoms as well, so that unburned hydrocarbon molecules can also come out of the tailpipe: These unburned hydrocarbons (plus any fuel hydrocarbon that evaporates from the fuel system before getting to be burned at all) react with nitrogen oxides (another pollutant from combustion) in the presence of sunlight to form ozone which is a lung irritant (the ozone layer in the stratosphere is a shield against the sun ultraviolet light, but a ground level ozone is the main component of the "photochemical Smog"). Carbon atom can also remain stuck to one another with few or no hydrogen atom attached, especially during incomplete combustion of diesel fuel producing soot.

This is one of the reasons alternative fuels are less polluting than gasoline (petrol) and diesel: their simpler molecules are easier to burn more completely in an engine, so that less CO, soot and unburned hydrocarbons come out of the tailpipe. In addition, any hydrocarbon that is produced is less reactive than those produced from the incomplete burning of petrol or diesel fuel and they produce less ground level ozone. Methane (CNG) particularly is almost incapable of producing smog.

2.3.4 Oxygen Content

Some alternative fuels are not hydrocarbon; alcohol and biodiesel- contain oxygen atoms as well as carbon and hydrogen. The chemical structure of common alcohols is as shown in Fig. 2.1.

Biodiesel molecules are monoalkyl esters, the ester part indicates that the molecules include oxygen atoms. RCOOR' where R – alkyl group, COO-ester, R'-metal **(Ababio, 2001).**

In many part of U.S.A, petrol is "oxygenated" i.e. oxygen bearing compounds are added to the fuel mixture, this is to promote more complete combustion so that less CO, soot unburned hydrocarbon come out of the tailpipe. Alcohols and biodiesels carry this one step further in that oxygen-bearing compounds are not additives at 5 – 10% level but major constituents of the fuel which increases the benefit of oxygenation.

2.3.5 Carbon Content

Even if, with the aid of electronic engine controls and efficient catalytic converters, a hydrocarbon fuel is completely burned to water and carbon (iv) oxides. There is now a growing concern about CO_2 as a green house gas, measures to cut back the production of CO_2 by automobiles without sacrificing performance focused on efficiency i.e. getting as much useful propulsive power out of a given amount of fuel as possible. Typical example involves replacing the traditional drive train of a piston engine driving the wheels through a gearbox with a more efficient design.

However, some fuels inherently produce less CO_2 when burned than gasoline or diesel fuel for example, counting the numbers of oxygen atoms it takes to burn up Isooctane molecule (gasoline) and methane (Natural Gas) molecule. One can calculate that 100 oxygen atoms will combine with four Isooctane molecules to produce 32 CO_2 and 36 water molecules, while, the same number of oxygen atoms will combine with 25 methane molecules to produce 25 CO_2 molecules and 50 water molecules. That is, a given amount of air (O_2) will produce about 25% less CO_2 if used to burn natural gas than if used to burn gasoline **(FUEL Chemistry.htm, 80).**

2.3.5.1 Avoiding CO_2 Emission Completely

The source of carbon in the fuel needs to be considered; if it came from the CO_2 in today's air to start with, like an alcohol fuel produced by fermenting Biomass (as opposed to a fossil fuel, whose carbon came out of the air when the dinosaurs were around), then returning it to the atmosphere (air) now adds nothing to the net flow of CO_2 into the atmosphere. Alcohol fuels, biodiesel produced from plant when burned just return to the air the CO_2 that those plants took out from the air while growing.

Finally as stated earlier, hydrogen is the only fuel in use that produce no CO_2 when combusted.

2.3.6 Air – Fuel Ratio In Combustion Process

When combustion is complete, carbon and hydrogen burn to CO_2 and H_2O respectively. Engine fuels seldom are chemical compounds, and combustion calculation serves only to illustrate general principles. Hydrocarbon gas (CH_4), the simplest of the paraffin series can be used to illustrate the combustion process. Atoms and molecular weight of some elements entering into the combustion process are given in table 2.2

The calculations of the masses of the compound are as follows:

$$CH_4 + 2O_2 + 7.6N_2 \Rightarrow CO = 2H_2O = 7.6N_2$$

$$(12 + 4) + (2 \times 32) + (7.6 \times 28) \Rightarrow (12 + 32) + 2(2 + 16) + (7.6 \times 28)$$

$$16 + 64 + 2128 \Rightarrow 44 + 36 + 212.8$$

$$292.8g = 292.8g$$

From the above calculations, it can be seen that 64 kg of oxygen is required for 16 kg of fuel since air is represented by oxygen plus nitrogen in the process. 276.8 kg of air is required for 16 kg of fuel. The theoretically correct air – fuel ratio becomes

$$= \frac{Air}{Fuel} = \frac{276.8}{16} = \frac{17.3}{1}$$

Calculating on a volume basis,

1 mole of CH_4 + 2 moles of O_2 + 7.6 mole of N_2 = 1 mole CO_2 + 2mol H_2O +7.7mol N_2

Or $1m^3$ of CH_4 $+2m^3$ O_2 $+7.6m^3$ $\rightarrow$ $1m^3CO_2$ $+2m^3H_2O$ + $7.6m^3$ N_2 **(Liljcdahl ct al, 1989).**

Table 2.2 – Atomic and Molecular weights of Common combustion Substances

Substance	Symbol for Element	Atomic Weight	Symbol for Molecules	Molecular Weight
Carbon	C	12	C	12
Hydrogen	H	1	H	2
Oxygen	O	16	O_2	36
Nitrogen	N	14	N_2	28
Sulphur	S	32	S	32
Air-apparent	-	-	-	29.0

(Lijcdahl ct al, 1989).

2.4.0. Thermodynamic Principles Of Internal Combustion Engine

Thermodynamics is the study of changes in which energy is involved. The first law of thermodynamics states that: "All forms of energy are mutually convertible. The energy of a closed and isolated systems remains constant".

2nd law of thermodynamic states that: "In real systems heat cannot be fully converted to work". i.e. no engine can convert all the energy supplied to it, in as much as a large part of the energy supplied is rejected in the form of unused heat.

In heat engines, work is done by changes in the volume of gas. Certain laws have been established concerning their pressure, volume and temperature relationship such laws for perfect gases. These include Boyle's law, Charle's law and the general gas law

$P_1V_1 = P_2V_2 = P_nV_n$....................(2.3) (Boyle's Law)

$\frac{V_1}{T_1} = \frac{V_2}{T_2} = \frac{V_N}{T_N}$ ....(2.4) Charle's Law

At constant pressure **(Liljedahl et al, 1989).**

And at constant volume, Charles's Law states that;

$\frac{P_1}{T_1} = \frac{P_2}{T_2} \Rightarrow \frac{P_1}{P_2} = \frac{T_1}{T_2}$ ....(2.5)

Combining equations (2.3) and (2.4) gives the general gas law

$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} = \frac{P_NV_N}{T_N}$ ....(2.6) General Gas Law

From (2.6) $\frac{PV}{T} = R \Rightarrow PV = RT$ where R is a constant called characteristic gas constant. When mass of gas considered is involved,
PV = mRT-----------------------(2.7)

Where R = 8.314J/gmoleK

2.4.1 Specific Heat

Specific heat is defined as the amount of heat necessary to change the temperature of a unit mass by one-degree. For a system of fixed energy, the situation may be represented by the following statements of the first law

$\Rightarrow Q = Mc\,(T_2 - T_1) = (U_2 - U_1) + W$------------------------ (2.8)

Q = heat gained or rejected,

M = Mass of Gas,

$U_2 - U_1$ = Change in internal energy,

W = Work done

C = Specific heat, and C could be C_v or C_p

C_V = Specific heat at constant volume,

C_p = Specific heat at constant pressure

$C_p/C_v = k$.

2.4.2. Entropy

Entropy as used in engineering thermodynamics has two principal applications when considering ideal or reversible processes. First, as a co-ordinate on a diagram in which heat transfer is shown graphically; second, as an index of unavailability of heat energy for conversion into work. ($TdS = dQ$)

For an ideal process, +S indicate or $+\Delta H^oC$ heat energy is added during a process and

-ΔH°C or – S indicate heat rejected and zero entropy change signifies no heat energy is added or rejected. For real processes TdS>dQ. The three energy quantities are related as follows

$$Q = (U_2 - U_1) + W \text{ --------------------------------(2.9)}$$

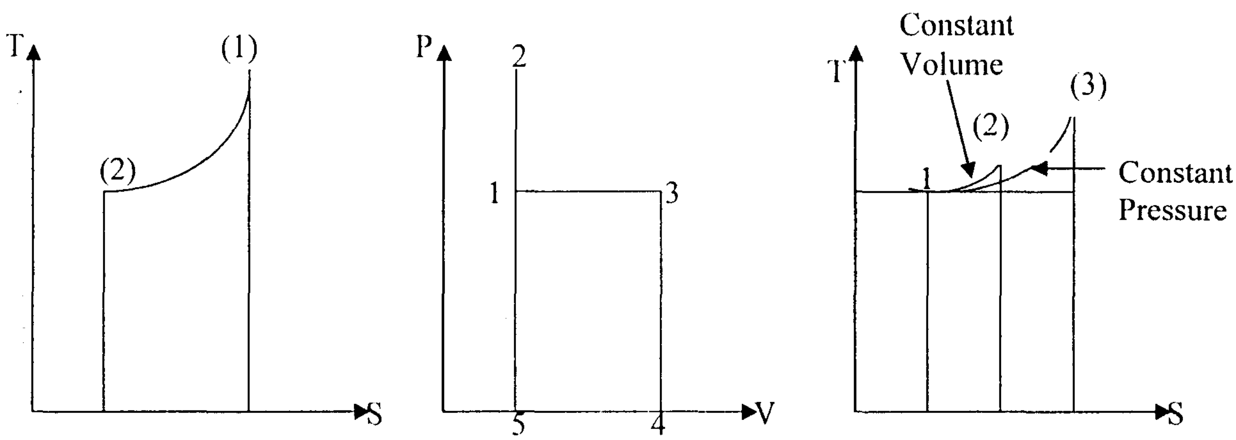


Fig. 2.4.2 (a): Heat energy represented as area on temperature – entropy plane.

Fig. 2.4.2 (b): Constant Volume and constant pressure curves on pressure – volume plane.

Fig. 2.4.2 (c): Constant Volume and constant pressure curves on temperature – entropy plane.

(Liljedahl et al, 1989)

Constant-volume heating is represented by the lines 1-2 in fig. 2.4.2 (b) & c. The heat supplied = C_v (T_2-T_1) = area 1-2-5-6. There is no work done by or upon the gas and hence, W = 0. And there is no area covered on the P-V plane. Substitution in the general energy equation gives $C_v (T_2 - T_1) = (U_2 - U_1) + \delta''$ ------------------------(2.10)

All heat supplied appears as an increase in internal energy i. e.

$$C_v (T_2 - T_1) = (U_2 - U_1) \text{ ----------------------------(2.11)}$$

Constant pressure heating is represented by the line 1-3 in Fig. 2.4.2 b & c. heat supplied = $C_p (T_3-T_1)$ = area 1-3-4-6 in Fig. 2.4.2c. The work done is $W = P(V_3-V_1)$ = area 1-3-4-5 in Fig. 2.4.2.b . Substitution in the general energy equation gives

$$C_p (T_3-T_1) = C_v (T_3-T_1) + P (V_3-V_1) \text{ ------------------------(2.12)}$$

But $P_3V_3 = RT_3$ & $P_1V_1 = RT_1 \Rightarrow P_1 = P_3$

$$\Rightarrow C_p (T3-T1) = C_v (T3-T1) + R (T3-T1) \text{------------------(2.13)}$$

$$\therefore \quad C_p = C_v + R \text{ ----------------------}$$

$$R = C_p - C_v \text{ -------------------------(2.14)}$$ **(Liljedahl et al, 1989)**

2.4.3 Isothermal Changes

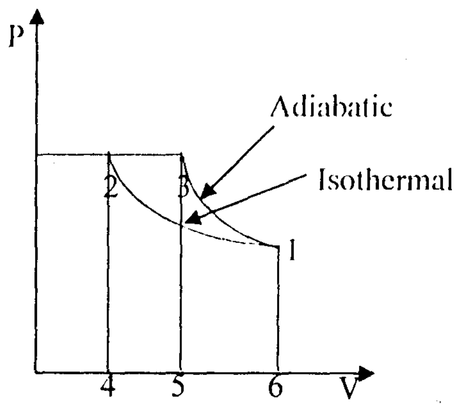


Fig. 2.4.3 (a): Isothermal And Adiabatic Compression lines on a P-V Plane.

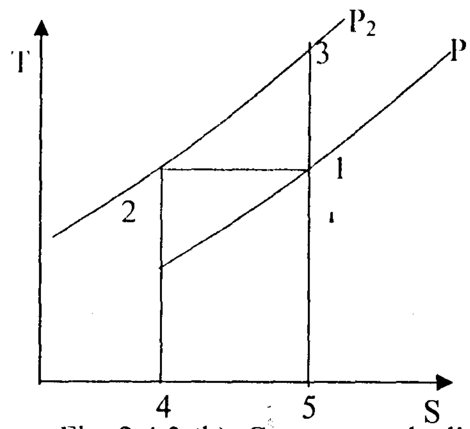


Fig. 2.4.3 (b): Constant and adiabatic Compression Line on temperature entropy plane.

For a perfect gas,

$$PV = C = \text{a constant} \text{ ---(2.15)}$$

Eqn 2.15 is a special case of the general equation $PV^n = C$ where $n = 1$ and the pressure is represented by the line 1-2 in fig 2.4.3 a & b. The external work is represented by the area under the line between 1 and 2. The work done during any change in volume from V_1 to V_2 is given by;

$$W = \int_{v_1}^{v_2} pdv \text{ --- } 2.16$$

From equation 2.15 p = c/v, substituting p = c/v in equation (2.16)

$$w = \int c/_{v}\, dv = c\int_{v_1}^{v_2} \frac{dv}{v} = c\left[\log V\Big|_{v_1}^{v_2}\right]$$

$$w = c\,(\log eV_2 - \log eV_1) \text{ .... } (2.17)$$

Since $C = P_1V_1 = PV$, substituting this expression into eqn (2.17) gives

$$W = P_1V_1 \log e\, V_2/V_1 \text{ .... } (2.18)$$

For any mass of gas other than unity, these may be substituted and

$P_1V_1 = MRT$ & $V_2/V_1 = P_1/P_2$ thus the work becomes

$$W = MRT \log e \frac{V_2}{V_1} = MRT \log e\, P_1/P_2 \text{ .... } (2.19)$$

Letting $r = V_1/V_2$ be the compression ratio or ratio of expansion, we have

$$W = MRT \log e \frac{1}{r} \text{ .... } (2.20)$$

In an isothermal expansion, temperature is constant; the gas receives or rejects an amount of heat to the work done by it or on it.

$$\therefore Q = W = MRT \log e\, 1/r = MRT \log e\, V_2/V_1 \text{ .... } (2.21)$$

2.4.4 Adiabatic Or Constant Entropy Change

In an adiabatic expansion or compression, the gas neither receives nor rejects heat. The line 1-3 in fig 2.4.3 a&b represent an adiabatic compression from a pressure P_1 to pressure $P_3 = P_2$. The work on the gas is represented in fig. 2.4.3 a by the area 1-3-5-6.

Since it is an adiabatic change, no heat is received or rejected to an outside body. Therefore, the line 1-3 in Fig.2.4.3 b covers no area.

Eqn (2.10) now becomes 0 $C_v (T_3 - T_1) + W$. ....(2.22)

$$W = -C_v(T_3 - T_1) = -C_v \frac{(P_3V_3)}{R} - \frac{(P_1V_1)}{R}$$

$$\Rightarrow \quad W = \frac{-1C_v}{R}(P_3V_3 - P_1V_1), R = C_p - C_V \text{ from eqn (2.14)}$$

$$\Rightarrow W = \frac{-C_V}{C_p - C_V}(P_3V_3 - P_1V_1) = P_1V_1 - P_3V_3.$$

$$\Rightarrow \quad W = \frac{P_3V_3 - P_1V_1}{1-k} = \frac{P_1V_1 - P_3V_3}{k-1}$$

]

For mass (m) unit of a gas

$$W = \frac{MR(T_1 - T_3}{K-1} \text{ ....(2.24)}$$

2.4.5 General Polytropic Change

Considering polytropic change (PV^n = a constant) from the condition of P_1 and V_1 to P_2V_2

$$w = \int pdv$$

By mathematical induction,

$$w = \frac{P_2V_2{}^nV_2{}^{1-n} - P_1V_1{}^nV_1{}^{1-n}}{1-n} = \frac{P_2V_2 - P_1V_1}{1-n} \text{.......... ....(2.25)}$$

$$w = \frac{P_2V_2 - P_1V_1}{1-k} \text{.......... ....(2.26)}$$

Liljedahl et al, 1989

Equations, 2.25 and 2.26 assured that work is done by the gas. If the solution gives a positive value, the gas has done work by expansion. If the result is negative, work has been done on the gas by compression.

2.4.6. Internal Combustion Engine Cycle

Internal combustion engine cycle (I.C.E) would be discussed under two specific cycles i.e.;

(i) The ideal –Air Otto cycle --spark ignition engine and (ii) The ideal air-standard diesel cycle. As discussed or mentioned earlier, the four-stroke engine cycle is applicable to both petrol and diesel cycle. Fig 2.4.6 (a) illustrates the ideal indicator diagram for a four-stroke cycle engine. MA represents the introduction of the charge; AB is the adiabatic compression; BC is instantaneous combustion of the fuel and heating of the change at constant volume; CD is adiabatic expansion of the hot gas. DA is an instantaneous drop of pressure following release and AM is the ejection of the remainder of the charge.

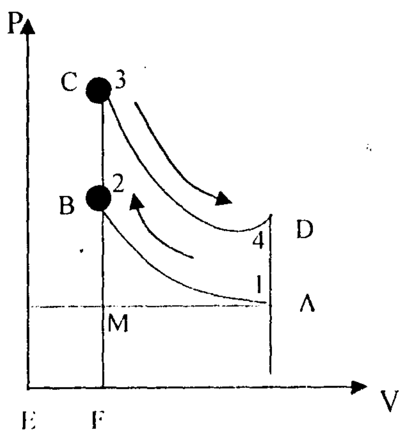


Fig. 2.4.6 (a): Ideal indicator diagram for a four -stroke cycle engine

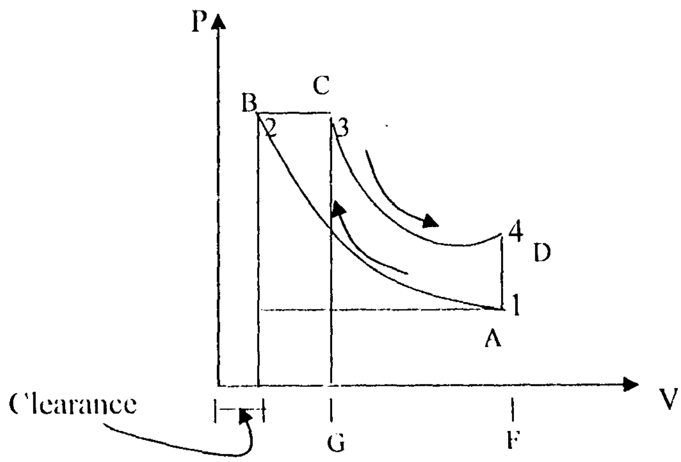


Fig. 2.4.6 (b): Ideal diesel cycle

2.4.6.1 The Ideal Air- Standard Otto Cycle

A lot of assumptions were made in considering an ideal cycle. The efficiency of the cycle may be calculated as follows. Heat added at constant volume during the cycle is;

$Q_{in} = MC_v (T_3 - T_2)$ ....(2.27)

$Q_{out} = Mc_v (T_4 - T_1)$ ....The heat rejected from D to A

The cycle efficiency is $e = \dfrac{Q_{in} - Q_{out}}{Q_{in}} = \dfrac{Mc_v (T_3 - T_2) - Mc_v (T_4 - T_1)}{Mc_v (T_3 - T_2)}$

$$\rightarrow e = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)}$$

since $\dfrac{T_2}{T_1} = \left(\dfrac{V_1}{V_2}\right)^{K-1}$ $and\ \dfrac{T_3}{T_4} = \left(\dfrac{V_4}{V_3}\right)^{K-1}$ & $\dfrac{V_1}{V_2} = \dfrac{V_4}{V_3}$

Therefore; $\dfrac{T_4}{T_1} = \dfrac{T_3}{T_2}$

Eqn 2.28 becomes e =

$$e = 1 - \left(\frac{1}{V_1/V_2}\right)^{K-1} \qquad \text{........................(2.29)}$$

$$\Rightarrow e = 1 - \frac{1}{r^{K-1}}$$

where r = compression ratio also

$$\Rightarrow e = 1 - \left(\frac{P_1}{P_2}\right)^{(K-1)/K} \qquad \text{..(2.30)}$$

Hence, it is apparent that the efficiency of an ideal Otto cycle increases as the compression ratio increases and is independent on the heat liberated by the fuel. It is the work done during each cycle that enables the engine to do work and develop power. For ideal Otto cycle illustrated in fig 2.4.6 a, the work of the cycle is equal to the algebraic sum of the work done under the compression and expansion curves. Where compression work

FABE is negative and expansion work FDCE is positive. From equation 2.27 the Cycle work may be calculated as;

$$w = \frac{P_1V_1 - P_2V_2}{k-1} + \frac{P_3V_3 - P_4V_4}{k-1} \quad \text{..........} \quad \text{..........} \quad \text{..........} \quad \text{......}(2.31)$$

Or from conservation of energy;

$$W = Q_{in} - Q_{out}$$

2.4.6.2 The Ideal Air- Standard Diesel Cycle

The heat quantities and efficiency of diesel cycle can be worked out in a manner similar to that used for Otto cycle. It will be noted that the efficiency of the diesel cycle is influenced not only by the compression ratio, but also by the length of time during which the heat is being added to consider compressed air.

With reference to fig 2.4.6.b, the following expressions will be obtained. No heat is added in adiabatic compression from 1 to 2. Heat is added from 2 to 3

$$Q_{in} = Mc_p (T_3 - T_2)$$

No heat is rejected in adiabatic expansion from 3 to 4. Heat is rejected from 4 to 1

$$Q_{out} = Mc_v (T_4 - T_1),$$

The efficiency of the cycle is;

$$e = \frac{Qin - Qout}{Qin} = 1 - \frac{Qout}{Qin}$$

$$\Rightarrow e = 1 - \frac{Mcv\,(T_4 - T_1)}{Mcp\,(T_3 - T_2)} = 1 - \frac{(T_4 - T_1)}{k(T_3 - T_2)} ------2.32 =$$

$$e = 1 - \frac{T_1/T_2\left[\frac{T_4}{T_1} - 1\right]}{k\left[\frac{T_3}{T_2} - 1\right]} \quad ....2.32$$

From T,V relation, we have

$$T_3 = \frac{T_2V_3}{V_2} \text{ at constant pressure}$$

Also, $T_4 = T_3\left(V_3/V_4\right)^{K-1}$ adiabatically

Thus,

$$T_4 = \frac{T_2V_3}{V_2}\left(\frac{V_3}{V_4}\right)^{K-1} = T_2\left(\frac{V_3/V_2}{V_4/V_2}\right)^{K} K-1$$

Since the compression from 1 to 2 is adiabatic $V_4 = V_1$, we have

$$T_2 = T_1\left(V_4/V_2\right)^{K-1}$$

Solving for T_1 and then adiabatic T_4 by T_1 we obtain

$$\frac{T_4}{T_1} = \frac{\left(V_4/V_2\right)^{KY}\left(V_3/V_2\right)^{K}}{V_4/V_2} = \left(\frac{V_3}{V_2}\right)^{K}$$

Substituting in equation 28, we have

$$e = 1 - \left\{ \frac{\left(\frac{V_2}{V_1}\right)^{K-1}\left(\frac{V_3}{V_2}\right)^{K-1}}{K\left(\frac{V_3}{V_2}\right)^{-1}} \right\} = 1 - \frac{1}{\left(\frac{V_1}{V_2}\right)^{K-1}} \left\{ \frac{\left(\frac{V_3}{V_2}\right)^{K} - 1}{K\left(\frac{V_3}{V_2} - 1\right)} \right\}$$

$$= 1 - \frac{1}{r^{k-1}} \left[\frac{\left(\frac{V_3}{V_2}\right)^{K} - 1}{k\left(\frac{V_3}{V_2}\right) - 1} \right] \qquad \text{.......................................(2.33)} \ \textbf{(Liljedahl, 1989)}$$

From the equations it may be seen that efficiency increases as the compression ratio V_1/V_2 increases and also as the fuel cut off ratio V_3/V_2 is diminished. In order words, maximum efficiency will be obtained with high compression and early fuel cut off. As the fuel cut off approaches zero, the efficiency of the diesel cycle approaches that of the Otto cycle for the same compression ratio and same Q_{in}, the diesel cycle is less efficient than the Otto cycle.

The external work per cycle in the case of the ideal diesel cycle consists also of the algebraic sum of the work under the compression and expansion curves from fig. 2.4.6b there is a negative work ABEF under adiabatic compression curve and positive compression work under constant pressure line BC and adiabatic curve CD.

$$w = \frac{P_1V_1 - P_2V_2}{K-1} + \frac{P_3V_3 - P_4V_4}{K-1} + P_{2,3}(V_3 - V_2) \text{..............(2.34)}$$

2.4.7 Combustion Processes Using Thermodynamics Principles

In flow processes such as in the combustion chamber of turbine engines in industrial furnaces and rocket engines, combustion is assumed to occur at constant pressure, this is also the case for the ideal air standard diesel internal combustion engines which is treated as a control mass system. In calorimetric equipment used for measurement of the energy released during reaction, and during very fast combustion reaction such as at top dead centre of I.C.E or during the combustion period of an explosion (followed by the expansion of the combustion gases, combustion is assumed to occur at constant volumes. Combustion at constant pressure and constant volume would be the important cases of discussion.

The two parameters of interest to engineers are the energy released and the temperature reached in the combustion process **(Howell and Buckius, 1992).**

2.4.7.1 Combustion At Constant Pressure

By examining the case of an isothermal constant flow reactor to which are fed the fuel at unit molar fuel rate and air to a rate that gives the molar ratio to the fuel specified by

$$C_l H_m O_n + A_1 O_2 + 3.76 A_1 N_2 \rightarrow l CO_2 + {}^{m}\!/_{2}\, H_2O + \frac{n + 2A_1 - 2l - {}^{m}\!/_{2}}{2} O_2 + 3.76 A_1 N_2$$

There is a heat transfer from the reactor in order for the control volume to remain isothermal. The first law to this control volume is

$$Q = \sum_p m_p h_p - \sum_r m_r h_r \text{ ....(2.35)}$$

Where the subscript r denotes reactants and the subscript p denote the products of the reaction. The reaction is assumed to take place at standard reference state of atmospheric pressure and temperature maintained at 25°C (77°F). On molar basis, equation (2.35) becomes.

$$Q^0 = \sum_p \dot{n}_p h_p - \sum_r \dot{n}_r h_r \text{ ....(2.36)}$$

Where, as before, the over bar on a quantity indicates that it is on a molar basis. Now divide through equation 2.36 by the molar flow rate of the fuel, n_f and the equation becomes;

$$\frac{Q^0}{n_f} = q^0 = \frac{1}{n_f}\left(\sum_p n_p h_p - \sum_r n_r h_r\right) \text{ ....(2.37)}$$

Equation (2.37) gives the amount of heat transferred required per mole of fuel from a reactor operating at standard reference conditions, which is equal to the net enthalpy differences between the products leaving the reactor (engine) and the enthalpy of the reactants entering $q^{\circ}$is the enthalpy of combustion or the heat of combustion and it is also denoted by h_{rp}.q° is fixed for a given fuel and it is a measure of the chemical energy content per mole of the fuel when burned systematically with oxygen (O_2) or (air) at constant pressure.

2.4.7.2 Determination Of Enthalpy Of Methane (CNG) Method 1

For methane (CH_4) the combustion equation in equation (2.1) becomes;

$$CH_4 + (1+1)\,O_2 + 3.76(1+1)\,N_2 \longrightarrow CO_2+2H_2O+3.76(1+1)N_2$$

Now, we determine the values of h° for each species in the reaction for use in equation 2. 37.

Table 2.3: Heat of combustion & Molecular Mass of Reactant & product of Methane Combustion.

Species	**Molar Mass**	**h°(Btu 1lbmol)**	**h° nij/kmol**
CH_4	16.042	-32.2x103	-74.9
O_2	32.00	0	0
N_2	28.016	0	0
CO_2	44.01	-169.3x103	-393.8
H_2O; Liquid Vapour	18.016	-104x103	-242

Source: (Howell and Buckius, 1992)

Substituting into equation 2.37 and assuming that the water is produced in the form of vapour we find;

$$\bar{q}^{0} = \frac{1}{nCH_4}\left[\left(n\bar{h}^{0}\right)CO_2 + \left(n\bar{h}^{0}\right)H_2O + \left(n\bar{h}^{0}\right)N_2 - \left(n\bar{h}^{0}\right)CH_4 - \left(n\bar{h}^{0}\right)O_2 - \left(n\bar{h}^{0}\right)N_2\right]$$

$$= \frac{1}{1}\left[-169.3\times10^{3} + 2\left(-104.\times10^{3}\right) + 0 - (1)32.2mo^{3} - 0 - 0\right]$$

= - 345,100Btu/1bmol= - 802,700kJ/kmol

$\therefore q^{-0}$ = -802,700kJ/mol

It should be noted that if we had assured that the water produced in thecombustion reaction was in the form of liquid rather than vapour, then the calculated enthalpy of combustion would have been larger (-383,100Btu/1bmol or -23,800 Btu/1bmol of the fuel).

This occurs because more of combustion energy is required to vapourize the water and less is available to remove from the combustion reactor in the form of heat transfer. The enthalpy of combustion when liquid water is produced is often called the "higher Heating Value (HHV) of the fuel (usually the minus signs is dropped when the name is used). And the enthalpy of combustion when the water vapour is produced is called thelower heating value (LHV). Hence the LHV is the only choice we have since combustion reactions in practice produce usually water vapour as a combustion product.

2.4.7.3 Combustion At Constant Volume

For constant volume combustion of a stoichemetric mixture of fuel and oxidant the first law for control mass gives

$$Q = \sum_p M_p U_p - \sum_r M_r U_r \;\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2.38)$$

Subscripts p and r denote product and reactants of the reaction respectively. Assuming the combustion is carried out at the reference state (1 atmosphere and 25°C) such that Q is the heat transfer necessary to maintain isothermal combustion and the products and reactants are both at the reference state, then equation 2.38 becomes.

$$\left(\frac{Q^0}{n_f}\right)_v = \left(q^0\right)_v = \frac{1}{n_f}\left(\sum_p n_p U_p^0 - \sum_r n_r U_r^0\right) \text{------------------------(2.39)}$$

We have divided through by nf (no of moles of the fuel). This equation (2.39) predicts the heat transfer that occurs per mole of fuel from a constant temperature, constant-volume combustion process at standard reference conditions. But from enthalpy of formation, the standardized reference internal energy is

$$U^{o} = h^{o} - P_{ref} V \; (T = 25^{o}C, \; P = 1 \text{ atm}) \quad \text{.......................(2.40)}$$

And assuming all product and reactants are ideal gases, then equation 2.39 becomes;

$$\left(\bar{q}^{0}\right)_{r} = \bar{q}^{0} - \frac{R\bar{T}^{0}}{n_{f}}\left(\sum_{p} n_{p} - \sum_{r} n_{r}\right) \quad \text{..............................(2.41)}$$

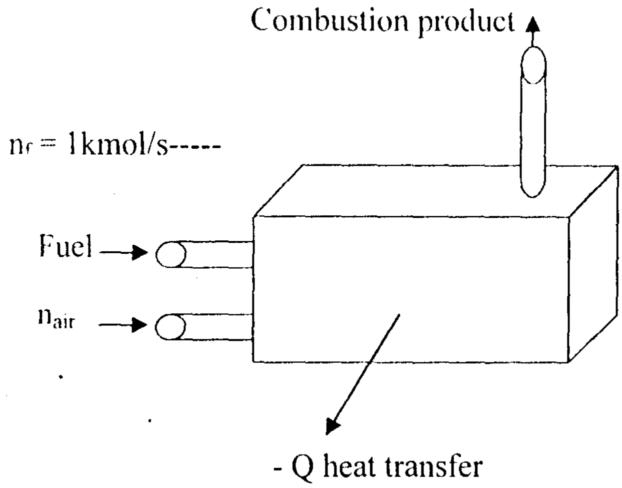


Fig. 2.4.7 (a) Isothermal Constant Flow

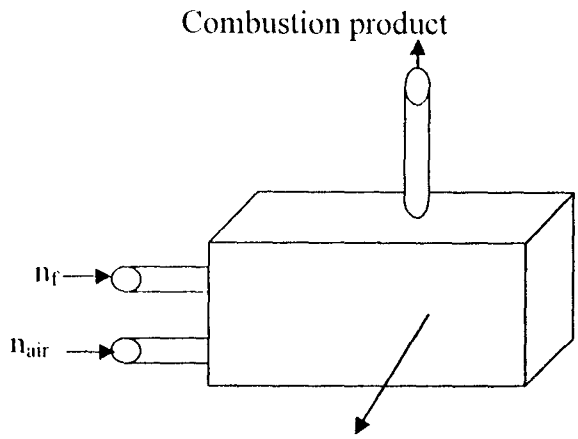


Fig. 2.4.7 (b): Adiabatic reactor
- Q = 0

2.4.7.4 Adiabatic Flame Temperature

Suppose that the combustion reactor in fig 2.4.7a is operated in a different mode from that used to determine the enthalpy of combustion, rather than removing sufficient energy from the reactor (engine) by heat transfer, so that the reactor is maintained at isothermal condition, the reactor is insulated heavily so that no heat transfer occurs and the reactor operates adiabatically, therefore equation 2. 35 now reduces to

$$\sum_{p} \dot{m}_{p} h_{p} = \sum_{r} \dot{m}_{r} h_{r} \quad \text{..(2.42)}$$

Suppose that the reactants enter the reactor at standard reference state involving equation 2.42 by n_s (no of moles of fuel) equation 2.42 becomes;

$$\frac{1}{nf}\sum_p n_p \bar{h}_p^0 = \frac{1}{nf}\sum_r n_r \bar{h}_r^0 \qquad (2.43)$$

If the enthalpies of combustion data are available for fuel being used, then it is convenient to substitute equation (2.39) into equation (2.40) to eliminate the reactants terms

$$\Rightarrow \frac{1}{nf}\sum_p n_p \bar{h}_p = \frac{1}{nf}\sum_p n_p \bar{h}_p^{0} - \bar{q}^{0} \qquad (2.44)$$

The enthalpy carried from the reactor by the products of combustion equals the enthalpy of the products at the standard reference state plus (since q°carries a negative value), the energy released by the combustion reactor.

Solution of equation 2.43 for the temperature of the combustion products (assumed to be the same for all products moving with the reactor) is iterative. A temperature must be assumed and the values of h_p are found from tables or computed. The h_p values are substituted into the product side of the equation (2.43) and the result is checked against the reactant side of equation (2.43). The process is repeated until a temperature is found for which the quality is satisfied.

For the case of no excess air, the temperature of the products is called "Theoretical Adiabatic Flame temperature" for the fuel. The theoretical adiabatic flame temperature is the highest temperature that can be obtained from the fuel used in the combustion reaction.

Fig 2.4.7c shows schematically the lines of enthalpy versus temperature for the reactants and products of typical combustion reaction. The lines are fairly straight because of the weak variation of specific heat with temperature. At a given temperature, the enthalpy of products is always smaller than that of the reactants because our sign convention is taken to mean that the enthalpy of combustion is removed from the reactant to maintain the temperature of the product at the given control temperature.

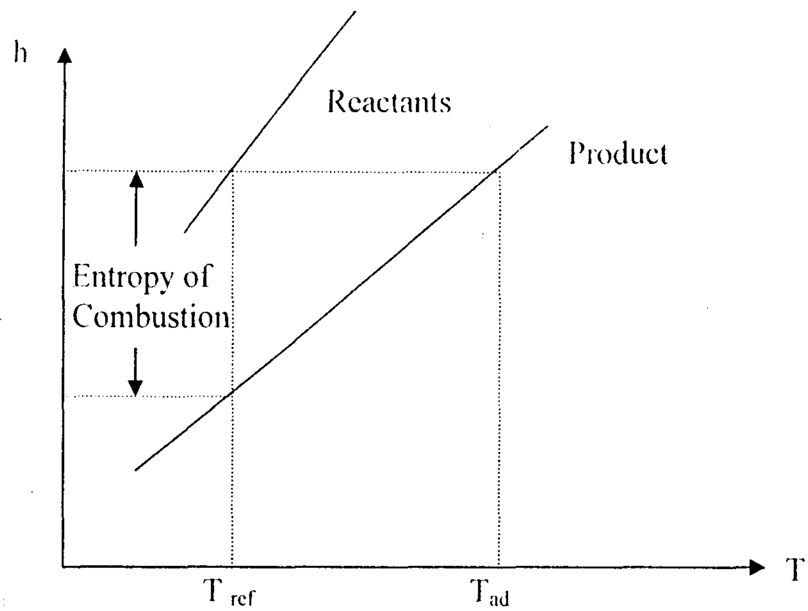


Fig. 2.4.7 (c): Relation of entropy of combustion to adiabatic flame temperature. **(Howell and Buckius, 1992).**

The values of theoretical adiabatic flame temperature of some common fuels are shown in Howell, and Buckius, (1992).

2.4.7.4.1 Determination Of The Theoretical Adiabatic Flame Temperature Of CNG (Methane)

The enthalpy of combustion of methane (for the production of water vapour) is found from 2.4.7.2 to be − 802, 700kj/kmol. The enthalpy of the standard reference states for the products are also given in that section. The chemical reaction is

$$CH_4 + 2O_2 + 7.52N_2 \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

Substituting the known values into equation (44) for 1 kmol of fuel results in

$$\sum_{p} n_p h_p = h^0 CO_2 + 2\overset{0}{h} H_2O - q_0 = -393.8x10^3 + 2(-242x10^3) - (-802,700)$$
$$= -75.100 kj/kmol.$$

(Howell and Buckius, 1992).

The RHS of this equation does not include N_2 because its standard reference state is zero. Then the terms

$$\sum_{r} n_p \bar{h}_r$$

on the LHS is

$$\sum_{p} n_p h_p = hCO_2 + 2hH_2O + 7.52hN_2$$
$$\sum_{r} n_p h_p o$$

The enthalpy of the N_2 becomes important and must be included since the last expression is at the temperature of the products leaving the reactor. Using the polynomial expression for specific heat to determine the adiabatic flame temperature the expression for the LHS by this method are as follow;

For CO_2;

$$\int_{T}^{Tp} C_p n dT = 22.26(T_p - 298) + (5.981\times10^{-2})\left(\frac{T_p^2 - 298^2}{2}\right) - (3.501\times10^{-5})\left(\frac{T_p^3 - 298^3}{3}\right)$$

$$+ 7.40\times10^{-9}\left(\frac{T_p^4 - 298^4}{4}\right)$$

$$For H_2 \Rightarrow 2\int_{P}^{TP} \bar{CP}(T)dT$$

$$2\left[\begin{array}{l} 32.24(TP - 298) + (0 - 1293\times10^{-2})\left(\frac{TP^2 - 298^2}{2}\right) + (1.055\times10^{-5})\left(\frac{Tp^3 - 298^3}{3}\right) \\ -(3.59\times10^{-9})\left(\frac{TP^4 - 298^4}{4}\right) \end{array}\right]$$

$$For N_2 \Rightarrow 7.52\int_{P}^{TP} \bar{CP}(T)dT = 7.52\int_{P}^{Tp} 28.90(TP - 298) - 0.1571\times10^{-2}\left(\frac{TP^2 - 298^2}{4}\right) +$$

$$(0.8081\times10^{-5})\left(\frac{TP^3 - 298^3}{3}\right) - (2.873\times10)\left(\frac{TP^4 + 298^4}{4}\right)$$

(Howell and Buckius, 1992)

We now choose values of T_P, evaluate the RHS of the first law relation by summing the three terms above and compare the RHS. Graphical representation of the iterative is usually in interpolating the results.

TABLE 2.40: Enthalpy Of Combustion Of Methane (CNG) At Different Guessed Temperature. Method 2

Tp (guessed) K,	CO_2	H_2O	N_2	Sum,kj	$\bar{q}$°kj
2300	110,700	174,200	499,200	784,100	802,700
2400	117,300	183,900	523,700	824,900	802,700
2346	113,700	178,700	510,500	802,900	802,700

The graph of sum ;

$$= \sum_{P} np\left[h(TP) - h_{p}^{0}\right]$$

Versus T_p(guessed is shown in fig 2.4.7d

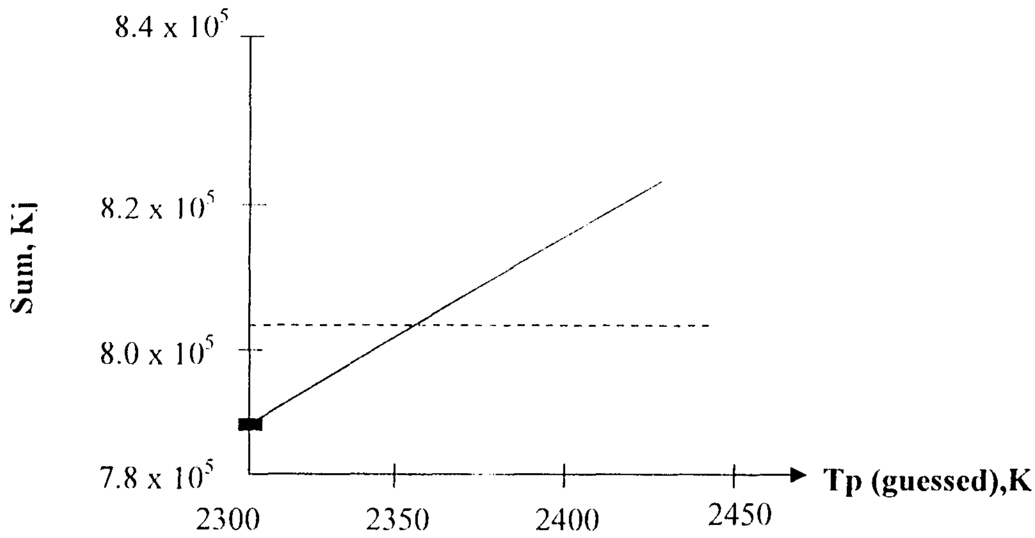


Fig. 2.4.7 (d) Graph of sum Kj Vs Tp (guessed) K.

It should be noted that, the guessed was chosen based on linear interpolation between the guessed at 2300k and 2400k and this yielded the correct result. This is often the case for combustion problems because $h^{-1}$is vary nearly linear with temperature over fairly large temperature ranges and the iterative procedure converges very quickly.

It should be noted also, that the largest portion of the product enthalpy is carried by Nitrogen rather than reaction products CO_2 and H_2O.

The computed value does not agree well with the tabulated values of 228K for the adiabatic flame temperature of methane shown in Howell and Buckius (1992). The discrepancies is partly due to the use of C_p (T) relation outside the temperature range for which they were

$$\sum_p n_p \left(h_p - h_p^0 \right) = -n_f q^0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.45)$$

included (The range is usually 273 to 1800k).

And then to replace the enthalpy changes in terms of the specific heat for constant pressure combustion, or by assuming the enthalpy to be independent of pressure (the ideal gas assumption), so that equation 2.44 becomes

$$\sum_{P} np \int_{P}^{TP} c\,p.pdT = -nf\,q^{0} \quad \text{...(2.46)}$$

Now, the polynomial expression of tables D1 **(Howell and Buckius, 1992)** can be used in the LHS and iteration used until convergence. It should be noted that for fixed T^o the individual integrals for each products species here form

$$\sum_{T^{0}}^{Tp} \bar{C}P(T)dT$$

We could define a temperature average value of $\bar{C}p$ by the relation

$$Cp*(T) = \frac{1}{T-T^{0}} \int_{T^{0}}^{T} C.1(T)dT = \frac{h(TP)-h(T^{0}}{T-T^{0}} \quad \text{............................2.47}$$

Eqn 2.46 reduces to

$$(T_P - T^{o})\sum_{P} np\bar{C}_P*, P(TP) = -nf\,q^{-0} \quad \text{..2.48}$$

A T_p value is assumed, value of $C_p*(T)$ is found that T_p for each species, and convergence is checked by testing the equality in equation 2.48. A graph of $C_p*(T)$ is presented in Fig 2.4.7.e. If the mean specific heat is computed from the polynomial expression of table D1 **(Howell and Buckius, 1992)** and E1 **(Howell and Buckius, 1992)** care must be taken that the temperature limits of equation 2. 47 are within the ranges of applicability of the polynomials. A table of Cp* values for varying ideal gases is given in table C.14 (**Howell and Buckius, 1992**), computed from the enthalpy value. Calculation of the adiabatic flame temperature of methane using the above method. Using the definition of mean specific heat,

$$\left(T_P - T^0\right)\left[\left(n\bar{C}p^*(Tp)\right)CO_2 + \left[n\bar{C}p^*(T_P)\right)H_2O + \left(n\bar{C}p^*(Tp)\right)N_2\right] = -q^{-0}$$

$$\Rightarrow T_P = T^0 - \frac{q^{-0}}{\left[n\bar{C}p^*(T_P)\right]CO_2 + \left(n\bar{C}p^*(T_P)\right)H_2O + \left[n\bar{C}p^*(T_P)\right]N_2}$$

we can write equation 2.48 for these cases as in the equation above.

Now, values of T_p are assumed and of Cp* (T_p) are taken from figure 2.4.7e or table C.14 and substitute into the RHS of the equations above. A new value of T_p is then completed from the equation, and the procedure is repeated until convergence.

T_p (guessed) K	Cp*(T)			
	CO_2	H_2O	N_2	RHS
2300	6.56R	5.31R	4.05R	2325

The first guess in this case produced a new value of T_p that is within the accuracy of the graph.

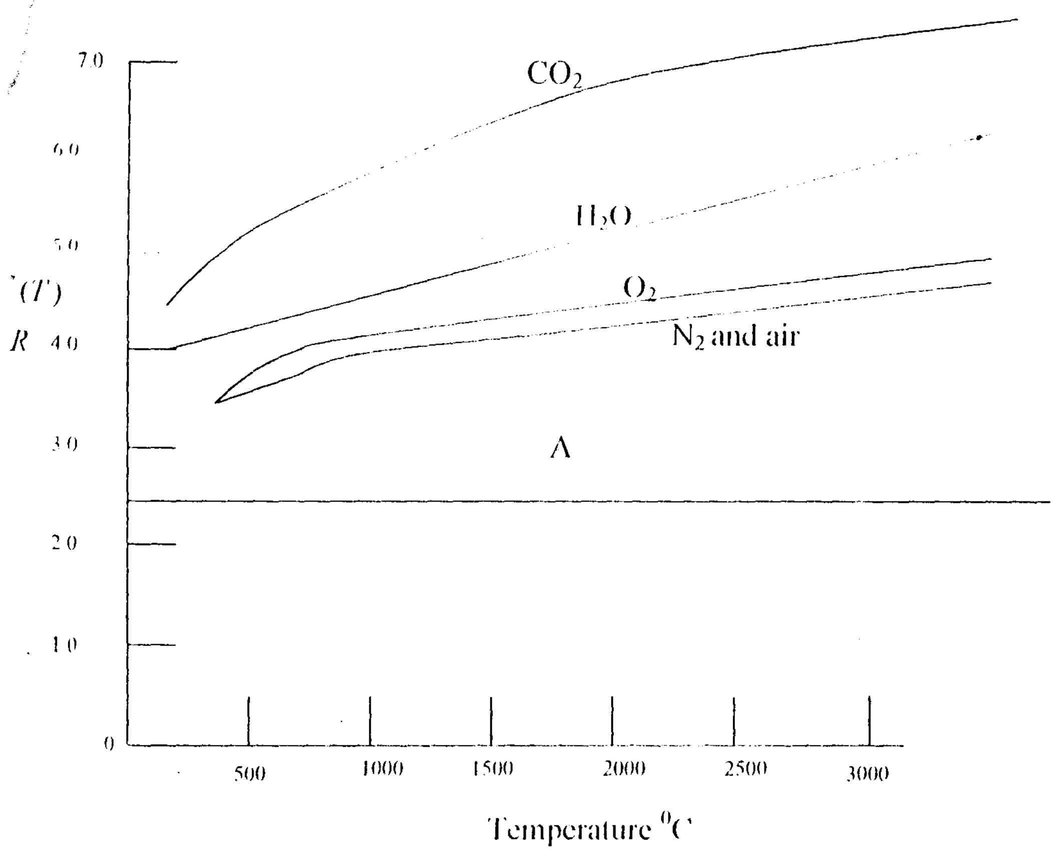


Fig 2.4.7.c: Mean Specific Heat of gases

Method 3:

Tabulated value of $\bar{h}(T_p) - h^{-0}$ form ideal gas tables are used directly in the equation.

$$(h - h^{-0})CO_2 + 2(h - h^{0})H_2O + 7.52\,(h - h^{0})N_2 = 802{,}700$$

Again, values of T_p are assumed and h-h^0 values are found directly from the tables. The process is continued until the equation is satisfied.

Table 2.5: Enthalpy Changes In terms Of Specific Heat During Combustion Of Methane

T_p (guessed) K	$h-h^0 = M(h-h^0)$ kJ/kmol			
	CO_2	H_2O	N_2	LHS, kj/kmol
2300	109,100	88,420	67,250	791,700
2350	112,300	91,070	68,860	812,300
2329	110,800	89,960	68,310	804,400

Linear interpolation between the first and third table entries gives T = 2325K. The three methods have produced theoretical flame temperature prediction for methane of 2346K, 2525K and 2325K respectively, while from table C14 (**Howell and Buckius, 1992**) gives 2285K.

2.4.7.5 Explosion Temperature (Constant Volume) For CNG (Methane)

When the products of combustion are contained at constant volume, the energy released by combustion causes an increase in temperature and usually presumes the final temperature reached in the constant volume system is referred to as the explosion temperature.

Determination of the stoichiometric explosion temperature and final pressure for methane combustion in air assume standard initial conditions. The energy equation for the explosion is by analogy with equation (2.39)

For the stoichiometric combustion of methane,

$$\sum_p np\left(\bar{u}_p - \bar{U}_p^0\right) = -nf\left(q^0\right)v; \text{ from } \quad \ldots\ldots\ldots \quad \ldots eq\ (2.41)$$

$$\left(q^0\right) = q_0 - \frac{RT^0}{nf}\left(\sum_p np - \sum_r nr\right)$$

$$CH_4 + 2O_2 + 7.52N_2 \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

Thus, for one mole of fuel equation 2.41 gives

$(q^0)_v$ = -802,700kJ/kmol – [8.3144kj/kmolk]298k(10.52 – 10.52)

= - 802,700kJ/kmol.

For this particular chemical reaction, because the number of moles of products equals number of moles of reactant, the internal energy from gas tables and paralleling the solution of method 3 of section 2.4.7.4, we guess temperature, compute the change in internal energy of the products and compare with internal energy of combustion, the equation being used is:

$$\left(\bar{U}-\bar{U}^0\right)CO_2+2\left(\bar{U}-\bar{U}^0\right)H_2O+7.52\left(\bar{U}-\bar{U}^0\right)N_2=802{,}700kJ/mol$$

Constructing a table of values gives

Table 2.6: Comparison of Change in Internal Energy of Product with Internal Energy of Combustion.

T_p(guessed)	$(\bar{U}-\bar{U}^0)$ = M $(\bar{U}-\bar{U}^0)$,kJ/kmol			
°C	CO_2	H_2O	N_2	LHS,kJ/kmol
2600	123,620	195,960	500,850	820,430
2500	118,270	186,750	479,650	784,670

Interpolating between the two entries guess T_p = 2550°C = 2823K. This temperature should be compared with the adiabatic flame temperature for the same case of 2325k. The higher explosion temperature results because the combustion products did not work against the atmospheric in the constant volume process. The final volume = initial volume V_1. For initial condition of one mole of methane (CH_4) and noting that in this case, the total numbers of moles of reactant equals the number of moles of products. The ideal gas law result is,

$$P_2=P_1\left(T_2/T_1\right)=1atm.\left(\frac{2823}{298}\right)=9.51atmos$$

Therefore, the pressure is 9.51 atmosphere.

2.5.0 Engine Operation

Engine can be classified into different types according to different criteria. Most importantly, engines are classified into internal combustion engine (I.C.E) and external combustion engine (steam engine). Modern technologies completely phasing out "external combustion engines" and most engines are now I.C.E. and these engines could be a four-stroke or two- stroke cycle. Two-stroke engine delivers one power stroke every two stroke instead of one power every four stroke in four stroke engine, this, it developed more power with the same displacement or can be lighter yet deliver the same power. For this reason, it is used in lawn mower, chain saws, small automobile, motorcycle, and out board marine engine. However, there are several advantages that restrict its use, for example, since there are twice as many power stroke during the operation of a 2- stroke engine, the engine tends to heat up more and this is likely to have a shorter life. Hence, most heavy-duty engines and tractors are four stroke engines.

Natural Gas (CNG, LNG) vehicles are also four stroke engines. Four-stroke engine cycle could be spark ignition or compression ignition, initially, spark ignition refers to petrol/gasoline engine while compression ignition refers to diesel engines, but with the latest technological developments, Natural gas can be made to suit either of the two.

2.5.1. Four Stroke Cycle

In most engines, a single cycle of operation (intake, compression, power and exhaust) takes place over four stroke of a piston, made in two engine revolutions. When an engine has more than one cylinder, the cycles are evenly staggered for smooth operation, but each cylinder will go through a full cycle in two engine revolutions. When the piston is at the top of the cylinder of the beginning of intake stroke, the intake valves open and descending piston draws in the air- fuel mixture.

At the bottom of the stroke, the intake valve closes and the piston start upward movement on the compression stroke, during which it squeezes the air-fuel mixture into small space of the top of the cylinder. The ratio of the volume of the cylinder when the piston is at the bottom to the volume when the piston is at the top is called compression ratio. The higher the compression ratio, the more powerful the engine and the higher it's efficiency. However, in order to accumulate air pollution, control devices manufacture have to lower compression ratios. Hence, the use of dual-fuel engine can be used to achieve high compression and less pollution.

Just before the piston reaches the top again, the spark plug fires, igniting the air-fuel mixture or (alternatively, the heat of compression ignite the mixture). The mixture in burning becomes hot, expanding gases forcing the piston down on its power stroke. Burning should be smooth and controlled.

Faster, uncontrolled burning sometimes occurs when hot spots in the cylinder preignite the mixture; these explosion are called engine knock and causes loss of power. As the piston reaches the bottom, the exhaust valve opens, allowing the piston to force the combustion products mainly CO_2, CO, Nitrogen oxides and unburned hydrocarbons out of the cylinder during the upward exhaust stroke.

The procedure is same for diesel or compression Ignition Engines, only that cleaned air comes in through the intake valve and it is this air that is compressed and the injector atomises the fuel to hot compressed air and this ignites the air during the power stroke.

2.5.2. Direct Injection Engines

Direct injection Engines are used to carry out research in relation to the combustion in Engines. Injection Engines could be spark-Ignition Direct Ignition (SIDI) or compression Ignition Direct Injection (CIDI) Engines.

2.5.2.1 Compression Ignition Direct Injection Engines

More commonly known as diesel Engines, the compression Ignition direct injection engine has the highest thermal efficiency of any internal combustion engine, therefore, produces the least greenhouse CO_2 from its exhaust, however, a number of disadvantages include a lower specific power than the petrol engines, significant amount of particulate matter (PM) and nitrogen oxides (NO_x) from the exhaust, noise, vibration and smell.

Recent advances in European high-speed automobile diesel Engines address some of the shortcomings and make such engines ideal candidates, for Hybrid Electric Vehicle (HEV) applications. These advances include, high-pressure injection, lead catalysts and sophisticated electronic controls. With a thermal efficiency greater and well-understood maintenance, reliability, manufacturing, operating characteristics, thus the high speed CIDI engines show great promise as near term hybrid propulsion.

A petrol engine mixes fuel and air, compresses the mixture and ignite it with a spark and in an advanced CIDI diesel engine, fuel is ignited directly into the cylinder near the top dead centre, during the compression stroke, where the heat of high compression immediately ignites it. Beside their sophisticated fuel injection equipment, CIDI engines are more costly than common petrol engines. Higher operating pressure and temperatures also increase the cost of the engine structure and additional emission control (in-cylinder or after treatment) add still to the cost.

CIDI engines become an especially attractive primary power source for HEVs when operated with either reformulated fuels (e.g. low sulphur fuel now available in California) that help catalyst converters work better at cleaning up pollutant or natural gas alternative fuels (for example Di methyl Ether or "Fischer Tropsch" synthetic fuels from, NG) that produce significantly lower levels of particulates. Thus, the development of CIDI engines technologies and new fuels is crucial to meeting emission targets.

2.5.2.2 Spark Ignition Direct Injection Engines. (SIDI)

SIDI engines offer significant potential for increasing the fuel economy and power density of gasoline engines. Goals of the projects include; Investigation of how fuel spray interacts with the air charge and piston bowl preparation of appropriate air-fuel vapour mixtures at the spark plug priors to ignition. Identification of the extent and the effect of Liquid fuel deposits on cylinder surfaces, characterization of flame development and identification of in cylinder processes that generate sooty unburned Hydrocarbon and NOx emissions.

2.5.3 Enhancement Of Natural Gas Heavy-Duty Engine Technology

Increasingly stringent standards on engine and vehicle emissions necessitate the development of more sophisticated control approaches for I.C.E. Natural gas has significant potential to reduce vehicle emissions but most dedicated or dual fuel (CNG-gasoline) engines currently in operation have fairly primitive system of controlling air-fuel ratio and spark timings. Natural gas composition varies significantly both geographically and seasonally, with methane content changing by up to 12%. This alone dictates that an interactive control system which can be monitored either by the natural gas composition or exhaust gas composition is essential if operation at too rich or too lean a mixture is to be avoided. Moreover, to avoid such emissions of oxides of nitrogen (NO_X), the air-fuel ratio must be controlled. Precisely at a point near the stoichiometric operation or else to the lean side of NO_X emission peak. Further control can be exercised by the use of exhaust gas re circulation (EGR) and through intake-boost control. Varying quantities of inert species in the natural gas can alter flame speed significantly while the presence of higher hydrocarbon can change the effective octane rating of the fuel.

2.5.4 Dual- Fuel, Bi-Fuel And Mono- Fuel Engines

There is a clear distinction between the dual-fuel, bi-fuel and mono-fuel (dedicated) engine. A bifuel engine is the one that allows the use of only one fuel at a time, dual-fuel engines are designed to run on combinations of alternatives fuels with petrol. Dual fuel systems inject both fuels into the combustion chamber at the same time. Dual-fuel systems are mostly used in heavy duty or diesel engine, while bi-fuel systems are used in passenger's cars or trucks.

Dedicated conversion systems run on only one fuel. Generally, dedicated vehicles have improved emission performance because they are tuned to optimize operation on only one fuel.

Dual-fuel engines are a clean air solution using alternative fuels. Traditionally, alternative fuels such as natural gas, in engines produce less power due to ignition problems.

When air and gas are compressed alone 11.5:1 ratio. Spark ignition can no longer constantly ignite the fuel **(:/ LIQUEFIED NATURAL GAS HTM. 2002).** Engines then were unable to reach the level of compression that makes today's diesels so powerful and efficient.

Dual-fuel technology changes that restriction by using a small amount of diesel fuel as an igniter, Dual-fuel equipment diesel engines can use the same compression common on diesel engines about 16:1

Presently, all dual-fuel engines will run on either CNG or LNG. Both fuel have relatively high octane number of about 140. The high octane produces high performance without knock and is very important to the operation of dual-fuel engine.

Propane at approximately 120 octane, does impact engine performance, the rating expected for dual-fuel engines, when available will be approximately 200bhp(150 kW) for the 3126 (tractor) with 400 lb. ft. torque (546.45 Nm), and 350bhp(262.50 kW) for C-12 engine with 1100 lb ft. torque (1502.73 Nm).

Application of dual-fuel Engines: - until recently, alternative fuels such as methanol, ethanol, propane (LPG) methane (CNG) created as many problems as their use could solve. Dual-fuel technology, however, is propelling many trucks owner to replace their diesel engines with caterpillar engines equipped with Dual-fuel technology that provides, the clean emissions of natural gas combined with efficiency and power of diesel.

Vehicles that use natural gas are handicapped by limited reserves of supply. Natural gas is widely used but refueling facilities are not yet common even in large cities. The constraint of limited operating range makes use of dual- fuel engines most practical for return to bare applications.

2.5.5. Components Of Conversion Kits

The components of the conversion kits for operation of CNG as an alternative fuel will be discussed broadly under three specific systems. They are: bi-fuel (CNG- Petrol), Dual-fuel (CNG AND DIESEL) and mono fuel (CNG only).

2.5.5.1 Components of bi-fuel conversion kits)

(i) **CNG Cylinder: -** these are high-pressure cylinders designed for storage of CNG at a pressure of 20 MPa. A typical tank capacity is 60 litres. The number of cylinders required depends on the vehicle.

(ii) **Vapour bag Assembly: -** this is made of PVC and is designed to cover the cylinder valves. It is tubular in shape and has a threaded flange at one end screwed onto the cylinder neck threads and a screwed cap at the other end to give access to the cylinder valve.

(iii) **CNG Pressure Regulator: -** it is a multi-stage pressure reducer in which the gas pressure reduced from that prevailing in the tank to a pressure just below the atmospheric pressure. This ensures that natural gas will not flow out of the pressure regulator when the engine is not running. The filling connection/valve is used in filling high -pressure gas from the CNG compressor to the tank. The electronic selector/change over switch activates the electrical circuits in the system to automatically change the mode of operation from diesel or petrol to CNG. Venturi is a gas and air mixing and metering device. It meters the gas flow proportionately to the engine speed.

2.5.5.2 Component Of Dual-Fuel Conversion Kits (For Diesel- CNG)

The components of a CNG conversion kit for dual-fuel operation in diesel engines are:

(i) Special filler valves for filling CNG storage tank;

(ii) Multi stage pressure regulator to regulate pressure from 20 bar to less than the atmospheric pressure;

(iii) Pneumatically operated safety valve: to close gas supply as the engine rpm. reaches beyond specific limits.

(iv) Linear load valve- connected to the accelerator paddle – control gas flow as per engine load,

(v) Rack limiter allows fuel load diesel flow up to certain engine rpm and reduces to pilot valve beyond specific speed.

(vi) Venturi - the gas mixing and metering device located down stream of the engine air filter.

2.5.5.3 Component Of Mono- Fuel Kit (Cng)

The components of the kit for mono-fuel operation are:

(i) High pressure cylinder- designed for storage of CNG at a pressure of 20 mPa

A typical tank capacity is 50 litres. The number of cylinders required depends on the vehicle.

Special filler valve for filling CNG storage tanks;

(ii) Pressure regulators to reduce the gas pressure from 20 mPa to just above the atmospheric pressure:

Specific air-valve diaphragm carburetor, six cylinders contact less distributor Ignition system with spark plugs located in the place of injector. Electronic governor (special) to reduce gas flow at second stage regulator and the specified rpm reached **(Indiamart, 2002)**

2.5.6. Operational Performance, Maintenance And Reliability For Natural Gas Vehicles (NGV_S)

Operational performance- vehicle ranges for CNG and LNG depends on fuel storage capacity, but generally, it is less than that of a comparable petrol- fuelled vehicles.

Power, acceleration and cruise speed are comparable with those of an equivalent internal Combustion Engines.

Cylinder locations and numbers may displace some payload capacity.

2.5.6.1 Maintenance And Reliability

High-pressure tanks require periodic inspection and certification.

Some fleet reports two or three year longer service life and extended time between required maintenances. However, manufacturers and converters recommend conventional maintenance intervals.

2.5.6.2 Safety Precaution For NGV_S

* Pressurize tanks have been designed to withstand severe impact, high external temperatures and automotive environmental exposure: they are as safe as petrol tanks. Design changes have resolved the problems responsible for earlier in-service failures.

* Adequate training is required to operate and maintain vehicles; training and certification of services technicians is required.

2.6 Tractor And Engine Performance

2.6.1 Tractor Performance Criteria

The performance of a farm tractor can be expressed in different ways. The criterion that best describes the performance depends largely upon the intended use of the tractor.

(i) The tractors size: - the number of ploughs it can pull under average condition.

(ii) The maximum drawbar pull is often used in carrying or evaluating tractors. Drawbar pull is seriously affected by the soil or test track conditions and also by the gear ratio and the ballast being carried. Power is a function of velocity and drawbar pull, hence, drawbar pull partly describes the ability to do work. Maximum drawbar power (P_{db}) is normally the most useful criterion for farm tractors.

(iii) The maximum Power –Take - off power (P_{PTO}) developed is a useful criterion for farmers who use a tractor extensively on machine requiring Pto drive.

(iv) Fuel consumption is another criterion that can be used to indicate directly or indirectly the efficiency of the tractor.

(v) Torque curve or lugging ability – it is a way of measuring the stability or pulling ability of an engine as the engine is slowed down because of increased load. For tractors the drawbar pull versus speed, for a single gear and open throttle is the most useful method of interpretation, since this method considers the effects of transmission and traction.

2.6.2 Power Measurement Methods

Definition of terms: -

Power: - The rate of doing work. A unit of power is Nm/s = watt. It is also measured in hp (horsepower) 1 hp= 0.746 kW, 1000 W= 1 kW.

Brake power: - the power output of the engine crankshaft. The engine may be stripped of past or all its accessories. It is otherwise called effective power and it is 10-12% less than the indicated power.

Pto Power: - the power delivered by a tractor through its Pto shaft.

Drawbar power: - the power of a tractor measured at the end of a drawbar, it is the product of the drawbar pull and the velocity of operation.

Friction power: - the power required to operate/run the engine at any given speed without production of useful work. It is usually measured with a suitable electric dynamometer that runs that engine. It represents the friction and the pumping losses of an engine.

Indicated power: - this is the power developed by the engine as a result of the pressure in the combustion chamber and the volumes produced by the reciprocating components of the engine or it is expressed mathematically as

$$P_i = \frac{P_e LANn}{60 \times C} \text{ -------------------(2.49)}$$

Where P_i = indicated power, P_e= mean effective pressure, p_e (N/m^2)

L= length of stroke, m; A= area of piston bore, m^2

N= engine speed in rpm; n= number of cylinder

C= 1 or 2 for 2 and 4 stroke engines respectively.

Gross indicated power = Net brake power + Friction power --------(2.50)

Maximum brake power: - is the maximum power an engine will develop with the throttle fully open at specific speed. With tractor engines, the maximum power is measured at rated speed.

Observed power: - the power obtained at the dynamometer without any correction for the atmospheric temperature, pressure and or vapour pressure.

Corrected power: - power obtained by correcting observed power to standard conditions of sea-level pressure (1.013×10^5 pa), 15.5^0 C temperature and zero vapour pressure.

Kilowatt-hour: - one kilowatt working for one hour. It is 3.6×10^6 joules of work.

Dynamometer: - an instrument for determining power, usually by independent measure of force, time and the distance through which the force is moved.

Dynamometer may be classified as brake, drawbar, or torsion, according to the manner in which work is being applied. Also, they may be classified as absorption or transmission, depending on the disposition of the energy.

2.6.2.1 Absorption Dynamometer

An absorption dynamometer measures the power applied and at the same time converts it to some other form of energy, usually, heat. Examples of absorption dynamometer are;

(i) Prony brake dynamometer- the most elementary form of absorption dynamometer.

(ii) Hydraulic dynamometer

(iii) The air brake or fan brakes dynamometer and

(iv) Eddy-current dynamometer. An absorption dynamometer which is used to measure power using the relation equation 2.51 **(Liljedahl et al, 1989)**

$$\text{Power} = \frac{2\pi lfn}{60{,}000}, \text{ kW} \quad \text{-------------------(2.51)}$$

Where l, in meters, f, in Newton and n, in revolution per minutes (speed, rpm)

Torque is measured by the prony brake and similar dynamometer. From equation 2.51, torque, T= fl and power becomes

$$P = \frac{2\pi nT}{60{,}000}, \text{ kW} \quad \text{--------------------(2.52)}$$

Where T measured in Nm (Newton meter).

2.6.2.2 Shop- Type Dynamometer

This type of dynamometer is used primarily as an indication of the condition of the engine. It is also used in the process of adjusting or turning an engine and in indicating to customers the improvement in a tractor engine as a result of overhaul, maintenance or adjustment. Shop type dynamometer generally employs a pressure gauge to measure the force on the resisting torque arm. The Pto speed is usually measured by a direct reading speed indicator.

2.6.2.3 Drawbar Dynamometer

Drawbar dynamometers are commonly employed to determine the drawbar pull of power units or to ascertain the draft of field implements. Examples of drawbar dynamometers are: - spring dynamometer. Hydraulic drawbar dynamometer, strain gauge dynamometer.

- Spring Dynamometer– the simplest and the most common types of drawbar dynamometers, unit consists of a spring that elongates under tension and shortens under compression. It is suitable for rough measurements of forces, because of rapid variations in loads such as are commonly found in connection with agricultural implements.
- Hydraulic Drawbar Dynamometer uses hydraulic cylinder to transmit power from the drawbar force to the dynamometer car. The pressure is measured by a pressure transmitter, the signal from which goes to the recorder and a computer. The hydraulic cylinder for measuring drawbar pull has an advantage over a spring dynamometer in that fluctuations can be damped by a throttling valve.
- Strain gauge dynamometer– one method of measuring the drawbar pull is by means of a dynamometer that uses electrical resistance strain gauge to sense the strain.

2.6.2.4 Torsional Dynamometer

Torsional dynamometers are developed as a result of the machinery operated by tractor power take off shafts. A typical example of Torsion dynamometers is the Torque meter (strain gauge type).

2.6.2.5 Chassis Dynamometer

The testing of tractors outdoors has some limitations due to weather. One method of avoiding some difficulties of outdoor testing is the use of chassis dynamometer. The tractor is restrained from forward movement, and the drive wheels are placed on a drum that is part of an absorption dynamometer. Temperature can be better regulated when testing a tractor on a chassis dynamometer.

2.6.3 Power Estimation: Field Method

It is often desired to know the approximate power being developed by a tractor in the field. If the accuracy of strain-gauge types of torque meter is not needed, an estimate of the tractor power output can be obtained by measuring the manifold pressure. A relationship between the manifold pressure and the power is first obtained by a dynamometer test. The curve is correct only for full throttle or governor setting. Since manifold pressure is not controlled on a diesel tractor, a relationship between manifold pressure and power cannot be obtained. For a given no load engine speed, a curve can be plotted of Pto power versus fuel consumption, a typical example is shown in Fig. 2.6.3a

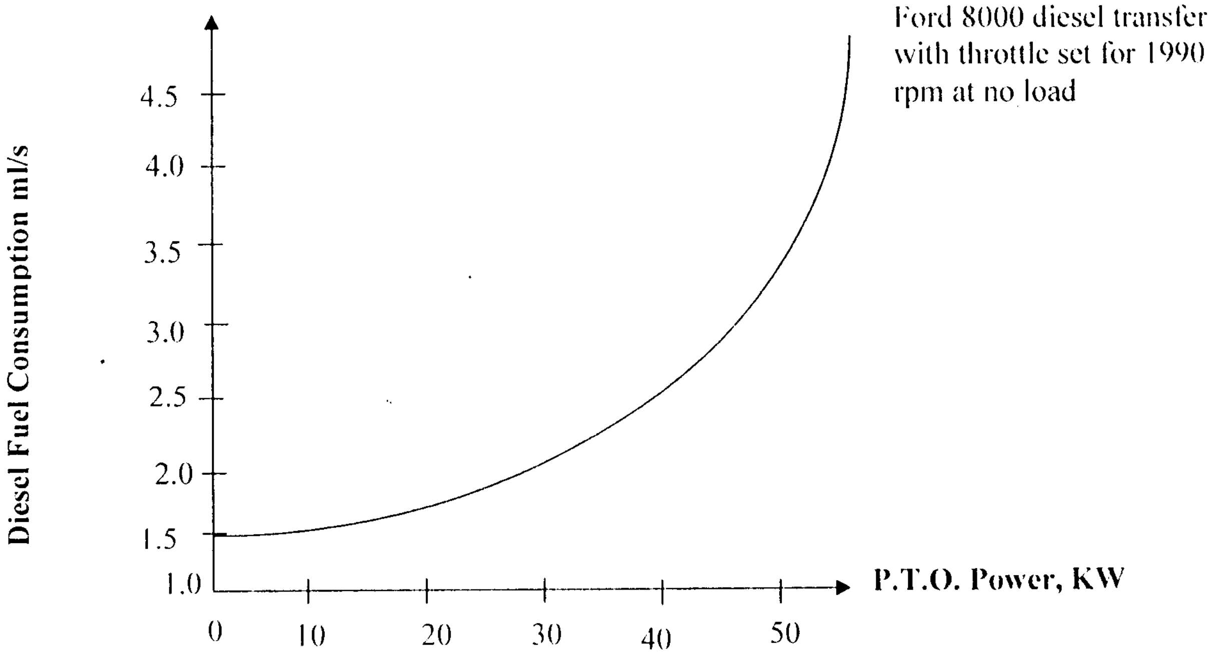


Fig. 2.6.3 (a) Diesel Fuel Consumption Vs P.T.O Power for Ford 8000 Diesel Tractor
(Liljedahl et al, 1989

2.6.4 Torque Curves

One performance criterion is the lugging ability of the engine or more precisely, the torque curve. It can be expressed as the torque in percentage of the maximum power torque versus the engine speed; also in percentages or it can be expressed as the engine torque in Nm versus the engine speed in rpm.

A desirable torque curve is one that increases significantly as speed decreases and is therefore stable. Such a torque curve results in a minimum of speed variation in the engine. A diesel tractor will normally have less speed variation for a given change in torque than a comparable petrol engine.

2.6.5 Engine Performance

Fig. 2.6.5a gives the results of a typical test of a tractor Engine whose crankshaft is attached directly to a dynamometer. The same engine when placed in its tractor chassis would have less power through the power take off because of losses due to gears, hydraulic pumps e.t.c.

The power rating of trucks and automobile engines usually are the results of a dynamometer test of the engine removed from its chassis.

2.6.5.1 Efficiency Of Tractor Engines

An important criterion of the engine performance is its Thermal Efficiency. It can be expressed in percentages, but it is easier to express the efficiency as the ratio of the mass of fuel burned per hour to the Pto power. Or it is a ratio of the amount of heat actually contained in the fuel. This is in the range of 0.26-0.37**(Adgidzi, 2002)**. About 30% of thermal Energy from fuel is converted to effective power, the rest (of thermal energy) is used in overcoming mechanical losses, 10% heating the cooling liquid, and the engine, 45% and thermal losses through exhaust gases, 15%.

Fig. 2.6.5b gives the fuel efficiency of tractors at Nebraska for petrol in 1976 and for diesel in 1984. The graph represent data from the Pto varying power and fuel consumption test, which indicate the tractors ability to convert potential energy (fuel) into useful work, since the efficiency is expressed as kWh/L, a high number means, high efficiency.

From the graph, it is clear that diesel engines have greater efficiency than the petrol engines. Analysis of the data indicates that diesel engines tractors are approximately 54% more efficient than petrol engine tractors. The fuel efficiencies of both fuels decreases as the power level percentage of the engine decreases.

$$\text{Eff (at Pto)} = \frac{\text{Output work (kW.s) x100}}{\text{Input work (kJ/kg.kgfuel)}} \text{ ---------------------------- } (2.53)$$

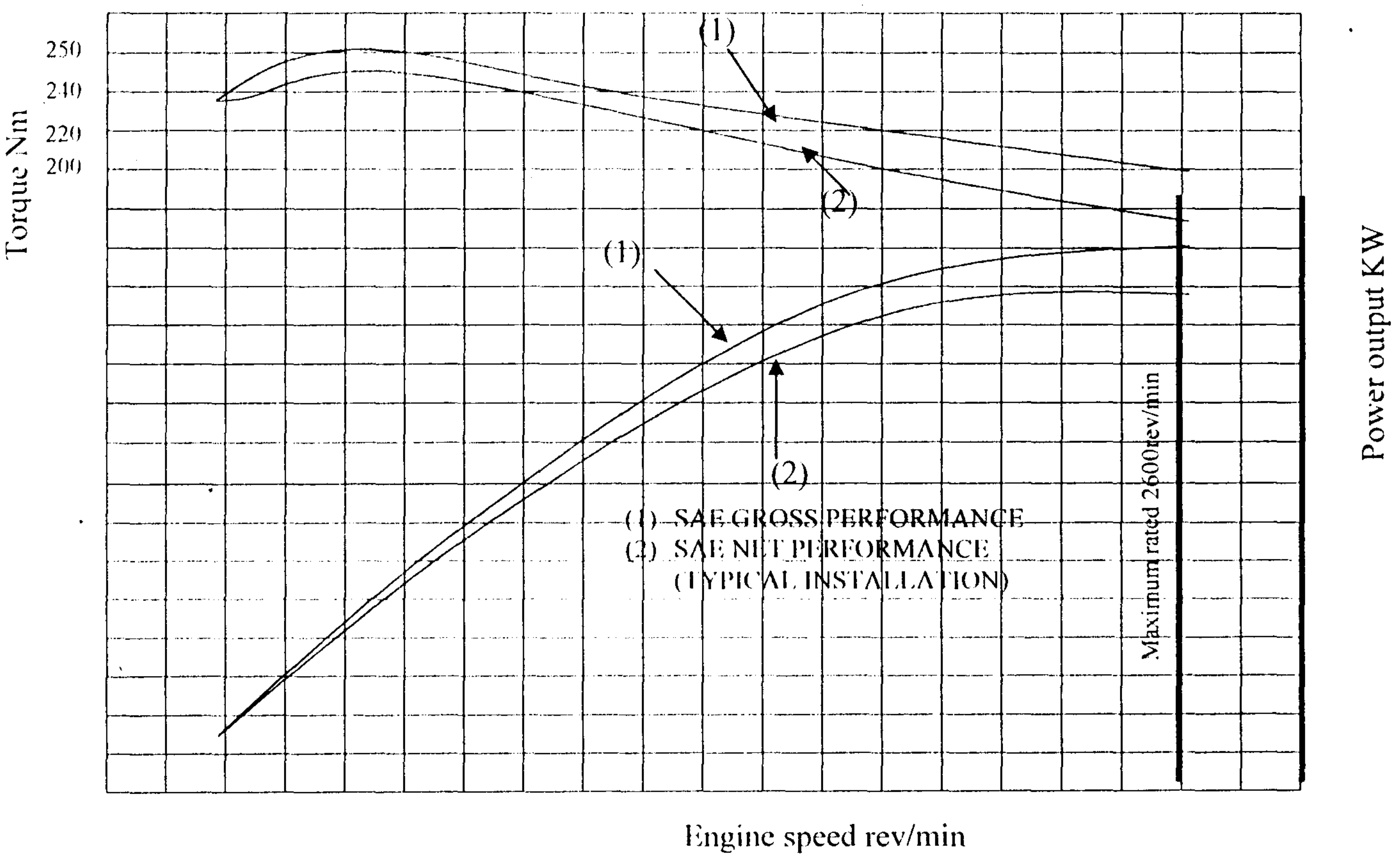


Fig. 2.6.5a: Performance curves for Perkins 4.236 Diesel Engine. (Courtesy, Perkins Engines Limited **(Liljedahi etal, 1989).**

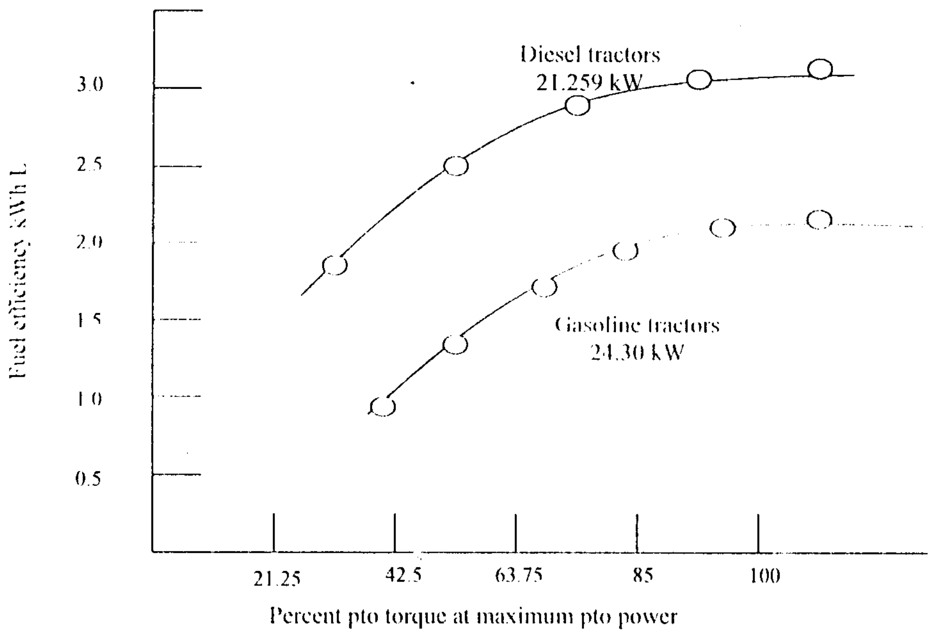


Figure 2.6.5b: Average fuel efficiency of gasoline tractors tested in 1976 and diesel tractors tested in 1984 at Nebraska **(Liljedahi etal, 1989).**

2.6.5.2 Fuel economy of a tractor

This is characterized by the specific fuel consumption determined by dividing the hourly fuel consumption of the engine by its effective power. In diesel engines it is approximately 72 kg/joules (200g/kW.hr).

2.6.5.2.1 Indicated Specific Fuel Consumption

The indicated specific fuel consumption is related as follows;

$$g_i = \frac{10^3 G_T}{N_i} \text{ (g/kWhr)} \ldots\ldots\ldots\ldots\ldots\ldots (2.54)$$

$$G_T = \frac{3600 N_i}{\eta_i H_u} \text{ (kg/hr)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2.55)$$

Where g_i = indicated specific fuel consumption

H_u = Specific heat of burnt gases (kj/kg)

η_i= Indicated efficiency, N_i = indicated power.

From equation 2.54 and 2.55 → $g_i = \frac{3.6 \times 10^6}{H_u \eta_i}$ -------------------(2.56)

g_i = 170 – 200g/kWhr for diesel engines

g_i = 240 – 340g/kWhr for petrol engines. (Adgidzi, 2002).

2.6.5.3 Mechanical Losses In An Engine

Effective Power = N_e, N_i - indicated power, N_m = Mechanical Losses.

$$N_e = N_i - N_m \text{ ....(2.57)}$$

Effective power of an engine is expressed through the mean effective pressure of the engine:

$P_e = P_i - P_m$; P_e = 0.6 – 1.0 mPa for 4 stroke petrol engines

P_e = 0.5-0.9 mPa for 4 stroke diesel engine.

Effective power, $N_e = \frac{P_e V_h n_i c}{30\tau}$.........................(2.58)

Effective engine efficiency

$$\eta_{eff} = \frac{3600\ Ne}{G_T H_u} = \frac{3.6 \times 10^6}{g_e H_u} \text{ ....(2.}$$

Engine efficiency =

$$\eta_e = \eta_t \eta_d \eta_m \text{ ....(2.60)}$$

Where η_t -- non eliminable heat loss

η_d – eliminable heat

η_m = Mechanical loss

Actual determination of specific fuel consumption

$$g_e = \frac{10^3 Gt}{Ni} = \frac{3.6 \times 10^6}{H_u \eta_e} \text{ ------------------------------------(2.61)}$$

g_e = 200 – 250g/kWh for 4 stroke diesel engine

& g_e = 250 – 320g/kW.hr for 4 stroke petrol engine **(Adgidzi, 2002)**

2.7.0 Engine Dynamics

This aims at determining the forces and moments acting on the crank mechanism of the engine and also used to determine moment of inertia and the mass of the flywheel.

2.7.1 Piston Crank Kinematics

Considering figure 2.7.1. the piston displacement is after the crank has turned 0° from top dead center is expressed as

$S = r + l - r\cos 0 - l\cos\phi$ ....2.62 **(Liljedahl et al, 1989)**

Where r and l are the crank radius and the connecting rod length, respectively since;

$$l^2 = bc^{-2} + ca^{-2} = r\sin^2\theta + l^2\cos^2\phi$$

then

$$l\cos\phi = \sqrt{l^2 - r^2\sin^2\theta}$$

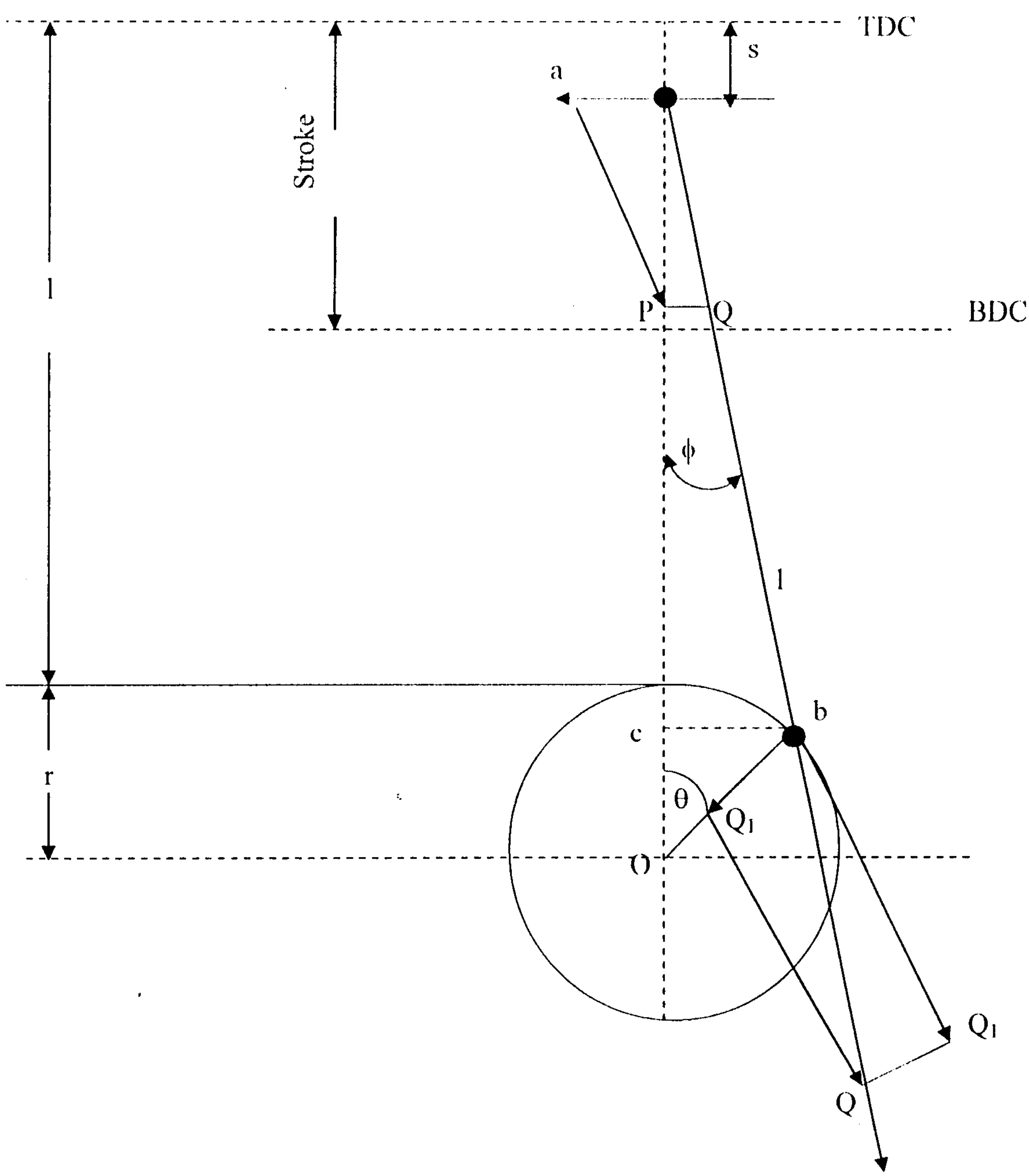
Substituting the later in equation 2.62 and simplifying, we obtain

$$s = r\left[(1 - \cos\theta) + \frac{l}{r}\left(1 - \sqrt{\frac{r^2}{l^2}\sin^2\theta}\right)\right] \text{--------------------------------(2.63)}$$

Adding $r^4\sin^4 0/4l^4$ to the terms under the radical to complete the square results in the approximate relation.

Similarly, $l \sin\phi = r \sin\theta$ ------------------2.64

$$S = r\left[(1-\cos\theta) + \frac{r}{2l}\sin^2\theta\right]$$
$$= r\left[\left(1-\cos\theta + \frac{r}{4l}\right)(1-\cos 2\theta)\right] \text{.........} \quad 2.64$$



'IG 2.7.1: Piston and crank relation. TDC= top dead center
BDC = bottom dead center **(Liljedahi etal, 1989)**

The piston velocity v at a given crank angle can be derived by differentiating equation 2.64 with respect to time t to obtain

$$V = \frac{ds}{dt} = \frac{ds}{d\theta} \bullet \frac{d\theta}{dt} = r\omega\left(\sin\theta + \frac{r}{2l}\sin 2\theta\right) \text{------------------------} (2.66)$$

Where ω is the angular velocity of the crank, the maximum velocity of the piston is attained When:

$$\frac{dv}{d\theta} = r\omega(\cos\theta + \frac{r}{}\cos 2\theta) = 0$$

From which cos θ =

$$\frac{L}{4r}\left(-1 \pm \sqrt{1 + \delta r^2/l^2}\right) \text{...} (2.67)$$

It is noticed that the value of θ at which the piston velocity is maximum or minimum depend upon the connecting rod to crank radius ratio. Usually, the piston attains its maximum velocity at 75°C and 85° from TDC at which the angle between the crank arm and connecting rod is close to perpendicular.

The acceleration of the piston can be obtained by differentiating the expression for piston velocity with respect to time. Thus,

$$a = \frac{dv}{dt} = \frac{dv}{d\theta} - \frac{d\theta}{dt} - r\omega^2\left(\cos\theta + \frac{r}{l}\cos 2\theta\right) \text{..............................} 2.68$$

The maximum and minimum atmosphere values are attained when;

$$\frac{d\phi}{d\theta} = 0$$

$$or$$

$$\sin\theta + \frac{2r}{l}\sin 2\theta = 0$$

From which sin θ = 0 , and cosθ = $-\frac{l}{4r}$

Maximum acceleration is attained at the angles of $\theta = 0$ and $\theta = 2\pi$

$$F = M_a = Mrw^2\left(\cos\theta + {}^{r}\!/_{c}\cos 2\theta\right) \ldots\ldots 2.69$$

(Liljedahl etal, 1989)

The piston and the upper part of the connecting rod are assumed to have reciprocating motion. If the mass of the reciprocating part is denoted by M the inertia force, F acting on the mass is as expressed in equation (2.69).

CHAPTER THREE

3.0 METHODOLOGY

Engine Design General

New engines correspond essentially to current models but contain changes or modifications made possible by improvement in design, fuel, lubricants or materials. The first step of any engine design problem is to select the speed type, number and size of cylinder and the arrangement of the cylinders for the required output

A tractor diesel engine was used as the (existing) engine, whose engine parameters served as a basis for the tractor CNG engine design. Having known the parameters of the diesel engine, the design procedure for internal combustion engine (I.C.E) were followed strictly to determine the basic engine parameters of the designed CNG engine and the theoretically regulated characteristics of both engines. Comparisons were made on the fuel economy or the economic effectiveness between the two engines.

3.1 Basic Engine Parameters Of A Named Tractor Diesel Engine

Prototype tractor: 8075a

Engine type: WD411.45

Effective power (diesel engine): 47kW

Rated engine Speed, n (rpm) – prot.t: 2,400rpm

Compression Ratio, E: 16:2:1

Number (and type) of cylinder: i 4p,

Stroke to Bore ratio (S/D); 110mm/100mm

Maximum torque at engine speed, Nm: 229 at 1400rpm

Mean piston speed, : 8.8m/s

Total piston displacement: $3456cm^3$

Injection sequence; 1 3 4 2

(STEYR DAIMLER PUCH; Akiengeselschell) steyr Wien Graz Austria

Source: Operating Instructions for Steyr 8075/8065 printed in Austria by Vereinsdrucken steyr

3.2 Design Procedure

3.2.1 Design Assumptions

The under listed assumptions were made during the process of the design of the tractor CNG engine.

λ = R/L where λ is a ratio between the crankshaft Radius and the connecting rod length, L.

R/L = (between 0.21------- 0.30) (Heywood, 1996)

For this design, λ is assumed to be 0.25 =0.25L= R

- The rated engine speed for the designed CNG = the rated engine speed of the diesel engine = 2400rpm.
- The stroke to Bore ratio of the two engines are equal = 110/100 i.e. S/D = 110mm/100mm
- Since the CNG operated engine would be a spark ignition engine **(:/ LNG, htm, 2002)** compression ratio of the engine (diesel) would be reduced to enhance the use of the CNG therefore compression ratio for tractor CNG engine for this design = 15 **(Technocab, 2003).**

  CNG contains mainly methane, i.e. about 95% of CNG is methane therefore it is assumed that the composition of CNG is only carbon C and Hydrogen H.
- η_m mechanical efficiency of the existing (diesel) engine = 0.72 **(Adgidzi, 1988)**

- η_m mechanical efficiency of the CNG. Engine = 0.85 **(Heywood, 1996)**

ξ of the Diesel engine is 0.8**(Adgidzi, 1988)**

ξ of the CNG. Engine is 0.89**(Heywood, 1996)**

3.2.2 Determination Of Parameters For Indicated (Curve) For Tractor Engines [Diesel & CNG]

For absolute pressure P and absolute temperature, T. The pressure and temperature for points at: a – completion if intake, c – completion of compression, z – completion of combustion and b - completion of expansion process are required in the construction of the indicated curves for both engines.

3.2.2.1 Pressure, P_a And Temperature, T_a At Completion Of Intake Process (And Commencement Of Compression)

P_a and T_a are determined as follows:

$$P_a = \frac{\eta_v P_o(\varepsilon - 1)T_0 + P_r T_0}{\varepsilon T_0} \quad \text{------3.00} \quad \textbf{(Adgidzi, 1988)}$$

Where η_v = filling co –efficient (of the cylinder with mixture) – same as the volumetric efficiency which related to density or density dependent.

For diesel engine, η_v= 0.80 - 0.90 and for this design η_v for the diesel is taken to be 0.85. Similarly η_v for CNG = 0.50 – 0.78 (Heywood, 1996) and for this design η_v CNG = 0.55 (Technocarb 2003)

ε = Compression ratio, ε_{diesel} = 16:2:1, ε_{CNG}=15:1

P_o and T_o – Atmospheric (surrounding) pressure and temperature respectively

P_o for both engine = 1 atmosphere (standard pressure) = 101.325kPa

i.e. $P_{o\ diesel} = P_{o\ CNG}$ = 101.325kPa

$T_{o\ diesel}$ ····= T_o= 27^0 = 300K (27 + 273)= Average atmospheric temperature in Nigeria air is inducted at 27^0 and 1 atmosphere (**Richard, 1985**)

P_r T_r = pressure and temperature as a results of the effects of residual gases in the cylinder

$P_{r\ diesel}$ = 1.06 mPa (Adgidzi, 1988)

$T_{r\ diesel} = 114^0 = 387K$, $P_{r\ diesel} = P_{r\ CNG} = 1.06$ mPa

$P_{r\ CNG} = 1.06$mPa, $T_{r\ CNG} = 115^0 = 388$ K (Richard, 1985)

Substituting these values into equation 3.00 for both the diesel and the CNG, we obtain the pressure $P_{a\ diesel}$ α $P_{a\ CNG}$ as follows:

$$P_a = \frac{0.85 \times 101.325 \times 300 + 1060 \times 300}{16.2 \times 300} = 146.24kPa$$

$$P_a = \frac{0.78 \times 101.325(14) \times 300 + 1060 \times 300}{15 \times 300} = 122.68kPa$$

The temperature, $$T_a = \frac{\varepsilon P_a T_0^1}{\eta_v(\varepsilon - 1)P_0 + \frac{T_0}{T_r} P_r} \text{------------------------------}3.01$$

Substituting the above values into eqn 3.0, $T_{a\ prot.t}$ and $T_{a\ CNG.}$ are determined where T_0 is temperature of mixture at the point of entry into the combustion chamber. ($T_o' = T_0 = 300K$)

Therefore, $$T_{a\ diesel} = \frac{16.2 \times 146.24 \times 300}{0.85 \times (15.2) \times 101.325 \frac{300}{387} \times 1060}$$

$$= 333.55K$$

$$T_{a\ CNG} = \frac{16.2 \times 122.68 \times 300}{0.55 \times (15.2) \times 101.325 \frac{300}{387} \times 1060}$$

$$= 345.08K$$

2.2.2 Pressure, P_c And Temperature, T_c At Completion Of Compression Process (Commencement Of Combustion)

$P_c = P_a \varepsilon^{n_1}$ mPa -------------------- 3.02

$T_c = T_a \bullet \varepsilon^{n_1 - 1}$, K ------------------3.03

Where n_1 is the polytropic index for compression, assume $n_1 = 1.39$

Therefore, $P_{c\,diesel} = 46.24 \times 16.2^{1.39} = 7.02$mPa

$P_{c\,CNG} = 122.68 \times 15^{1.39} = 5.29$mPa

$T_{c\,diesel} = 323.55 \times 16.2^{0.39} = 988.26$K

$T_{c\,CNG}$ $345.08 \times 15^{0.39} = 992.19$K

Specific volume V_c at the completion of compression

$$V_c = \frac{V_a}{\varepsilon} \text{ ----------------------------------3.04}$$

$$\text{Where } V_a = \frac{8.314 \times 10^{-3} T_a}{\mu_a P_a} \text{ ----------------------------------3.05}$$

$$\therefore \quad V_{a\,diesel} = \frac{8.314 \times 10^{-3} \times 333.55}{28.97 \times 146.24} = 6.55 \times 10^{-4} m^3$$

$$V_{a\,CNG} = \frac{8.314 \times 10^{-3} \times T_{CNG}}{28.97 \times 58.087} = \frac{8.314 \times 10^{-3} \times 345.08}{28.97 \times 122.68} = 8.07 \times 10^{-4} m^3$$

$$V_{c\,diesel} = \frac{V_{a diesel}}{\varepsilon_{diesel}} = \frac{6.55 \times 10^{-4}}{16.2} = 4.04 \times 10^{-5} m^3$$

$$V_{c\,CNG} = \frac{V_{aCNG}}{\varepsilon_{CNG}} = \frac{8.07 \times 10^{-4}}{15} = 5.38 \times 10^{-5} m^3$$

2.2.2.1 Quantity Of Gas (Air) In The Cylinder At The End Of Compression (TDC) i.e. (Diesel) With Composition; C, H, O And CNG With Composition; C, H)

The theoretical quantity of air required to burn one kilogram of fuel i.e. (diesel) with composition; C, H, O and CNG with composition C, H)

$$L_o^1 = \frac{1}{0.23}\left(\frac{8}{3}C + 8H - O\right)kg/kg \text{ ----------------------3.06 (Adgidzi, 1988)}$$

$$L_o = \frac{L_o^1}{29} \text{ ---3.07}$$

Where C, H, O weight per element per kg of diesel fuel i.e. C= 0.857, H= 0.133 and O =0.01 and for CNG C= 0.75 and H= 0.25 (Adgidzi, 1988)

Substituting these values into equations 3.06& 3.07

$$L_{0diesel}^1 = \frac{1}{0.23}\left(\frac{8}{3}(0.557)\right) + 8(0.133) - .01) = 14.52kg/kg$$

$$L_{o\,diesel} = \frac{L_{o\,diesel}^1}{29} = \frac{14.52}{29} = 0.500kmol/kq$$

$$L_{0CNG}^1 = \frac{1}{0.23}\left(\frac{8}{3}\right)(0.75) + 8(0.25) = 17.39kg/kg$$

$$L_{0CNG} = \frac{L_{0CNG}^1}{29} = \frac{17.39}{29} = 0.600kmol/kg$$

Actual quantity of air required for combustion of 1kg of fuel is given as

$L = \alpha L_0$---------------------------------3.08

Where α= excess air co-efficient (Assume α= 1.9)

For $L_{diesel} = \alpha L_{o\,diesel}$ = 1.9 x 0.500kmol/kg = 0.95kmol/kg

$L_{CNG} = \alpha\ L_{o\,CNG}$ = 1.9 x 0.600kml/kg = 1.14kmol/kg

Quantity of fresh mixture, M_1 is determined as follows:

$$M_1 = \alpha l_o + \frac{1}{\mu d}$$ ----------------------------------3.09

Where $\lambda \mu_d$ molar mass of fuel (diesel CNG)

For diesel $1/\mu_d$ is negligible, for methane $1/\mu_d = 0.0625$

$\Rightarrow M_{1\ diesel} = \alpha L_{o\ diesel} = L_{diesel}$ ----------------------------------3.09

$M_{1\ diesel} = 0.95$kmol/kg

$M_{1\ CNG} = 1.14 + 0.0625 = 1.2025$ kmol/ kg

Apart from the in-coming air (mixture) into the cylinder, there are residue gases in the cylinder (left over of previous burnt gases), therefore, quantity of residual gases in the cylinder was determined as:

$M_r = \gamma_r \times \alpha L_o$, kmol/kg----------------------------------3.10

Where γ_r – coefficient of residual gases which is related by equation (3.11) below:

$$\gamma_r = \frac{P_r T_o}{P_o T_r \eta_v (\Sigma - 1)}$$ --3.11

Substituting the already known values of P_r, T_r, P_o, T_o, and ε for each engine into equation 3.11 yields:

$$\gamma_{r\,diesel} = \frac{1060 \times 300}{101.325 \times 387 \times 0.85 \times 15.2} = 0.63$$

$$\gamma_{r\,CNG} = \frac{1060 \times 300}{101.325 \times 388 \times 0.55 \times 14} = 1.05$$

$M_{r\ diesel} = \gamma_{r\ diesel} \times \alpha L_{odiesel}$

$= 0.63 \times 0.950 \cong 0.60$ kmol/kg

$M_{r\ CNG.} = \gamma_{r\ CNG.} \times M_{1\ CNG} (\alpha L_{o\ CNG})$

$= 1.05 \times 1.2025 = 1.26$kmol/kg

Total quantity of mixture in the cylinder at the compression process

$M_t = M_1 + M_r$--3.12

$M_{t\ diesel} = M_{1\ diesel.} + M_{r\ diesel}$

$= 0.95 + 0.63$

$= 1.58 kmol/kg$

$M_{t\ CNG} = M_{1\ CNG} + M_{r\ CNG}$

$= 1.2025 + 1.26 \cong 2.46 kmol/kg$

3.2.2.2.2 Quantity Of Gases In The Cylinder At The Completion Of Combustion

For the combustion of 1kg of fuel with excess air & >1. The components of the burnt gases consist of CO_2, water vapour, excess oxygen and nitrogen, this quantity was denoted by M_2 and calculated as: for diesel

$M_2 = \alpha L_o + H/4 + O/32$ (kmol/kg) ------------------------------------3.13

$\Rightarrow M_{2diesel} = 0.95 + 0.133/4 + 0.01/32$

$= 0.984 kmol/kg$

$\Rightarrow M_{2\ CNG} = \alpha L_{o\ CNG} + 0.25/4 = 1.2025 + 0.25/4$

$= 1.265 kmol/kg$

Taking into consideration, the residual gases (the quantity of gases in the combustion chamber) after was given as

$M_z = M_z + M_r$ (kmol/kg) ------------------------------------3.14

$M_{z\ diesel} = M_{2\ diesel} + M_{r\ diesel}$

$= 0.984 + 0.6$

$= 1.584$ kmol/kg

$M_{z\,CNG} = M_{2\,CNG} + M_{r\,CNG}$

$= 1.265 + 1.26$

$= 2.53$kmol/kg

Coefficient of molecular change in the mixture is given as:

$$\mu = \frac{M_z}{M_t} \text{--3.15}$$

$$\mu_{disel} = \frac{M_{z\,diesel}}{M_{t\,diesel}}$$

$$\mu = \frac{1.584}{1.58} = 1.00$$

$$\mu_{CNG} = \frac{M_{z\,CNG}}{M_{t\,CNG}}$$

$$\mu = \frac{2.53}{2.46} = 1.03$$

3.2.2.3 Pressure, P_z And Temperature, T_z At The Completion Of Combustion

This is at the commencement of expansion process.

Denoting the level of increase in pressure at the end of combustion by λ_p

$P_z = \lambda_p \times P_c$, mPa --3.16

Taking $\lambda_p = 1.90$, the lower the coefficient of gases, the higher the increase in pressure, λ_p

$P_{z\,diesel} = 1.90 \times 7.02$

$= 13.34$ mPa

$P_{z\,CNG} = 1.90 \times 5.29$

$= 10.05$ mPa

The temperature T_z is determined from combustion equation for a four-stroke diesel engine as (Adgidzi, 1988):

$$\left(\mu C'_v + 8.28\lambda_p\right)T_c + \frac{\xi H_u}{\alpha L_0\left(1+\gamma_r\right)} = \mu \bullet \mu_{cp} \bullet T_z \text{ ------------------------------------ } 3.17$$

Where ξ- co - efficient of effective combustion of fuel

$\xi = \delta\varphi$, δ - Co - efficient of heat loss (for α, 1), but $\alpha > 1$ $\delta = 1$

Where $\xi = 0.8$ for diesel ($\xi_{diesel} = 0.8$) and 0.89 for CNG (ξ_{CNG}) (Adgidzi, 1988)

μC_v – mean molar heat capacity of air at constant volume in kJ/kmol K

μC_p – mean molar capacity of burnt gases under constant pressure kJ/kmol.K

H_u heat of combustion ($H_{u\ diesel}$ = 42, 500kJ/kg for diesel)

$H_{u\ CNG}$ 80300kJkmol /16.004 = 50,175kJ/kg

$\mu_{cv} = 20.16 + 1.738T_c \times 10^{-3}$ ---------------------------------3.18

$\mu_{cv\ diesel} = 20.16 + 1.738 \times 10^{-3}\ T_{c\ diesel} = 20.16 + 1.738 \times 988.26 \times 10^{-3}$

$= 21.88$

$\mu_{cvCNG} -= 20.16 + 1.738 \times 10^{-3}\ T_{c\ CNG} = 20.16 + 1.738 \times 10^{-3} \times 922.19$

$= 21.88$

$\mu_{cp} = 8.28 + (20.1 + 0.921/\alpha) + (13.82/\alpha + 15.49) \times 10^{-4}T_z$ -----------------------3.18

Substituting $\alpha = 1.9$ into equation 3.19 gives

$\mu_{cp} = 28.865 + 22.764 \times 10^{-4}$ -----------------------------------3.19

diesel engine

$(\mu C_v + 8.28\ \lambda_p)T_c = (21.88 + 8.28 \times 1.9)\ 988.26$

$= 37,170.44$

$$\frac{\xi H_u}{\alpha L_o(H\gamma_r)} = \frac{0.8 \times 42{,}500}{0.95(1.60)} = 22368.42$$

$\mu \, x \mu_{cp} \, x \, T_z = 1.00\,(28.865 + 22.764 \times 10^{-4})\, T_z\,)T_z$

$= 28.865T_z + 2.2764 \times 10^{-3}\, T_z$

$\therefore 2.2764 \times 15)T_z^2 + 28.865T_z = 37.170.44 + 2236.42$

$\rightarrow 2.2764 \times 10^{-3}\, T_z^2\, 28.865T_z - 59538.86 = 0$

$T_{z1} = 1.805.48$, $T_{z2} = -\ 14{,}486.06K\ T_z \neq T_{z2}$

$\therefore T_{z\ diesel} = 1805.48$

For the CNG engine

$(\ \mu e_v + 8.28\lambda_p)T_c = (21.88 + 8.28 \times 1.9)\ 992.19 = 37318.25$

$$\frac{\xi H_u}{\alpha L_o(H\gamma_r)} = \frac{0.89 \times 50{,}175}{1.14 \times 22.05} = 19{,}108.15$$

$\mu \times \mu_{cp}\, T_z = 1.03T_z\,(28.865 + 22.764 \times 10^{-4} T_z)$

$= 29.73T_z + 2.345 \times 10^{-3}\, T_z^2$

$\therefore\ 2.345 \times 10^{-3}\, T_z^2 + 29.73\, T_z = 37{,}318.25 + 19{,}108.15$

$\rightarrow 2.345 \times 10^{-3} + 29.73T_z - 56.426.40 = 0$

$T_{z1} = 1{,}758.78$ and , $T_{z2} = -\ 14{,}353.94K\ T_z \neq T_{z2}$

$\therefore T_{z\ CNG} = 1{,}758.78K$

2.4 Pressure P_b and temperature, T_b At The Completion Of Expansion (i.e. Commencement of intake process):

a. Pre- expansion index

The pre- expansion index is as stated below: where the temperatures T_z for both fuels are determined.

$$\rho = \frac{\xi \; T_z}{\lambda_p \; T_c} = 1.06 \text{-----------------------------------} 3.20$$

$$T_{z\ diesel} = \frac{1.06 \times \lambda_p \times T_{cCNG}}{\xi} = \frac{1.06 \times 1.9 \times 988.26}{0.80} = 2487.95 \cong 2488K$$

$$T_{z\ CNG} = \frac{1.06 \times 1.9 \times 992.19}{0.89} = 2245.25K$$

The ratio of the compression ratio to the pre- expansion index, δ is used to determine P_b and T_b for both Diesel and CNG.

(b). $\delta = \varepsilon / \rho$---3.21

$$\delta_{diesel} = \frac{16.2}{1.06} = 15.28$$

$$\delta_{diesel.} = \frac{15}{1.06} = 14.15$$

(c) **Pressure, P_b At Completion Of Expansion Process (Commencement Of Induction Process):**

The pressure, P_b is determined as follows:

$$P_b = \frac{P_z}{\delta^{n_2}} = P_z\left(\rho/\varepsilon\right)^{n_2} \text{--------------------------------} 3.22$$

$n_2 = 1.24$

$$V_z = \rho V_c, \quad \frac{V_z}{V_a} = \frac{1}{\delta}$$

$$P_{b\,diesel} = \frac{P_{diesel}}{\delta^{n_2}},$$

$$= \frac{13.34}{15.28^{1.24}} = 0.453mPa$$

$$P_{b\,CNG} = \frac{P_{z\,CNG}}{\delta^{n_2}{}_{CNG}}$$

$$= \frac{10.05}{14.15^{1.24}} = 0.376 \text{ mPa}$$

d Temperature T_b At The Completion Of The Expansion Process (Commencement Of Induction Process):

The temperature, T_b is determined as follows:

$$T_b = \frac{T_z}{\delta^{n_2-1}} \text{ ---3.23}$$

$$T_{b\,diesel} = \frac{2488}{15.28^{0.24}} = 1293.19K$$

$$T_{b\,CNG} = \frac{T_z}{\delta_{des}} = \frac{2245.25}{14.15^{0.24}} = 1,188.60K$$

Volumes at intake, compression combustion and expansion

V_a, V_c, V_z and V_b respectively

From equation 3.05, $V_{a\,diesel} = 6.55 \times 10^{-4}\ m^3$

$V_{c\,diesel} = 4.04 \times 10^{-5}\ m^3$

Also, $V_{a\,CNG} = 8.07 \times 10^{-4}$

$V_{c\,CNG} = 5.38 \times 10^{-5}$

$V_z = \rho V_c$, $V_a = \varepsilon V_c$

Where $\frac{V_z}{V_a} = \frac{\rho}{\varepsilon} = \frac{1}{\delta}$

Volumes at intake, compression, combustion and expansion

V_a, V_c, V_z and V_b respectively.

From polytropic compression equation

$$P_x = P_a\left(\frac{V_a}{V_x}\right)^{n_1} \text{------------------------3.24}$$

$$\text{And } P_x = P_b\left(\frac{V_a}{V_x}\right)^{n_2} \quad \text{(for expansion)-------------------3.25}$$

$$\frac{V_a}{V_x} = \frac{V_b}{V_x}$$

$$\Rightarrow V_a = V_b \text{ ----------------------------------3.26}$$

$\therefore V_{z\,diesel} = P_{diesel} \text{ X } V_{c\,diesel}$

$= 1.06 \text{ x } 404 \text{ x } 10^{-5} m^3$

$= 4.28 \text{ x } 10^{-5} m^3$

$V_{z\,CNG} = \rho_{CNG} \text{ X } V_{c\,CNG}$

$= 1.061 \text{ x } 5.38 \text{ x } 10^{-5} m^3 = 5.70 \text{ x } 10^{-5} m^3$

From $V_{a\,diesel} = V_{b\,diesel}$

$= 6.55 \text{ x } 10^{-4} m^3$

$V_{a\,CNG} = V_{a\,CNG} \qquad = 8.07 \text{ x } 10^{-4} m^3$

$V_{a\,CNG} = 8.07 \text{ x } 10^{-4} m^3$	$V_{a\,diesel} = 6.55 \text{ x } 10^{-4} m^3$
$V_{b\,CNG} = 8.07 \text{x } 10^{-4} m^3$	$V_{b\,diesel} = 6.55 \text{ x } 10^{-4} m^3$
$V_{c\,CNG} = 5.38 \text{ x } 10^{-5} m^3$	$V_{c\,diesel} = 4.04 \text{ x } 10^{-5} m^3$
$V_{z\,CNG} = 5.70 \text{ x } 10^{-5} m^3$	$V_{z\,diesel} = 4.28 \text{ x } 10^{-5} m^3$

Procedure followed in the construction of indicated curves. The various pressure points (P_x) and the volumes V_x were used for both polytropic compression and expansion curves.

The volume axis, V was divided into nineteen – equal halves from the V_c to V_a at indicated interval for both engines and the corresponding values of P_x obtained using equations 3.24 &3.25 for polytropic compression and expansion respectively.

$$P_x = P_a\left(\frac{V_a}{V_x}\right)^{n_1} \text{ --3.24}$$

$$P_x = P_b\left(\frac{V_b}{V_x}\right)^{n_2} \text{ ---3.25}$$

Where $n_1 = 1.39$, $n_2 = 1.24$ and P_a, P_b, V_a, V_b are obtained from table 4.10 for both engines.

3.2.2.5 Determination Of Mean Indicated Pressure From The Indicated Diagrams

The mean indicated pressure was given as

$$P_i^1 = \frac{\mu F}{l} \text{ mPa -- 3.27}$$

Where F= indicated diagram area, mm^2

l = length of indicated diagram, mm^2

μ = Pressure scale (1mm = mPa)

For diesel F = 3.580mm^2 from fig 3.1.0

l = 153mm

μ = 0.05 mPa(1mm, = 0.05 mPa)

$$P^{1}_{i\,diesel} = \frac{3580mm^2}{153} \times 0.05mPa$$

$= 1.170$ mPa

$P^{l}_{i\ CNG}$, F = 3300 mm^2

l = 150 mm

μ = 0.04 mPa (1mm = 0.04 mPa)

$$P^{l}_{i\,CNG} = \frac{3300}{150} \times 0.04mPa$$

= 0.88 mPa

The value of the theoretical pressure by the analytical method was given by the formula:

$$P^{cal}_{i} = \frac{P_c}{\varepsilon - 1}\left[\lambda_p(\rho - 1) + \frac{\lambda_p \rho}{n_2 - 1}\left(1 - \frac{1}{\delta^{n_2 - 1}}\right) - \frac{1}{n_1 - 1}\left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right)\right] \text{-------------}3.28\text{(Adgidzi, 1988)}$$

$$P^{cal}_{i\,diesel} = \frac{7.02}{15.2}\left[1.9(1.06 - 1) + \frac{1.9 \times 1.06}{0.24}\left(1 - \frac{1}{15.28^{0.24}}\right) - \frac{1}{0.39}\left(1 - \frac{1}{16.2^{0.39}}\right)\right]$$

$= (0.114 + 4.03 - 1.70)\ 0.462$

$1.129 \cong 1.13$ mPa

$$P^{cal}_{i\,CNG} = \frac{5.29}{14}\left[1.9(1.06 - 1) + \frac{1.9 \times 1.06}{0.24}\left(1 - \frac{1}{14.15^{0.24}}\right) - \frac{1}{0.39}\left(1 - \frac{1}{15^{0.39}}\right)\right]$$

$= 0.378(0.114 + 3.949 - 1.672)$

= 0.904 mPa

The accuracy of construction of indicated diagram is given by the error co-efficient as

$$\delta_{id} = \frac{P^{cal}_{i} - P^{l}_{i}}{P^{cal}_{i}} \times 100\% \text{ ----------------------------}3.29$$

$$\delta_{id\,diesel} = \frac{1.13 - 1.17}{1.13} \times 100 = \frac{-0.04}{1.13} \times 100\% = -3.54\%$$

$$\delta_{id\,CNG} = \frac{0.904 - 0.88}{0.904} \times 100\% = 2.66\%$$

δ_{id} should not be higher than 3 – 4 % (Adgidzi, 1988)

The actual or real mean indicated pressure is given as:

$P_i = P_i\ \gamma$--------------------------------3.30

Where γ is co-efficient of incomplete indicated diagram ($\gamma = 0.93$)

$\Rightarrow Pi_{\ diesel} = 1.170\ x0.93$

$= 1.088$ mPa

$P_{i\ CNG} = 0.88 \times 0.93$

$= 0.818$ mPa

3.2.3 Determination Of Engine Parameters And Fuel Economy Indices

3.2.3.1 Mean Effective Pressure

The mean effective pressure is given as:

$P_e = p_i \cdot \eta\ (mp_a)$..................3.31 (Adgidzi, 1988)

Where η_m is mechanical efficiency, this is the level by which indicated parameters of p_i can be reduced through losses $\eta_m = 0.7 - 0.82$ for stroke diesel engine, $\eta = 0.85 - 0.90$ for four stroke petrol engine.

Assume $\eta_{m\ CNG} = 0.85$

P_e diesel $= \eta_m \cdot p_i$ diesel

$= 0.72 \times 1.088$

$= 0.783$ mPa

$P_{e\,CNG} = \eta_m CNG\; p_{i\,CNG}$

0.85 x 0.818

0.695 mPa

3.2.3.2 Engine (Displacement) Capacity

The Engine displacement capacity $V_l = V_{hi}$

Where i = no of cylinders = 4 and is related as follows:

$$V_l = V_{hi} = \frac{60\tau N_e}{P_e n} \text{ --3.32}$$

Where $\tau = 2$ for four- stroke engine

The rated engine power, the $Ne^r_{diesel} = 47$ kW

∴ The engine displacement capacity, V_l is determined to be

$$V_l = \frac{60\times 6\times 47\times 10^3}{0.783\times 10^6 (prot.t.P_e)\times 2400} = 3.00\times 10^{-3} m^3$$

Assume $V_{l\,diesel} = V_{l\,CNG}$ or $V_l = V_{hi}$ for CNG = $(8.07 \times 10^{-4} - 5.38 \times 10^{-5})\ m^3$

$= 7.532 \times 10^{-4} \times 4\ m^3$

$= 3.0128 \times 10^{-3}\ m^3$

∴ $V_{l\,diesel} = V_{l\,CNG} = 3.0 \times 10^{-3}\ m^3$

Assume S/D diesel = S/D CNG = 110/ 100

The rated effective power for the designed tractor CNG engine is

$$Ne^r_{CNG} = \frac{P_{e\,CNG} \times n^r_{CNG} \times V_{l\,CNG}}{60\times 6}$$

$$= \frac{0.695\times 10^6 \times 2400\times 3.00\times 10^{-3}}{60\times 2}$$

= 41.70 kW

The diameter, D of the cylinder is determined from the relation as:

$$D = \sqrt[3]{\frac{V_l}{\pi\left(S/D\right)}} \text{------------------------------}3.33 \text{ (Adgidzi, 1988)}$$

$$D_{diesel} = D_{CNG} = \sqrt[3]{\frac{3.00 \times 10^{-3}}{\pi(1.1)}}$$

$$= 0.0954\text{m}$$

$$\cong 0.09\text{m} \cong 90 \text{ mm}$$

Since S/D = 1.1 $\Rightarrow$ S = 1.1D

The length of stroke S, = 1.1 x 95.4 mm

$$\cong 104.5 \text{ mm}$$

The crank radius is 0.5S $\Rightarrow$ r = 0.5 (104.5) mm

$\therefore$ r = 52.25mm

r $\cong$ 52 mm = 0.052m

3.2.3.2 Indicated And Effective Specific Fuel Consumption

The indicated and effective fuel consumption (for rated power for engine design)

$$g_i = \frac{3600}{\eta_i Hu} \bullet 10^3 \quad \text{g/kW.hr} \text{--------------------------------------}3.34$$

Where η_i = (0.28 ....0.33) (Adgidzi, 1988)

$$g_{i\ diesel} = \frac{3600 \bullet 10^3}{0.30 \times 42{,}500}$$

$$= 282.35\text{g/kWhr}$$

where Hu_{diesel} = 42,500kj/kg, for $\eta_{i=diesel}$ =0.30

$$g_{i\,CNG} = \frac{3600 \bullet 10^3}{\eta_{iCNG}\,Hu_{CNG}}, assume\,\eta_{iCNG} = 0.32$$

$H_{u\,CNG} = 50,175$ kJ/kg

$$g_{i\,CNG} = \frac{3600 \bullet 10^3}{0.32 \times 50,175}\ \text{g/kW.hr}$$

$= 224.22$g/kW.hr

The effective specific fuel consumption, g_e is determined to be

$g_e = g_i \,/\, \eta_m$ --3.35

$$g_{e\,diesel} = \frac{g_{i\,diesel}}{\eta_{m\,diesel}}$$

$\eta_m = 0.72$, $g_i = 282.35$g/kWhr

$$= \frac{282.35}{0.72}$$

$= 392.15$g/kWhr

$$g_{e\,CNG} = \frac{g_{iCNG}}{\eta_{mCNG}}$$

$\eta_{m\,CNG} = 0.85$, $g_{i\,CNG} = 224.22$

$$= \frac{224.22}{0.85}$$

$= 263.79$g/kWhr

Effective indicated efficiency, $\eta_e = \eta_i\,\eta_m$ --------------------------3.36

$\eta_{e\,diesel} = \eta_{i\,diesel} \times \eta_{m.\,diesel}$

$= 0.30 \times 0.72 = 0.216 = 21.6\%$

$\eta_{e\,CNG} = \eta_{i\,CNG} \times \eta_{m\,CNG}$

$= 0.32 \times 0.85$

$= 0.272 = 27.2\%$

3.2.3.4 Specific Power – To- Volume Ratio

The power – to – volume ratio is the effective power produced by one volume of the engine operating volume.

$$\mu_i\,(\mu_{p-v}) = \frac{N_e}{V_h \times i_c}\ \text{kW/litre} \quad \text{--3.37}$$

Where $V_h = \frac{\pi D^2}{4} \bullet S$ D = 0.095m, S= 0.1045m

$$V_{h\,diesel} = V_{h\,CNG} = \frac{\pi(0.095)^2}{4} \times 0.1045 m^3$$

$= 7.4072 \times 10^{-4} m^3 = 0.74072$ litres

$$\therefore N_{i\,(diesel)} = \frac{N_{e\,diesel}}{V_h \bullet ic(V_l)} = \frac{47kW}{0.7407214}$$

$= 15.86$ kW/litre

$$N_{i\,CNG} = \frac{41.7kW}{3.litres} = \frac{N_{e\,CNG}}{V_l}$$

$= 13.90$ kW/litres

3.2.3.5 Specific Power –To – Piston Ratio

This is the effective power produced by $1dm^2$ area of the piston

$N_{p-p} = N_e/F_{Pic}$ --3.38

F_p = cross sectional area of piston head

$$F_p = \frac{V_h}{S} = \frac{7.4072 \times 10^{-4} m^3}{0.1045 m}$$

$= 7.088 \times 10^{-3} m^2$ ($100\ dm^2 = 1 m^2$)

$= 7.088 \times 10^{-3} \times 10^2\ dm^2$

$= 0.7088\ dm^2$

$$\therefore \mu_{p-p\ diesel} = \frac{N_{e\,diesel}}{F_p \times ic}$$

$$\mu_{p-p\ diesel} = \frac{47kW}{0.7088 \times 4}$$

$= 16.58\ kW/dm^2$

$$\mu_{p-p\ CNG} = \frac{41.7}{0.7088 \times 4}$$

$= 14.71\ kW/dm^2$

3.2.3.6 Mean Piston Velocity

The mean piston velocity is determined from

$$V_p = \frac{n'S}{30} \qquad \text{------------------------------------} 3.39$$

$V_{p\ diesel} = V_{p\ CNG}$ since $S_{diesel} = S_{CNG}$ & $n_{diesel} = n_{CNG}$

$$\therefore V_p = \frac{2400 \times 0.1045}{30}$$

$= 8.36 m/sec$

3.2.4 Theoretically Regulated Characteristics Of A Tractor Diesel Engine

This should be a comparison between the existing diesel (diesel) and the CNG engine.

Constructing the regulated characteristics as a function of revolution frequency (rotational speed).

3.2.4.1 Maximum Revolution Frequency (Rotational Speed) Without Load (Idle running)

$$n_{idle}^{max} = \frac{2+\delta p}{2-\delta p} n' \cong (1+8p)n', \text{1/min} \text{ --3.40}$$

Where δp – degree (level) of irregular control (regulation), take (δp = 0.06)

Then $n_{idle}^{max}(diesel) = \frac{2+\delta p}{2-\delta p} n^r diesel$ =2544rpm

Since $n^r{}_{diesel} = n^r{}_{CNG}$

$$n_{idle\ prot\ I}^{max} = n_{idle\ des}^{max} = 2544 rpm$$

3.2.4.2 Rotational Speed Under Maximum Torque

$$n_{idle}^{max} = \frac{n^r}{k_{o\delta}} \quad \text{min}^{-1} \text{ --3.41}$$

k_{od} = 1.3 – Engine fixture co –efficient (Adgidzi, 1988)

Since $n^r{}_{diesel}$ $n^r{}_{CNG}$ =2400rpm

$\Rightarrow n_{tk}^{max}{}_{diesel} = n^r{}_{CNG}$ = 2400/1.3

= 1846.15 rpm

At n_{tk}^{max} torque, $\mu_e = 0$. But at rated operating condition the torque is given as

$$t_k^r = \frac{9550Ne^r}{n^r} \text{ ---3.42}$$

$$t_k^r = \frac{9550 Ne^r_{diesel}}{n^r_{diesel}}$$

$$t_k^r = \frac{9550 \times 47}{2400} = 186.04 N.m$$

$$t_k^r = \frac{9550 Ne^r_{CNG}}{n^r_{CNG}}$$

$$t_k^r = \frac{9550 \times 41.7}{2400}$$

$= 165.93$ Nm

3.2.4.2 Maximum Torque (At The Max Speed, n^{MAX})

$t_k^{max} = t_k^r \bullet k_m$, Nm------------------------------------3.43

Where k_m is the torque fixture by moment. - $k_m = 1.2$

$\Rightarrow t_k^{max}{}_{diesel} = t^r_{k\,diesel} \bullet k_m$

$= 186.04 \times 1.2$ Nm

$t_k^{max} diesel \quad = 223.25$ Nm

$t_k^{max}{}_{CNG} = t^r_{k\,CNG} \bullet k_m$

$= 165.93 \times 1.2$ Nm

$t_k^{max}{}_{CNG} \quad = 199.116 \cong 199.12$ Nm

3.2.4.3 Hourly Fuel Consumption

The hourly fuel consumption is denoted by G_T^r and its as given below

$$G_T^r = \frac{g_e N_e^r}{10^3} \text{ kg/hr} \text{ ------------------------------------ } 3.44$$

To construct the graphs of hourly fuel consumption upon rotational speed, 3 – 4 values of GT are determined i.e. $GT = f(x)$

$$G_{T\ prot.t}^r = \frac{ge_{diesel} \times Ne^r_{diesel}}{10^3} \text{ kg/hr}$$

$$= \frac{392.15 \times 47}{10^3} kg/hr$$

$$= 18.43 \text{ kg/hr}$$

$$G_{T\ CNG}^r = \frac{g_{eCNG} \times Ne^r_{CNG}}{10^3}$$

$$= \frac{263.79 \times 41.70}{10^3}$$

$$= 11.00 \text{kg/hr}$$

In operating maximum condition (i.e. n_{idle}^{max})

G_{Tidle} = (0.22 ------------------------0.27) G_T^r --------------3.45(Adgidzi, 1988)

Taking $G_{Tidle} = 0.25 G_T^r$ for this design

$G_{Tidle\ (diesel)} = 0.25 G_T^r{}_{diesel}$

$= 0.25 \times 18.43$

$= 4.61$ kg/hr

$G_{Tidle\ (CNG)} = 0.25 G_T^r{}_{CNG}$.

$\Rightarrow G_{Tidle} = 0.25 \times 11.00$ kg/hr

$= 2.75$ kg/hr

At max torque. (i.e when n_{idle}^{max})

$$\therefore G_{Tk\ (diesel)}^{max} = \frac{1.1G_T{}'_{diesel}}{kod}$$

$$= \frac{1.1 \times 18.43 \times 1.2}{1.3}$$

$= 18.71$ kg/hr

$$G_{Tk\ (CNG)}^{max} = \frac{1.1G_T{}'_{CNG} \times k_m}{kod}$$

$$= \frac{1.1 \times 11 \times 1.2}{1.3}$$

$= 11.17$ kg/hr

3.2.4.4 Effective Fuel Consumption

Effective fuel consumption is donated by g_e

To construct the graph of effective fuel consumption upon rotation speed, i.e. $g_e = f(x)$.

3 – 4 values of g_e are determined using

$$g_e = \frac{G_T}{Ne} \times 10^3\, g / kW.hr \text{ --- } 3.47$$

Further values of G_T and if they are taken from the graphs and calculated as:

$$g_e{}^r{}_{diesel} = \frac{G_T{}^r{}_{diesel}}{Ne^r{}_{diesel}}$$

$$= \frac{18.43}{47} \bullet 10^3$$

$= 392.15$g/kWhr

$$g_{e\,max(diesel)} = \frac{G_T{}^{max}{}_{diesel}}{Ne_{max\,diesel}} \bullet 10^3$$

$$N_{emax\,(diesel)} = \frac{t_k^{max} \bullet n_k^{max}}{9550} = \frac{223.25 \times 1846.15}{9550} = 43.16kW$$

$$\Rightarrow g_{e\,max\,tk\,(diesel)} = \frac{18.71}{43.16} \times 10^3$$

$$= 433.50 g/kWhr$$

$$g_{e\,idle\,diesel} = \frac{G_{T\,idle}}{Ne_{idle}} \bullet 10^3\, g/kWhr$$

From the graph, Let N_e=10 kW for idle running

g_e at Ne = 10kW i.e. Ne $_{idle}$ 10 kW for both diesel and CNG.

Therefore $g_{e\,idle\,(diesel)} = \frac{G_{T\,idle\,diesl}}{Ne_{r\,diesel}} \times 10^3$

$$= \frac{4.61 \times 10^3}{10} = 461 g/kWhr$$

$$g^r_{e(CNG)} = \frac{G^r_{T(CNG)} \times 10^3}{N^r_e}$$

$$= \frac{11.00}{41.70} \times 10^3$$

$$= 263.79 g/kWhr$$

$$g^{max}_{etk\ (CNG)} = \frac{G^{max}_{TCNG} \times 10^3}{Ne^{tk}_{max(CNG)}}$$

$$Ne^{tk}_{max\,CNG} = \frac{t_k^{max} \bullet n_{tk}^{max}}{9550} = 38.49 \text{ kW}$$

$$\Rightarrow g^{max}_{etk(CNG)} = \frac{199.12 \times 1846.15}{9550}$$

$$= 290.21 g/kWhr$$

$$\Rightarrow ge^{max}_{tk\ (CNG)} = \frac{11.17}{38.49} \times 10^3$$

$$= 290.21 g/kWhr$$

$g_{e\ idle\ (CNG)}$ at $N_{e\ idle} = 10$ kW

Therefore $g_{eidle\ CNG} = \frac{G_{T\ idle(CNG)}}{Ne_{idle}} \times 10^3$

$$= \frac{2.75}{10} \times 10^3$$

$$= 275 g/kWhr$$

To construct the graph of control characteristics as a function of effective power: taking the various points at idle running, $N_e=0$, rated power, Ne^r and power under maximum Torque. Also, in the construction of the control characteristics as a function of torque, i.e. (n, N_e, G_T, g_e). The points are $t_k=0$ for the idle running, t_k^r and t_k^{max}.

The values used in the construction of control characteristics graphs are summarized in the table 4.3a and 4.3b:

3.2.4.5 Economic Effectiveness Of The Tractor Engine

This could be achieved or determined through the analysis of the cost of fuel consumption per hour or litres /km under various conditions.

3.2.4.5.1. Tractor Diesel Engine

The hourly fuel consumption is measured in kg/hr, but diesel is sold in litres, hence the G_T could be converted to litres/ hr by dividing the G_T by the density of diesel.

At idle running, G_T= 5.7625 litres/ hr

1 litre of diesel costs ₦ 40, therefore cost of using diesel per hour =5.7625x ₦ 40= ₦ 230.50

At rated condition, G_T=18.43/0.8 =23.037 litres/hr x40 = ₦ 921.50

At rated condition, cost of operating the tractor diesel engine = ₦ 921.50/hr

At maximum torque, G_T=18.71/0.8 =23.3575 litres/hr.

Cost of operating a tractor diesel engine under maximum torque = ₦ 985.50/hr

At maximum Torque, i.e. during a ploughing operation for instant, the speed of ploughing operation is 5.3km/hr in Nigeria (Yisa, 1997). Therefore the fuel consumption in litres per km is determined as follows;

$$\text{Fuel consumption in litres/km} = \frac{G_T \text{ at max. torque}}{\text{Speed of operation}}$$

$$= \frac{23.3x75\text{litres/hr}}{5.3\text{km /hr}}$$

$$= 4.413 \text{ litres/km} = ₦\ 176.52/\text{ Km}$$

3.2.4.5.2 Tractor CNG Engine

For automotive purpose, the fuel is measured by weight (in kilograms) when sold. Although measurement of kilogram is very accurate, the conversion to cents/litres is only meant to give on approximate ideal as to the quantity of useful energy compared to gasoline pumped into the vehicle, it is not to be taken as an accurate means of measurement (Technocarb, 2003).

In Nigeria, the sale of CNG has not be commercialized, but it is known to be 50 – 60% less than the cost per litres of traditional fuel (NGC release).

Assume 1litre of CNG cost $^1/_2$ the cost of 1 litre of diesel. (₦20)

G_T litres/hr = G_T kg/hr / (density)

Density of methane = 0.55kg/litre

At the idle running, G_T = 2.75/ 0.55 = 5 litres/hr =₦ 100/hr

At rated condition, G_T = 11.00/ 0.55 = 20 litres/hr =₦ 400/hr

Under maximum torque, G_T = 11.17 /0.55 = 20.31 litres /hr=₦ 406.20

Fuel consumption in litres/km under max torque = 3.83 litres/km = ₦ 76.60/km

3.2.5 Engine Dynamic Design

This is required in the determination of forces and moments acting on the crank mechanism of the engine and also the determination of moment of inertia and the mass of the flywheel.

3.2.5.1 Determination Of Forces Acting On The Gudgeon Pin

(i) The gas pressures, P_r and forces of inertia, P_j

$$P_r = (P_x - 0.1)\,\pi D^2 /4 \times 10^6 \quad \text{--}3.48$$

Where P_x – running gas pressure (at points considered) taken from the indicated diagram.

D =cylinder diameter = 0.095m = 95 mm

From table 4.2, the values of the running gas pressures are stated for both engines.

Between 0 – 180^0 crank angle rotation, intake takes place at constant gas pressure, Also, between 550 – 720^0 crank angle rotation of Exhaust/expansion process takes place at constant gas pressure.

The process starts from point a (intake) on the indicated diagram through c(compression) z (combustion) and b (expansion) and back to a. This completes the cycle here. The values of P_{x1} and Px_2 for both engines are substituted into eqn 3.48 and the values for gas pressures, P_r is determined for each angle of crankshaft rotation α (radian) between 0^0 to 720^0 at 10^0 intervals.

The P_r of the two engines and as stated in the table 3.4a and 3.4b respectively.

.ii) Forces of inertia

This is denoted by P_j and it is divided into two i.e. 1st and 2nd order force of inertia

i.e. $P_j = P_{j1} + P_{j2}$

Where $P_{j1} = - M\,\omega^2 r\cos\alpha$ (force of inertia of the 1st Order) ------------------------3.50

$Pj_2 = -M\omega^2 r\lambda\cos2\alpha$(force of inertia of the 2nd order)------------------------3.51

Therefore $P_j = -M\omega^2 r(\cos\alpha + \lambda\cos2\alpha)$ ---3.52

Where M = mass of the reciprocating components (of the crank mechanism) given by the relationship

$M = M_p + \lambda M_{rod}$, kg---3.53

Where $\lambda = 0.25$ for this design

M_p = mass of piston component, kg

M_{rod} = mass of the connecting rod, kg

ω = angular velocity of crankshaft (under rated operating condition)

$$\omega = \frac{\pi n'}{30} \text{ rad/s} \text{ -------------------------- } 3.54$$

and r = radius of the crankshaft determined to be 52 mm = 0.052m. The forces of inertia P_{j1}, P_{j2}, & P_j are constructed in this order where $0^0 < \alpha$ 720^0 at interval of 10^0 measured in radians

The mass of the reciprocating components is taken to be

$$M = M_p + 0.25 M_{rod} \quad (\lambda = 0.25)$$

Where M_p =2.544kg and M_{rod} =2.7kg (Adgidzi, 1988)

Therefore, M = [2.544+ 0.25(2.70)] kg

= 3.219 kg

r = 0.052m

The angular velocity ω is calculated for both engines as follows

Since $n'_{prot.t.} = n'_{CNG} = \omega_{prot.t} = \omega_{CNG}$

$$\therefore \omega = \frac{\pi n'}{30} = \frac{\pi \times 2400}{30} = 251.327 rad/sec$$

$$M\omega^2 r = -3.219 \times 251.327^2 \times 0.052 = -10573.11N$$

Substituting the value $M\omega^2 r$ into eqn 3.50 and 3.51 to determine the forces of inertia acting on the engine

The value of the forces of inertia were determined in this order and the results is as presented in tables: 3.4 a & b of appendix 1 with α(crank angle of rotation) ranges between $0^0 - 720^0$ at 10^0 interval

Since the parameter required for the determination of the inertia forces for the engine remain the same.

$$P_{j\ diesel} = P_{j\ CNG}$$

The resultant force, P_{res} acting on the gudgeon pin is given by:

$P_{res} = P_r + P_j$ --3.55

The P_{res} is also calculated for each angle of rotation of the crankshaft at interval of 10^0

Since $P_{r\ diesel} \neq P_{r\ CNG} \Rightarrow P_{res.\ diesel} \neq P_{res.\ CNG}$

The results are as presented in tables 3.4a and b of appendix 1

3.2.5.2 Determination Of Forces Acting On The Crank Pin (Of Crankshaft):

Two forces acting on the connecting rod

$$P_{CR} = \frac{P_{res.}}{\cos\beta}, \text{N} \text{ --------------------------------------- } 3.56$$

Cos β - deflecting angle between connecting rod axis and cylinder axis as the crankshaft rotates with angle α

From the geometry of piston and crank relation Fig.2.7.1 i.e. equation 2.64

$$\beta = \sin^{-1} r/_l \sin\alpha, \quad r/l = 0.25$$

(ii) ⇒ $\beta = \text{Sin}^{-1}(0.25 \sin\alpha)$

Equation 3.56 now becomes

$$P_{CR} = \frac{Pres}{\cos[\sin^{-1}(0.25\sin\alpha)]}$$

Substituting the values of P_{res} and α for α = 0 to 720 in radian, for both tractor engines (CNG and diesel) to obtain PCR values. The results are presented in tables 3.4 a & b of appendix 1. The centrifugal force of inertia, P_c is determined from the relation as:

$P_c = -(1-\lambda)\ Mrod\ r\omega^2$, N, $\lambda = 0.25$

$P_c = -0.75M\ rod\ \omega^2 r$ --3.57

$P_{c\ diesel} = P_{c\ CNG}$

Therefore, $P_c = -0.75 \times 2.70 \times 251.327^2 \times 0.052$

$= -6651.30N$

The geometric sum of P_{CR} and P_c gives R` (resultant force) for a single crank pin and a cylinder. The result is as shown in tables 3.4 a & b of appendix 1

The connecting rod force, P_{CR} consists of two components as the force Z_1 acting through the crank radius and the tangential fore, T, acting perpendicular to the crank radius.

$$Z_1 = \Pr es.\frac{\cos(\alpha + \beta)}{\cos\beta} \quad \text{--3.58}$$

$$T = \Pr es.\frac{\sin(\alpha + \beta)}{\cos\beta} \quad \text{---3.59}$$

The resultant force, R therefore can be calculated by the given formula:

$$R = \sqrt{(Z_1 + R`)^2 + T^2}, \qquad N \text{ -----------------3.60}$$

These forces are determined for both diesel engine and the CNG engine and the results are presented in tables 3.4 a & b of appendix 1

3.2.5.3 Determination Of Moment Of Inertia Of A Flywheel

The tangential force, T diagram with mean ordinates of diagram (Y – co ordinate) is drawn and the mean P is determined as

$$P = \frac{\sum F_{positive} - \sum F_{negative}}{l_d} \quad \text{--------------------------------------3.61}$$

Where $\sum F_{positive}$ – total area in the diagram above the x – axis, mm^2

$\sum F_{negative}$ – total area in the diagram below x – axis, mm^2

l_d – length of the diagram, mm

From the diagram; Figs. 4.5.2.3 a&b for diesel & CNG respectively

For the diesel engine, P = (2500 – 840) mm^2/204 mm

= 8.137 mm

For the CNG engine, P = (2825 – 875) mm^2/203 mm

= 9.606 mm

For multi –cylinder engine, the sum of tangential forces diagram is constructed for all the number of cylinders and the average of the P_i (in mm) is used. (where i = no of cylinder)

Therefore, $T_{av} = P.\, i\, \mu_I N$ --------------------------------------3.62

Where $\mu_I = T/l_h$, N/mm ----------------------------3.63

μ_I – scale on Y axis, l_h = height of diagram

$\mu_{I\ diesel}$ =230.04N/mm, $\mu_{I\ CNG}$ = 195.55N/mm

$\Rightarrow T_{av\ (diesel)}$ = 8.134 x 4 x 230.04 = 7487.405N

$T_{av\ (CNG)}$ = 9.606 x4 x 195.55 = 7513.970N

The values of P_i is used to check for accuracy of construction of the sum of the tangential forces diagram and the determination of the dynamic characteristics of the engines as:

$$\frac{Pi\mu rn^r}{9550}\eta_m = N_e^{rated} \quad \text{--3.64}$$

Where r is the crank radius, r = 0.052m

η_m – mechanical efficiency of the engines $\eta_{m\ diesel}$ = 0.72, and $\eta_{m\ CNG}$ =0.85

$\Rightarrow$ For diesel = $8.137x4x230.04x0.052x\,{}^{2400}\!/_{9550}\,x0.72 = 70.08kW$

Ne^r =47kW, a difference of (32.93%)

For the CNG = $9.606x4x195.55x0.052x\,{}^{2400}\!/_{9550}\,x0.85 = 83.03kW$

Ne^r = 41.7 kW, a difference of 49.6%

The ordinate P.i is laid on the sum of tangential forces diagram and then on the maximum value (from the diagram) is determined as the excess area (F_{ex}) from where the excess work is determined as

$L_{ex} = \mu F_{ex}$ ---3.65

Where $\mu = \mu_1 . \mu_2$. Nm/mm^2 scale area ---------------------3.66

The x – axis scale of area is given as

$$\mu_2 = \frac{4\pi r}{l_d} \text{ --3.67}$$

From figs 4.5.2.3a & b

$\mu_{2\ diesel} = 0.0032 m/mm$, $\mu_{2\ CNG} = 0.0032 m/mm$

$\mu = \mu_1 . \mu_2$

$\mu_{diesel} = 230.04 \times 0.0032 = 0.7361\ Nm/mm^2$

$\mu_{CNG} = 195.55 \times 0.0032 = 0.6258\ Nm/mm^2$

$\Rightarrow L_{ex\ (diesel)} = 0.7361 \times 1680 = 1236.65\ Nm$

$L_{ex(CNG)} = 0.6258 \times 1750 = 1095.15\ Nm$

Given the level of irregular rotation of the crank shaft ($\delta_{CR} = 0.03$)

The moment of inertia of the flywheel is given as:

$$j_{fw} = \frac{0.8 L_{ex}}{\delta_{CR}\left(\frac{\pi n^r}{30}\right)^2} \text{ ,kg/m}^2 \text{ --------------------3.68 (Adgidzi, 1988)}$$

$j_{fw\ (diesel)} = 0.5219\ kg/m^2$

$j_{fw(CNG)} = 0.4622\ kg/m^2$

Having known the moment of inertia of the flywheel, j_{fw}, the mass of the flywheel is determined as:

$$M = \frac{4 j f_w}{D_{av}^2} \text{ -------------------------------- } 3.69 \text{ (Adgidzi, 1988)}$$

Taken $D_{av} = 0.40$m

$$M_{diesel} = \frac{4 \times 0.5219}{0.4^2}$$

$$= 13.05 \text{ kg}$$

$$M_{CNG} = \frac{4 \times 0.4622}{0.4^2}$$

$$= 11.56 \text{ kg}$$

CHAPTER FOUR

4.0 RESULTS AND DISCUSSIONS

4.1.0 Indicated Diagram Parameters

The summary of results of the indicated diagram parameters is as presented in table 4.1.0 i.e. the indicated parameters at the completion of intake, compression, combustion and exhaust (expansion). These include pressure, temperature and volume.

Table 4.1.0. Summary of Results for indicated parameters.

Indicated Parameter	Diesel Engine	CNG Engine
Volume (m^3): V_a	$6.55 \times 10^{-4}\ m^3$	$8.07 \times 10^{-4}\ m^3$
V_c	$4.04 \times 10^{-5}\ m^3$	$5.38 \times 10^{-5} m^3$
V_z	$4.28 \times 10^{-5}\ m^3$	$5.70 \times 10^{-5} m^3$
V_b	$6.55 \times 10^{-4} m^3$	$8.07 \times 10^{-4} m^3$
Temperature (K) : T_a	333.55K	345.08K
T_c	988.26K	992.19K
T_z	2488K	2245K
T_b	1293.19K	1188.60K
Pressure (P_a): P_a	146.24 kPa	122.68 kPa
P_c	7.02 mPa	5.29 mPa
P_z	13.34 mPa	10.05 mPa
P_b	0.453 mPa	0.376 mPa
P_0	101.325 KPa	101.325kPa
P_r	1.06 mPa	1.06 mPa

4.1.1 Pressure Pa And Temperature Ta At Completion Intake (And Commencement Of Compression)

For the diesel engine, the pressure at intake is 0.146mPa (146.24kPa) while the CNG engine has a pressure of 0.123mPa (122.68kPa). The difference between the intake pressure is as a result of the difference in the filling co-efficient of the cylinder with mixture, η_v of the two fuels engines.

The temperature at intake for the diesel engine is 333.55k while that of the CNG engine is 345.08k. The difference is as a result of the effect of the residual gas temperature.

The volume at intake of the diesel engine is $6.55 \times 10^{-4} m^3$ while that of the CNG engine is $8.07 \times 10^{-4} m^3$. This implies that more of the air-fuel mixture was compressed by the tractor CNG engine.

Statistical analysis show that these calculated parameters are statistically correlated with the existing diesel engine parameters with a standard error of less than 1%, except the intake temperature of about 6% error (Appendix 2).

4.1.2 Pressure P_c, Temperature, T_c, Volume V_c At Completion Of Compression Process (Commencement Of Combustion)

The pressure P_c at compression for the diesel engine is 7.02 mPa and the volume V_c is $4.04 \times 10^{-5} m^3$. For the design engine, the pressure at compression, P_c is 5.29 mPa at a volume Vc of $5.38 \times 10^{-5} m^3$. In both cases, the pressures are increased and the volumes are reduced, hence Boyle's law was obeyed. The P_c CNG is lower than the P_c diesel, this may be because gases are compressible. The temperature T_c at compression for the diesel engine is

988.26k while that of the CNG engine is 922.19K. This is because the compression ratio of diesel engine is higher and it is self ignited thus the higher temperature.

Statistical analysis show that the tractor CNG engine parameters are statistically correlated with the diesel engine parameters with standard error between 0-2 (Appendix 2).

4.1.3 Pressure P_z, Temperature T_z And Volume V_z At The Completion Of Combustion Process (Commencement Of Expansion Process)

For the diesel engine, the pressure at completion of combustion process is 13.34 mPa at a volume $4.28 \times 10^{-5} m^3$; the design engine has pressure of 10.05 mPa at a volume of 5.70×10^{-5} at the completion of the combustion process. The increase in pressure is as a result of the work done (heat energy) released on the combustion chamber from the combustion of fuel, the increase in volume is as a result of the effect of the residual gases. The temperature T_z of the diesel engine is 2,488K and 2,245K for the CNG engine. The high temperature may be due to the heat of combustion in the combustion chamber. The temperature during the combustion process of a diesel engine is above 2000^0C (Adgidzi, 2002).

The calculated parameters for the CNG engine is correlated with the existing diesel engine parameters with a standard error of between 0-2% except the temperature at the completion of the combustion process with 121.2% error. (Appendix 2).

4.1.4 Pressure, p_b, and temperature, T_b, and volume, V_b, at the completion of expansion process (commencement of intake process).

The pressure at the completion of the expansion process for the diesel engine is 0.453 mPa and that of the CNG engine is 0.376 mPa. It is stated by Liljedahl et al, 1989 that exhaust stroke takes place above atmospheric pressure of 101.3 kPa. This is because before the end of expansion process intake is already taking place, hence the decrease in pressure. The temperature at the completion of the expansion process for the diesel engine is 1,293.19k and the CNG engine has a temperature of 1188.60k at the completion of expansion process. This difference may be due to the effect of the residual gases temperature. The volume V_b At completion of expansion is the same as volume V_a at intake for both tractor engines.

The expansion volumes and pressures have a standard error of less than 2% while the temperature at the completion of the expansion process has an error of 52%. This may be due to the larger values of the temperatures.

These indicated parameters were used in construction of the indicated curves and this is shown in Fig 4.10 and Fig 4.20 for tractor diesel and CNG engines respectively.

Table 4.2: Values of V_x and P_x used in the construction of indicated diagram for both diesel & the design engine

DIESEL			CNG		
Volume (V_x) x $10^{-5}m^3$	Px_1 Compression (Pressure)	Px_2 Expansion (Pressure)	Volume (V_x) x $10^{-5}m^3$	Compression Pressure (Px_1)	Expansion Pressure (Px_2)
7.28	3.094	6.906	9.34	2.464	5.451
10.52	1.855	4.375	13.30	1.508	3.517
13.76	1.277	3.136	17.26	1.049	2.546
17.00	0.952	2.413	21.22	0.788	1.970
20.24	0.747	1.943	25.18	0.621	1.594
23.48	0.608	1.616	29.14	0.507	1.330
26.72	0.508	1.377	33.10	0.425	1.135
29.96	0.433	1.195	37.06	0.363	0.987
33.20	0.375	1.052	41.02	0.315	0.870
36.44	0.330	0.937	44.98	0.277	0.776
39.68	0.293	0.843	48.94	0.247	0.699
42.92	0.263	0.765	52.90	0.221	0.635
46.16	0.238	0.699	56.86	0.200	0.580
49.40	0.216	0.643	60.82	0.182	0.534
52.64	0.198	0.594	64.78	0.167	0.494
55.88	0.182	0.552	68.74	0.154	0.459
59.12	0.168	0.514	72.70	0.142	0.428
62.36	0.156	0.481	76.66	0.132	0.400
65.60	0.146	0.453	80.62	0.123	0.376

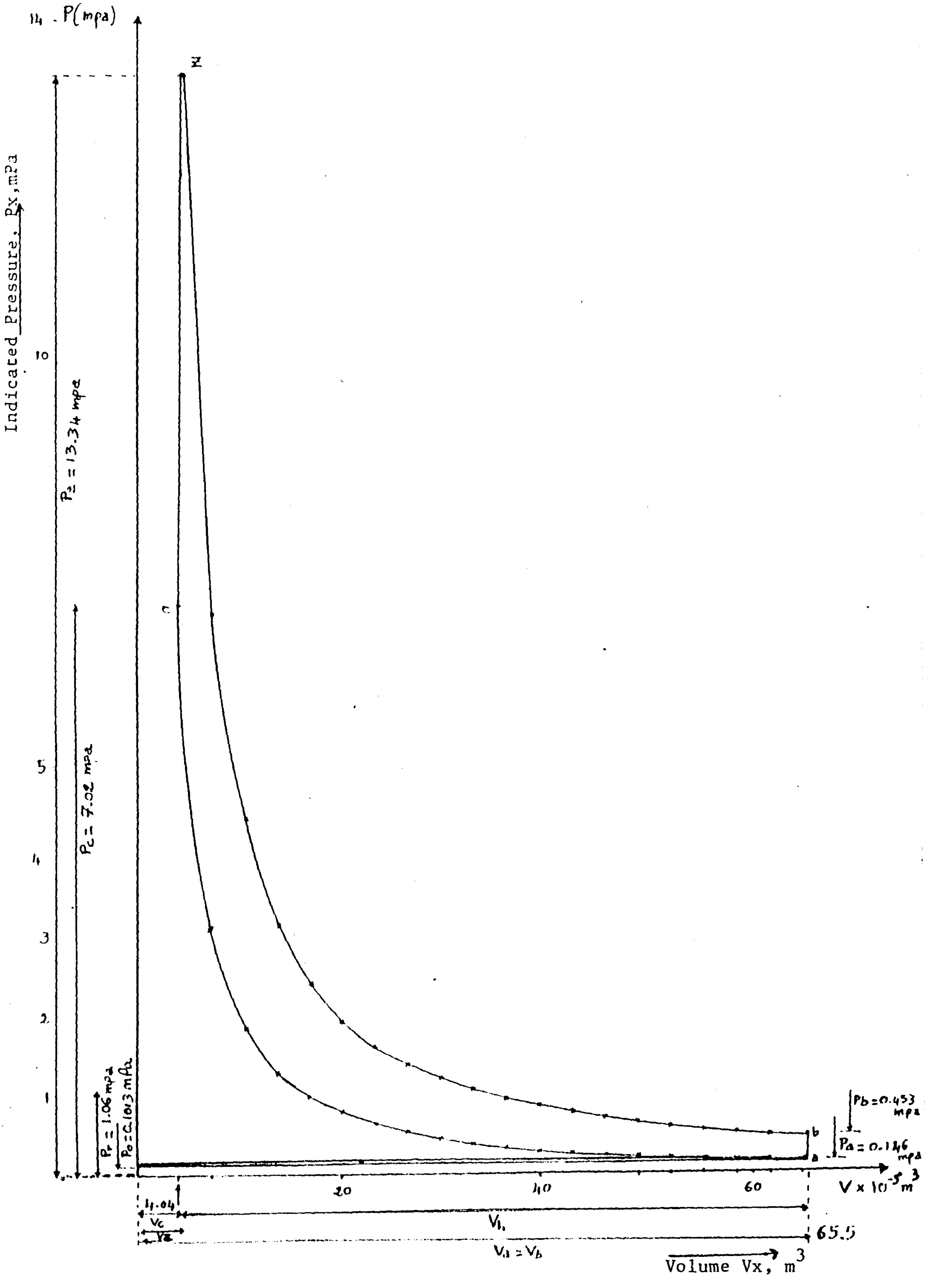


Fig: 4.1.0 Indicated Diagram for Tractor Diesel Engine

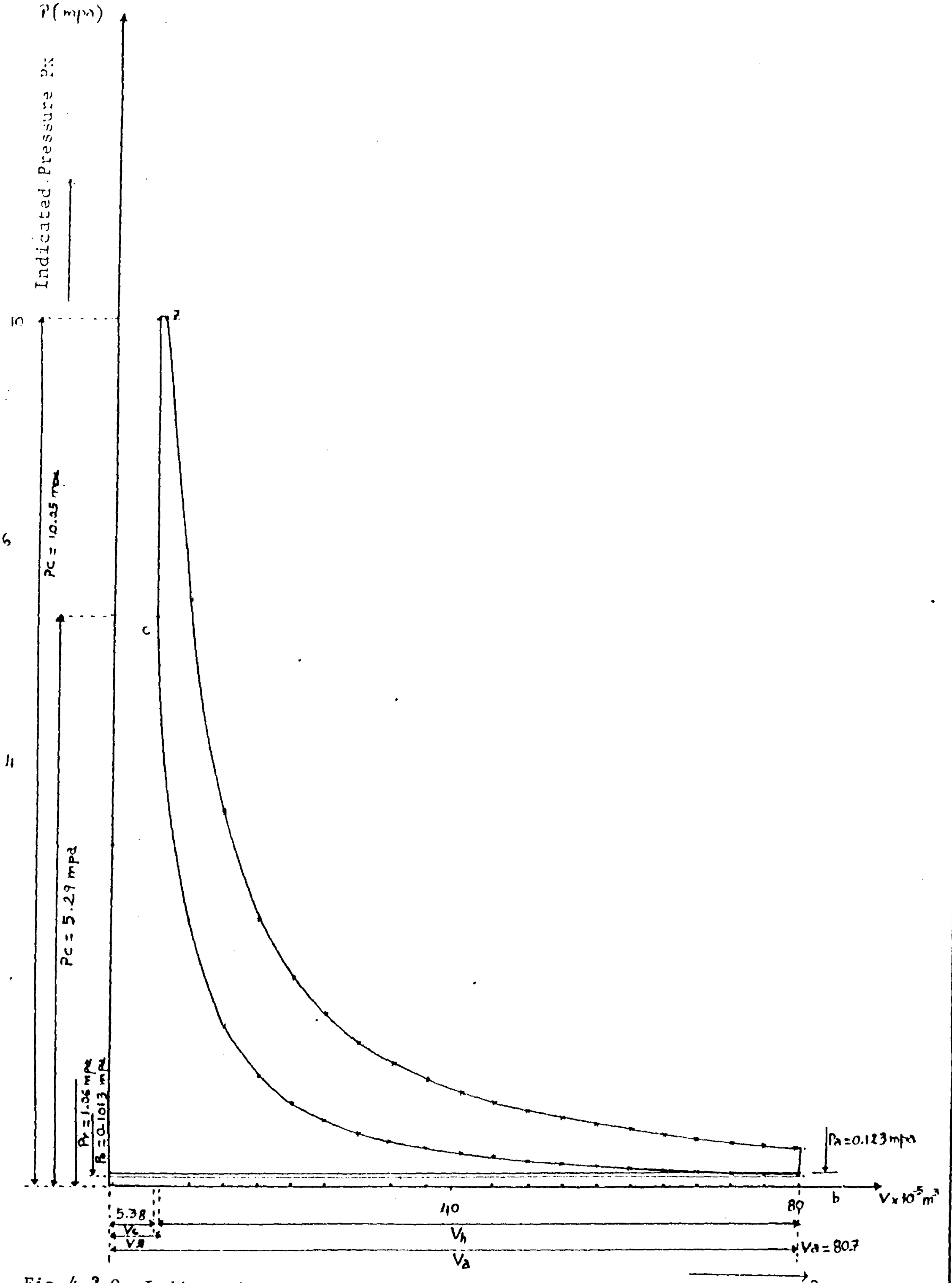


Fig 4.2.0 Indicated Diagrame for Tractor CNG Engine Volume, Vx, m^3

4.1.5 Mean Indicated Pressure From The Indicated Curve

The mean indicated pressures of the two engines were determined from the area under the indicated curves, the length of the curve and the scale of construction. The mean indicated pressure of the diesel engine is 1.170 mPa and the mean indicated pressure of the CNG engine is 0.880 mPa. The difference in the mean indicated pressure is due to the lager area of the indicated curve of the diesel engine (3580 mm^2) and scale of construction, scale of 1mm=0.05 mPa while the CNG engine has area of 3300 mm^2 and scale of 1mm=0.04 mPa

4.2 Engine Parameters And Fuel Economy Indices

4.2.1 Mean Effective Pressure

The mean effective pressure of the diesel engine is determined to be 0.783 mPa. The mean effective pressure of diesel engine ranged between 0.6-0.9 mPa. The mean effective pressure falls within the range. The mean effective pressure of the CNG engine is 0.695 mPa. This is less than that of the diesel engine, and it is a reflection of the values of the indicated parameters i.e. the pressure values from intake process through the exhaust process of the diesel engine is greater than the values of the CNG engine. But the mechanical efficiency of the CNG engine is greater than that of the diesel engine due to the complete combustion of the CNG fuel. The mean effective pressure is the major factor in determining the effective power of the engine, the diesel engine should generate more power than the CNG engine

The mean effective pressure of the tractor CNG engine is statistically correlated with the existing diesel engine with a standard error of less than 1%.

4.2.2 The Engine Displacement Capacity.

The engine displacement capacity of the diesel engine was determined to be $3.0 \times 10^{3} m^{3}$ taken into consideration, the effective rated power, 47kW, and the mean effective pressure 0.783 mPa the engine speed (rated), 2400rpm and a factor of 2 for four- stroke engine.

During the design process, the engine displacement capacity was assumed to be the same for the two tractor engines since the stroke to bore ratio was assumed to be the same for the two engines. Hence, the rated effective power (N_e^r CNG.) for the CNG engine was determined to be 41.7 kW. From the effective power rating of the two engines, they both fall under the same category of tractor (i.e. e.g. standard or row crop tractor, power ranges from 15 to 150 kW, utility tractor or light industrial tractor, power ranges from 15 to 100 kW) courtesy of Ford Motor Co. Also we have skid-steer tractor, power ranges between 15 kW to 60 kW (Courtesy J.I Case Co).

The effective power of the tractor CNG engine has a standard error of 2.65% compared with the existing diesel engine. Statistically, they are correlated and are in the same category of tractor classification.

The diameter of the Cylinder, D, is determined to be 95 mm (0.095m) for the two engines, and the stroke length is 104.5 mm = 0.1045m. The radius of the crankshaft(crank radius)is determined to be 52 mm = 0.052m.

4.2.3 Indicated and Effective Specific Fuel Consumption

4.2.3.1 Indicated Fuel Consumption

For the diesel engine, the indicated fuel consumption (g_i) was determined to be 282.35g/kWhr at an indicated efficiency of 0.30. The CNG engine has an indicated fuel consumption of 224.22g/kWhr at an indicated efficiency of 0.32. The heat of combustion of the CNG engine is greater than the diesel and the efficiency is also greater than that of the diesel, hence, the indicated fuel consumption of the CNG engine is less than that of diesel Therefore, the CNG engine is more economical.

4.2.3.2 The Effective Specific Fuel Consumption

The effective specific fuel consumption, g_e for the diesel engine is 392.15g/kWhr while that of the CNG engine is 263.79g/kWhr. This is as a result of the mechanical efficiency of the diesel engine is 0.72 while that of CNG engine is of 0.85. More fuel will be converted to heat energy leading to thermal loss and exhaust gas generation in the diesel engine. Therefore CNG engine is more economical than the diesel engine.

4.2.3.3 Effective Indicated Efficiency

The diesel engine has an effective indicated efficiency of 21.6% while the CNG engine has indicated efficiency of 27.2%. This implies that the CNG engine is more economical than the diesel engine.

4.2.3.4 Specific Power – To – Volume Ratio

The specific power- to volume of the prototype engine was determined to be 15.86kW/litres, and the design tractor engine has 13.90 kW/litres. This implies that more power to volume is experienced in the diesel engine than the CNG engine.

4.2.3.5 Specific Power - To - Piston Ratio

For the diesel engine, the effective power produced by $1dm^2$ area of the piston is 16.58 kW while the CNG engine has a specific power – to – piston ratio of 14.71 kW/dm^2.

4.2.3.6 Mean Piston Velocity

Since the two engines have the same stroke length, and equal rated engine speed of 2400rpm, their mean piston velocity is the same and it is determined to be 8.36m/s.

4.3 Theoretically Regulated Characteristics Of Tractor Engines

4.3.1 Maximum Revolution Frequency (Rotational And Speed) Without Load (Idle Running)

Since the diesel and CNG engines have the same rotational speed of 2400rpm and the two assumed to have the same level or degree of irregular control (regulation) of 0.06, the maximum revolution frequency for the two engine at idle running and n_{idle}^{max} was determined to be 2544rpm.

4.3.2 ROTATIONAL SPEED UNDER MAXIMUM TORQUE

The diesel engine and the CNG engines have the same engine fixture co- efficient, K_{on} = 1.3.The two engines have the same rotational speed under maximum torque and it is determined to be 1846.15 rpm. As stated in section 3.2.4.2, that at maximum rotational speed without load, the torque is 0.But at rated operating condition, the torque developed by the diesel engine was 186.04 Nm, while the CNG engine developed a torque of 165.93 Nm.

4.3.3 Maximum Torque (At The Maximum Speed, n^{MAX})

The two engines have the same engine fixture by moment, k_m equals 1.2, hence, the maximum torque at the maximum speed for prototype engine and design engine was 223.25Nm and 199.12 Nm respectively. This implies that at equal speed of rotation of the Pto shaft, the power developed at the Pto (Pto power) of diesel engine was greater than that of the CNG engine.

Table 4.3a: Summary of values used in the construction of control characteristics graph for the tractor Diesel engine

Condition of operation	Engine speed n,(rpm)	Torque t_k (Nm)	Engine effective power, N_e, (kW).	Hourly fuel consumption G_T,(kg/hr)	Effective fuel consumption, g_e, (g/kWhr)
Idle running	2554	0.00	0.00, 10	4.61	∞, 461.00
Rated condition	2400	186.04	47.00	18.43	392.15
Max. Torque	1864.15	223.25	43.16	18.71	433.50

Table 4.3b: Summary of values used in the construction of control characteristics graph for the tractor CNG engine

Condition of operation	Engine speed n, (rpm)	Torque, t_k (Nm)	Engine effective power, N_e, (kW).	Hourly fuel consumption G_T, kg/hr	Effective fuel consumption g_e (g/kWhr)
Idle running	2554	0.00	0.00, 10	2.75	∞, 275
Rated condition	2400	165.93	41.70	11.00	263.89
Max. Toque	1864.15	199.12	38.49	11.17	290.21

4.4.0 Hourly Fuel Consumption

The hourly fuel consumption for each engine was determined under various conditions such as idle running, under rated condition and under maximum torque. The hourly fuel consumptions for the diesel engine under the stated condition were 4.61 kg/hr, 18.43 kg/hr and 18.71 kg/hr respectively. Similarly, the CNG engine has an hourly fuel consumption of 2.75 kg/hr, 11.00kg/hr and 11.17 kg/hr under idle running, rated condition and under maximum torque respectively. This implies that the CNG engine is more economical in terms of fuel consumption than the diesel engine. Considering also the fuel consumption in litres/hours and the economy implication i.e. the cost per hour, for a specific farm operation say ploughing where the engines were under maximum torque, the fuel consumed per km by the CNG engine (3.832=₦76.60 litre/km) is less than that of the diesel engine (4.413litres/km ₦176.52/km).

From the graphs, the hourly fuel consumption was maximum under maximum torque for both engines, this implies that more fuel is consumed per hour under maximum torque.

The statistical analysis show that the two tractor engines have correlated Hourly fuel consumption with standard error between 1-4% depending on the condition of operation of the two engines (Appendix 2).

4.4.1 Effective Fuel Consumption

The effective fuel consumption, g_e (g/kW.hr) under various conditions of idle running, rated condition and under maximum torque was determined for both the tractor diesel and the CNG engine. The tractor diesel engine has an effective fuel consumption of 392.15g/kWhr under rated condition, 433.950g/kWhr under maximum torque and 461g/kWhr under idle running at engine power N_e = 10 kW. The calculated effective fuel consumption of CNG engine is 263.79g/kWhr at rated condition, 290.21g/kWhr under maximum torque and 275 g/kWhr under no load of 10 kW engine power gotten from the graph. The summary of values used in the construction of control characteristics graphs is as presented in table 4.3a &b. It implies that the tractor CNG engine is more economical than the tractor diesel engine.

The effective fuel consumption of the diesel engine is at maximum under idle running condition when the effective power equals 10 kW. At rated condition it is at minimum, under maximum torque, the effective fuel consumption increased to 433g/kWhr. But in the CNG engine, the effective fuel consumption followed a different pattern, the effective fuel consumption is maximum under maximum torque while at rated condition it has minimum value of 263.79g/kWhr. Under maximum torque, the effective power of the engines (for both diesel and the CNG) dropped from 47kW to 43.11 kW and 41.7 kW to 33.49 kW respectively, hence the increase in the effective fuel consumption.

The specific fuel consumption for practical application for a 4 stroke diesel engine ranges between 200 – 250g/kWhr (Adgidzi, 2002) but for 4 stroke petrol engine g_e = 280 – 340g/kWhr while the effective fuel consumption determined at rated condition is 392.15 kg/kWhr. But the design engine, using CNG as alternative fuel has its specific fuel consumption within the range of the petrol engine.

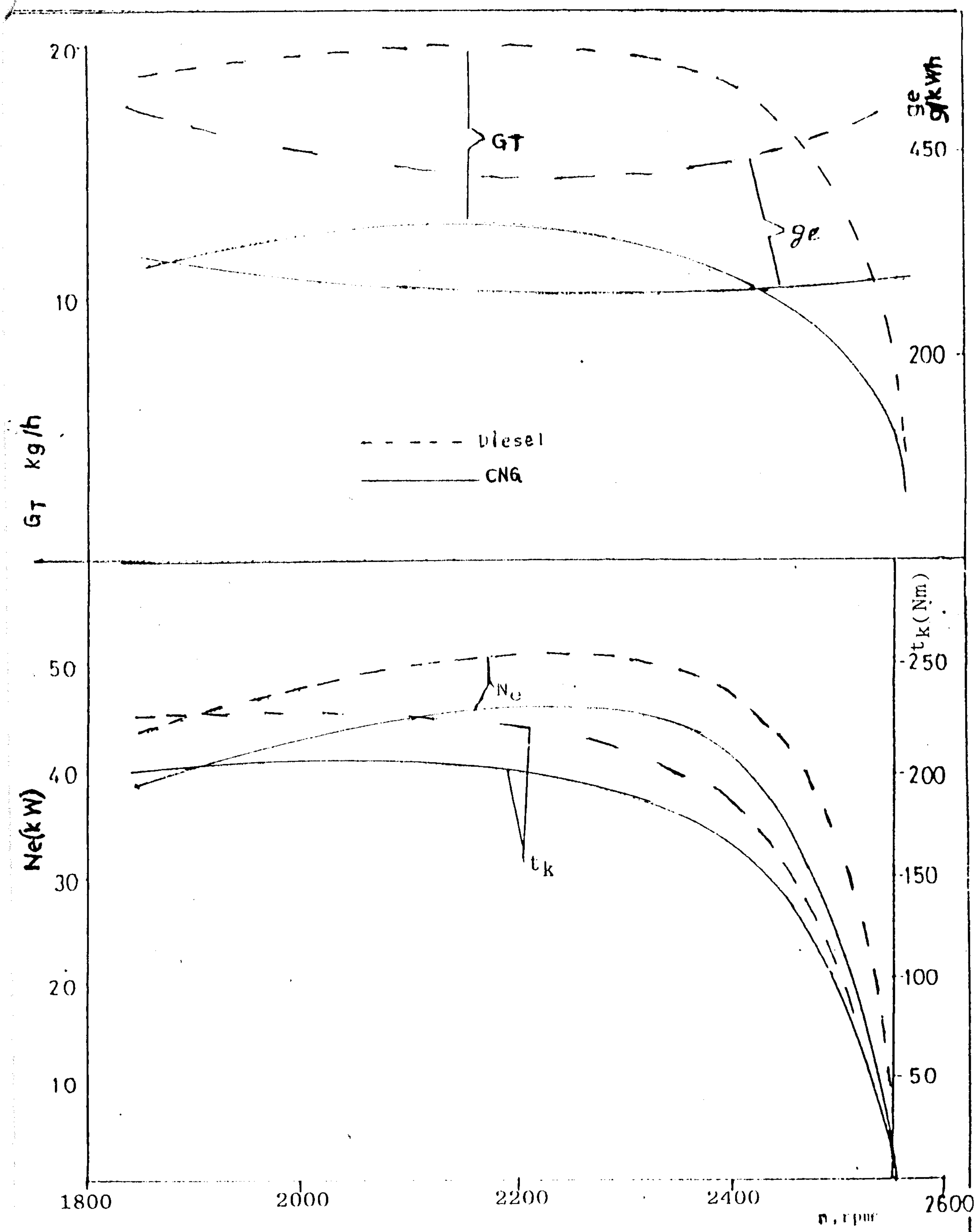
Statistical analysis show that the specific fuel consumption of the CNG engine is correlated (99%) with the existing engine but the standard error is greater and ranged between 64 – 93% depending on the condition of operation of the engines. Hence, the CNG engine has lower values of specific fuel consumption, which lead to the higher standard error. This implies that the tractor CNG engine is economical (Appendix 2).

Figs 4.41 a -d show the control characteristics graphs for both engines. These graphs include: Fig 4.41a - graph of hourly fuel consumption as a function of rotational speed, i.e. n = f(N_e, t_k , G_T, g_e)

Figs 4.41b - Control characteristics as a function of torque i.e. t_k = f(N_e , n, g_e, G_T)

Figs 4.41c - Control characteristics as a function of effective power i.e. N_e = f(n, g_e)

Fig.4.4.1d - Control characteristics as a function of effective power i.e. N_e= f(t_k, G_T)



G: 4.4.1a: CONTROL CHARACTERISTICS AS A FUNCTION OF ROTATIONAL SPEED

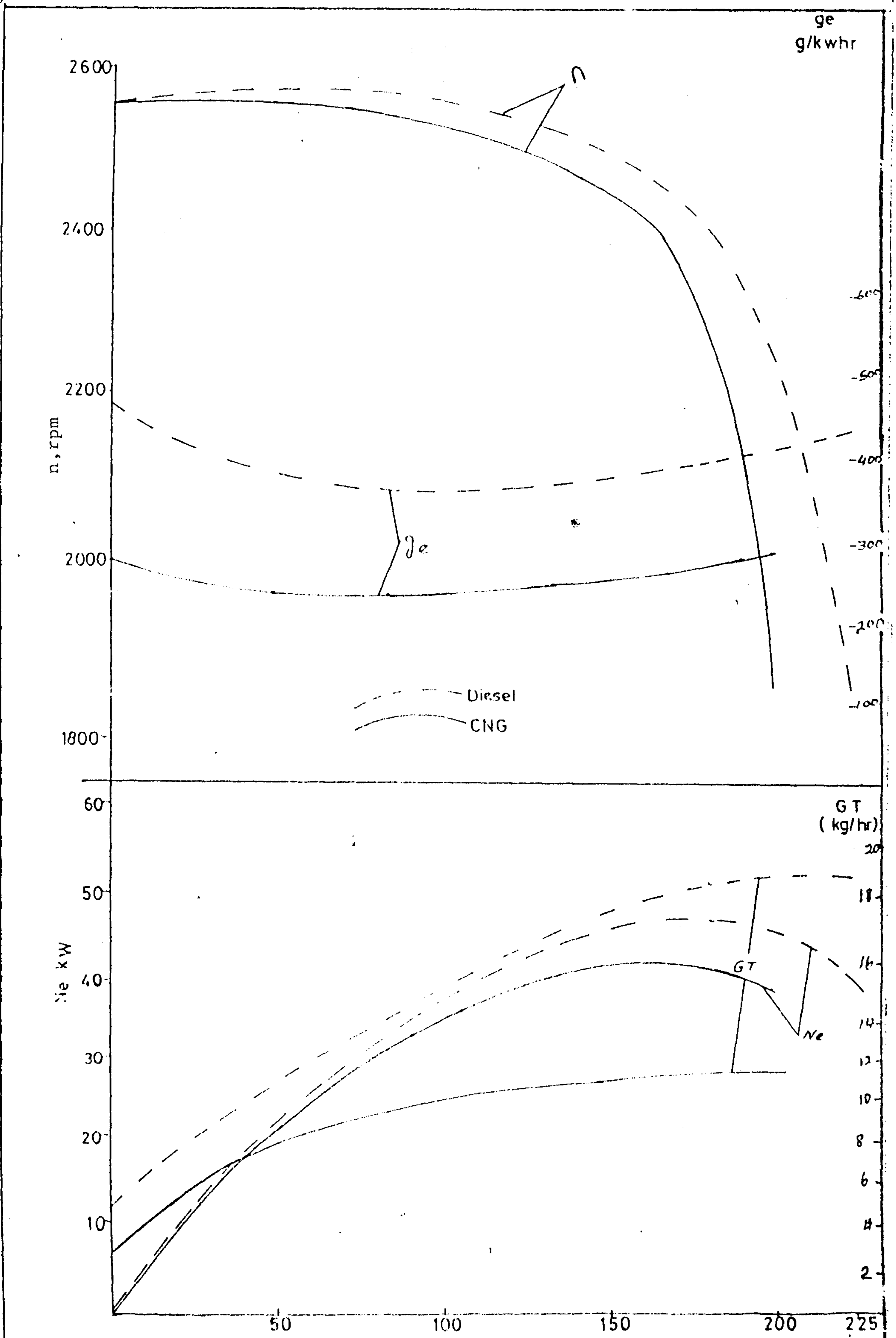


FIG:4.4.1b: CONTROL CHARACTERISTICS AS FUNCTION OF TORQUE

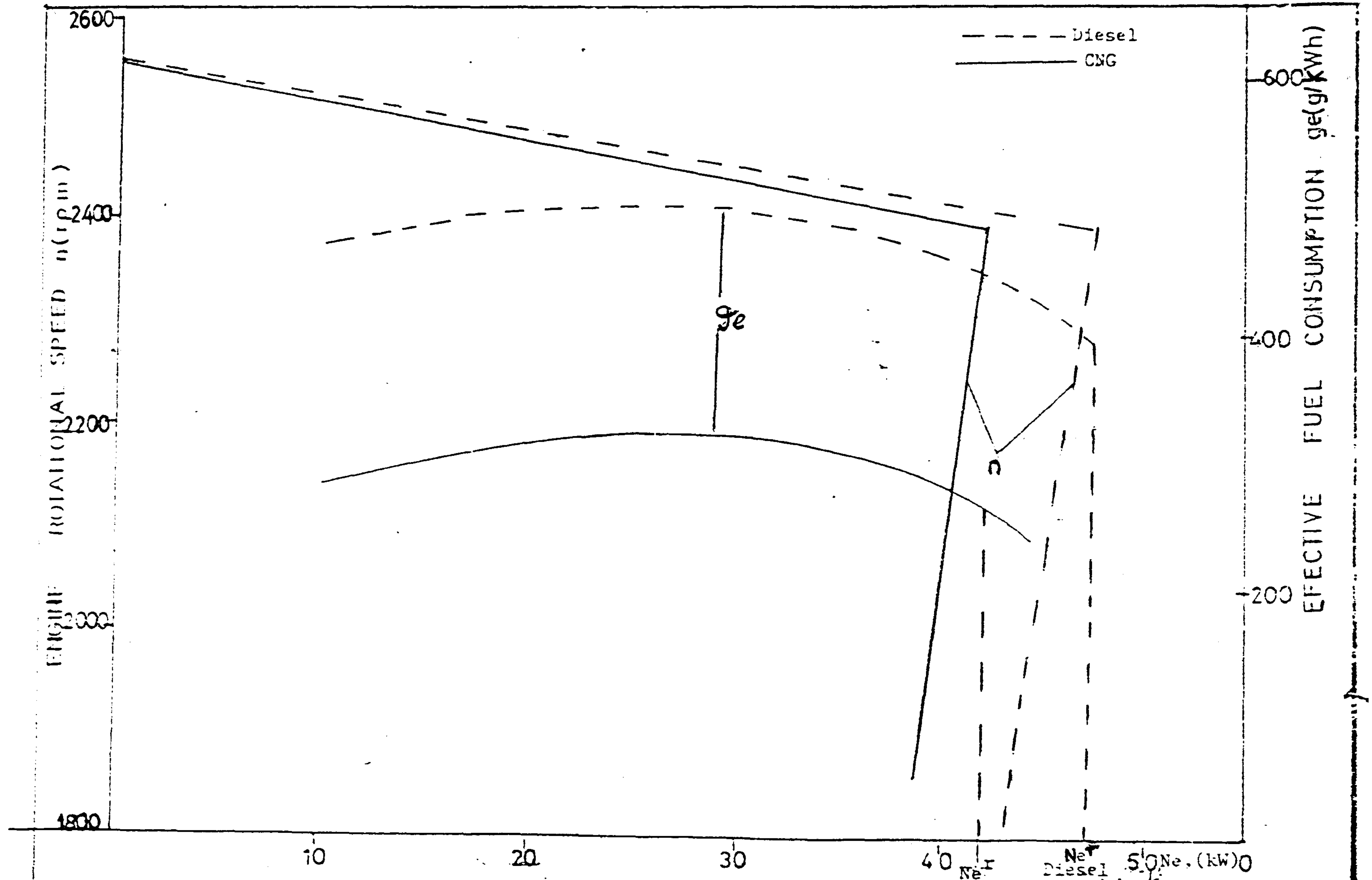


FIG: 4.4.1c – CONTROL CHARACTERISTICS AS A FUNCTION OF EFFECTIVE POWER, Ne

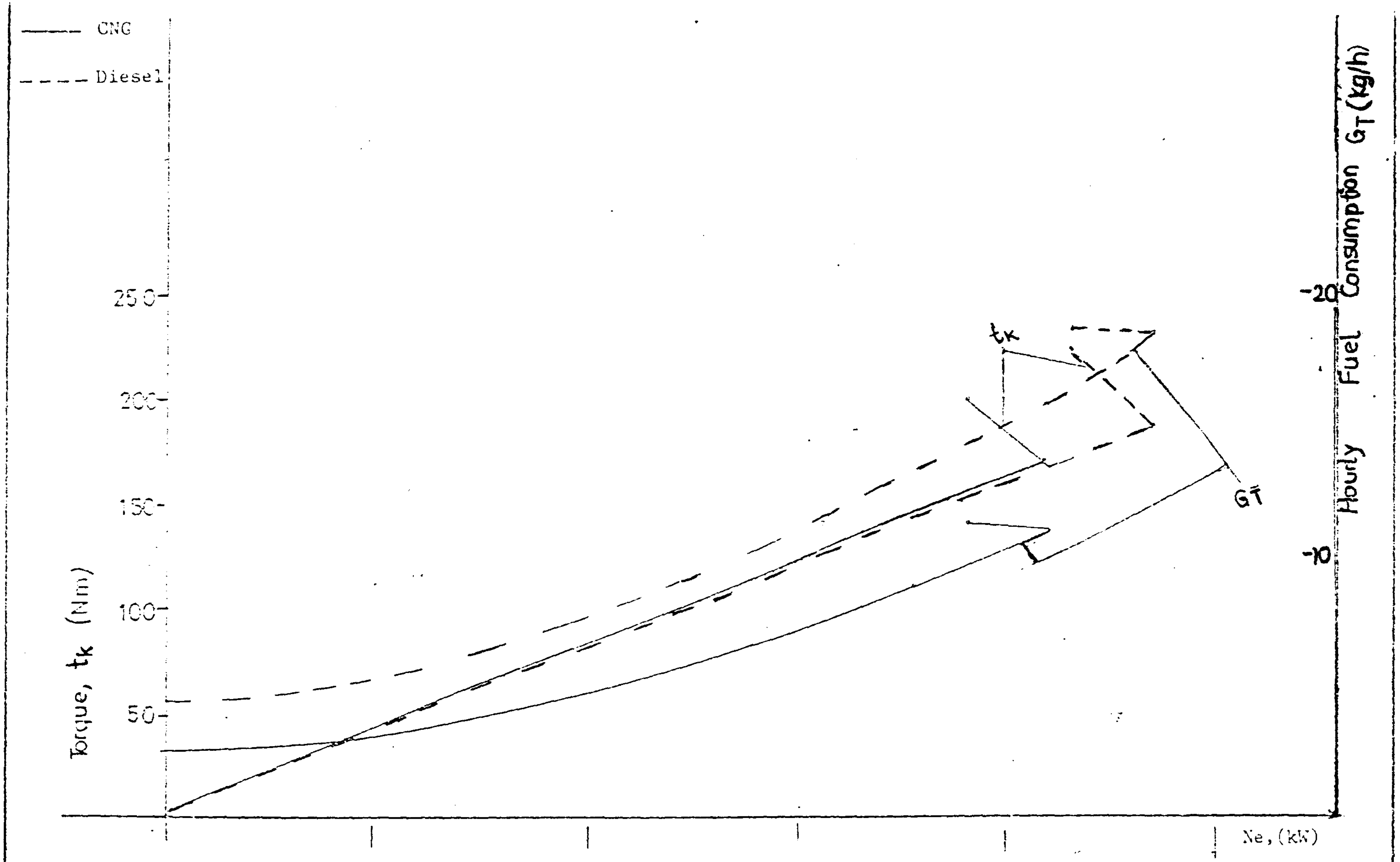


FIG: 4.4.1d: CONTROL CHARACTERISTICS AS A FUNCTION OF EFFECTIVE POWER, Ne

4.5 Engine Dynamic Design

This was required in the determination of forces and moments acting on the crank mechanisms of the engine and also in the determination of the moment of inertia and mass of the flywheel. The results of the forces and moment acting on the crank mechanism of the engines are presented in Tables 3.4a and b of appendix 1 for tractor diesel and CNG engine respectively.

4.5.1. Forces Acting On The Gudgeon Pin.

Two forces act on the gudgeon pins, they are the gas pressure P_r, and the forces of inertia P_i.

4.5.1.1 The Gas Pressure P_r

For the diesel engine, the gas pressure, P_r at intake i.e. between $0^0 - 180^0$ crankshaft rotation is the same at 270.27N and at compression from $180^0 - 360^0$, the P_r forces increased with an increase in the angle of rotation of the crankshaft and it is at maximum at the beginning of combustion process ($360^0 - 540^0$) (i.e. at 370^0, P_r = 86,204.36N). This reduces till the end of the combustion process, the forces reduced to 2,315.23N and during the expansion/exhaust process, the forces reduced to 131.36N. The value of P_r at 370^0 implies that pressure was highest at that point and hence the force P_r is also at maximum. Pressure at combustion P_z for the diesel engine was determined to be 13.34 mPa. At exhaust the pressure is reduced to 0.453 mPa, hence the lower values of P_r during the exhaust process.

For the CNG engine, the gas pressure at intake was 135.14N, this is low compared to the diesel engine and this may be as a result of the difference in their pressures at intake. The P_r value is at maximum of 67,775.44N at the commencement of the combustion process and reduces to 1823.61N at the completion of the process, the P_r value during expansion process reduces to 102.71N and this may be also due to the low value of the pressure at exhaust (0.376 mPa).

Greater forces of P_r act on the gudgeon pin of the diesel engine compared to the CNG engine. Therefore, the diesel engine is exposed to more wear and tear hence the need for frequent maintenance and services of the engine component. According to Mr. Stone, **(African review, 2002)**, a diesel engine will need a change between 250 – 500 hours while a gas unit will need a change only after approximately 3000 hrs. Another significant maintenance cost is a major overhaul, with gas engine only requiring one every 60,000 hrs, fewer spare part and less cost to the customer.

4.5.1.2 Forces Of Inertia

The two engines have the same rotational speed of 2,400 rpm at rated condition hence, they have the same angular velocity, ω of 251.327 rad per second, the mass of reciprocating component are the same and is equal to 3.219 kg and the crank radius is 0.052m. Therefore the two engines have the same force of inertia of 1st order and second order at angles of rotation of the crankshaft ($\alpha = 0$) 10573.11N and -2643.28N respectively. This implies that the two engines are exposed to the same forces of inertia. The angle of rotation ranges between 0 - 720^0 at 10^0 interval in radians and the results are as presented in the table 3.4a &b. The resultant inertia forces, P_j ranged between -13216.38N at 0^0 to -13216.38N at 720^0 with the maximum 7929.83 at 180^0. Fig. 4.5.1.2 shows the forces of inertia, P_{ji}, P_{j2} and the resultant P_j against the angle of rotation of the crank α(in radians.)

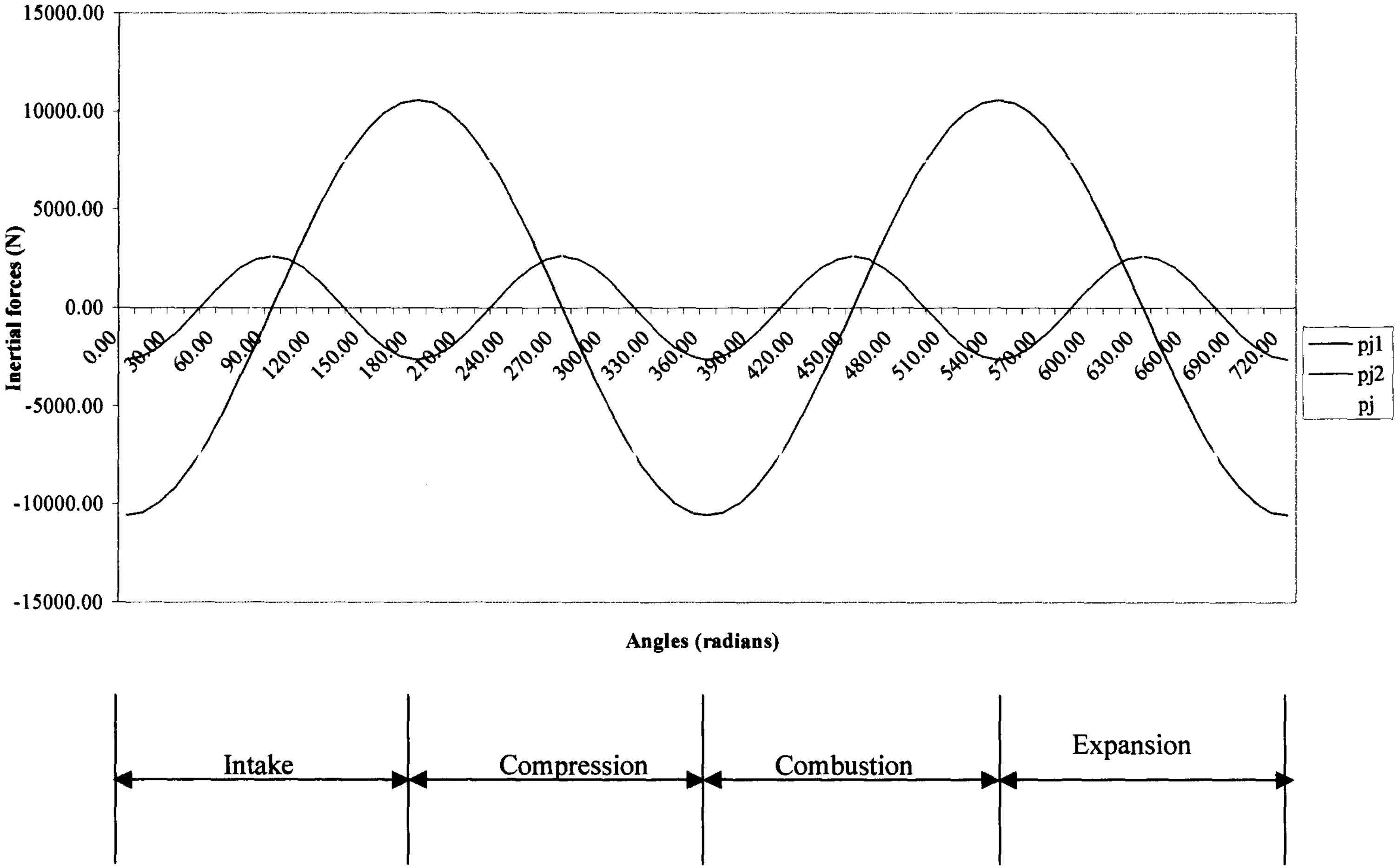


Fig. 4.5.1.2 Inertia forces (p_{j1}, p_{j2}, p_j) against angles of rotation of crank

4.5.1.3 The Resultant Force, Pres. Acting On The Gudgeon Pin

The resultant force is the sum of the gas pressure P_r and the inertia force, P_j at specific crank angle rotation. Since the two engines have the same force of inertia, but different in the resultant force of the engines. Since the diesel engine has a greater P_r value, the resultant of the diesel engine is also greater than the CNG engine. This is also presented in tables 3.4a & b of appendix 1. Figs 4.5.1.3a & b show the graphs of the force of inertia, P_j, gas pressure, P_r and the resultant force P_{res} against the angle of rotation of the crankshaft for the two engines respectively. The resultant force ranged between -12946.114N& 73.30801N(maximum at 370^0) for diesel and between 13081.24N& 54930.88N (maximum at 370^0) for the CNG. It shows that the diesel engine is exposed to more force (load) than the CNG engine, and thus it needs to be maintained more frequently than the tractor CNG engine.

4.5.2 Force Acting On The Crank Pin (Of The Crankshaft)

4.5.2.1 Force Acting On The Connecting Rod

The forces acting on the connecting rod, Pc_R was determined as the resultant forces Pres per the cosine of the reflecting angle between the connecting rod axis and cylinder axis as the crankshaft rotates at angle α.

The result is as presented in tables 3.4 a & b of appendix 1 for both engines. The forces acting on the connecting rod ranges between -1,946.11N and 73,377.19N for diesel engine and -13,081.24N and 54,930.88N for the design engine. But the forces acting on the connecting rod of the diesel engine is greater than the one acting on the connecting rod of the CNG engine due to the differences in the gas pressure forces and the resultant force.

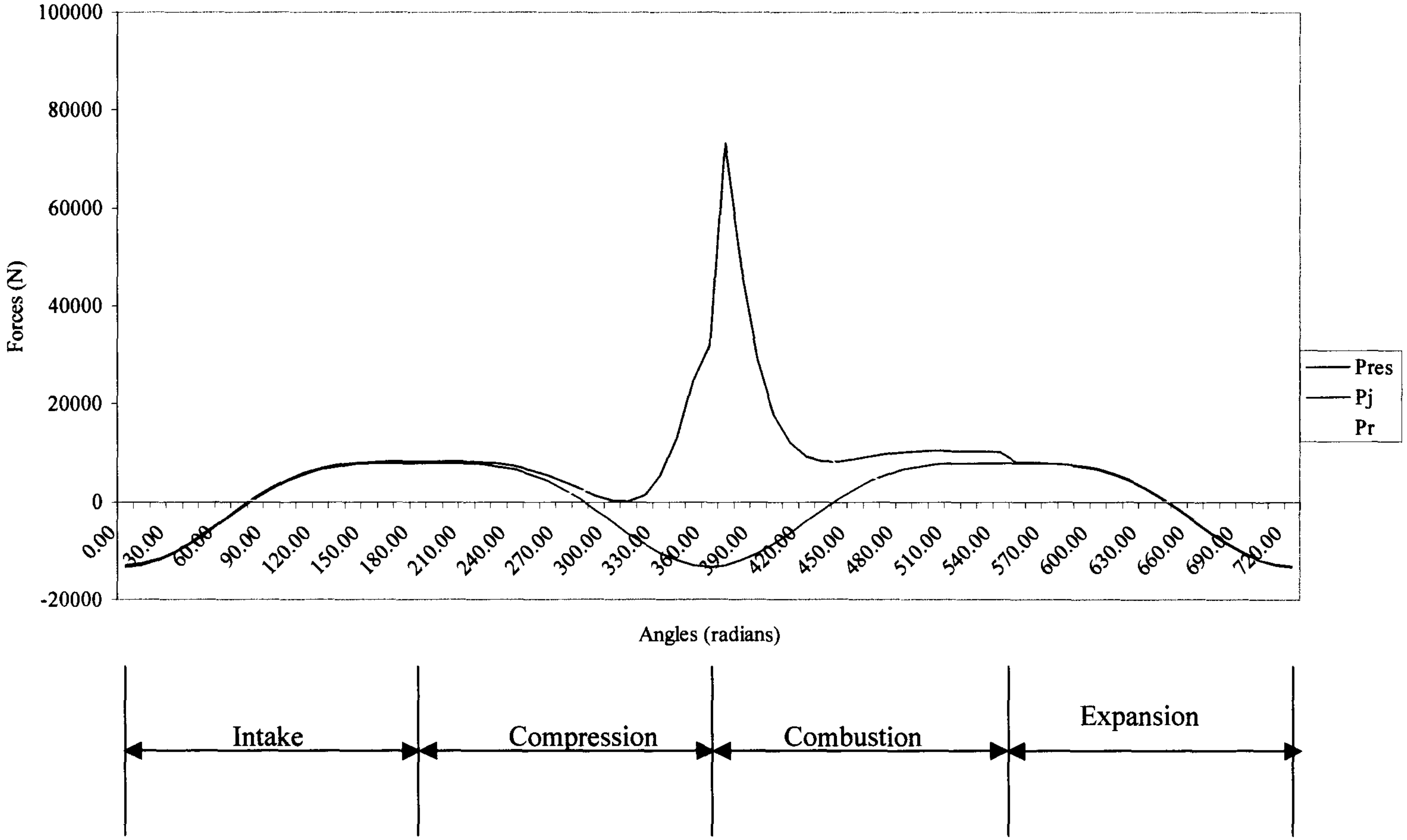


Fig. 4.5.1.3a Gas pressure force (P_r), resultant inertia force (p_j) and resultant force (P_{res}) acting on the gudgeon pin against angles of rotation of the crank (for tractor diesel engine)

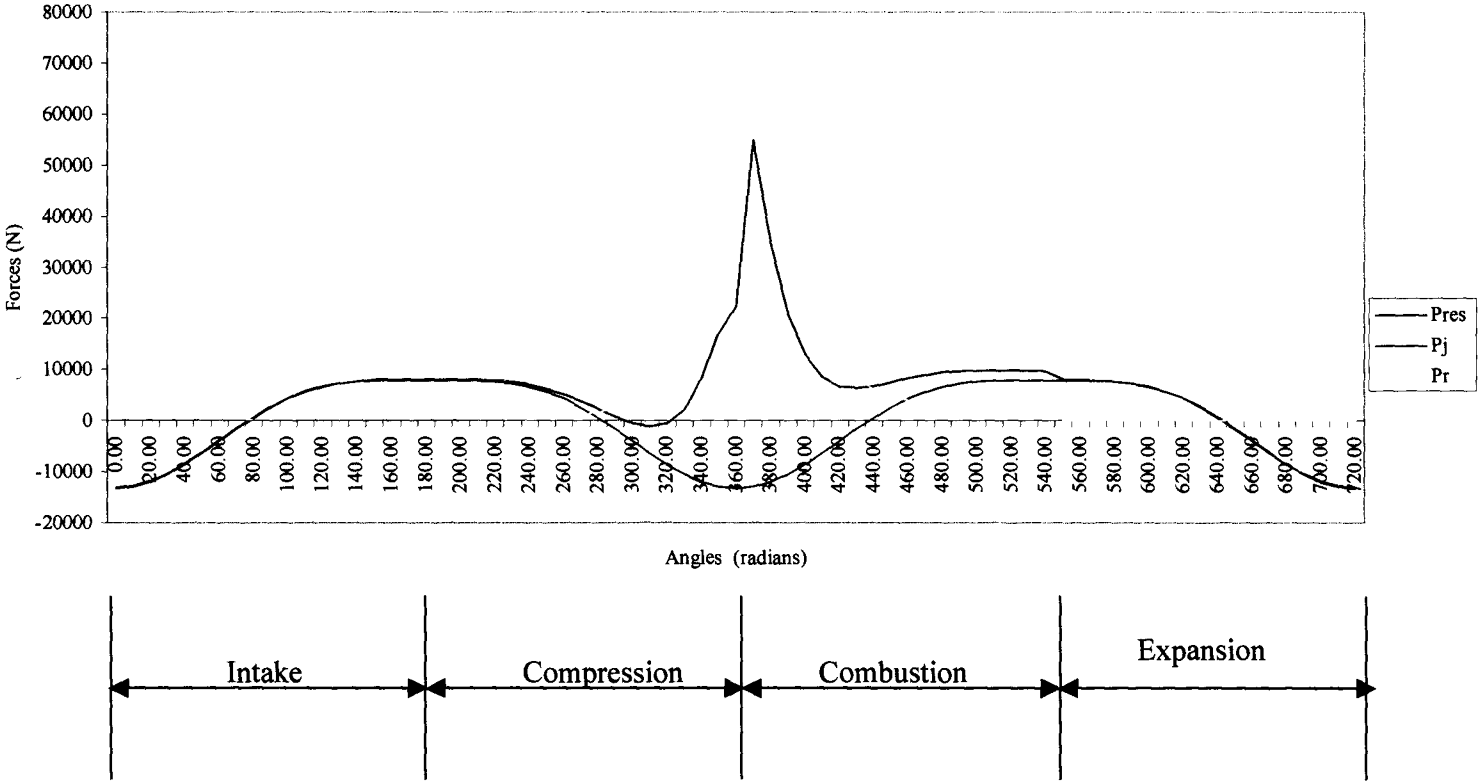


Fig. 4.5.1.3b: Gas pressure force (P_r), resultant inertia force (P_j) and resultant force (P_{res}) acting on the gudgeon pin against angles of rotation of the crank (for CNG engine)

4.5.2.2 Centrifugal Force Of Inertia

The two engines have the same centrifugal force of inertia of -6651.30N, since the two engines have the same angular velocity, ω, crank radius r and the Mass of connecting rod M = 2.70 kg. The results are as presented in tables 3.4 a & b of appendix 1.

The geometric sum of the P_c (centrifugal force of inertia) and Pc_R (force acting on the connecting rod) gives the resultant force R^1 for a simple crank pin and a cylinder. The resultant force R^1 varies from -19597.41N and 66725.89N for the tractor diesel engine and –19,732.75N and 48,2779.58N for the tractor CNG engine. The difference may be as a result of the cumulative effect of the gas pressure forces and the resultant force.

4.5.2.3 Component Of The Connecting Rod Force

The component forçes are two: forces Z_1 acting through the crank radius and the tangential force T, acting perpendicular to the crank radius.

The values of the force Z_1 acting through the crank radius at different angles of rotation is also presented in tables 3.4 a & b of appendix 1 for the two engines. It ranged between -12,946.11N and 71,641.15N for the diesel engine and between –13081.24 and 53,631.27N for the CNG engine.

The tangential force, T_1 acting perpendicular to the crank radius also ranged between –6361.27N and 20,243.69N for the diesel engine and between –6464.99N and 14862.11N for the CNG engine. The differences between the values of T and Z may be also as a result of the cumulative effect of the difference in the resultant force Pres.

Figs 4.5.2.3a & b show the graphs of the tangential force for the two engines, which were used in the determination of the moment of inertia and mass of the flywheels.

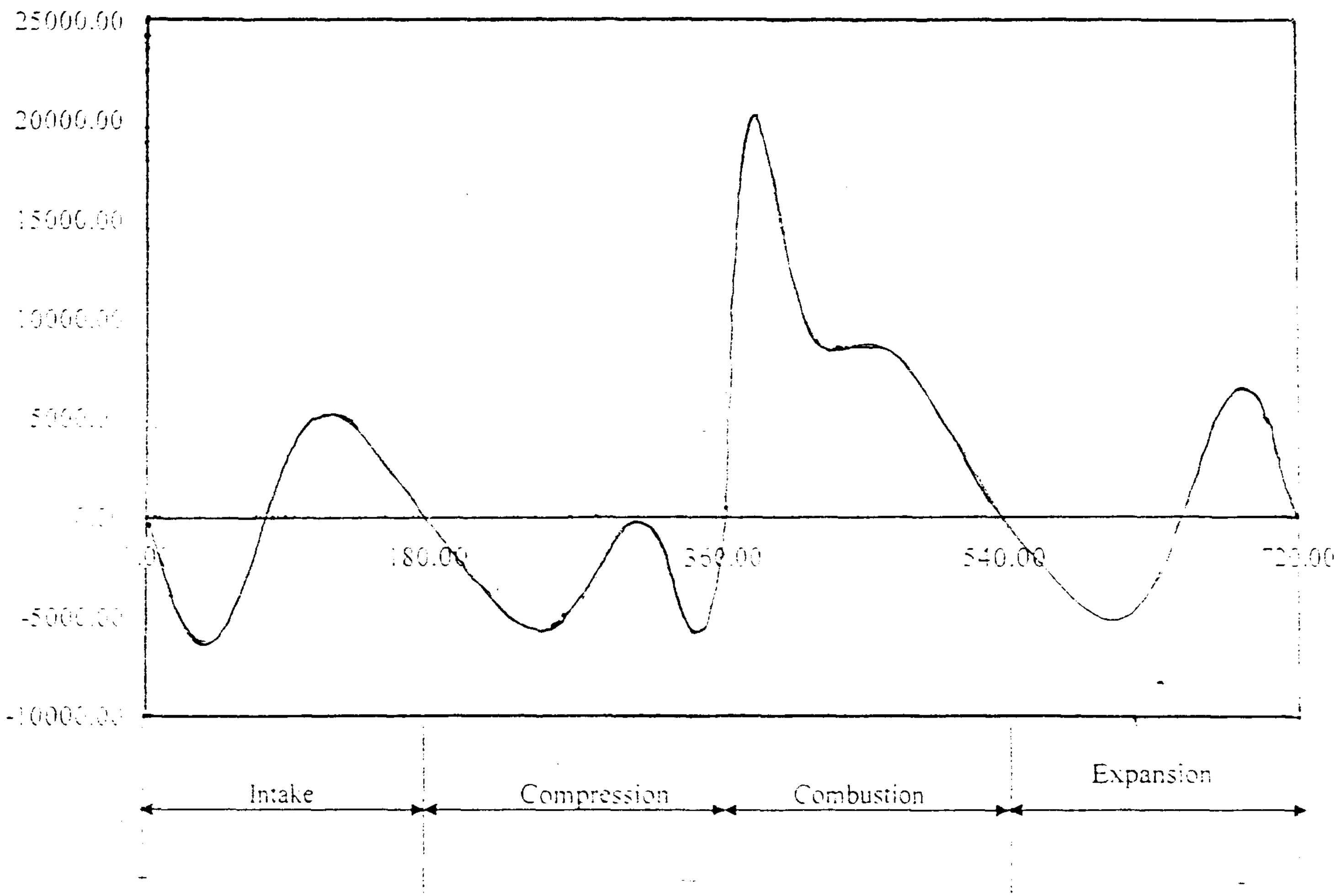


Fig. 4.5.2.3a: Tangential force acting perpendicular to the crank radius of tractor diesel engine

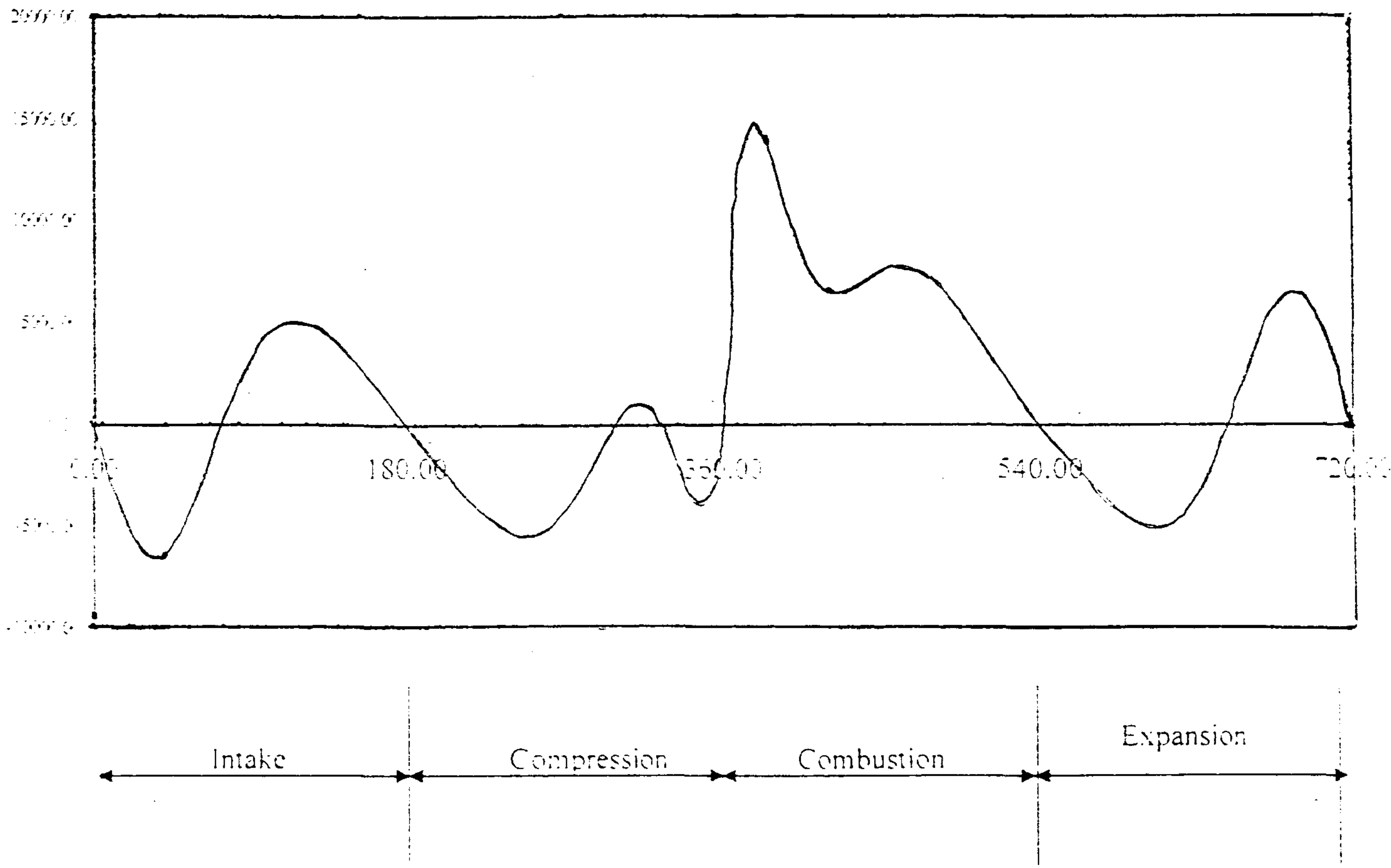


Fig. 4.5.2.3b: Tangential force acting perpendicular to the crank radius of tractor CNG engine (design)

The resultant force, R was also determined for the various angles of rotation of the crankshaft and the result is also presented in tables 3.4 a & b of appendix 1 for both engines. Figs 4.5.2.3c&d show the graphs of the resultant force, R acting on the crank pin against the angle of rotation of the crankshaft (α). The values ranged between 4,474.64N at 330^0 crank angle rotation and 139,273.82N maximum at 370^0 crank angle rotation for the diesel engine and between 2873.32N at 330^0 crank angle rotation and 102600.73N maximum at 370^0 for the tractor CNG engine. The resultant force on diesel engine is larger than that on the design engine. This calls for frequent maintenance of the diesel engine.

4.5.3 Moment Of Inertia Of The Flywheel

The moment of inertia of the flywheel of the diesel engine is determined to be 0.5219 kg/m^2 and the moment of inertia of the flywheel of the tractor CNG engine is 0.4622 kg/m^2. This implies that more torque is developed by the diesel engine than the CNG engine This is as a result of the different in the resultant forces acting on the crank mechanism of the two engines. The resultant force on the diesel engine is greater than that of the CNG engine. The tangential force exerted on the crank pin of the diesel engine is greater than that exerted on the CNG engine from Figs. 4.5.2.3 a & b respectively, hence the excess area where excess work is determined from the graphs. The excess work determined to be 1236.65 Nm and 1095.15 Nm, for the diesel and the CNG engines respectively. This may account for the difference in the moment of inertia of the two engines.

Statistically, the moment of inertia of the tractor CNG engine compared with the tractor diesel engine is correlated (100%) and has a standard error of less than 1% (Appendix 2).

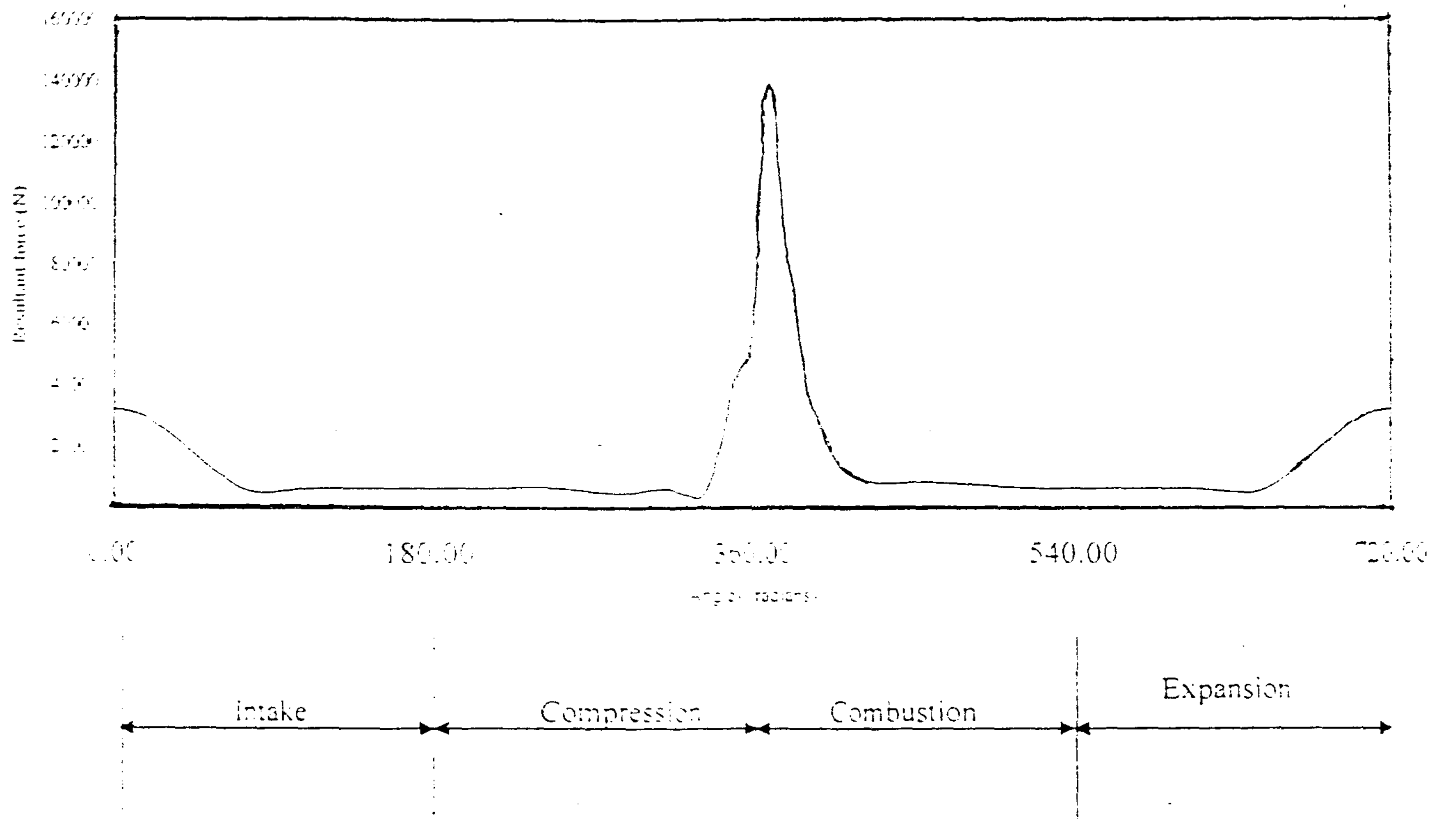


Fig. 4.5.2.3c: Graph of resultant force (R) acting on the crank pin against angles of rotation of the crank (for tractor diesel engine)

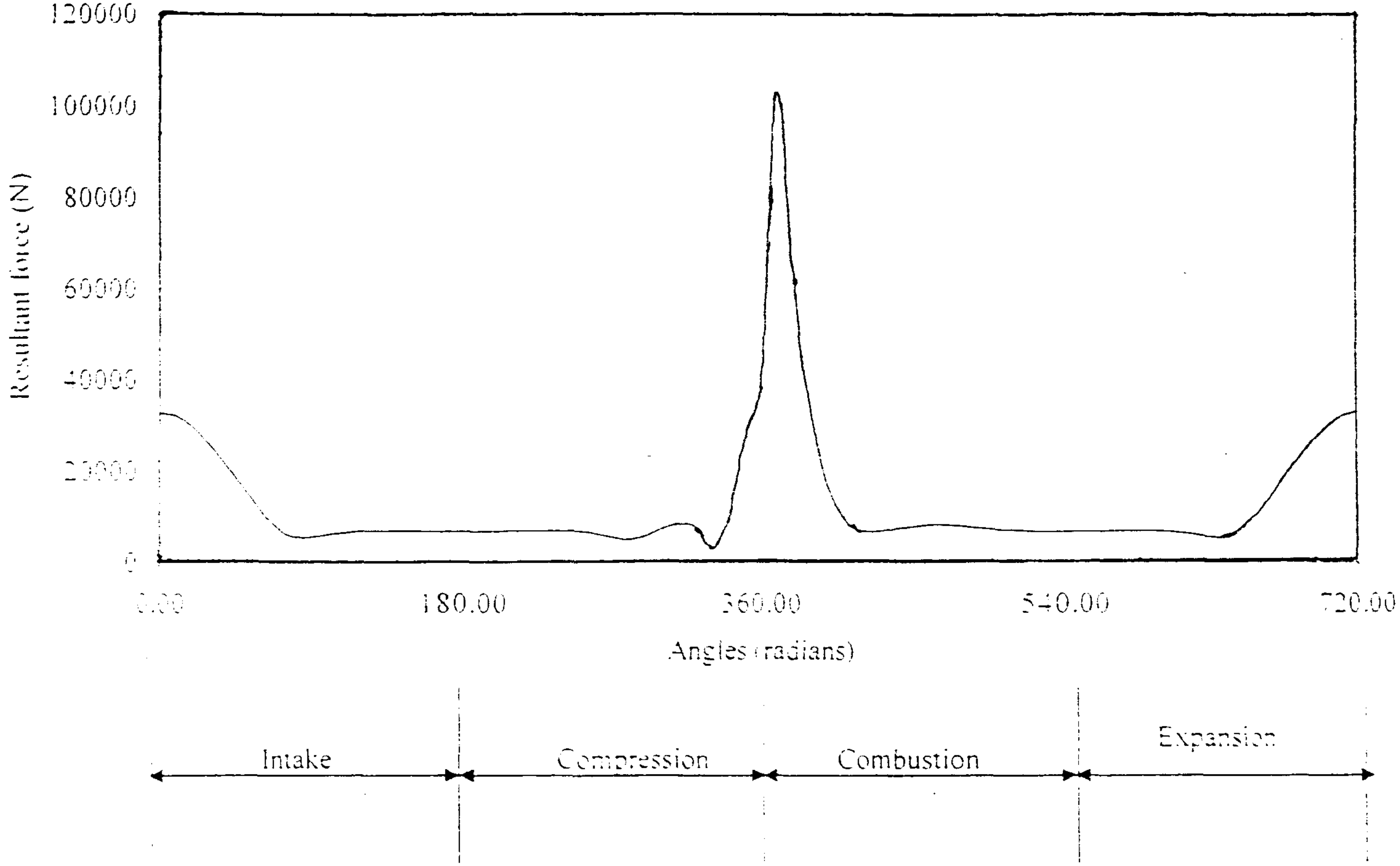


Fig. 4.5.2.3d: Graph of resultant force (R) acting on the crank pin against angles of rotation of the crank (for tractor CNG engine)

4.5.3.1 Mass of the Flywheel

The mass of the flywheel of the diesel engine was 13.05 kg, while that of the design engine is 11.55 kg. The diameter of the flywheel for the two engines is taken to be the same (0.40m). The difference in the mass of the flywheels of the two is also due to the difference in the moment of inertia of the two engines. In the design of the flywheel, it must not be excessively large because such a flywheel may cause the engine to slow in accelerating. It would be needlessly heavy and expensive. It will also increase the gyroscopic moment during turning, and for machine not protected by slip clutches and shear pins.

4.5.4 Interpretation Of The Dynamics And Kinematics Graph Of Tractor Engines

Fig. 4.5.1.2 shows the graph of the forces of inertia of the 1st order P_{j1}, P_{j2} for second order and the resultant inertia forces P_j against the angle of rotation of Crank. At intake the maximum inertia force of the 1^{st} order is at 180^0 (10573.11N) i.e. at the end of intake process. Similarly, the total resultant inertia force is maximum at 180^0 (7929.83N) crank angle while the 2^{nd} order force of inertia is at maximum at 90^0 (2643.28N). The graph is sinusoidal and during the compression stroke the minimum values of P_{j1} and P_j are obtained at 360^0 crank angle rotation (-10573.11N and -13216.38N) respectively while the second order force of inertia P_{j2} was at minimum at the completion of intake, compression, combustion and expansion processes. It implies that at intake and combustion the forces of inertia P_{j1} and P_j are at maximum while at compression and expansion they are at minimum. The forces of inertia acting on the Gudgeon are the same for the two engines.

Figs. 4.5.1.3 (a) and (b) show the gas pressure force, P_r, resultant inertia force P_j and the resultant force P_{res} acting on the Gudgeon Pin of the two engines. The two graphs are similar, the inertia force P_j at maximum point (7929.83N for diesel Engine and CNG Engine at the completion of intake and combustion processes. Gas pressure p_r and the P_{res} (Resultant Force) are at maximum point during the combustion process i.e. 86024.36N and 73308.01N respectively for the diesel engine and 67775.44N and 54879.09N for the design engine.

Figs. 4.5.2.3 (a) and (b) show the tangential force, T acting on the crank pin of the two engines. The two graphs are similar, during the intake, compression and expansion processes, the work done on the engine crank pin is less compared to the work done on the crank pin during the combustion process. This implies that the crank pin was subjected to more tangential force. But CNG engine has lower tangential force at combustion compared with diesel engine.

Figs 4.5.2.3 (c) and (d) show the resultant force, R acting on the crank pin and during the combustion process of the two engines. Resultant force was at the maximum, but the resultant R acting on the crank pin of the CNG engine is lower in magnitude than the resultant force acting on the crank pin of the diesel engine. This may be due to the difference in their gas pressures.

4.5.5 Constructional Features Of A Tractor CNG Engine

The tractor CNG engine operates on the principle of spark ignition engine and a tractor diesel engine is a compression ignition engine. Therefore, a tractor diesel engine that is converted to a tractor CNG engine would have the following constructional features to suit its operation as a spark ignition engine.

CNG cylinder with values and vapour box: this is for the storage of fuel, a hose for the refilling of the cylinder. These are high-pressure cylinders designed for storage of

CNG at a pressure of 20 mPa. A typical tank capacity is 60 litres, the number of cylinders required depends on the size of the tractor. A tractor is a heavy weight machine and can carry more cylinders placed behind the driver's seat or placed in the floor of the Roll-over Protective Structure (R.O.P.S) and any other place adjudged to be appropriate.

The vapour bag assembly: This is made up of PVC and it's designed to cover the cylinder valve. It is tubular in shape and has a threaded flange at one end screwed into the cylinder neck threads and a screwed cap at the other end to give access to the cylinder valves.

TNI Electronic Regulator: Pressure regulator that reduces the gas pressure.

ON-Off Refueling Electronic Valve: Opens and controls gas flow from 20 mPa to just above the atmospheric pressure.

Mixer: special air valve diaphragm carburetor that mixes the CNG and the pre-cleaned air.

A gaseous metering system installed.

Ignition system and ignition trigger – the fuel injector of a tractor diesel engine is replaced with six-cylinder contact-less distributor ignition system with spark plugs located in place of injector.

Fuel Electronic Gauge: to be installed on the vehicle dashboard to control gas level in the cylinder.

CHAPTER FIVE

5.0 CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

The effective power of the tractor CNG engine was determined to be 41.7 kW, which falls within the same category as the diesel engine (its Effective power is 47kW). This shows that with the rated power, the tractor CNG engine can be used to do most farm operations.

From the fuel economy indices and analysis, the tractor engine using Compressed Natural Gas, as alternative fuel is more economical than its diesel counterpart, hence the use of Compressed Natural Gas as alternative fuel for tractor engine is to be advocated.

The dynamic characteristics of the engines show that the tractor CNG engine crank mechanism (reciprocating components) are exposed to a lesser force compared with the tractor diesel engine, thereby reducing the susceptibility of the tractor CNG engine to frequent repair and maintenance services unlike the tractor diesel engine.

The commissioning of the two CNG stations at Warri and Egbin near Lagos by the Nigeria Gas Company (NGC) has further boosts efforts at utilization of natural gas as a source of cheap automobile fuel in Nigeria (NGC, release, 1990).

5.2 RECOMMENDATIONS

The numerous advantages of the use of CNG alternative fuel for tractor engines and automobiles generally include: production of emission free exhaust (low or no pollutant), it is economical, and has low cost of maintenance for CNG engine to mention a few. It is therefore recommended that the use of CNG as alternative fuel for tractors and automobiles be encouraged.

However, there are some hindrances that may affect its usage in totality especially in the third–world countries like Nigeria. Such hindrances are as stated below and the appropriate recommendations made.

Vehicles that use natural gas are handicapped by the limited sources of supply. Natural gas is widely used but refueling facilities are not yet common even in larger cities. It is therefore recommended that at state capitals major cities and local government headquarters, Government should provide refueling facilities to be at the reach of the users. Like in Nigeria for instance, we only have refueling stations in Warri and Egbin, that of Egbin is presently not functioning. Therefore, more refueling stations need to be provided and adequate maintenance should be adopted.

It is widely known that Natural gas is highly inflammable and it's therefore needed to be properly stored. A well designed storage and dispensing facilities should be put in place. Farmers in the rural areas using tractor CNG engines could refill from stations and keep in a cool environment for onward installation during operation when the need arises.

The positioning of the CNG cylinder on the tractor is another area that researcher and designer should look into, because it is an established fact that the CNG cylinder / tank is an added weight to the vehicle, unlike diesel or petrol driven automobiles whose tank has been designed and appropriately installed. Vehicle or automobile to run on pure CNG should be designed to have a storage tank, which could be used for a long distance before refueling.

Finally, lack of awareness is another hindrance facing the use of CNG as alternative fuel. In the developed world, the use of Natural Gas Vehicle (NGV_s) is not new. But in this part of the world, popular for as in seminars and workshop on the use of CNG as alternative fuels for automobiles generally must be employed to consolidate gains already made for both living in the cities and people living in rural areas where bulk of the population resides.

REFERENCES

Ababio, O.Y. (2001): Chemistry for Senior Secondary School. Third edition, First Publisher Limited, Nigeria. Pp 476

Adgidzi, D. (1988): Improvement of the Operational Indices of Tractors using Turbo-super charging. Unpublished M. Sc. Thesis submitted to the Department of Tractors and Automobiles, Agro-Technical University, Minsk, Minsk.

Adgidzi, D. (2002): Lecture Note on Advanced Tractor design (Unpublished). Agric. Engineering Department, FUT. Minna.

African Review (2002): African Review of Business and Technology, Pp 44.

Bassey, E , Barley, .B. & Jaeger, S. (1993): LPG Conversion and HC Emissions Specification of Light Duty Vehicles. SAE Paper no 932745, Society of Automotive Engineers, Warrandale, Pa

Bechtold, R.L. (1997): Alternative Fuels Guidebook, properties, storage, Dispensing and Vehicle facility modifications. SAE International, Warandale, Pa, U.S.A.

Chemical Engineering News (1996): "Growth of Top 50 Chemicals Slowed in 1995 from very high 1994 rates". Vol. 74 No 15; facts & Figure for the U.S. Chemical Industry.

Clean Fuels Report (1991): As reported in the Clean Fuel Reports, Volume 3, No. 3. Pp 75 – 77, 96 –97.

Cowart, J.S. (1995): Powertrain Development of the 1996 Ford Flexible Fuel Taurus. SAE paper No 952751, Society of Automobile Engineers, Warrandale Pa. USA.

Doe Poe + 0100P (1991): Assessment of Costs and Benefits of Flexible and Alternative Fuel use in the U.S Transportation Sector. Technical Report Seven: Environmental Health and Safety Concern" Pp ix.

Duke J.A and Bagby, M.O (1982): Comparison of Oil seed yields: A Preliminary Review Proceeding of the International Conference on Plants and vegetable oils as fuel. August 2 – 4, 1982, American Society of Agricultural Engineers St. Joseph, Mich.

Fuel Cell Vehicle Htm. (1980): Alternative Fuel Vehicles Retrieved April 3rd, 2002 from www.bath.ac.uk/enotahel/alternat.htm.3k from http:// www.google.com/search.

Fuel Chemistry. Htm (1980): Retrieved April 3rd, 2002 from www./altfuels.org/fuelchemistry.html from www.google.com/search.

Health Effects Institute, HEI (1994): "HEI Strategic plan for Vehicle Emissions an Fuel, 1994-1998."

Heywood J. B. (1996): Internal Combustion Engine Fundamentals (1st Edition). R. R. Donnelly and Sons Company', USA. Pp 43-58, Pp. 823-899.

Howell J.R. and Buckius, R.O (1992): Fundamental of Engineering Thermodynamics, (2nd Edition). Published by McGraw Hill, Inc. USA.

Indiamart (2002): India Travel Portals Retrieved April, 3rd, 2002 from http://auto.indiamart.com/going-green/alternative.fuel.html.

Kris (2002): Alternative Fuel: College Term paper. Com. Department of Energy http://www.even.doe.gov/Encarta © 2001 Retrieved April, 30th 2002 from.. //college Term papers – science – Alternative Fuel - Free Term papers

Liljedahl, J.B, Turnquist, P.K Smith, D.W and Hoki, M. (1989): Tractor and Their Power Units. 4th edition. Van Nostrand Reinhold New York.

Liquefied Natural Gas.Htm (1/4/80): Retrieved April 3rd, 2002 from http://www.google.com/search

LNG Review (2003): Diesel Conversions to Natural Gas engine on:/LNG.htm Retrieved July 3rd, 2003 from http://www.ask.com.

Nuclear Energy, (1961): Power Engineering in Liljedahi et al (1989) Pp 14

Oppenhelmer, E.J. (1981): Natural Gas; the New Energy Leader, Pen and Podium Production, New York, N.Y. (in Bechtold, R.L.1987), Pp. 20

Nigeria Gas Company (1990); Staff Information Bulleting, NGC

Nigeria Gas Company Limited, a subsidiary of NNPC, Nigeria, Volume 1. Pp 2-6.

Peter, J. (2001): Relative Economics of Environmentally- Friendly Fuel. Financial daily from the Hindus group of publications. Retrieved April. 30th. 2002 from //A: Relative Economic of environmentally-Friendly Fuels. Htm.

Ray' D.L. (1973): The Nation's Energy Future' WASH 1282'U.S. Atomic Energy Commission. Washington D.C.

Richard S. (1985): Fundamental of Internal combustion Engine Macmillan Publisher Ltd. Hondmills. Basing stoke, Hampshire RG21 2xs. and London.

SAE (1994)-By Bums V. R etal: emission with reformulated Gasoline and Methanol Blends in 1992 and 1993 Model Year Vehicles "SAE paper No 94/969. society of Automobile Engineers, Warrandale, Pa.

Scarlott, C.A. (1987): Changing Energy Science, Science Month Pp 221ff in Liljedahl et al (1989): Tractors and their power units.

Sporn, P (1957) in Liljedahi et al (1989): Energy Requirement and the Role of energy in an expanding economy, Agric Engineering, Pp 657 ff

Steyr (1980) Operating Instruction for Steyr 8075/8065 5th Edition printed in Austria by Vereindruekel Steyr.

Swain, M.R. , ADT, R.R., Jr and Pappas, J.M (1983): Experimental Hydrogen Fueled Automotive Engine design Data -Base Project – US Department of Energy report DOE/CS/51212.1, vol. 3 May 1983.

Technocarb (2003): Alternative Fuel Power, Natural Gas Properties Comparison to other Fuel retrieved July 3rd, 2003. from http://www.technocarb.com.nat.gas.properties.htm

US Department of Energy, (1996): Monthly energy Review – DOE/EIA35/96/07). Energy Information Administration Office of Energy Market and End use

US Department of Energy (1994): Alternative to Traditional Transportation Fuel: Green House Gas Emission. Volume 2, Pp 1-2

Wang M.Q (1996): Development and Use of the GREET Model to estimate Fuel Cycle energy use and Emissions of Various Transportation Technologies and Fuels. Argonne Natural Laboratory Argonne, Ill

Williams Bartok and Adelf Sarofun (1991): Fossil Fuel Combustion: A source book. Wiley & Sons Incorporation Publisher, Canada

Yisa, M.G. (1997): Lecture Note on Farm Power & Machinery, Unpublished, Agric Engineering Dept. FUT., Minna.

Table 3.4a: Dynamic characteristics values of the Tractor Diesel Engine

ANGLE (deg)	ALPHA (rad)	BETA (rad)	P_r (N)	P_{j1} (N)	P_{j2} (N)	P_Σ (N)	P_{re} (N)	$P_{c'}$ (N)	P_c (N)	R'	Z	T	R
0.00	0.00	0.00	270.27	-10573.11	-2643.28	-13216.38	-12946.11	-12946.11	-6651.30	-19597.41	-12946.11	0.00	32543.53
10.00	0.17	0.04	270.27	-10412.48	-2483.87	-12896.35	-12626.08	-12637.99	-6651.30	-19289.29	-12338.99	-2732.80	31746.12
20.00	0.35	0.09	270.27	-9935.47	-2024.87	-11960.34	-11690.07	-11733.04	-6651.30	-18384.34	-10641.94	-4940.97	29443.81
30.00	0.52	0.13	270.27	-9156.58	-1321.64	-10478.22	-10207.95	-10288.64	-6651.30	-16939.94	-8197.30	-6217.75	25894.82
40.00	0.70	0.16	270.27	-8099.47	-459.00	-8558.47	-8288.20	-8397.33	-6651.30	-15048.63	-5481.74	-6361.27	21493.30
50.00	0.87	0.19	270.27	-6796.26	459.00	-6337.26	-6066.99	-6181.41	-6651.30	-12832.71	-2992.94	-5408.52	16724.33
60.00	1.05	0.22	270.27	-5286.55	1321.64	-3964.92	-3694.65	-3784.41	-6651.30	-10435.71	-1137.75	-3609.33	12123.20
70.00	1.22	0.24	270.27	-3616.22	2024.87	-1591.35	-1321.08	-1359.11	-6651.30	-8010.41	-151.80	-1350.61	8273.21
80.00	1.40	0.25	270.27	-1836.00	2483.87	647.87	918.14	947.30	-6651.30	-5704.00	-70.25	944.69	5851.02
90.00	1.57	0.25	270.27	0.00	2643.28	2643.28	2913.55	3009.10	-6651.30	-3642.20	-752.27	2913.55	5272.59
100.00	1.75	0.25	270.27	1836.00	2483.87	4319.87	4590.14	4735.92	-6651.30	-1915.38	-1945.35	4317.93	5792.22
110.00	1.92	0.24	270.27	3616.22	2024.87	5641.08	5911.35	6081.55	-6651.30	-569.75	-3364.34	5066.21	6414.32
120.00	2.09	0.22	270.27	5286.55	1321.64	6608.19	6878.46	7045.57	-6651.30	394.27	-4760.28	5194.22	6785.42
130.00	2.27	0.19	270.27	6796.26	459.00	7255.26	7525.53	7667.45	-6651.30	1016.15	-5962.18	4821.02	6906.91
140.00	2.44	0.16	270.27	8099.47	-459.00	7640.47	7910.74	8014.90	-6651.30	1363.60	-6887.87	4098.28	6878.48
150.00	2.62	0.13	270.27	9156.58	-1321.64	7834.94	8105.21	8169.28	-6651.30	1517.98	-7529.90	3168.25	6795.66
160.00	2.79	0.09	270.27	9935.47	-2024.87	7910.60	8180.87	8210.94	-6651.30	1559.64	-7927.63	2138.29	6717.40
170.00	2.97	0.04	270.27	10412.48	-2483.87	7928.61	8198.88	8206.62	-6651.30	1555.32	-8136.19	1072.87	6667.75
180.00	3.14	0.00	270.27	10573.11	-2643.28	7929.83	8200.10	8200.10	-6651.30	1548.80	-8200.10	0.00	6651.30
190.00	3.32	-0.04	280.84	10412.48	-2483.87	7928.61	8209.45	8217.20	-6651.30	1565.90	-8146.67	-1074.25	6667.88
200.00	3.49	-0.09	313.35	9935.47	-2024.87	7910.60	8223.95	8254.18	-6651.30	1602.88	-7969.38	-2149.55	6719.58
210.00	3.67	-0.13	369.66	9156.58	-1321.64	7834.94	8204.60	8269.46	-6651.30	1618.16	-7622.23	-3207.11	6806.94
220.00	3.84	-0.16	446.59	8099.47	-459.00	7640.47	8087.06	8193.54	-6651.30	1542.24	-7041.39	-4189.63	6913.29
230.00	4.01	-0.19	550.24	6796.26	459.00	7255.26	7805.50	7952.70	-6651.30	1301.40	-6183.99	-5000.38	6988.81
240.00	4.19	-0.22	669.59	5286.55	1321.64	6608.19	7277.78	7454.60	-6651.30	803.30	-5036.63	-5495.76	6937.18
250.00	4.36	-0.24	889.93	3616.22	2024.87	5641.08	6531.01	6719.05	-6651.30	67.75	-3717.01	-5597.28	6681.81
260.00	4.54	-0.25	1179.65	1836.00	2483.87	4319.87	5499.52	5674.18	-6651.30	-977.12	-2330.75	-5173.38	6140.51

Table 3.4a: Dynamic characteristic values of the Tractor Diesel Engine

270.00	4.71	-0.25	1576.65	0.00	2643.28	2643.28	4219.93	4358.32	-6651.30	-2292.98	-1089.58	-4219.93	5408.28
280.00	4.89	-0.25	2144.38	-1836.00	2483.87	647.87	2792.25	2880.93	-6651.30	-3770.37	-213.65	-2872.99	4911.87
290.00	5.06	-0.24	2995.32	-3616.22	2024.87	-1591.35	1403.97	1444.39	-6651.30	-5206.91	161.33	-1435.36	5245.77
300.00	5.24	-0.22	4313.40	-5286.55	1321.64	-3964.92	348.48	356.95	-6651.30	-6294.35	107.31	-340.44	6196.39
310.00	5.41	-0.19	6509.56	-6796.26	459.00	-6337.26	172.30	175.55	-6651.30	-6475.75	85.00	-153.60	6392.60
320.00	5.59	-0.16	9933.45	-8099.47	-459.00	-8558.47	1374.98	1393.08	-6651.30	-5258.22	909.40	-1055.31	4475.03
330.00	5.76	-0.13	15840.10	-9156.58	-1321.64	-10478.22	5361.88	5404.27	-6651.30	-1247.03	4305.76	-3265.97	4474.64
340.00	5.93	-0.09	25353.86	-9935.47	-2024.87	-11960.34	13393.52	13442.75	-6651.30	6791.45	12192.67	-5660.96	19810.18
350.00	6.11	-0.04	37493.62	-10412.48	-2483.87	-12896.35	24597.27	24620.49	-6651.30	17969.19	24037.99	-5323.86	42343.19
360.00	6.28	0.00	45130.98	-10573.11	-2643.28	-13216.38	31914.60	31914.60	-6651.30	25263.30	31914.60	0.00	57177.89
370.00	6.46	0.04	86204.36	-10412.48	-2483.87	-12896.35	73308.01	73377.19	-6651.30	66725.89	71641.15	15866.86	139273.82
380.00	6.63	0.09	59855.82	-9935.47	-2024.87	-11960.34	47895.48	48071.53	-6651.30	41420.23	43601.21	20243.69	87398.24
390.00	6.81	0.13	39512.56	-9156.58	-1321.64	-10478.22	29034.34	29263.87	-6651.30	22612.57	23315.49	17685.08	49215.32
400.00	6.98	0.16	26586.21	-8099.47	-459.00	-8558.47	18027.74	18265.12	-6651.30	11613.82	11923.37	13836.46	27302.88
410.00	7.16	0.19	18428.81	-6796.26	459.00	-6337.26	12091.55	12319.58	-6651.30	5668.28	5964.94	10779.22	15859.49
420.00	7.33	0.22	13265.62	-5286.55	1321.64	-3964.92	9300.70	9526.67	-6651.30	2875.37	2864.10	9085.94	10746.90
430.00	7.50	0.24	9913.37	-3616.22	2024.87	-1591.35	8322.02	8561.63	-6651.30	1910.33	956.27	8508.06	8978.00
440.00	7.68	0.25	7654.49	-1836.00	2483.87	647.87	8302.36	8566.03	-6651.30	1914.73	-635.24	8542.44	8637.73
450.00	7.85	0.25	6092.08	0.00	2643.28	2643.28	8735.36	9021.84	-6651.30	2370.54	-2255.46	8735.36	8736.11
460.00	8.03	0.25	4973.50	1836.00	2483.87	4319.87	9293.37	9588.52	-6651.30	2937.22	-3938.62	8742.25	8799.42
470.00	8.20	0.24	4170.05	3616.22	2024.87	5641.08	9811.13	10093.61	-6651.30	3442.31	-5583.83	8408.44	8676.87
480.00	8.38	0.22	3581.92	5286.55	1321.64	6608.19	10190.11	10437.68	-6651.30	3786.38	-7052.12	7694.98	8359.30
490.00	8.55	0.19	3145.74	6796.26	459.00	7255.26	10401.00	10597.15	-6651.30	3945.85	-8240.30	6663.11	7927.13
500.00	8.73	0.16	2827.79	8099.47	-459.00	7640.47	10468.26	10606.10	-6651.30	3954.80	-9114.70	5423.25	7485.73
510.00	8.90	0.13	2597.47	9156.58	-1321.64	7834.94	10432.41	10514.88	-6651.30	3863.58	-9691.91	4077.94	7113.30
520.00	9.08	0.09	2448.40	9935.47	-2024.87	7910.60	10359.00	10397.08	-6651.30	3745.78	-10038.34	2707.60	6850.35
530.00	9.25	0.04	2347.03	10412.48	-2483.87	7928.61	10275.64	10285.34	-6651.30	3634.04	-10197.07	1344.62	6699.35
540.00	9.42	0.00	2315.23	10573.11	-2643.28	7929.83	10245.06	10245.06	-6651.30	3593.76	-10245.06	0.00	6651.30
550.00	9.60	-0.04	131.36	10412.48	-2483.87	7928.61	8059.97	8067.58	-6651.30	1416.28	-7998.34	-1054.69	6666.03
560.00	9.77	-0.09	131.36	9935.47	-2024.87	7910.60	8041.96	8071.52	-6651.30	1420.22	-7793.02	-2101.98	6710.50

Table 3.4a: Dynamic characteristic values of the Tractor Diesel Engine

570.00	9.95	-0.13	131.36	9156.58	-1321.64	7834.94	7966.30	8029.28	-6651.30	1377.98	-7400.85	-3113.96	6780.24
580.00	10.12	-0.16	131.36	8099.47	-459.00	7640.47	7771.83	7874.16	-6651.30	1222.86	-6766.92	-4026.32	6851.85
590.00	10.30	-0.19	131.36	6796.26	459.00	7255.26	7386.62	7525.92	-6651.30	874.62	-5852.13	-4732.03	6867.87
600.00	10.47	-0.22	131.36	5286.55	1321.64	6608.19	6739.55	6903.29	-6651.30	251.99	-4664.14	-5089.32	6735.60
610.00	10.65	-0.24	131.36	3616.22	2024.87	5641.08	5772.44	5938.64	-6651.30	-712.66	-3285.28	-4947.16	6360.65
620.00	10.82	-0.25	131.36	1836.00	2483.87	4319.87	4451.23	4592.60	-6651.30	-2058.70	-1886.48	-4187.26	5753.05
630.00	11.00	-0.25	131.36	0.00	2643.28	2643.28	2774.64	2865.63	-6651.30	-3785.67	-716.41	-2774.64	5288.41
640.00	11.17	-0.25	131.36	-1836.00	2483.87	647.87	779.23	803.97	-6651.30	-5847.33	-59.62	-801.76	5961.11
650.00	11.34	-0.24	131.36	-3616.22	2024.87	-1591.35	-1459.99	-1502.02	-6651.30	-8153.32	-167.77	1492.63	8453.90
660.00	11.52	-0.22	131.36	-5286.55	1321.64	-3964.92	-3833.56	-3926.69	-6651.30	-10577.99	-1180.52	3745.03	12340.50
670.00	11.69	-0.19	131.36	-6796.26	459.00	-6337.26	-6205.90	-6322.94	-6651.30	-12974.24	-3061.46	5532.36	16963.21
680.00	11.87	-0.16	131.36	-8099.47	-459.00	-8558.47	-8427.11	-8538.07	-6651.30	-15189.37	-5573.61	6467.89	21747.07
690.00	12.04	-0.13	131.36	-9156.58	-1321.64	-10478.22	-10346.86	-10428.65	-6651.30	-17079.95	-8308.85	6302.36	26159.34
700.00	12.22	-0.09	131.36	-9935.47	-2024.87	-11960.34	-11828.98	-11872.46	-6651.30	-18523.76	-10768.40	4999.68	29715.78
710.00	12.39	-0.04	131.36	-10412.48	-2483.87	-12896.35	-12764.99	-12777.03	-6651.30	-19428.33	-12474.74	2762.87	32022.48
720.00	12.57	0.00	131.36	-10573.11	-2643.28	-13216.38	-13085.02	-13085.02	-6651.30	-19736.32	-13085.02	0.00	32821.35

Table 3.4b: Dynamic characteristics values of the Tractor CNG engine

ANGLE (deg)	ALPHA (rad)	BETA (rad)	P_r (N)	P_{j1} (N)	P_{j2} (N)	P_j (N)	P_{res} (N)	P_{gr} (N)	P_c (N)	R'	Z_1	T	R
0.00	0.00	0.00	135.14	-10573.11	-2643.28	-13216.38	-13081.24	-13081.24	-6651.30	-19732.54	-13081.24	0.00	32813.79
10.00	0.17	0.04	135.14	-10412.48	-2483.87	-12896.35	-12761.21	-12773.25	-6651.30	-19424.55	-12471.04	-2762.05	32014.96
20.00	0.35	0.09	135.14	-9935.47	-2024.87	-11960.34	-11825.20	-11868.66	-6651.30	-18519.96	-10764.96	-4998.08	29708.37
30.00	0.52	0.13	135.14	-9156.58	-1321.64	-10478.22	-10343.08	-10424.84	-6651.30	-17076.14	-8305.82	-6300.06	26152.14
40.00	0.70	0.16	135.14	-8099.47	-459.00	-8558.47	-8423.33	-8534.24	-6651.30	-15185.54	-5571.11	-6464.99	21740.16
50.00	0.87	0.19	135.14	-6796.26	459.00	-6337.26	-6202.12	-6319.09	-6651.30	-12970.39	-3059.60	-5528.99	16956.71
60.00	1.05	0.22	135.14	-5286.55	1321.64	-3964.92	-3829.78	-3922.82	-6651.30	-10574.12	-1179.36	-3741.34	12334.58
70.00	1.22	0.24	135.14	-3616.22	2024.87	-1591.35	-1456.21	-1498.13	-6651.30	-8149.43	-167.33	-1488.76	8448.96
80.00	1.40	0.25	135.14	-1836.00	2483.87	647.87	783.01	807.87	-6651.30	-5843.43	-59.91	805.65	5958.06
90.00	1.57	0.25	135.14	0.00	2643.28	2643.28	2778.42	2869.54	-6651.30	-3781.76	-717.38	2778.42	5287.90
100.00	1.75	0.25	135.14	1836.00	2483.87	4319.87	4455.01	4596.50	-6651.30	-2054.80	-1888.08	4190.81	5754.06
110.00	1.92	0.24	135.14	3616.22	2024.87	5641.08	5776.22	5942.53	-6651.30	-708.77	-3287.43	4950.40	6362.08
120.00	2.09	0.22	135.14	5286.55	1321.64	6608.19	6743.33	6907.16	-6651.30	255.86	-4666.76	5092.17	6736.93
130.00	2.27	0.19	135.14	6796.26	459.00	7255.26	7390.40	7529.77	-6651.30	878.47	-5855.12	4734.45	6868.92
140.00	2.44	0.16	135.14	8099.47	-459.00	7640.47	7775.61	7877.99	-6651.30	1226.69	-6770.21	4028.28	6852.56
150.00	2.62	0.13	135.14	9156.58	-1321.64	7834.94	7970.08	8033.09	-6651.30	1381.79	-7404.36	3115.43	6780.66
160.00	2.79	0.09	135.14	9935.47	-2024.87	7910.60	8045.74	8075.32	-6651.30	1424.02	-7796.68	2102.97	6710.69
170.00	2.97	0.04	135.14	10412.48	-2483.87	7928.61	8063.75	8071.36	-6651.30	1420.06	-8002.09	1055.18	6666.07
180.00	3.14	0.00	135.14	10573.11	-2643.28	7929.83	8064.97	8064.97	-6651.30	1413.67	-8064.97	0.00	6651.30
190.00	3.32	-0.04	160.48	10412.48	-2483.87	7928.61	8089.09	8096.72	-6651.30	1445.42	-8027.23	-1058.50	6666.38
200.00	3.49	-0.09	193.54	9935.47	-2024.87	7910.60	8104.14	8133.93	-6651.30	1482.63	-7853.28	-2118.23	6713.57
210.00	3.67	-0.13	243.44	9156.58	-1321.64	7834.94	8078.38	8142.24	-6651.30	1490.94	-7504.97	-3157.77	6792.65
220.00	3.84	-0.16	305.32	8099.47	-459.00	7640.47	7945.79	8050.41	-6651.30	1399.11	-6918.39	-4116.44	6885.31
230.00	4.01	-0.19	388.96	6796.26	459.00	7255.26	7644.22	7788.38	-6651.30	1137.08	-6056.21	-4897.06	6941.11
240.00	4.19	-0.22	506.95	5286.55	1321.64	6608.19	7115.14	7288.00	-6651.30	636.70	-4924.07	-5372.94	6873.87
250.00	4.36	-0.24	668.04	3616.22	2024.87	5641.08	6309.12	6490.77	-6651.30	-160.53	-3590.72	-5407.11	6580.94
260.00	4.54	-0.25	898.49	1836.00	2483.87	4319.87	5218.36	5384.09	-6651.30	-1267.21	-2211.59	-4908.90	6016.59

Table 3.4b: Dynamic characteristic value of the Tractor CNG Engine

270.00	4.71	-0.25	1213.08	0.00	2643.28	2643.28	3856.36	3982.83	-6651.30	-2668.47	-995.71	-3856.36	5319.56
280.00	4.89	-0.25	1676.52	-1836.00	2483.87	647.87	2324.39	2398.21	-6651.30	-4253.09	-177.85	-2391.60	5035.18
290.00	5.06	-0.24	2365.67	-3616.22	2024.87	-1591.35	774.32	796.62	-6651.30	-5854.68	88.98	-791.63	5819.80
300.00	5.24	-0.22	3435.92	-5286.55	1321.64	-3964.92	-529.00	-541.85	-6651.30	-7193.15	-162.90	516.78	7374.18
310.00	5.41	-0.19	5215.32	-6796.26	459.00	-6337.26	-1121.94	-1143.10	-6651.30	-7794.40	-553.47	1000.17	8407.57
320.00	5.59	-0.16	7998.95	-8099.47	-459.00	-8558.47	-559.52	-566.89	-6651.30	-7218.19	-370.06	429.44	7600.39
330.00	5.76	-0.13	12791.07	-9156.58	-1321.64	-10478.22	2312.85	2331.14	-6651.30	-4320.16	1857.29	-1408.78	2837.32
340.00	5.93	-0.09	20441.83	-9935.47	-2024.87	-11960.34	8481.49	8512.67	-6651.30	1861.37	7721.05	-3584.82	10231.01
350.00	6.11	-0.04	30080.35	-10412.48	-2483.87	-12896.35	17184.00	17200.22	-6651.30	10548.92	16793.28	-3719.32	27594.01
360.00	6.28	0.00	35634.48	-10573.11	-2643.28	-13216.38	22418.10	22418.10	-6651.30	15766.80	22418.10	0.00	38184.89
370.00	6.46	0.04	67775.44	-10412.48	-2483.87	-12896.35	54879.09	54930.88	-6651.30	48279.58	53631.27	11878.09	102600.73
380.00	6.63	0.09	47123.29	-9935.47	-2024.87	-11960.34	35162.95	35292.20	-6651.30	28640.90	32010.27	14862.11	62445.55
390.00	6.81	0.13	31835.90	-9156.58	-1321.64	-10478.22	21355.68	21524.50	-6651.30	14873.20	17149.28	13007.94	34563.65
400.00	6.98	0.16	21494.26	-8099.47	-459.00	-8558.47	12935.79	13106.12	-6651.30	6454.82	8555.61	9928.34	17996.80
410.00	7.16	0.19	14939.03	-6796.26	459.00	-6337.26	8601.77	8763.99	-6651.30	2112.69	4243.38	7668.19	9959.96
420.00	7.33	0.22	10763.00	-5286.55	1321.64	-3964.92	6798.08	6963.24	-6651.30	311.94	2093.43	6641.11	7063.30
430.00	7.50	0.24	8034.73	-3616.22	2024.87	-1591.35	6443.38	6628.90	-6651.30	-22.40	740.40	6587.42	6626.43
440.00	7.68	0.25	6200.49	-1836.00	2483.87	647.87	6848.36	7065.85	-6651.30	414.55	-523.99	7046.40	7047.25
450.00	7.85	0.25	4927.42	0.00	2643.28	2643.28	7570.70	7818.98	-6651.30	1167.68	-1954.75	7570.70	7611.50
460.00	8.03	0.25	4016.83	1836.00	2483.87	4319.87	8336.70	8601.46	-6651.30	1950.16	-3533.18	7842.31	8000.49
470.00	8.20	0.24	3361.86	3616.22	2024.87	5641.08	9002.94	9262.15	-6651.30	2610.85	-5123.86	7715.80	8114.72
480.00	8.38	0.22	2881.69	5286.55	1321.64	6608.19	9489.88	9720.44	-6651.30	3069.14	-6567.52	7166.21	7974.54
490.00	8.55	0.19	2520.79	6796.26	459.00	7255.26	9776.05	9960.42	-6651.30	3309.12	-7745.18	6262.75	7674.68
500.00	8.73	0.16	2260.15	8099.47	-459.00	7640.47	9900.62	10030.98	-6651.30	3379.68	-8620.45	5129.17	7333.08
510.00	8.90	0.13	2071.67	9156.58	-1321.64	7834.94	9906.61	9984.93	-6651.30	3333.63	-9203.43	3872.41	7032.08
520.00	9.08	0.09	1944.64	9935.47	-2024.87	7910.60	9855.24	9891.47	-6651.30	3240.17	-9550.17	2575.93	6815.54
530.00	9.25	0.04	1859.49	10412.48	-2483.87	7928.61	9788.10	9797.34	-6651.30	3146.04	-9713.25	1280.82	6690.95
540.00	9.42	0.00	1823.01	10573.11	-2643.28	7929.83	9752.84	9752.84	-6651.30	3101.54	-9752.84	0.00	6651.30
550.00	9.60	-0.04	102.71	10412.48	-2483.87	7928.61	8031.32	8038.90	-6651.30	1387.60	-7969.91	-1050.94	6665.68
560.00	9.77	-0.09	102.71	9935.47	-2024.87	7910.60	8013.31	8042.77	-6651.30	1391.47	-7765.26	-2094.49	6709.10

Table 3.4b: Dynamic characteristic values of the Tractor CNG Engine

570.00	9.95	-0.13	102.71	9156.58	-1321.64	7834.94	7937.65	8000.40	-6651.30	1349.10	-7374.23	-3102.76	6777.12
580.00	10.12	-0.16	102.71	8099.47	-459.00	7640.47	7743.18	7845.14	-6651.30	1193.84	-6741.97	-4011.48	6846.44
590.00	10.30	-0.19	102.71	6796.26	459.00	7255.26	7357.97	7496.73	-6651.30	845.43	-5829.43	-4713.68	6859.95
600.00	10.47	-0.22	102.71	5286.55	1321.64	6608.19	6710.90	6873.94	-6651.30	222.64	-4644.32	-5067.69	6725.52
610.00	10.65	-0.24	102.71	3616.22	2024.87	5641.08	5743.79	5909.17	-6651.30	-742.13	-3268.97	-4922.61	6349.89
620.00	10.82	-0.25	102.71	1836.00	2483.87	4319.87	4422.58	4563.04	-6651.30	-2088.26	-1874.33	-4160.31	5745.46
630.00	11.00	-0.25	102.71	0.00	2643.28	2643.28	2745.99	2836.04	-6651.30	-3815.26	-709.01	-2745.99	5292.39
640.00	11.17	-0.25	102.71	-1836.00	2483.87	647.87	750.58	774.41	-6651.30	-5876.89	-57.43	-772.28	5984.36
650.00	11.34	-0.24	102.71	-3616.22	2024.87	-1591.35	-1488.64	-1531.50	-6651.30	-8182.80	-171.06	1521.92	8491.36
660.00	11.52	-0.22	102.71	-5286.55	1321.64	-3964.92	-3862.21	-3956.04	-6651.30	-10607.34	-1189.35	3773.02	12385.37
670.00	11.69	-0.19	102.71	-6796.26	459.00	-6337.26	-6234.55	-6352.13	-6651.30	-13003.43	-3075.60	5557.90	17012.50
680.00	11.87	-0.16	102.71	-8099.47	-459.00	-8558.47	-8455.76	-8567.10	-6651.30	-15218.40	-5592.56	6489.88	21799.41
690.00	12.04	-0.13	102.71	-9156.58	-1321.64	-10478.22	-10375.51	-10457.53	-6651.30	-17108.83	-8331.86	6319.81	26213.90
700.00	12.22	-0.09	102.71	-9935.47	-2024.87	-11960.34	-11857.63	-11901.21	-6651.30	-18552.51	-10794.48	5011.79	29771.87
710.00	12.39	-0.04	102.71	-10412.48	-2483.87	-12896.35	-12793.64	-12805.71	-6651.30	-19457.01	-12502.74	2769.07	32079.48
720.00	12.57	0.00	102.71	-10573.11	-2643.28	-13216.38	-13113.67	-13113.67	-6651.30	-19764.97	-13113.67	0.00	32878.65

Appendix 2

Intake Volume (cubic meter)

Va-diesel	6.55E-04
Va-CNG	8.07E-04

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	5.776E-09

Descriptive Statistics

Column1	
Mean	0.000731
Standard Error	7.6E-05
Median	0.000731
Mode	#N/A
Standard Deviation	0.00010748
Sample Variance	1.1552E-08
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.000152
Minimum	0.000655
Maximum	0.000807
Sum	0.001462
Count	2

Compression Volume (cubic meter)

Vc-diesel	4.04E-05
Vc-CNG	5.38E-05

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	4.489E-11

Descriptive Statistics

Column1	
Mean	0.0000471
Standard Error	6.7E-06
Median	0.0000471
Mode	#N/A
Standard Deviation	9.47523E-06
Sample Variance	8.978E-11
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.0000134
Minimum	0.0000404
Maximum	0.0000538
Sum	0.0000942
Count	2

Combustion Volume (cubic meter)

Vz-diesel	4.28E-05
Vz-CNG	5.70E-05

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	5.041E-11

Descriptive Statistics

Column1	
Mean	0.0000499
Standard Error	7.1E-06
Median	0.0000499
Mode	#N/A
Standard Deviation	1.0041E-05
Sample Variance	1.0082E-10
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.0000142
Minimum	0.0000428
Maximum	0.000057
Sum	0.0000998
Count	2

Intake Pressure (mPa)

Pa-diesel	1.46E-01
Pa-CNG	1.23E-01

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	0.00013225

Descriptive Statistics

Column1	
Mean	0.1345
Standard Error	0.0115
Median	0.1345
Mode	#N/A
Standard Deviation	0.01626346
Sample Variance	0.0002645
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.023
Minimum	0.123
Maximum	0.146
Sum	0.269
Count	2

Compression Pressure (mPa)

Pc-diesel	7.02E+00
Pc-CNG	5.29E+00

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	0.748225

Descriptive Statistics

Column1	
Mean	6.155
Standard Error	0.865
Median	6.155
Mode	#N/A
Standard Deviation	1.223295
Sample Variance	1.49645
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	1.73
Minimum	5.29
Maximum	7.02
Sum	12.31
Count	2

Combustion Pressure (mPa)

Pz-diesel	1.33E+01
Pz-CNG	1.01E+01

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	2.706025

Descriptive Statistics

Column1	
Mean	11.695
Standard Error	1.645
Median	11.695
Mode	#N/A
Standard Deviation	2.326381
Sample Variance	5.41205
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	3.29
Minimum	10.05
Maximum	13.34
Sum	23.39
Count	2

Intake Temperature (K)

Ta-diesel	3.34E+02
Ta-CNG	3.45E+02

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	33.23522

Descriptive Statistics

Column1	
Mean	339.315
Standard Error	5.765
Median	339.315
Mode	#N/A
Standard Deviation	8.152941
Sample Variance	66.47045
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	11.53
Minimum	333.55
Maximum	345.08
Sum	678.63
Count	2

Compression Temperature (K)

Tc-diesel	9.88E+02
Tc-CNG	9.92E+02

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	3.861225

Descriptive Statistics

Column1	
Mean	990.225
Standard Error	1.965
Median	990.225
Mode	#N/A
Standard Deviation	2.77893
Sample Variance	7.72245
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	3.93
Minimum	988.26
Maximum	992.19
Sum	1980.45
Count	2

Combustion Temperature (K)

Tz-diesel	2.49E+03
Tz-CNG	2.25E+03

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	14762.25

Descriptive Statistics

Column1	
Mean	2366.5
Standard Error	121.5
Median	2366.5
Mode	#N/A
Standard Deviation	171.8269
Sample Variance	29524.5
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	243
Minimum	2245
Maximum	2488
Sum	4733
Count	2

Expansion Temperature (K)

Tb-diesel	1.29E+03
Tb-CNG	1.19E+03

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	2734.767

Descriptive Statistics

Column1	
Mean	1240.895
Standard Error	52.295
Median	1240.895
Mode	#N/A
Standard Deviation	73.9563
Sample Variance	5469.534
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	104.59
Minimum	1188.6
Maximum	1293.19
Sum	2481.79
Count	2

Expansion Pressure (mPa)

Pb-diesel	4.53E-01
Pb-CNG	3.76E-01

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	0.001482

Descriptive Statistics

Column1	
Mean	0.4145
Standard Error	0.0385
Median	0.4145
Mode	#N/A
Standard Deviation	0.054447
Sample Variance	0.002965
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.077
Minimum	0.376
Maximum	0.453
Sum	0.829
Count	2

Effective Power (kW)

Ne-diesel	47
Ne-CNG	41.7

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	7.0225

Descriptive Statistics

Column1	
Mean	44.35
Standard Error	2.65
Median	44.35
Mode	#N/A
Standard Deviation	3.74766594
Sample Variance	14.045
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	5.3
Minimum	41.7
Maximum	47
Sum	88.7
Count	2

Mean Effective Pressure

Pe-diesel	0.783
Pe-CNG	0.695

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	0.001936

Descriptive Statistics

Column1	
Mean	0.739
Standard Error	0.044
Median	0.739
Mode	#N/A
Standard Deviation	0.062225397
Sample Variance	0.003872
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.088
Minimum	0.695
Maximum	0.783
Sum	1.478
Count	2

Moment of Inertia of Flywheel

jfw-diesel	0.5219
jfw-CNG	0.4622

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	0.000891023

Descriptive Statistics

Column1	
Mean	0.49205
Standard Error	0.02985
Median	0.49205
Mode	#N/A
Standard Deviation	0.042214275
Sample Variance	0.001782045
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	0.0597
Minimum	0.4622
Maximum	0.5219
Sum	0.9841
Count	2

Fuel Economic Indices

Specific Fuel Consumption(g/kWhr)

ge idle diesel	461
ge idle CNG	275

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	8649

Descriptive Statistics

Column1	
Mean	368
Standard Error	93
Median	368
Mode	#N/A
Standard Deviation	131.5219
Sample Variance	17298
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	186
Minimum	275
Maximum	461
Sum	736
Count	2

Specific Fuel Consumption(g/kWhr)

ge rated diesel	392.15
ge rated CNG	263.89

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	4112.657

Descriptive Statistics

Column1	
Mean	328.02
Standard Error	64.13
Median	328.02
Mode	#N/A
Standard Deviation	90.69352
Sample Variance	8225.314
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	128.26
Minimum	263.89
Maximum	392.15
Sum	656.04
Count	2

Specific Fuel Consumption(g/kWhr)

ge max tk diesel	433.5
ge max tk CNG	290.21

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	5133.006

Descriptive Statistics

Column1	
Mean	361.855
Standard Error	71.645
Median	361.855
Mode	#N/A
Standard Deviation	101.3213
Sample Variance	10266.01
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	143.29
Minimum	290.21
Maximum	433.5
Sum	723.71
Count	2

Fuel Economic Indices

Hourly Fuel Consumption(kg/hr)

GT idle diesel	4.61
GT idle CNG	2.75

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	0.8649

Descriptive Statistics

Column1	
Mean	3.68
Standard Error	0.93
Median	3.68
Mode	#N/A
Standard Deviation	1.3152186
Sample Variance	1.7298
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	1.86
Minimum	2.75
Maximum	4.61
Sum	7.36
Count	2

Hourly Fuel Consumption(kg/hr)

GT rated diesel	18.43
GT rated CNG	11

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	13.80123

Descriptive Statistics

Column1	
Mean	14.715
Standard Error	3.715
Median	14.715
Mode	#N/A
Standard Deviatior	5.253803
Sample Variance	27.60245
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	7.43
Minimum	11
Maximum	18.43
Sum	29.43
Count	2

Hourly Fuel Consumption(kg/hr)

GT max tk diesel	18.71
GT max tk CNG	11.17

Correlation

	Column 1
Column 1	1

Covariance

	Column 1
Column 1	14.2129

Descriptive Statistics

Column1	
Mean	14.94
Standard Error	3.77
Median	14.94
Mode	#N/A
Standard Deviation	5.331585
Sample Variance	28.4258
Kurtosis	#DIV/0!
Skewness	#DIV/0!
Range	7.54
Minimum	11.17
Maximum	18.71
Sum	29.88
Count	2